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Adaptive idling control scheme and its experimental validation for gasoline engines

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Abstract In this paper, the idle speed control problem is investigated for spark-ignition (SI) engines. To scope with physical model parameter-free control scheme, a nonlinear adaptive control law is proposed for the speed regulation loop. The stability analysis result shows that the idle speed converges to the set value and the estimated parameter converges to an equivalent static value of the intake manifold. Furthermore, the proposed fuel injection control law consists of a feedforward and a simple feedback loop. Finally, the proposed control scheme is validated based on a full-scaled six-cylinder gasoline engine test bench.

Keywords SI engine, idle speed control, adaptive control, Lyapunov design, model-based design

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1 Introduction

For combustion engines, the idling is an essential operating mode in which the basic requirement is to keep the rotational speed of the engine at a desired value under the load changes such as air conditioner, pump, on/off turning of the generator and the variation of thermal condition. In order to regulate the speed, throttle opening is typically used as the actuation and feedback control has been widely applied for the idle speed control problem in the last three decades.

At first, as most conventional feedback control strategy, proportional-integral (PI) controller has been widely used for idle speed regulation problem [1]. Compared with PI control, fuzzy control and variable structure control approaches have been also proposed to the idling speed control problem and the performance comparisons can be found in the literatures [2]. Furthermore, advanced dynamical system theory-based control approaches, such as adaptive control [3], model predictive control [4, 5], and neural network-based control [6], have been also developed to improve idling control performance. In the meantime, to improve the intake-to-power delay which is a basic characteristic of combustion engines, the spark advance (SA) is involved to compensate transient response of the idle speed control loop [7].

However, the advanced control designs rely on exact or simplified linear models of the dynamical behavior of the throttle opening, the torque generation and the mechanical rotational dynamics. To follow

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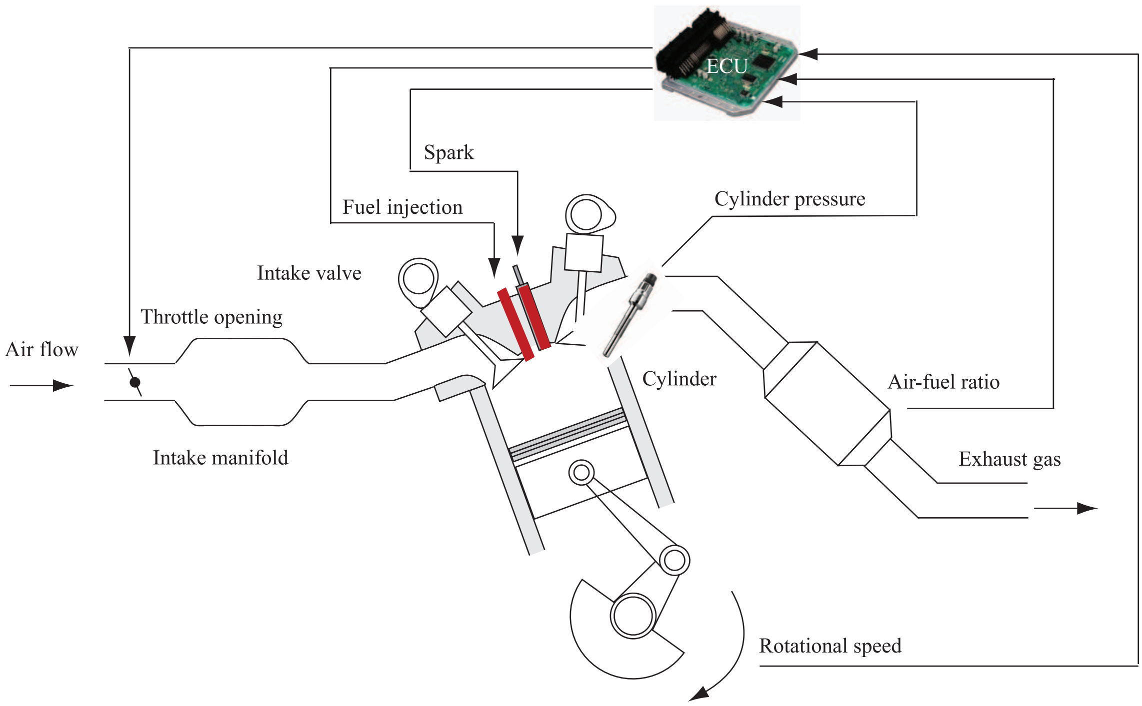

Figure 1 (Color online) Sketch of engine control system.

