ADAPTIVE IDLE SPEED CONTROL FOR SPARK-IGNITION ENGINES

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ABSTRACT

Due to the nonlinear time-varying nature of the spark-ignition engine, an adaptive multi-input single-output (MISO) controller based on self-tuning regulator (STR) is proposed for idle speed control in this paper. The spark timing and idle air control are simultaneously employed as control inputs for maintaining the desired idle speed, and are designed based on P and PI type STR, respectively. The Recursive Least Square technique is employed to identify the engine as a first-order MISO linear model. Pole placement technique is then used to design the adaptive MISO controller. Performances of the proposed algorithm are evaluated using a nonlinear engine model in Matlab/Simulink. The system parameters with 10% uncertainties are also utilized to perform the associated robustness analysis. Preliminary simulation results show significant reduction of speed deviations under the presence of torque disturbances and model uncertainties.

INTRODUCTION

Vehicles consume about 30 percent of fuel during idling in city driving scenarios [1]. An effective Idle Speed Control (ISC), which maintains the idle speed close to the designed value, can not only reduce fuel consumption and emission, but also increase combustion stability.

Takahashi [2] proposed a linear quadratic optimal control using a two-state linear engine model, which is identified using the least square method with idle air control (IAC) and engine speed as the input and output, respectively. The performance index consists of the speed deviation and control effort. The experimental results showed that the controller can immediately suppress the effect of various torque disturbances, such as air conditioner on-off, clutch engagement, coasting, etc. Butts [3] proposed a controller consisting of the feedback linear quadratic Gaussian (LQG) optimal control and the feedforward $\ell_1$ optimal controller. Lu [4] proposed a discrete sliding mode controller, and used IAC as an actuator for the ISC. The simulation and experimental results showed that the proposed algorithm possesses a smooth and quick transient response without steady-state errors under the presence of the torque disturbances.

Since the effect of spark timing control on the dynamic response of engine speed is faster than the IAC, Deur [5] simultaneously employed spark timing control and IAC for ISC, and utilized the least-squares estimation to identify the engine as a third-order model with the throttle angle and engine speed as input and output, respectively. PI, PID, and polynomial controllers are designed according to the identified linear engine model for the throttle control. Proportional control is then used for spark timing control. The simulation results showed that the polynomial controller contains lower noise sensitivity and better torque disturbance rejection. Yang [6] proposed a sliding mode control scheme to coordinate throttle and spark advance for ISC. The simulation results show that the proposed strategy can minimize idle speed variations under the rapid torque disturbances, system nonlinearities, and uncertainties. Sun [7] proposed an optimal control algorithm and utilized the spark timing and IAC as the actuators for ISC. The performance index consists of the speed deviations, spark timing deviated from maximum brake torque (MBT) timing, and IAC valve opening rate. The simulation results showed that the proposed algorithm under the load disturbance has faster settling time and smaller speed deviation than the traditional PID controller using only IAC.

An adaptive multi-input single-output (MISO) controller based on self-tuning regulator (STR) is proposed for ISC in this paper. The spark timing and IAC are simultaneously employed as control inputs for maintaining the desired idle speed, and are designed based on P and PI type STR, respectively. The Recursive Least Square (RLS) technique is employed to identify the engine as a first-order MISO linear model. Pole placement technique is then used to design the adaptive MISO controller for maintaining the idle speed at 700 rpm. A combination of different load torque patterns is used as disturbances to evaluate the performances of the traditional PI controller using only IAC, the MISO controller without the adaptation,
and the proposed adaptive MISO controller. The system parameters with 10% uncertainties are also utilized to perform the associated robustness analysis.

The remainder of this paper is organized as follows. The nonlinear engine dynamics is introduced in Section 2, followed by the proposed algorithm in Section 3. In Section 4, a nonlinear engine model is used to evaluate the performance of proposed algorithm under the presence of torque disturbance and model uncertainties. Finally, conclusions are made in Section 5.

