

Application of Adaptive Idle Speed Control on V2 Engine

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ABSTRACT

This paper develops an adaptive idle speed control strategy for a V2, 1000 cc four-stroke, water-cooled, port injection SI engine. In order to verify the proposed strategy, the non-dimensional engine model including charging and torque dynamics is established in Matlab/Simulink software based on previously experimental verification. The integration of dynamics above will be a multi-input-single-output (MISO) system, which inputs are throttle angle and spark advance angle, and the output is engine speed. The proposed adaptive controller is developed on the model-based structure. The system parameters are updated by recursive least square (RLS) method so the system is able to represent the actual operation. The updated system parameters adjust control gain by derivation of closed-loop gain and pole placement. This research also applies RLS method in torque load identification, and uses feedforward control module to calculate the desired throttle opening for the corresponding engine torque. The feedforward throttle control makes controller less angle of spark advance, therefore, the tracking time could be reduced. In order to verify the proposed adaptive idle speed controller performance, the conventional proportional integral derivative (PID) idle speed controller is established. The results show the proposed feedforward adaptive controller can obviously improve about 40% of tracking time with ramps load. In addition, when the water pump load is added, the proposed strategy can also improve about 20%. Finally, the results of the proposed strategy are also improved comparing to the conventional PID control as deposits happen.

KEYWORDS: RLS, Adaptive Control, Deposits, Feedforward Control

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INTRODUCTION

In the city, there is 30% of whole driving time under engine idle operation. Idle speed control plays an important role in engine management system (EMS). The great idle speed control can make engine speed stable, reduce pollution and improve fuel economy. Carbon deposits in intake system will make engine idle speed unstable due to the engine dynamics change. Therefore, there should be a strategy to solve deposits problem under engine idle operation.

Engine idle speed control has been studied around the world. The GM in American used the variable valve timing and ignition advance in the idle speed control system [1]. In addition, in HONDA, by-pass valve was applied to control air flowing into cylinder according to throttle opening, coolant temperature and engine speed [2]. Gängopadhyahy and Meckl [3] employed multivariable proportional integral (PI) controller and Nichols tuning method to design idle speed controller, the engine speed error reduced while adding the external load. Continuous action reinforcement learning automata (CARLA) which could automatically correct parameters was also used mainly for cost function optimization [4]. Min [5] pointed out the optimization control whose input was idle control valve and spark angle could have similar result of estimating engine speed comparing

with vehicle test based on nonlinear system model. There will be better fuel economy and lower variation of engine speed with linear quadratic regulator (LQR) and closed-loop control. Noble [6] adopted model predictive control (MPC) and cost function as engine speed error and quadratic form of idle speed valve-opening. The linear model whose output is engine speed get from system identification and inputs are idle by-pass valve and engine load made cost function minimize by using control rule extracted from optimal control. The response of ignition advance control was faster than idle air control (IAC) and electric throttle control. Deur [7] indicated polynomial controller can effectively resist noise and reduce load influence. Yang [8] combined spark and throttle control to utilize sliding mode controller. Hsieh [9] controlled idle by-pass valve and spark advance to stabilize engine idle with adaptive controller. Lin [10] applied RLS method to design a MISO model predictive controller whose inputs are idle control valve and spark timing and output is engine speed.

To increase engine stability, this paper proposes adaptive idle control on model-based structure for deposits problem. The mean value engine model (MVEM) is established first, and then the adaptive idle controller is developed by applying system

identification method and adaptive controller. In addition, the throttle feedforward control is also used to improve traditional carbon deposit problem on throttle valve.

ENGINE MODEL

Instantaneous Engine Model

It can reduce cost and developed times on control strategy development stage if model can verify feasibility and performance of strategy. In this research, the instantaneous engine model composed of charging dynamic, fueling dynamic, combustion dynamic, crankshaft dynamic and sensors dynamic, and established using modulation method as shown in [figure 1](#). V2 1000 cc four-stroke, water-cooled, port injection SI engine is selected as target engine, and the specification is listed in [Table 1](#).