the physics of the engine, it is possible to obtain more exact model such as in-cylinder models [8] that try to give exact representation of the engine combustion phenomena, the cylinder-to-cylinder imbalance and the cycle-to-cycle variation. But, it will cause unnecessary complexity of the model structure. A typical trade-off between the model complexity and handling the main characteristics of the engine is the mean-value model. However, it is usually not easy to obtain the exact mathematical representation of the engine dynamics due to the strong nonlinearity in the air path and the fuel injection path, and the complexity in the fluid dynamics and the combustion phenomenon. The difficulty in the engine modeling damages the idling performance of the advanced model-based controllers. This paper challenges an engine model parameter-free design scheme for the idle speed control problem. An adaptive controller which relies on the structure of the engine model is proposed, however, the controller does not depend on any exact value of the model parameters. Following the proposal of the controller, the stability of the idle speed regulation error is proofed with the Lyapunov stability theory. Finally, experimental validation results which are conducted on a transient control test-bench of full-scaled six-cylinder gasoline engine are shown to demonstrate the control performance.

2 Modeling and control problem

The following presents a model-based idle speed control system design approach for gasoline engines. Combustion engine is a complex system concerning thermal engineering, fluid dynamics, chemical and dynamical engineering. This paper exploits a control-oriented mean-value model for the sake of simplicity of the resulting control law.

2.1 Control-oriented engine model

Consider a four-stroke multi-cylinder combustion engine with direct injection system. Figure 1 shows a sketch of the engine control system, where only one cylinder is sketched for simplicity. For such an engine system, thermal energy is generated by the combustion of the air-fuel mixture that is compressed in each cylinder, and it converts to the force acting on the piston to drive the crankshaft. The generated engine torque determines the performance of the idling operation. According to the operating mechanism, management of the engine torque generation depends on the air intake, the fuel injection and the spark timing. The engine control unit (ECU) provides the commands for the corresponding actuators. The schemes for the idle speed control are usually constructed with the feedback of the engine speed and the intake manifold pressure. In this work, the feedback of air-fuel ratio (A/F) measured by the λ sensor and the in-cylinder pressure are taken into account the idle speed control design problem. On the other hand, for a multi-cylinder engine, it should be noted that the engine torque is generated serial by each cylinder.

By the ideal gas law, the dynamics equation of the air intake path is expressed as

$$\dot{p}_m = \frac{RT_m}{V_m}(\dot{m}_i - \dot{m}_o). \tag{1}$$

The rotational movement equation is deduced as follows according to the Newton's law:

$$J_e \dot{\omega}_e = \tau_e - \tau_f - \tau_l. \tag{2}$$

In (1) and (2), p_m, T_m and V_m denote the pressure, temperature and volume of the intake manifold, respectively, R denotes the gas constant, $\dot{m}_i$ denotes the air mass flow rate through the throttle, $\dot{m}_o$ denotes the air mass flow rate into the cylinders, J_e denotes the engine moment of inertia, ω_e denotes the engine speed, and τ_e, τ_f and τ_l denote the engine torque, the friction torque and the load torque, respectively.

As described above, exact consideration of the characteristics of each cylinder causes the complexity of the engine model. To avoid nonsmooth representation of the engine torque, cycle-based mean-value engine model is proposed by averaging the air-mass flow and engine torque equivalently. By the ideal gas law, the cycle-based mean-value air charge flow rate is usually modeled by

$$\dot{m}_o = c_v p_m \omega_e, \tag{3}$$

where c_v is a coefficient depending on the volumetric efficiency of the engine system. With the above air flow rate and the energy balance law, the mean-value engine torque can be formulated as

$$\tau_e(t) = c\frac{\dot{m}_o(t - t_d)}{\omega_e(t)}, \tag{4}$$

where t denotes the current time, c denotes the maximum torque capacity depending on the fuel conversion efficiency and t_d denotes the intake-to-power delay. Combining with the modeling (3), Eq. (4) can be rewritten as

$$\tau_e = c_\tau p_m(t - t_d), \tag{5}$$

with $c_\tau = c \cdot c_v$.

Furthermore, note that the pressure ratio between the ambient pressure p_a and p_m satisfies

$$\frac{p_m}{p_a} < \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa}{\kappa - 1}}, \tag{6}$$

during idling operation, where κ is the specific heat ratio and $\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$ denotes the critical pressure ratio [9]. Then, consider the process of air mass flow passing through a nozzle, the air mass flow rate through the throttle can be modeled by

$$\dot{m}_i = s_0(1 - \cos\phi)\frac{p_a}{\sqrt{RT_a}}\sqrt{\kappa\left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa+1}{\kappa-1}}}, \tag{7}$$

where s_0 denotes the area of the throttle valve, ϕ denotes the throttle opening and T_a denotes the ambient temperature.