MODELING

A 1200 c.c. four-stroke spark ignition engine model [8,9], which consists of the charging, torque, friction, and crankshaft dynamics, is established in Matlab/Simulink (see Figure 1) to simulate the nonlinear engine dynamics.

**Figure 1.** Block diagram of the engine model

AIR FLOW DYNAMICS

The air flow dynamics can be represented using the filling and emptying model [8] based on the one-dimensional isentropic compressible flow equation. It consists of non-choked and choked flow dynamics as shown in Equations (1) and (2), respectively.

\[
\dot{m}_{\text{air,th}} = \frac{C_{d,\text{ai th}} P_{\text{aim}}}{\sqrt{R_T \text{aim}}} \left( \frac{P_{\text{im}}}{P_{\text{aim}}} \right)^{\frac{1}{\gamma}} \left( 2 \frac{\gamma_T}{\gamma_T - 1} \left[ 1 - \left( \frac{P_{\text{im}}}{P_{\text{aim}}} \right)^{\frac{\gamma_T - 1}{\gamma_T}} \right] \right)^{\frac{1}{2}},
\]

for \( P_{\text{im}} > \left( \frac{2}{\gamma_T + 1} \right)^{\frac{\gamma_T}{\gamma_T - 1}} \)

\[
\dot{m}_{\text{air,th}} = \frac{C_{d,\text{ai th}} P_{\text{aim}}}{\sqrt{R_T \text{aim}}} \left( \frac{2}{\gamma_T + 1} \right)^{\frac{\gamma_T + 1}{2(\gamma_T - 1)}},
\]

for \( P_{\text{im}} \leq \left( \frac{2}{\gamma_T + 1} \right)^{\frac{\gamma_T}{\gamma_T - 1}} \)

where \( m_{\text{air,th}} \) is the mass airflow through the throttle body; \( C_{d,\text{ai th}} \) is the discharge coefficient of the intake manifold; \( R_T \) is the ideal gas constant of air; \( \gamma_T \) is the specific heat ratio of air; \( T_{\text{aim}} \) and \( P_{\text{aim}} \) are the temperature and pressure of the atmosphere, respectively. The cross-sectional area of the throttle body \( A_h \) is defined as a function of throttle opening angle \( \theta_a \), and can be expressed as follows:

\[
A_h = \frac{D}{2} \left[ 1 - \left( \frac{d}{D} \right)^{2} \right]^{\frac{1}{2}} + \frac{D^2}{2} \left[ 1 - \left( \frac{d}{D} \cos(\theta_a) \right)^{2} \right]^{\frac{1}{2}} + \frac{D^2}{2} \sin^{-1} \left( 1 - \left( \frac{d}{D} \right)^{2} \right) \left( \frac{d}{D} \cos(\theta_a) \right)^{2} \right]^{\frac{1}{2}}
\]

When \( \theta_a \geq \cos^{-1} \left( \frac{d}{D} \cos(\theta_{a,0}) \right) - \theta_{a,0} \), \( A_h \) can be expressed as follows:

\[
A_h = \frac{D^2}{2} \sin^{-1} \left( 1 - \left( \frac{d}{D} \right)^{2} \right) \left( \frac{d}{D} \cos(\theta_{a,0}) \right)^{2} \right]^{\frac{1}{2}} \]

where \( D \) and \( d \) are the bore diameter and shaft diameter of the throttle, respectively; and \( \theta_{a,0} \) is the closed throttle angle.

The intake manifold pressure \( P_{\text{im}} \) is obtained from the state equation of the ideal gas as follows:

\[
\frac{d P_{\text{im}}}{d t} = \frac{R_T \text{aim} m_{\text{a}}}{V_{\text{im}}}
\]

where \( V_{\text{im}} \) and \( T_{\text{aim}} \) are the volume and temperature of the intake manifold, respectively; and \( m_{\text{a}} \) is the air mass in the intake manifold that can be expressed as follows:

\[
m_{\text{a}} = m_{\text{a}} + m_{\text{air,th}} - m_{\text{air,th}}
\]
where $m_{\text{IACV}}$ is the mass airflow through the IAC valve; and $m_{\text{in}}$ is the mass airflow from the intake manifold into the cylinder, which can be expressed as follows:

$$m_{\text{in}} = \frac{P_{\text{B}, \text{IAC}}}{R_{\text{in}}} \omega \eta$$  \hspace{1cm} (7)

where $V_{\text{d}}$ is the displaced cylinder volume; $\omega$ is the engine speed; and $\eta$ is the volumetric efficiency [9].

