

Reduced-Order Modeling of Intake Air Dynamics in Single-Cylinder Four-Stroke Engine

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ABSTRACT

This study deals with reduced-order modeling of intake air dynamics in single-cylinder four-stroke naturally-aspirated spark-ignited engines without surge tanks. It provides an approximate calculation method for embedded micro computers to estimate intake manifold pressures in real time. The calculation method is also applicable to multi-cylinder engines with individual throttle bodies since the engines can be equated with parallelization of the single-cylinder engines. In this paper, we illustrate the intake air dynamics, describe a method to estimate the intake manifold pressures, and show experimental results of the method.

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INTRODUCTION

Background

This paper proposes a real-time calculation method for automotive embedded computers to estimate transient responses of intake manifold pressures in motorcycle engines: single-cylinder four-stroke engines or multi-cylinder four-stroke engines equipped with individual throttle bodies (Fig. 1-left). The paper starts from calculation methods to estimate intake manifold pressures of four-wheel automotive engines [1, 2, 3, 4] (Fig. 1-right), and modifies them for motorcycle engines.

The intake manifold pressures are vital to control air-fuel ratios in port-injection gasoline engines; the pressures are strongly associated with masses of intake air sucked into cylinders, and moreover, they determine injection amounts per unit time of injectors. For example, usual electronic control units of four-wheel automotive engines estimate the intake manifold pressures from air mass flows through throttle valves, next they calculate the required fuel amounts from the estimated intake manifold pressures, and finally they compute the injection times from the required fuel amounts. In short, estimating the pressure, the control units compensate differences between the air masses through the throttle valves and the air masses sucked into the cylinders.

There are three big differences between motorcycle engines and four-wheel automotive engines (Fig. 1). The

differences disallow us to estimate the intake manifold pressures in the motorcycle engines by utilizing the estimation methods for the four-wheel automotive engines [1, 2, 3, 4].

- Motorcycle engines are rarely equipped with surge tanks between their throttle valves and intake valves. Hence, their intake manifold volumes are often smaller than their swept volumes. On the other hand, four-wheel automotive engines usually have surge tanks, therefore, their intake manifold volumes are much larger than their swept volumes.
- Each cylinder has its own throttle valve in motorcycle engines. Thus, the air flow through the throttle valve changes during one engine cycle. On the other hand, common four-wheel automotive engines are even-firing single-throttle engines. In such an engine, the air flow through the throttle valve is continuous and smooth since any of its cylinders are in their intake strokes.
- Equivalent moments of inertia about crank shafts of motorcycle engines are much smaller than those of four-wheel automotive engines. Therefore, engine rotational velocities of motorcycle engines increase or decrease much more sharply than those of four-wheel automotive engines. Especially in single-cylinder motorcycle engines, their rotational velocities are perturbed significantly during engine cycles.

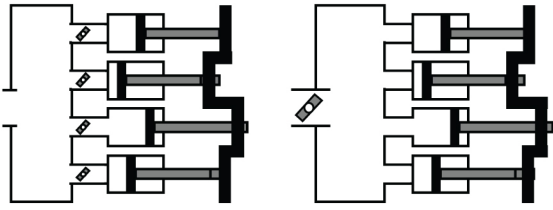


Figure 1. A common motorcycle engine with individual throttle bodies (left) and a common four-wheel automotive engine with a single throttle body (right).

The widely-used estimation methods [1, 2, 3, 4, 5] assume that the intake air flows are independent from crank angles of the engines. This assumption is validated by the above-mentioned characteristics of the four-wheel automotive engines. However, the intake air flows in the motorcycle engines depend strongly on crank angles; the intake manifold pressures decrease quickly in intake strokes and increase slowly in other strokes. Such pressure changes are caused by the characteristics of the motorcycle engines, and they preclude applications of the estimation methods for the four-wheel automotive engines.

Remark that we can estimate the air flows from the first-principle simulations [6, 7]. However, such simulations require too rich computational resource for automotive embedded computers to calculate them in real time. To compute the air flows in real time, it is required to replace the physical principles with statistical approximations.