Moreover, when the engine is operating (i.e. $\omega_e > 0$), the engine friction torque can be represented by the following equation [10]:

$$\tau_f = \mathrm{D}\omega_e + \mathrm{D}_0, \quad \omega_e > 0, \tag{8}$$

where D and D_0 are constants which can change at different engine operating modes.

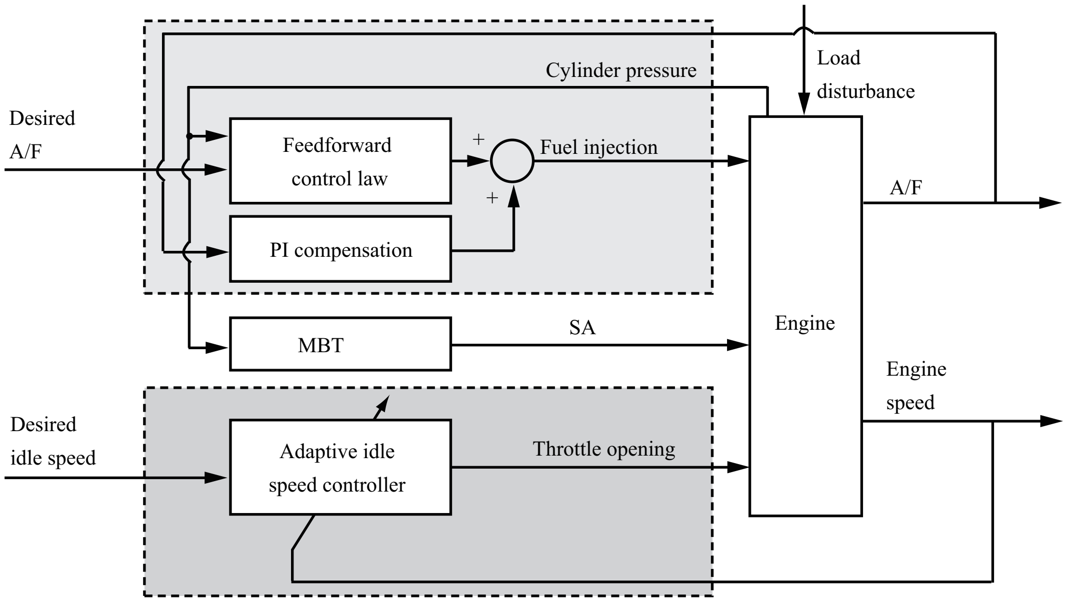

Figure 2 Structure of the idle speed control system.

2.2 Control problem

As summary, the mean-value model for the idle speed control design is as follows:

$$\begin{cases} J_e\dot{\omega}_e = c_\tau p_m - \mathrm{D}\omega_e - T_D(\tau_l); \\ \dot{p}_m = u_{th} - ap_m\omega_e, \end{cases} \tag{9}$$

where $T_D = \mathrm{D}_0 + \tau_l$, $a = RT_m/V_m c_v$ and $u_{th} = RT_m/V_m \cdot s_0\psi(p_a)(1-\cos\phi)$ which is the control input with respect to the throttle opening. The first equation in the above model is obtained by replacing the engine torque $\tau_e(t)$ by $c_\tau p_m(t)$, and this means that the mean-value engine torque is calibrated by the response of the intake manifold. Moreover, for the sake of simplicity of the proposed control law, the intake-to-power delay is ignored; however, the validity of this treatment is demonstrated by the experimental results latter.

The problem considered in this paper is to develop a control algorithm for the ECU in the feedback control system shown in Figure 1 without the requirement of exact values of the physical parameters J_e, c_τ, D and T_D, in the model (9), i.e. we will propose a controller based on the model structure of (9) but not the parameters. Precisely, the control design problem is as follows: for a given set-value of the idle speed, find a real-time control algorithm for the throttle opening related input u_{th} and the fuel injection command u_f such that the closed-loop system under the structure shown in Figure 1 is stable, and the idle speed converges to the set value asymptotically. For the SA, the decision is out of scope of this paper. Suppose that SA is simply decided based on pre-calibrated map that provides MBT (minimum SA for best torque) for the spark timing.