BY-PASS AIR FLOW DYNAMICS

The IAC valve controls the amount of by-pass air flow into intake manifold for maintaining idle speed, as shown in Figure 2.

![IAC valve](image)

**Figure 2. IAC valve**

The by-pass air flow dynamics can be modeled as a one-dimensional isentropic compressible flow, which is similar to Equations (1) and (2) with the $C_{d, \text{IAC}}$ and $A_{\text{in}}$ replaced by the discharge coefficient $C_{d, \text{IAC}}$ and cross-sectional area $A_{\text{IAC}}$ of the IAC valve. The $A_{\text{IAC}}$ can further be expressed using Equations (3) and (4) with $D_{\text{IAC}}$, $d_{\text{IAC}}$, and $\theta_{\text{IAC}}$ replaced by the bore diameter $D_{\text{IAC}}$, the shaft diameter $d_{\text{IAC}}$, and opening angle $\theta_{\text{IAC}}$ of the IAC valve, respectively.

TORQUE DYNAMICS

The indicated torque $T_{\text{ind}}$ is defined as a function of engine speed, air mass flow rate into cylinder, air-fuel ratio, and spark advance angle, and can be expressed as follows [9].

$$T_{\text{ind}}(t) = c_{r} \frac{m_{\text{in}}(t-\Delta t_{\text{cu}})}{\omega(t-\Delta t_{\text{cu}}) AFi(t-\Delta t_{\text{cu}})} SI(t-\Delta t_{\text{cu}})$$  \hspace{1cm} (8)

where $T_{\text{ind}}$ is the indicated torque; $\Delta t_{\text{cu}}$ is the intake to torque production delay; and $\Delta t_{\text{cu}}$ is the spark to torque production delay. $AFi$ is the normalized air fuel ratio influence, which is a function of air-to-fuel ratio $\lambda$.

The normalized spark influence $SI$ is a function of spark advance/retard from MBT ignition timing, as follows:

$$SI = (\cos(\text{SA}-MBT))^{2.75}$$  \hspace{1cm} (9)

where $\text{SA}$ is spark advance angle. The torque constant $c_{r}$ can be used to represent the maximum engine torque capacity with $AFi = 1$ and $SI = 1$.

ENGINE ROTATIONAL DYNAMICS

The engine rotational dynamics is modeled as:

$$T_{\text{load}} - T_{\text{fric}} - T_{\text{eng}} = I_{e} \alpha + b_{e} \omega$$  \hspace{1cm} (10)

where $T_{\text{load}}$ is the external load torque; $T_{\text{fric}}$ is the friction torque [9], which is a function of engine speed; $\alpha$ is the angular acceleration of the engine; $I_{e}$ is the engine rotational inertia; and $b_{e}$ is the damping constant of the crankshaft bearing.

CONTROL ALGORITHM

The proposed ISC consists of two parts: system identification and adaptive controller, as shown in Figure 3. The spark advance angle $\text{SA}$ and IAC valve opening angle $\theta_{\text{IAC}}$ are simultaneously employed as the control inputs for maintaining the designed idle speed.

