International Journal of Automotive Technology, Vol. 8, No. 4, pp. 477−481 (2007)

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1229−9138/2007/035−10

ENGINE CONTROL USING SPEED FEEDBACK

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(Received 20 March 2006; Revised 28 February 2007)

ABSTRACT−In this article we present a new, reference model based, unified strategy for engine control. Three main modes are considered: first is the driver control mode where the driver controls the engine via the pedal position; second is the dashpot mode, that is, when the driver takes his foot off the pedal; and, lastly is the idle speed control mode. These modes are unified so that seamless transitions between modes now becomes possible. The unification is achieved due to the introduction of a reference model for the engine speed whereby only the desired engine speed is different for different modes while the structure of the control system remains the same for all the modes. The scheme includes an observer that estimates unknown engine load torque. A proof of robustness with respect to unknown load disturbances both within the operating modes and during intermode transitions is given.

KEY WORDS : Engine control, Reference model, Control modes, Engine load observer

1. INTRODUCTION

In order to meet acceleration performance demanded by drivers, the engine torque produced by a spark ignition automotive engine should be controlled on the basis of a driver command signal. There are several methods available in the literature devoted to torque control of throttled engines, e.g., (Zhang and Streib, 1996; Wild *et al.*, 2000). These methods have a number of elements in common. To meet the acceleration performance of the vehicle, the output torque produced by the engine and speed change ratio of the transmission are controlled satisfying the driver's commands given by the pedal position. The desired indicated engine torque is determined according to the accelerator pedal position and possibly transmission speed and wheel speed together with their minimum and maximum attainable values. Then actual engine torque is controlled by the throttle and spark advance such that they deliver the desired indicated torque. Engine torque control systems described in the publications above suffer from drawbacks that come from mode change events. During mode change events, for example, from the driver control mode to the dashpot mode and to the idle speed control mode, the structure of the control system changes. The functionality, which is difficult to calibrate, must necessarily be used in existing systems in order to achieve smooth intermode transitions. In other words, since indicated engine torque was chosen as the main variable during the driver control mode and engine speed as the main variable for the idle speed control mode the transition between modes involves changing the main controlled variables and, therefore, requires special functionality. Moreover, during the driver control mode, the control system ensures the convergence of the indicated torque estimate to the desired indicated torque without paying attention to the convergence of engine speed to the desired speed considering inaccuracies in engine indicated torque estimate (e.g., due to different fuel heating values). These difficulties could be avoided by the introduction of a reference model in the system which is meant to assure the convergence of both engine acceleration and engine speed to the desired acceleration and desired speed values. In this article we present a new, unified, reference model based control system for a spark ignition automotive engine. A general structure that unifies several operating modes of the engine, including driver control mode, dashpot mode and idle speed control mode, is developed. Unification is achieved by the introduction of a reference model whose input is the desired engine speed, which is different for different modes, while the structure of the model and the control system remain the same. The model ensures the convergence of the engine speed to the desired engine speed, which is necessary for the dashpot and the idle speed control modes. The model also ensures the convergence of the acceleration to the desired acceleration,

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which is necessary for the driver control mode. The reference model and the structure of the system do not change for different modes or during mode transitions, only the desired speed changes allowing easy and seamless transitions between modes. Moreover, unknown load torque is estimated by an observer and compensated for in the control system. This guarantees robustness with respect to unknown load disturbances or errors in estimating engine torque, not only within certain operating modes, but also during mode transitions.

2. PROBLEM STATEMENT

The rotational dynamics of the engine are modeled as follows:

$$J\dot{\omega}=T_{ind}-T_f-T_p-T_{Lk}-\Delta T_L \tag{1}$$

where T_{ind} is the indicated engine torque, T_f is the friction torque, T_p is the pumping torque, J is the moment of inertia, ω is the engine speed, T_{Lk} is the known (mapped or accurately estimated) part of external load torque, and ΔT_L is the unknown part of the external load torque or errors in the models used for estimating friction torque T_f, pumping torque T_p, or indicated engine torque T_{ind}.