3 Proposed control scheme

3.1 Overview

The proposed control scheme is shown in Figure 2. The structure of the control scheme includes two main control loops for the idle speed regulation and the A/F control, respectively. Regarding the A/F control loop, although many strategies for A/F control have been reported, such as the predictive-based method in [11] and a numerical method in [12], to significantly deal with the idle speed control problem, a proper fuel injection control strategy is essential with the consideration of the transient performance of the control system. In this work, a new A/F control approach is proposed by using the cylinder pressure. The proposed method gives an estimation of the fresh air charge estimation which actually provides a feedforward control law for the A/F control loop. Then, the A/F controller is constructed by combining a PI control compensation to the feedforward controller. For the idle speed control loop, an adaptive control law is designed with the feedback of engine speed only.

3.2 Control algorithm

First, the control law for the fuel injection consists of feedforward and feedback loop. The feedforward fuel injection is designed by focusing on the individual air charge estimation of the current cycle k, i.e.

$$\mathrm{u_{ff}}(k+1) = \frac{m_a(k)}{\lambda_d}, \tag{10}$$

where k denotes the cycle-based sampling index, $\mathrm{u_{ff}}$ denotes the feedforward fuel injection command, $m_a(k)$ denotes the fresh air charge estimation in the current cycle k and λ_d denotes the desired A/F value. To get the feedforward compensation by (10), the fresh air charge should be decided. Actually, the cylinder air mass is composed of three components, the fresh air charge, the fuel injection and the residual gas mass. For simplicity, consider that the residual gas ratio is 7%. Then, the fresh air charge of the current cycle can be calculated as

$$m_a(k) = 0.93\, M_a(k) - u_f(k), \tag{11}$$

where $M_a(k)$ and $u_f(k)$ denote the total cylinder air mass and the fuel injection mass of the current cycle, respectively.

Regarding the total air mass in the cylinder which cannot be measured directly, an effective estimation approach is by using the cylinder pressure p_c and the following equation gives the expression of the estimation law [13], i.e.

$$M_a(k) = \frac{(p_{c2}(k) - p_{c1}(k))V_{c1}(k)}{RT_{c1}(k)} \left[\left(\frac{V_{c2}(k)}{V_{c1}(k)} \right)^m - 1 \right]^{-1}, \tag{12}$$

where the subscripts $c1$ and $c2$ denote two points of the crank angle during the compression stroke, $p_\cdot$, $V_\cdot$ and $T_\cdot$ are the corresponding values of the cylinder pressure, volume and temperature, respectively, and m denotes a coefficient.

Then, the A/F controller is constructed by combining the feedforward compensation law (10) with a PI feedback controller, i.e.

$$u_f(k+1) = \mathrm{u_{ff}}(k+1) + \mathrm{k_{pf}}(\lambda_d - \lambda(k)) + \mathrm{k_{if}} T_c \sum_{i=0}^{k} (\lambda_d - \lambda(k)), \tag{13}$$

where λ denotes the measured A/F, T_c denotes cycle-based sampling rate, and $\mathrm{k_{pf}}$ and $\mathrm{k_{if}}$ are given constants. It should be noted that to perform this fuel path controller, the in-cylinder pressure measurement is needed. The benefit of this cylinder pressure sensor (CPS)-based engine control is that the influence of combustion-to-exhaust delay in the A/F control loop and the longer response delay in the lambda sensor that cause constraint in improving the transient control can be avoided. As a case study of CPS-based engine management, here CPS-based A/F control is proposed.

Finally, for the speed control loop, the following adaptive control law is developed:

$$\begin{cases} u_{th} = (1+\gamma)\mathrm{k_p} e_\omega + a\omega_e \hat{p}_m; \\ \dot{\hat{p}}_m = \gamma \mathrm{k_p} e_\omega, \end{cases} \tag{14}$$

where ω_r denotes the desired idle speed value, $\hat{p}_m$ denotes an adaptive gain which adaptation law is given by the proportional of the speed error which is defined as $e_\omega = \omega_r - \omega_e$, and both $\mathrm{k_p}$ and γ are any given positive constants.