![Block diagram of the proposed algorithm](image)

**Figure 3. Block diagram of the proposed algorithm**

SYSTEM IDENTIFICATION

A simple discrete two-input single-output first-order model is chosen to approximate the engine dynamics from $\theta_{\text{IAC}}$ and $\text{SA}$ to $\omega$, and can be expressed as follows:
\[
\alpha(k) = \frac{b_1 z^{-k} \theta_{\text{IAC}} + b_2 z^{-k} \text{SA}}{1 + a z^{-k}} = \frac{\text{B}(z^{-k})}{\text{A}(z^{-k})} u(k)
\]  

(11)

where \( z^{-1} \) denotes the delay operator; \( u = [\theta_{\text{IAC}} \text{SA}]^T \) is the control input vector; \( a \) is the parameter of the system dynamics; \( b_1 \) and \( b_2 \) are the parameters of the input dynamics; \( \text{A}(z^{-1}) = [1 + a z^{-1}] \) and \( \text{B}(z^{-1}) = [b_1 z^{-1} b_2 z^{-1}] \) denote for the polynomial matrices of system and input dynamics, respectively. The Auto-Regressive Moving Average (ARMA) model [10] is then utilized to represent the Equation (11) as follows:

\[
\alpha(k) = \Psi^T(k) \Theta(k-1)
\]  

(12)

where \( \Psi(k) = [-\alpha(k-1) \theta_{\text{IAC}}(k-1) \text{SA}(k-1)]^T \) is the regression vector; and \( \Theta(k-1) = [a \ b_1 \ b_2]^T \) is the system parameter vector.

The RLS algorithm is suitable for on-line system parameter identification, and can be expressed as:

\[
\hat{\Theta}(k) = \hat{\Theta}(k-1) + K(k)e(k)
\]  

(13)

\[
e(k) = y(k) - \hat{y}(k) = y(k) - \Psi^T(k)\hat{\Theta}(k-1)
\]  

(14)

\[
K(k) = P(k-1)\Psi(k)[I + \Psi^T(k)P(k-1)\Psi(k)]^{-1}
\]  

(15)

\[
P(k) = P(k-1) - P(k-1)\Psi(k)[I + \Psi^T(k)P(k-1)\Psi(k)]^{-1}\Psi^T(k)P(k-1)
\]  

(16)

where \( e(k) \) is the output prediction error; and \( P(k) \) is the projection operator.

The pseudorandom binary input signals (PRBS) with the amplitudes of 20 deg \( \theta_{\text{IAC}} \) and 20 deg \( \text{SA} \) (see Figure 4) are employed to obtain the open-loop system parameters. The identification results are shown in Figure 5. Steady-state values in Figure 5a will be used as the initial parameter values for the proposed algorithm. The speed response of the proposed linear MISO model is very close to the actual response. After system identification is converged, the maximum speed error of the proposed model is only about 7 rpm.

ADAPTIVE MISO CONTROLLER

A MISO adaptive controller based on STR is then derived using pole placement. It can on-line adjust the control gains according to the estimated parameters using the RLS technique, such that the desired dynamic response can be achieved.

The IAC and spark timing control are designed based on PI and P type STR, respectively. The IAC with PI type STR can be used to maintain the desired engine speed and reduce the steady-state error. Since the effect of spark timing control on the dynamic response of engine speed is faster than the IAC, the spark timing control with P type STR can be used to improve transient dynamic response of engine speed. The STR can be expressed as follows:
\[
R(z^{-1})u(k) = T(z^{-1})\omega_n(k) - S(z^{-1})\omega(k)
\]  

(17)

where \(u(k)\) is the control input; \(\omega_n(k)\) is the desired engine speed; \(T(z^{-1}) = [t_1, t_2]^T\) is a constant matrix; \(R(z^{-1}) = \begin{bmatrix} 1 - z^{-1} & 0 \\ 0 & r_2 \end{bmatrix}\) and \(S(z^{-1}) = \begin{bmatrix} s_0 + s_1z^{-1} \\ s_2 \end{bmatrix}\) are the matrices with first order polynomials. The closed-loop transfer function derived from Equations (11) and (17) can be expressed as follows:

\[
\frac{\omega(k)}{\omega_n(k)} = \frac{h_1r_1z^{-1} + h_2r_2z^{-1}(1-z^{-1})}{r_2(1-z^{-1})(1+a\omega_n) + h_1z^{-1}(s_0 + s_1z^{-1}) + h_2z^{-1}(1-z^{-1})}
\]

(18)