The indicated engine torque is modeled as follows:

$$T_{ind}=am_e(t-t_d)a_{fi}(t-t_d)f_s(t-t_s) \tag{2}$$

where a is a parameter which represents maximal torque capacity, $m_e(t-t_d)$ is the engine load where $m_e=m_{cyl}/\omega$ and m_{cyl} is the air flow into the engine cylinders, $a_{fi}(t-t_d)$ is the air-to-fuel ratio influence, $f_s(t-t_s)$ is the spark advance influence, t_d is the intake-to-torque production delay, and t_s is the spark-to-torque production delay.

Taking into account that

$$m_{cyl}=k\omega p\,,$$

where p is the intake manifold pressure relative to atmospheric pressure and k is a parameter, the indicated torque can be written in the following form:

$$T_{ind}=akp(t-t_d)\,f_s(t-t_s)\,. \tag{3}$$

For simplicity, but without loss of generality, it is assumed that $a_{fi}(t-t_d)=1$.

Introducing the intake manifold pressure relative to atmospheric pressure similar to (Stotsky *et al.*, 2000; Cho and Hendrick, 1989; Kao and Moskwa, 1995) one obtains the following,

$$\dot{p}=k_1(a_1u_1p_r-k\omega p) \tag{4}$$

with $k_1=T/(V_mT_0\rho_0)$, and $k=V_{cyl}T_0\rho_0\eta/(4\pi T)$ where T is the temperature of the air in the intake manifold, V_m is the intake manifold volume, η is the volumetric efficiency, V_{cyl} is the displacement volume of the engine cylinders, and T_0, ρ_0 are, respectively, the ambient air temperature and density. Furthermore, a_1 is the characteristic of the flow past the throttle, u_1 is the normalized throttle position characteristic and p_r is the normalized pressure influence.

Our problem statement is as follows. We want to find a robust control strategy which allows tracking the desired engine acceleration in the presence of unknown load torque. The strategy should be able to stabilize engine speed at idle and handle dashpot (closed-pedal) operation and with overspeed protection in a seamless way.

3. DESIGN PROCEDURE

The design procedure includes the following steps. First, we design an observer for the unknown engine load torque. Then, we specify the reference model that reflects the desired behaviour of the engine variables, assuming that the desired engine acceleration and engine speed are prescribed. At the next step, we specify the desired indicated torque, which guarantees that the closed loop system follows the reference model specified in a previous step. The desired indicated torque calculation uses the output of the observer for unknown engine load torque. The next step is to choose the throttle position and the spark advance so as to achieve the desired indicated torque. The throttle is used as a main tool to deliver the torque while spark advance is used to compensate for the intake-to-torque production delay. The final step is to specify the desired engine speed during the driver control mode, during dashpot (closed-pedal) mode, during idle mode and during overspeed protection.

3.1. Observer Construction for the Load Torque

We define the following error

$$e=\Delta T_L-(\varepsilon-\alpha_0J\omega)=\Delta T_L-\Delta\hat{T}_L \tag{5}$$

where α_0 is a design parameter, $\Delta\hat{T}_L=\varepsilon-\alpha_0J\omega$ will turn out to be an estimate of ΔT_L (see below) and ε is the solution of the following differential equation

$$\dot{\varepsilon}=-\alpha_0\varepsilon+\alpha_0T_{ind}-\alpha_0T_f-\alpha_0T_p-\alpha_0T_{Lk}+\alpha_0^2J\omega-Ke_e$$
$$\varepsilon(0)=\alpha_0J\omega(0) \tag{6}$$

where $e_d=\omega-\omega_d$ is the tracking error, ω_d is the desired engine speed (see Subsection 3.6), and K is a tunable parameter. Notice that (6) is a numerical differentiation observer with an additional input e_d, which is helpful in achieving good convergence of the tracking error. Evaluating the derivative of e, one obtains

$$\dot{e}=-\alpha_0e+\Delta\dot{T}_L+Ke_d\,. \tag{7}$$

Assuming that $|\Delta\dot{T}_L|$ is bounded by a positive constant $\overline{\Delta T}_L$ and $Ke_d=0$, one can derive an upper bound on the estimation error as

$$|e(t)| \leq \sqrt{e^2(0)e^{-\alpha_0 t} + \frac{\overline{\Delta T_L}^2}{\alpha_0^2}}. \tag{8}$$