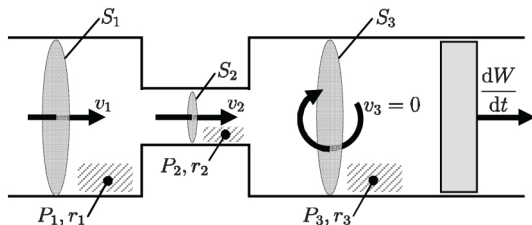


Figure 2. The flows and stagnation in the intake system.

Key Points

This paper derives a simple computational model of the intake air dynamics from a view point of enthalpy flows. The followings are the differences from the conventional estimation methods [1, 2, 3, 4] and the first-principle simulations [6, 7]. They are key points to simplify the computational model of the intake air dynamics.

1. This paper exploits structures inherent in typical motorcycle engines: small ratios of their intake manifold volumes to their swept volumes, large ratios of opening areas of their intake valves to their combustion chamber volumes, and so on. Due to these structures, the intake manifolds and cylinders reach states of thermodynamic equilibria in negligibly short times.
2. We focus on factors of importance for air-fuel ratio controls: (1) the intake manifold pressures from the intake

valve closing (IVC) timings to the intake valve opening (IVO) timings, (2) the intake manifold pressures at the IVC timings, and (3) residual gas densities at the exhaust valve closing (EVC) timings. Factors other than those listed above (e.g. the intake manifold pressures during periods of valve overlaps) are of secondary importance.

3. Periods of valve overlaps are approximated by zeros in the computational model. Instead, amounts of residual gas and shortcut gas are estimated by using look-up tables which are experimentally calibrated.

4. Intake pulsations are calculated independently from the air flows, and they are modeled as damping oscillations with four parameters: interval, damping coefficient, initial amplitude, and initial phase. These parameters are determined from look-up tables which are adjusted experimentally.

5. Changes of engine rotational velocities are predicted from cylinder pressures, inertia torques, back torques, and frictions. The prediction compensates sharp increase or decrease of the engine rotational velocities.

Compared to the estimation methods [1, 2, 3, 4], the proposed method requires richer computational resource to calculate the air flow and the engine rotational velocity. Nevertheless, it keeps feasibility of real-time computing due to the abovementioned simplifications.

Applications

The proposed method is of service to the following technical areas:

- The method can be utilized as a soft-sensor of the intake manifold pressure. Some of low cost motorcycles with fuel injection systems equip only throttle position sensors, and they equip neither intake manifold pressure sensors nor air flow sensors. Since the intake manifold pressures are vital for the fuel injection controls, the proposed method is useful for such motorcycles.

- Combining the method with a prediction method of a throttle position provides a predictive value of the intake manifold pressure, which is informative for computation of asynchronous injection. Especially for motorcycles with electronic throttle control systems, the predictive value can be accurately calculated by exploiting response lags of the throttle controls.

- The method increases accuracy of backfire detection. Even weak backfires are detectable by comparison of measured intake manifold pressures with estimated intake manifold pressures. In particular, the method is suitable for the backfire detection under unstationary driving conditions since it is applicable to estimation of transient intake manifold pressures.

Organization

Chapter “FLOWS AND STAGNATION” gives a mathematical description of flows and stagnation in the intake manifolds. Chapter “AIR MODEL” illustrates the intake air model. Chapter “CALIBRATION OF AIR MODEL” sketches how to adjust our method. Chapter “EXPERIMENTAL RESULTS” shows our results of experiments. The final chapter concludes the paper.

FLOWS AND STAGNATION

This chapter provides mathematical descriptions on flows and stagnation illustrated in Fig. 2. In the figure, the index 2 denotes a throttle valve, and the indices 1 and 3 denote its upstream and downstream, respectively.