For the closed-loop pole placement, the designed closed-loop poles can be expressed as a second order polynomial, as follows:

\[
A_m(z^{-1}) = [1 + a_{m_1}z^{-1} + a_{m_2}z^{-2}]
\]

(19)

The designed poles can be selected according to the standard second order form as follows:

\[
s^2 + 2\zeta\omega_n s + \omega_n^2 = 0
\]

(20)

where \(\zeta\) and \(\omega_n\) are the desired damping ratio and natural frequency, respectively. The reference model in discrete-time domain can then be obtained with sample time (frame time of the target ECU) \(t_s = 10\) ms, as shown below.

\[
z^2 + a_{m_1}z + a_{m_2} = 0
\]

(21)

where

\[
a_{m_1} = -2\cos(\omega_n t_s \sqrt{1 - \zeta^2 e^{-2\zeta\omega_n}})
\]

(22)

\[
a_{m_2} = e^{-2\zeta\omega_n}
\]

(23)

If we match the polynomial coefficients of the denominator of Equation (18) with Equation (19), the control parameters \(r_1, s_0,\) and \(s_1\) can be solved as follows:

\[
r_1 = 1
\]

(24)

\[
s_0 = \frac{a_{m_1} - a + b_2 s_2}{b_1}
\]

(25)

\[
s_1 = \frac{a_{m_2} + a + b_2 s_2}{b_1}
\]

(26)

Moreover, the coefficients \(s_0\) and \(s_1\) are function of \(s_2\), which is an adjustable design parameter and is designed as P control for spark timing.

Generally, \(T(z^{-1})\) is used to reduce the steady-state error without increasing the system order, thus it can be solved by setting the DC gain of the closed-loop transfer function in Equation (18) to be 1. \(T(z^{-1})\) can then be obtained as

\[
T(z^{-1}) = \begin{bmatrix} r_1 \\ r_2 \end{bmatrix} = \begin{bmatrix} s_0 + s_1 \\ \square \end{bmatrix}
\]

(27)

where \(\square\) is a arbitrary real number.

The spark advance angle \(SA\) is always maintained at MBT for maximum torque output. When load torques are imposed on the engine, the engine speed reduces and the \(SA\) should be advanced to increase the torque output. However, if \(SA\) is advanced more than the MBT, it will reduce the torque output according to Equation (9). Therefore, the \(SA\) is saturated at MBT for the spark timing control in this paper.

Since the P type controller is used for spark timing, \(t_2 = s_2\) is selected in this paper. The adaptive MISO control law can then be expressed as:

\[
u(k) = \begin{bmatrix} t_1\omega_n(k) + u(k-1) - s_0\omega_n(k) - s_2\omega_n(k-1) \\ s_2(\omega_n(k) - \omega_n(k-1)) \end{bmatrix}
\]

(28)

**SIMULATION RESULTS**

A nonlinear engine model is established in Matlab/Simulink to evaluate the performances of the traditional PI controller using only IAC, the MISO controller without the adaptation, and the proposed adaptive MISO controller. The desired natural frequency \(\omega_n\) and damping ratio \(\zeta\) of the reference model are selected to be 1.2 rad/sec and 1, respectively. The identified system parameters from Section 3 are used as the initial conditions for the RLS, i.e. \(\hat{\Theta}_0 = [-0.997 \quad 2.5 \quad -0.009]^T\).

If the adaptation of the proposed algorithm is turned off, the associated simulation results are referred as MISO only in the following figures. The same reference model and identified parameters \(a\) and \(b_1\) are also used to design the traditional PI controller (using only IAC) by the pole placement technique. The external load torque
profile is a combination of ramp, step, and sinusoidal patterns, as shown in Figure 6.

Simulation results of the identified system parameters and engine speed are shown in Figures 7 and 8, respectively. The zoom-in plot of the speed responses under the sinusoidal disturbances is shown in Figure 9.

As can be seen from Figure 7, the torque disturbance effect is implicitly lumped into the identified system dynamics using the RLS algorithm. Because the MISO controller without adaptation and the proposed adaptive algorithm are both designed based on STR structure, they have similar speed responses (see Figures 8 and 9). Although smaller speed deviation can be achieved with the adaptation, the effect of adaptation is not obvious without the model uncertainties.