This upper bound can be made arbitrarily small by amplifying the design parameter α_0, even though in the actual implementation a judicious choice of α_0 is necessary due to measurement noise and sampling data approximation errors, the effect of which may be amplified for large α_0. The estimate of the load torque can thus be written as follows:

$$\Delta\hat{T}_L = \varepsilon - \alpha_0 J\omega. \tag{9}$$

3.2. The Reference Model of the System

At the next step of the design, it is assumed that the desired engine acceleration and desired engine speed have been specified at the present time instant. The details will be discussed in Subsection 3.6. To unify all operating modes of the engine, including driver control mode, dashpot (closed-pedal) mode, idling mode and overspeed protection, a reference model must be found that ensures the convergence of the engine speed to the desired engine speed, which is necessary for the idle speed control, the dashpot function and overspeed protection, as well as the convergence of the engine acceleration to the desired acceleration, which is necessary for the driver control mode. Such a reference model is specified below. We define the following error

$$s_1(t) = \gamma(\omega - \omega_d) + J(\dot{\omega} - \dot{\omega}_d) \tag{10}$$

where ω_d is the desired engine speed, $\dot{\omega}_d$ is the desired engine acceleration, and γ is a positive design parameter. The reference model for the system is given by the following equation:

$$s_1(t) = 0. \tag{11}$$

If (11) can be enforced through engine control, then

$$\dot{e}_{d1} = -\frac{\gamma}{J} e_{d1} \tag{12}$$

where $e_{d1} = \omega - \omega_d$, and ω exponentially converges to ω_d, while $\dot{\omega}$ exponentially converges to $\dot{\omega}_d$. Moreover, the engine acceleration converges to the desired acceleration. A control law, which is derived to enforce (11), is discussed in Subsection 3.3. It achieves exponential convergence of the engine speed and engine acceleration if ΔT_L is constant. If ΔT_L is not constant, the errors may not converge to zero but can be made arbitrary small (under idealized assumptions) through the appropriate selection of design parameters.

Similarly, for the desired pressure dynamics, the reference model is:

$$s_2(t) = \alpha_2(p - p_d) + (\dot{p} - \dot{p}_d) \tag{13}$$

so that enforcing $s_2(t)=0$ provides an exponential convergence of the relative intake manifold pressure, p, to its desired value. The desired values for p and p_d, will be defined in Subsection 3.5.

3.3. Desired Indicated Torque

We define the desired indicated torque as a solution of the following equation:

$$\gamma(\omega - \omega_d) - J\dot{\omega}_d + T_{indd} - T_f - T_p - T_{Lk} - \Delta\hat{T}_L = 0 \tag{14}$$

where T_{indd} is the desired indicated torque. Note that (14) is (11), where the load torque ΔT_L is substituted by its estimate $\Delta\hat{T}_L$. Equation (14) defines the indicated torque that the engine must deliver:

$$T_{indd} = -\gamma(\omega - \omega_d) + J\dot{\omega}_d + T_f + T_p + T_{Lk} + \Delta\hat{T}_L. \tag{15}$$

It is worth noting that in existing engine torque control strategies the pedal position is translated directly into the desired indicated torque, thereby attention is not paid to the engine speed. On the contrary, in the strategy (15) the desired indicated torque realizes the reference model (11), which in turn guarantees the convergence of the acceleration to the desired acceleration as well as convergence of the engine speed to the desired engine speed. Moreover, the desired indicated torque depends on the estimate of unknown external load torque, that in turn ensures the automatic compensation of unknown loads. Note also, that $J\dot{\omega}_d + T_f + T_p + T_{Lk}$ may be viewed as a reference model for the desired indicated torque, which comes from (1). The strategy (15) has an additional term, $-\gamma(\omega - \omega_d)$, which is responsible for the convergence of the engine speed to the desired speed, and also the term $\Delta\hat{T}_L$, which compensates for unknown load disturbances.

If the engine torque realizes the indicated torque (15), then $J\dot{e}_d = -\gamma e_d - e$, $\dot{e} = -\alpha_0 e + Ke_d + \Delta\dot{T}_L$. The design parameters γ, α_0, K can be determined to ensure good response properties of this closed-loop system with $\Delta\dot{T}_L$ viewed as a disturbance input. The selection of $\gamma > 0$, $\alpha_0 > 0$, $K \geq 0$ is sufficient to guarantee stability.