3.3 Stability analysis

With respect to the equilibrium (ω_r, p_m^*) of the engine system (9), the dynamics of the closed-loop system of (9) under the control law (14) can be represented with the errors e_ω, $\hat{e}_p$, $\tilde{e}_p$ as follows:

$$\begin{cases} J_e\dot{e}_\omega = -\mathrm{D}e_\omega + c_\tau\tilde{e}_p + c_\tau\hat{e}_p; \\ \dot{\hat{e}}_p = \dot{\hat{p}}_m - u_{th} + a\omega_e\hat{p}_m - a\omega_e\hat{e}_p; \\ \dot{\tilde{e}}_p = -\dot{\hat{p}}_m, \end{cases} \tag{15}$$

where p_m^* is expressed by the following equation according to the engine model (9):

$$p_m^* = \frac{\mathrm{D}\omega_r + T_D(\tau_l)}{c_\tau}, \tag{16}$$

and the errors $\hat{e}_p$ and $\tilde{e}_p$ are defined as $\hat{e}_p = \hat{p}_m - p_m$ and $\tilde{e}_p = p_m^* - \hat{p}_m$, respectively. Then, taking the physics of the engine system that the model parameters J, c_τ and D are positive into account and applying the Lyapunov stability theory, the convergence of the system (15) can be obtained as follows.

Proposition 1. For the system (15), consider that the load disturbance is an unknown constant and the parameters J_e, c_τ, D and a are uncertain. For any given desired idle speed ω_r, the closed-loop system (15) is stable at the equilibrium $(e_\omega, \hat{e}_p, \tilde{e}_p)$ and $e_\omega \to 0$ as $t \to \infty$.

Proof. Choose a candidate Lyapunov function as

$$V = \frac{J_e\mathrm{k_p}}{2c_\tau}e_\omega^2 + \frac{1}{2}\hat{e}_p^2 + \frac{1}{2\gamma}\tilde{e}_p^2, \tag{17}$$

and along the trajectory of the control system (15), the time derivative is

$$\dot{V} = -\mathrm{k_p}\frac{\mathrm{D}}{c_\tau}e_\omega^2 - a\omega_e\hat{e}_p^2 \leqslant 0. \tag{18}$$

This meas that the control system is stable, and since the set $\Omega := \{(e_\omega, \hat{e}_p, \tilde{e}_p) \mid \dot{V} = 0\} = \{e_\omega = 0, \hat{e}_p = 0\}$, conclusion of $e_\omega \to 0$ can be obtained by LaSalle invariance principle. This completes the proof.

Remark 1. The above proof demonstrates the convergence of the engine speed and $\hat{p}_m \to p_m$. Note that the controller (14) employs the model parameter a theoretically. However, if an imprecise a is used in the feedback controller, the controller satisfies $u_{th} \to \bar{a}\hat{p}_m\omega_e$ when $e_\omega \to 0$, where $\bar{a}$ denotes a constant that closes to the precise value a. This can equivalently means that $u_{th} \to ap_m\omega_e$. In this case, $e_\omega \to 0$ although $\hat{p}_m$ cannot converge to p_m. This observation is later confirmed by the experimental validation results.

4 Experimental results

4.1 Experimental set-up

The designed nonlinear feedback idle speed control strategy with the proposed A/F control law is validated on a 3.5 L V6-type gasoline engine which connects with an eddy current dynamometer. The test bench is shown in Figure 3. The specifications of the test bench is shown in Table 1. Additional in-cylinder pressure and optical encoder on crankshaft are equipped for in-cylinder pressure measurement with respect to the crank angle. The engine prototype ECU is equipped to achieve real-time engine control, and a rapid prototype controller (dSPACE 1006) is employed as a parallel controller of the original engine ECU via the bypass interface. Regarding the proposed control laws, throttle opening command is delivered by time scale per 8 ms. The fuel injection is controlled based on event. At the end of each compression stroke, the trapped air mass is calculated by (12), and apply (13) with (10) to obtain the desired fuel injection in the next cycle.

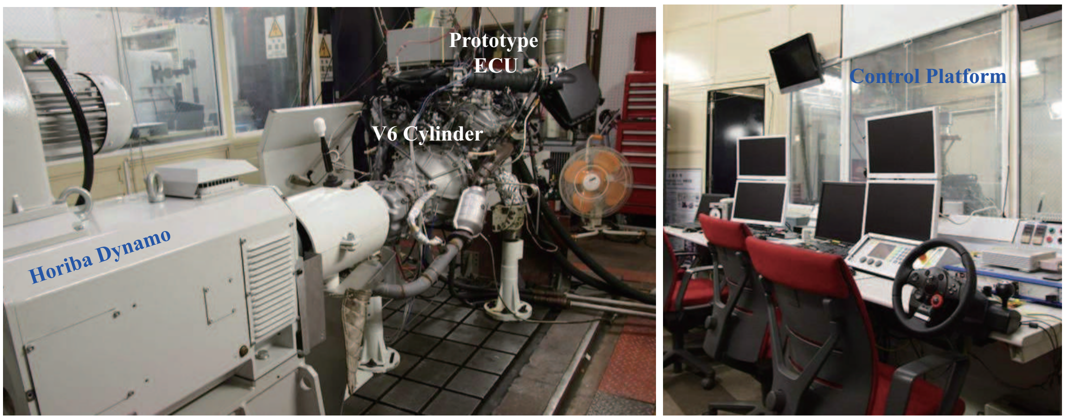

Figure 3 (Color online) Experimental test bench of engine control.