The speed deviations of these two algorithms are smaller than that of the traditional one. If the torque disturbance is suddenly introduced, such as step disturbance, it results in abrupt undershoot speed response. This is mainly due to the slow response time of the IAC. The charging dynamics of the by-pass air is not fast enough to compensate the sudden change of the load conditions. Similar overshoot speed response can be observed for suddenly removed disturbance, especially for the traditional controller using IAC only. However, this overshoot response can be significantly reduced if both IAC and spark timing are employed simultaneously for maximum control authorities.

The IAC valve opening angles of the traditional PI controller, the MISO controller without adaptation, and the proposed adaptive controller are shown in Figure 10. Since IAC valve openings of the MISO controller without adaptation and the proposed adaptive algorithm are both designed based on STR structure, they have similar speed responses (see Figures 8 and 9). Although smaller speed deviation can be achieved with the adaptation, the effect of adaptation is not obvious without the model uncertainties.

The IAC valve opening angles of the traditional PI controller, the MISO controller without adaptation, and the proposed adaptive controller are shown in Figure 10. Since IAC valve openings of the MISO controller without adaptation and the proposed adaptive algorithm are larger than that of the traditional algorithm, it can bring the speed
back to the desired value faster, as shown in the zoom-in plot on Figure 8.

The spark advance angle retarded from MBT, i.e. $SA = SA - MBT$, of the proposed algorithm is shown in Figure 11. As mentioned in Section 3, the $SA$ is saturated at MBT for the spark timing control. Therefore, $SA$ is lower saturated at 0 deg. The large spark timing controls of the MISO controller without adaptation and the proposed adaptive algorithm can retard the $SA$ from the MBT quickly. Thus reduce the torque output promptly and bring the speed back to the desired value faster than the traditional control using only IAC. This eases the remaining work for the IAC to regulate the engine speed.

If the IAC valve and the damping of the crankshaft bearing are affected by carbon deposition and wear, respectively, the performance of the MISO controller without adaptation maybe degraded due to model uncertainties. The system parameters $A_{IAC}$, $C_{z,IAC}$, and $b_r$ with 10% error are employed to assess the robustness of each control algorithm. The simulation results are shown in Figures 12 and 13. The speed deviation of proposed algorithm is smaller than that of the traditional PI and MISO only algorithms. Since the proposed adaptive algorithm can on-line estimate system parameters and adjust the control gain, it can mitigate the impact of the model uncertainties and thus maintain the desired control performance. Because the traditional PI and MISO only algorithms are designed based on open-loop identified system parameters without adaptation, they are significantly affected by the model uncertainties. Thus results in larger speed deviations.
CONCLUSION

An adaptive MISO controller is proposed for ISC of the spark-ignition engine in this paper. The spark timing and IAC are simultaneously used as the control inputs for maintaining the desired idle speed, and are designed based on P and PI type STR, respectively. The RLS approach is employed to identify the engine dynamics as a first-order two-input single-output model from IAC valve opening angle and spark advance angle to engine speed. Pole placement technique is then used to design the adaptive MISO controller. The simulation results show that the torque disturbance effect can be implicitly lumped into the identified system dynamics using the RLS algorithm. The speed deviations of the MISO controller without adaptation and the proposed adaptive algorithm are smaller than that of the traditional PI controller using only IAC. This is mainly due to that both IAC and spark timing control are employed simultaneously for maximum control authorities. Furthermore, because the traditional PI controller and the MISO controller without adaptation are designed based on open-loop identified system parameters without adaptation, they are significantly affected by the model uncertainties. Thus results in larger speed deviations. Since the proposed adaptive algorithm can on-line estimate system parameters and adjust the control gains, it can mitigate the impact of the model uncertainties and thus maintain the desired control performance. This preliminary simulation study shows promising results for ISC. The prototype ISC is being established in the lab and the associated experiment will be conducted soon to verify the proposed algorithm.

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