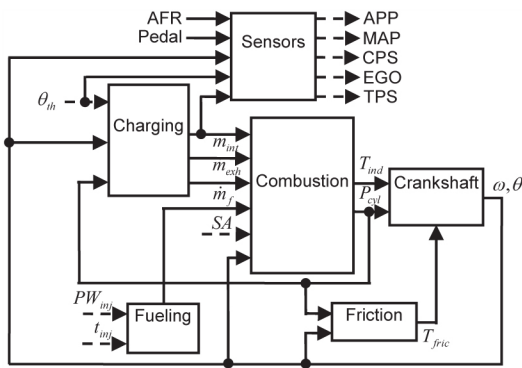


Figure 1. Instantaneous engine model

Table 1. The specification of target engine

Engine Type	Four-stroke, water cooled
Bore x Stroke	86mm x 86mm
Fuel System	Electronic fuel injection
Number of Valves	4 (2-In, 2-Ex) per cylinder
Displacement Volume	1000cc
Compression Ratio	9.9
Idle Speed	1200rpm
Ignition Type	SI(Spark Ignition)

Mean Value Engine Model

In order to the computation of instantaneous engine model is complicated. This study proposes a multi-input-single-output (MISO) engine model mainly containing intake manifold pressure and torque dynamics whose input is electric throttle opening and ignition advance, and engine speed is output. The intake manifold pressure (P_m) dynamic can be defined as:

$$\dot{P}_m(t) = \frac{R_{im} T_{im}}{V_{im}} \dot{m}_{net}(t) \quad (1)$$

$$\dot{m}_{net}(t) = \dot{m}_{th}(t) - \dot{m}_{int}(t) \quad (2)$$

Where R_{im} is intake manifold gas constant, T_{im} is intake manifold temperature, V_{im} is intake manifold volume, $\dot{m}_{th}$ and $\dot{m}_{int}$ represent separately throttle intake air mass flow rate and the intake air mass flow rate as follow:

$$\dot{m}_{th} = \frac{C_d A P_0}{\sqrt{R T_0}} \sqrt{k} \left(\frac{2}{k+1} \right)^{\frac{k+1}{2(k-1)}} \quad (3)$$

$$\dot{m}_{int} = \frac{P_{im} V_d}{R_{cyl} T_{im}} \eta_v \cdot \frac{\omega}{60} n_{event} \quad (4)$$

Where C_d is flow coefficient obtained from flow test, A is orifice effective area, R is gas constant, P_0 is upstream pressure of orifice, T_0 is upstream air temperature of orifice, k is air specific heat ratio of orifice, P_{im} is pressure of intake manifold, V_d is displacement volume, R_{cyl} is gas constant cylinder, T_{im} is internal intake manifold temperature, η_v is volume efficiency, ω is engine speed, n_{event} is power stroke number per engine operation, 0.5 is set in this research due to the target engine is V2 engine.

Finally, function variables of intake manifold dynamic can be defined as follow:

$$\dot{P}_m = f_p(P_m, \omega, \theta_{th}) \quad (5)$$

In torque dynamics model, the brake torque output can be defined as follow:

$$T_{brak} = I_e \cdot \dot{\omega} = T_{ind} - T_{fric} - T_{aux} \quad (6)$$

Where I_e is crankshaft moment of inertia got from engine experiment, $\dot{\omega}$ is angular acceleration, T_{ind} is indicated torque, T_{fric} is friction torque, T_{aux} is auxiliary torque,

In this study, T_{ind} is calculated by intake air flow according to fuel consumption as shown in [equation \(7\)-\(8\)](#). Furthermore, the [equation \(9\)](#) is used to get indicated torque under minimum advance for best torque (MBT) condition. Finally, the [equation \(10\)](#) can be obtained by using [equation \(7\)-\(8\)](#) to substitute [\(9\)](#).

$$\dot{m}_{int} = \frac{P_{im} V_d}{R_{cyl} T_{im}} \eta_v \cdot \frac{\omega}{60} n_{event} \quad (7)$$

$$m_f = \frac{1}{AFR_{des}} \cdot \frac{60 \cdot \dot{m}_{int}}{\omega n_{event}} \quad (8)$$

$$T_{ind, MBT} = \frac{m_f Q_{HV} n}{4\pi} \eta_f$$

$$T_{ind,MBT} = \frac{1}{AFR_{des}} \cdot \frac{P_{im} V_d}{R_{cyl} T_{im}} \eta_v \cdot \frac{Q_{HV} n}{4\pi} \eta_f \quad (10)$$

Where m_f is fuel consumption, AFR_{des} is desired air fuel ratio, $T_{ind,MBT}$ is indicated torque under MBT, Q_{HV} is fuel heating value, n is cylinder quantity, η_f is fuel efficiency obtained from engine experiment.

In normal engine idle operation, the engine ignition timing will delay for achieving immediate torque requirement via ignition control. The relationship between ignition timing and torque can be represented as equation (11)-(12) and shown as figure 2.

$$\frac{15.06 - 13.14}{36 - 14.5} = \frac{T_{ind,MBT} - T_{ind}}{MBT - SA} \quad (11)$$

$$T_{ind} = T_{ind,MBT} - 0.06(MBT - SA) \quad (12)$$

T_{ind} is actual output torque, SA is ignition advance angle.