Table 1 Specifications of the engine test bench

Engine specification		Dynamo technical data	
Cylinder type	V-type 6 cylinders	Rated power (Absorbing/Driving)(kW)	250/225
Fuel system	SFI D-4S	Rated torque (Absorbing/Driving)(N·m)	480/442
Displacement (L)	3.456	Rated speed (Absorbing/Driving)(rpm)	4980/4860
Compression ratio	11.8:1	Maximum speed(rpm)	10000
Maximum torque (N·m)	375	Moment of inertia(kg·m^2)	0.36

4.2 Validation results

The main result of this paper is to propose a physical parameter-free adaptive control law. If the model parameter values are exactly know, the control performance must be expected. To compare the proposed control scheme with the design case with known parameters, the following feedback controller is designed where the pressure p_m is measurable and parameter a is an exactly known parameter. Let $e_p = p_m^* - p_m$. The engine system (9) can be represented equivalently by the following error dynamics:

$$\begin{cases} J_e\dot{e}_\omega = -\mathrm{D}e_\omega + c_\tau e_p; \\ \dot{e}_p = -u_{th} + ap_m\omega_e. \end{cases} \tag{19}$$

Design the feedback controller as

$$u_{th} = ap_m\omega_e + \mathrm{k_p}e_\omega, \tag{20}$$

with a positive feedback gain $\mathrm{k_p}$. Then, employing the following candidate Lyapunov function:

$$V = \frac{J_e\mathrm{k_p}}{2}e_\omega^2 + \frac{c_\tau}{2}e_p^2, \tag{21}$$

along the trajectory of the control system, its time derivative can be calculated as

$$\dot{V} = -\mathrm{D}\mathrm{k_p}e_\omega^2 \leqslant 0. \tag{22}$$

This inequality (22) guarantees that the control system is stable at the equilibrium $(e_\omega,\ e_p) = (0,\ 0)$ by the Lyapunov stability theory. Moreover, it implies that the set $\Omega := \{(e_\omega, e_p) \mid \dot{V} = 0\} = \{e_\omega = 0\}$. Therefore, $e_\omega \to 0$ as $t \to \infty$ follows by the LaSalle invariance principle.

Then, the following shows the experimental tests in which the proposed control laws have to reject disturbances with respect to load changes. Consider that the target idle speed is at 800 rpm. In the testing experiments, the feedback gains in the A/F control loop are chosen as $\mathrm{k_{pf}} = 0.6$, $\mathrm{k_{if}} = 0.2$. For the speed control loop, the feedback control law (20) and the adaptive control law (14) are implemented to the engine system, respectively.

In the test by using the controller (20), a $0 \to 10 \to 0$ N·m step change in load is added to the engine to simulate the disturbances caused by the change of friction, power steering, air conditioning compressor,