3.4. Desired Spark Advance

The desired indicated engine torque can be realized by throttle and spark advance. Similar to (Stotsky, 2000), it is reasonable to compensate the intake-to-torque production delay with the spark advance. Neglecting t_s, which is smaller than t_d, and assuming sufficient torque reserve, we require

$$akp(t - t_d)f_s(t) = akp(t)c \tag{16}$$

where c is a constant corresponding to the "base" value of spark advance. From (16), the required spark advance influence $f_s(t)$ can be calculated as

$$f_s(t) = \frac{p(t)}{p(t - t_d)} c. \tag{17}$$

Assuming sufficient spark reserve, the spark advance can be calculated to deliver the spark advance influence in (17), thereby compensating the intake-to-torque production delay.

3.5. Desired Throttle Position

The desired throttle position is defined using the desired spark advance and the desired relative pressure. The desired relative pressure is defined such that $s_1(t)=0$ where $s_1(t)$ is defined by (10). Using (3) and (16), we obtain:

$$p_d=\frac{1}{cak}T_{indd}. \tag{18}$$

The desired throttle position is calculated from $s_2(t)=0$, where s_2 is defined by (13), as

$$u_1=\frac{k}{a_1k_1p_r}\omega p+\frac{1}{a_1k_1p_r}(-\alpha_2(p-p_d)+\dot{p}_d) \tag{19}$$

Similar to (Stotsky, 2000), it can be shown that the overall closed loop system is stable.

3.6. Desired Engine Speed and Acceleration

We now describe how desired engine speed and engine acceleration can be prescribed for different modes of the engine, including the driver control mode, dashpot (closed-pedal) operation, idling and overspeed protection.

In the driver control mode, a lookup table, which defines the desired acceleration as a function of at least the pedal position, reflects the desired response that the driver demands from the engine. The desired acceleration also typically depends on the engine speed, and it may, in addition, depend on the wheel speed and the transmission signals. The desired acceleration profile defines, in turn, the desired engine speed:

$$\omega_d=\int_0^t \dot{\omega}_d(\tau)d\tau.$$

In the idle speed control mode, $\omega_d=\omega_0=const$ is the required engine speed. Similarly, in the overspeed protection mode, $\omega_d=\omega_{\max}=const$, where $\omega_{\max}$ is the engine speed set-point commanded in the overspeed protection mode when ω exceeds $\omega_{\max}$.

In the dashpot (closed-pedal) mode, which refers to the control of the transient from high engine speed to low idling speed, ω_d must be specified. We select the following relation to define the desired engine speed:

$$\dot{\omega}_d=-\gamma_d(\omega_d-\omega_0)-\gamma_s\text{sign}(\omega_d-\omega_0) \tag{20}$$

where ω_d is the desired engine speed, ω_0 is a constant speed at idling and $\gamma_d>0$, $\gamma_s>0$ are design parameters. Equation (20) represents a stable system and ensures that the engine speed converges to the desired speed at idle in a finite time. The parameters $\gamma_d>0$, $\gamma_s>0$ can be selected to guarantee a specified time, D, of convergence of ω_d to ω_0 without requiring negative indicated torque values to follow the determined desired engine speed trajectory. Finally, to prevent the vehicle from surging ahead, a logic-based reset may be used to reinitialize ω_d to ω if ω falls below ω_d, for instance, due to brake application or transmission shift.

3.7. Summary of the Proposed Strategy

In this section the proposed strategy is summarized. Since the strategy for all the modes is the same except for the desired engine speed, the description begins with the desired engine speed for different modes:

– The driver control mode. The desired engine speed and acceleration are defined as

$$\dot{\omega}_d=f(pedal\_position,\ \omega)$$

$$\omega_d=\int_0^t \dot{\omega}_d(\tau)d\tau.$$

– The dashpot mode. The desired engine speed and accelereation are defined according to

$$\dot{\omega}_d=-\gamma_d(\omega_d-\omega_0)-\gamma_s\text{sign}(\omega_d-\omega_0).$$

– The idle speed control mode. The desired engine speed is constant at $\omega_d=\omega_0$, $\dot{\omega}_d=0$.