Subsonic Flow

Consider an enthalpy balance along a diabatic subsonic flow from the upstream to the throttle valve:

$$\frac{1}{2} \left(\frac{v_1}{C} \right)^2 + \frac{\kappa}{\kappa-1} \frac{P_1}{r_1} = \frac{1}{2} \left(\frac{v_2}{C} \right)^2 + \frac{\kappa}{\kappa-1} \frac{P_2}{r_2} \quad (1)$$

where C is a discharge coefficient, κ is a specific heat ratio, v is a flow velocity, P is a static pressure, and r is a density. In the above equation, the discharge coefficient indicates the enthalpy balance. If the flow is lossless, then the coefficient is equal to the unity.

Next, let S_1 and S_2 denote a cross-sectional area of the upstream and a valve-opening area of the throttle, respectively. From the equation of continuity, we have

$$S_1 r_1 v_1 = S_2 r_2 v_2. \quad (2)$$

Moreover, modeling the diabatic flow as a polytropic process, we get

$$\frac{P_2}{P_1} = \left(\frac{r_2}{r_1} \right)^n \quad (3)$$

with n denoting the polytropic index. Various thermodynamical processes can be represented by the polytropic process; an isobaric process by $n = 0$, an isothermal process by $n = 1$, an isentropic process by $n = \kappa$, and an isochoric process by $n = \infty$. In case of an adiabatic expansion process, $n = \kappa$ if all of its internal energy is converted to a work, $1 < n < \kappa$ if a part of its internal energy is converted to a work, and $n = 1$ if none of its internal energy is converted to a work.

Choked Flow

The flow is caused by propagation of the pressure difference, but the propagation velocity can not exceed the sonic velocity. Therefore, the mass flow rate becomes saturated, that is “choked”, when the flow velocity reaches the propagation velocity.

Let $\gamma := P_2/P_1$. Then, solving Eqs. (1)(2)(3), we can see that the mass flow rate is saturated when γ satisfies

$$0 = \frac{n-1}{2} \left(\frac{S_2}{S_1} \right)^2 \gamma^{1+\frac{1}{n}} - \frac{n+1}{2} \gamma^{1-\frac{1}{n}} + 1. \quad (4)$$

Let γ^* denote the solution of the above equation. Then, γ^* corresponds to the critical pressure ratio; if $P_2/P_1 > \gamma^*$, the flow is subsonic, otherwise, it is choked. On the assumption that $S_2/S_1 \approx 0$, the solution can be approximated by

$$\gamma^* \simeq \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}}. \quad (5)$$

The mass flow rate of the choked flow is equal to that of the saturated subsonic flow. Hence, it is given by solving the following equations:

$$\frac{1}{2} \left(\frac{v_1}{C} \right)^2 + \frac{\kappa}{\kappa-1} \frac{P_1}{r_1} = \frac{1}{2} \left(\frac{v_s}{C} \right)^2 + \frac{\kappa}{\kappa-1} \frac{P_s}{r_s}, \quad (6)$$

$$S_1 r_1 v_1 = S_2 r_s v_s, \quad (7)$$

$$\frac{P_s}{P_1} = \left(\frac{r_s}{r_1} \right)^n, \quad (8)$$

$$P_s = \gamma^* P_1 \quad (9)$$

where P_s , r_s , and v_s denote the static pressure, density, and velocity of the saturated subsonic flow, respectively.

Stagnation of Subsonic Flow

When the subsonic flow stagnates in the intake manifold, the enthalpy balance is described by

$$\frac{1}{2} v_2^2 + \frac{\kappa}{\kappa-1} \frac{P_2}{r_2} = \frac{\kappa}{\kappa-1} \frac{P_3}{r_3}. \quad (10)$$

And, the equation of the continuity is given by

$$\frac{d}{dt} (V_3 r_3) = S_2 r_2 v_2 \quad (11)$$

with V_3 being a volume of the stagnation region. Moreover, the enthalpy of the stagnation region changes as

$$\frac{d}{dt} \left(\frac{\kappa}{\kappa-1} P_3 V_3 \right) + \frac{dW}{dt} = \left(\frac{1}{2} v_2^2 + \frac{\kappa}{\kappa-1} \frac{P_2}{r_2} \right) S_2 r_2 v_2. \quad (12)$$