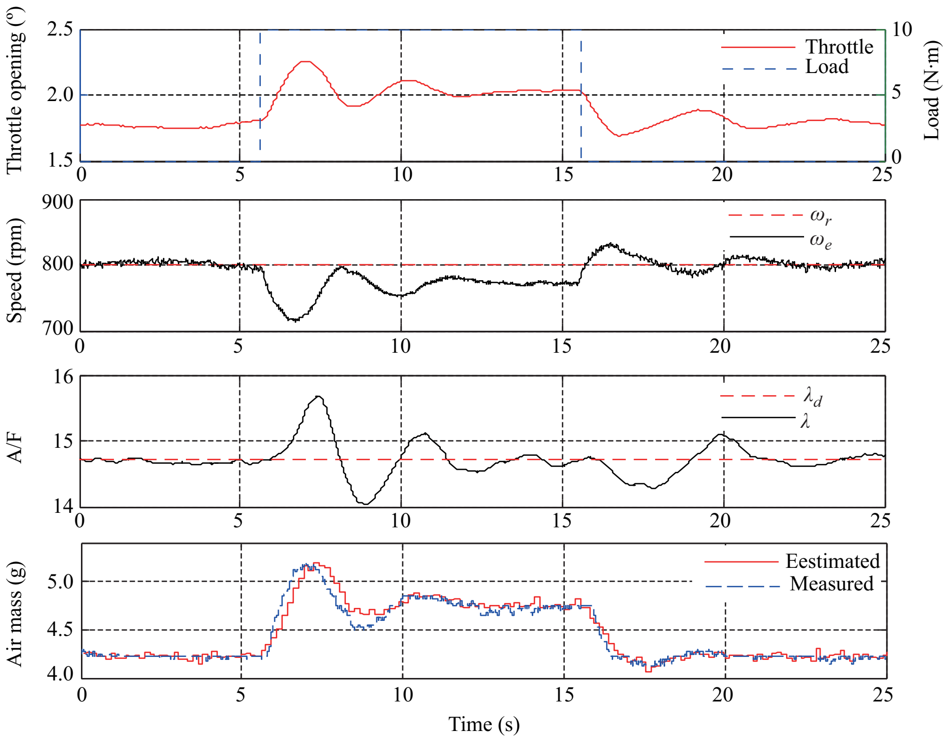

Figure 4 (Color online) Validation result under control law (20).

etc. The feedback gain is chosen as $k_p = 0.4$ which makes a trade off between fast dynamics and overshoot of the closed-loop system. Moreover, based on model identification, the nominal value a in the model (9) can be obtained when the engine operates around the reference idle speed without any additional load. The experimental responses of engine states and control input are shown in Figure 4. It is clear from the Figure 4 that in the first 5 s, engine speed tracks the reference idle speed 800 rpm quite well on the benefit of model identification. When an additional load is added at 5.6 s, engine speed is dragged down with a %11 overshoot, while the transient process cannot cause the engine to stall. Under the effect of feedback control, the speed stays at 780 rpm stably after 7 s. When the load is removed at 15.6 s, engine speed returns to the reference idle speed in a few seconds. Moreover, it can be observed from Figure 4 that the A/F is regulated to the stoichiometric ratio at steady condition. But, the performance deteriorates a little at transition period. It should be noted that since the air mass M_a is estimated by (12) according to the pressure data of last cycle, there is a cyclic delay for M_a estimation. The cyclic estimation and measurement using air flow sensor of the cylinder air mass are also given in Figure 4. It is noticeable that the proposed method can estimate the air mass into the cylinder quite well at steady condition, but when the engine speed has big variation, the air mass estimation M_a is changed cyclically. Hence, there is an inherent error. The above result shows that the control law (20) cannot compensate the load disturbance for idle speed control.

The adaptive control law (14) is implemented under same load disturbances to the above test for control law (20). Choose the feedback gains as $k_p = 0.4$ and $\gamma = 0.04$. Experimental responses of the engine states and control input are shown in Figure 5. It can be seen that there is no tracking error at steady condition. Under a 10 N·m change in load, the speed response has 8.8% overshoot and the settling time is 4 s. It is also noticeable that the estimated $\hat{p}_m$ is not equal to the real p_m, even there is no engine load added to the engine. This error is due to using an imprecise parameter a in the control law (14). However, as the notes in Remark 1, the condition $u_{th} \to \bar{a}\hat{p}_m\omega_e$ guarantees $e_\omega \to 0$. Additionally, the trajectory of Lyapunov function $V(t)$ shown in Figure 5 demonstrates the stability of the adaptive closed-loop control system. Since the nonzero $\hat{e}_p$ and $\tilde{e}_p$, $V(t)$ is not equal to zero during the steady state. Furthermore, Figure 6 shows a testing result with the feedback gains chosen as $k_p = 0.4$ and $\gamma = 0.06$.