– The overspeed protection mode. The desired engine speed is maximum engine speed $\omega_{\max}$ if $\omega\geq\omega_{\max}$, i.e., $\omega_d=\omega_{\max}$, $\dot{\omega}_d=0$.

The control strategy, which is the same for all the engine operating modes, can be summarized as follows.

The desired indicated torque is computed as:

$$T_{indd}=-\gamma(\omega-\omega_d)+J\dot{\omega}_d+T_f+T_p+T_{Lk}+\Delta\hat{T}_L.$$

External load torque is estimated via the following observer:

$$\Delta\dot{\hat{T}}_L=\varepsilon-\alpha_0J\omega$$

$$\dot{\varepsilon}=-\alpha_0\varepsilon+\alpha_0T_{ind}-\alpha_0T_f-\alpha_0T_p-T_{Lk}+\alpha_0^2J\omega-Ke_d$$

$$e_d=\omega-\omega_d$$

where $K>0$ is a tunable parameter to improve the response.

The desired spark advance influence and the throttle position are calculated as:

$$f_s(t)=\frac{p(t)}{p(t-t_d)}c$$

$$u_1(t)=\frac{k}{a_1k_1p_r}\omega p-\frac{1}{a_1k_1p_r}(\alpha_2(p-p_d)-\dot{p}_d).$$

3.8. Simulation Results

The control system has been simulated for a scenario involving an acceleration and then transition into idle (see Figure 1). The simulation starts with an acceleration,

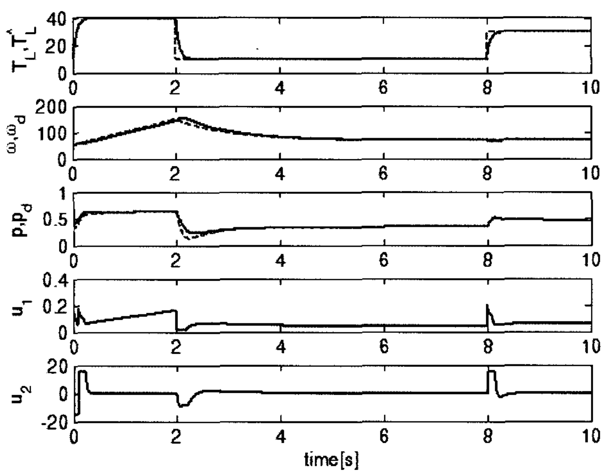

Figure 1. Simulation results for an acceleration-deccele-ration scenario. Additionally, at t = 8, a load disturbance resulting in load torque increase by 20 Nm is applied. The control system adjusts the throttle position u_1 and the spark-advance u_2 such that the desired engine speed trajectory is followed. The first plot shows time histories of load torque (dashed) and load torque estimate (solid) in Nm. The second plot shows engine speed (solid) and desired engine speed (dashed) in rad/s. The third plot shows manifold pressure relative to atmospheric (solid) and desired manifold pressure relative to atmospheric (dashed) in bars. The fourth plot shows throttle position. The fifth plot shows spark advance.

where the pedal map is delivering a desired engine acceleration of 50 rad/s^2. After two seconds the driver releases the pedal and the dashpot functionality drives the engine speed to its idle value. At eight seconds a load disturbance is simulated, e.g. air-conditioning start. The controller rejects the disturbance by adjusting the throttle position u_1 and the spark-advance u_2, where the spark advance influence is given by $f_s=\cos(-b+u_2)^{2.875}$ and where b is the position of the spark advance from the point of maximum brake torque (MBT).

4. CONCLUSIONS

This article proposes a new, reference model based, unified strategy for engine control. Three main modes are considered: the driver control mode, the dashpot mode, and the idle speed control mode, which are all unified under one single control algorithm. The unification is achieved due to the introduction of a reference model for the engine speed, thereby the desired engine speed is different only for different modes, while the structure of the control system remains the same for all the modes. The advantages of the proposed scheme can be summarized as follows:

(1) Unified scheme allows easy and seamless transitions between engine operating modes;
(2) Automatic load torque compensation in all the modes is provided due to the observer in the loop;
(3) The dashpot (closed-pedal) mode allows convergence to the idle speed in a specified time.

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