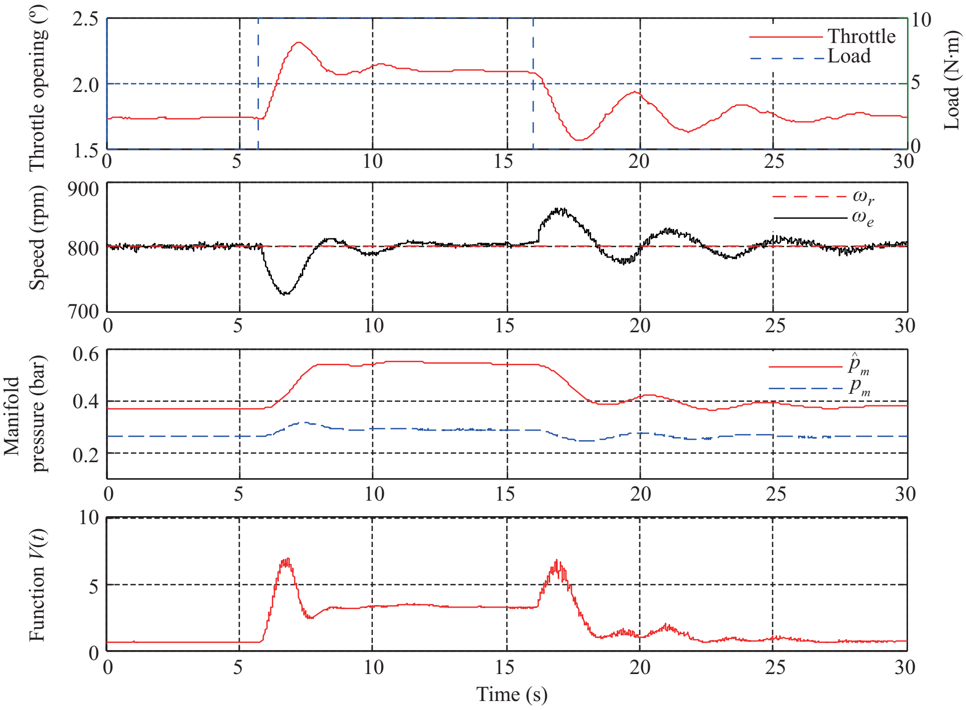

Figure 5 (Color online) Validation result under the adaptive control law: $\mathrm{k_p} = 0.4$, $\gamma = 0.04$.

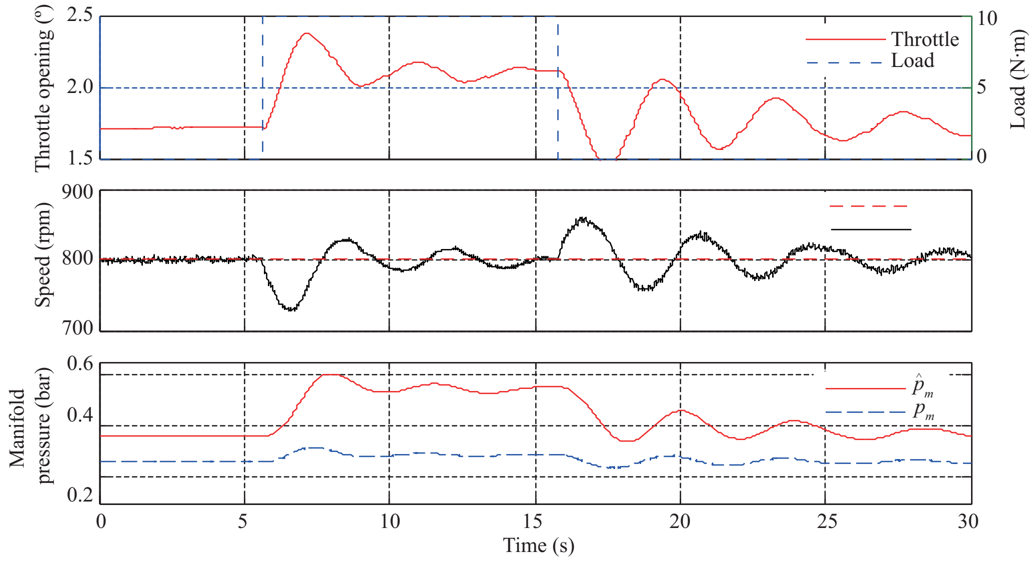

Figure 6 (Color online) Validation result under the adaptive control law: $\mathrm{k_p} = 0.4$, $\gamma = 0.06$.

The increased oscillation of the speed transient response indicates that the feedback control gains effect the performance significantly.

5 Conclusion

As is well-known, involving the dynamical model of controlled plant is effective to get good transient performance of the control system. This causes a lot of challenges to apply the model-based advanced control design method in the idle speed control problem. This paper targets on the defect of the model-based control strategy that requires the exact values of the model parameters. An adaptive control scheme is proposed with guarantee of stability of idle speed regulation error. The experimental validation results demonstrate the feasibility and effectiveness of the proposed controller since the experiments are performed at an industrial level engine control test bench.

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Conflict of interest The authors declare that they have no conflict of interest.

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