Function variable of engine speed dynamic can be defined as follow:

$$\dot{\omega} = f_{\omega}(P_{im}, \omega, SA) \quad (13)$$

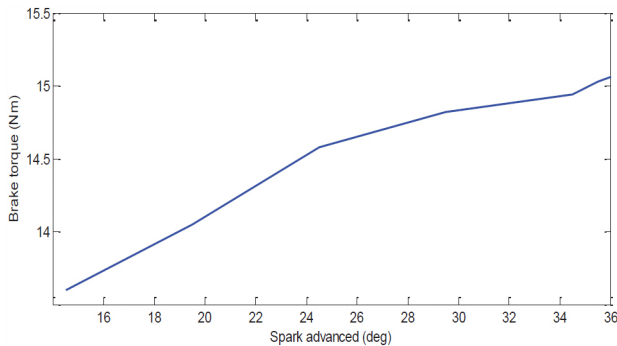


Figure 2. Ignition angle and actual output torque

REGRESSION MODEL AND SYSTEM IDENTIFICATION METHOD

For executing linearization control in target point, the nonlinear engine dynamics should be linearized with above-mentioned MVEM. In this study, the linearized MVEM is applied to verify RLS method whether system parameters can be identified. First, the engine dynamics variation near the equilibrium point defined in desired engine idle speed can be regarded as the linear relation. Therefore, using linearization method to substitute equation (1), (2), (3), (4), (6), (7), (8), (9), (10), (11), (12) can get equation (14)-(15). After discretizing equation (14)-(15), the equation (16)-(17) can be expressed as follow:

$$\Delta \dot{P}_{im} = \frac{\partial \dot{P}_{im}}{\partial P_{im}} \Delta P_{im} + \frac{\partial \dot{P}_{im}}{\partial \omega} \Delta \omega + \frac{\partial \dot{P}_{im}}{\partial \theta_{th}} \Delta \theta_{th} \quad (14)$$

$$\Delta \dot{\omega} = \frac{\partial \dot{\omega}}{\partial P_{im}} \Delta P_{im} + \frac{\partial \dot{\omega}}{\partial \omega} \Delta \omega + \frac{\partial \dot{\omega}}{\partial SA} \Delta SA \quad (15)$$

$$\Delta P_m(k+1) = D_{id_1} \Delta P_m(k) + D_{id_2} \Delta \omega(k) + D_{id_3} \Delta \theta_{th}(k) \quad (16)$$

$$\Delta \omega(k+1) = D_{id_4} \Delta P_m(k) + D_{id_5} \Delta \omega(k) + D_{id_6} \Delta SA(k) \quad (17)$$

The equation (18)-(19) can be obtained by transforming difference equation into auto-regressive moving average (ARMA) form:

$$\Delta P_{im}(k+1) = \begin{bmatrix} \Delta P_{im}(k) & \Delta \omega(k) & \Delta \theta_{th}(k) \end{bmatrix} \begin{bmatrix} D_{id1} \\ D_{id2} \\ D_{id3} \end{bmatrix} \quad (18)$$

$$\Delta \omega(k+1) = \begin{bmatrix} \Delta P_m(k) & \Delta \omega(k) & \Delta SA(k) \end{bmatrix} \begin{bmatrix} D_{id4} \\ D_{id5} \\ D_{id6} \end{bmatrix} \quad (19)$$

This research applies system identification as recursive least square (RLS). Before applying RLS method, inferring the engine model to least square including unknown-factor and known-factor is first. The unknown-factor got from least square can make response of least square model similar to real engine operation so it can be employed to infer adaptive structure. This chapter represents the derivation process of least square from MVEM and induction about RLS.

The RLS algorithm is defined as follow:

$$P(k) = P(k-1) - P(k-1)\varphi(k) \cdot \left[I + \varphi^T(k)P(k-1)\varphi(k) \right]^{-1} \cdot \varphi^T(k)P(k-1) \quad (20)$$

$$K(k) = P(k-1)\varphi(k) \left[I + \varphi^T(k)P(k-1)\varphi(k) \right]^{-1} \quad (21)$$

$$e(k) = Y(k) - \hat{Y}(k) = Y(k) - \varphi^T(k)\hat{\theta}(k-1) \quad (22)$$

$$\hat{\theta}(k) = \theta(k-1) + K(k)e(k) \quad (23)$$

Where $P(k)$ is projection operator, $e(k)$ is forecasts inaccuracy output.

The convergent system with system identification stands for the system response is closed to real operation and here resulting equations can be applied in controller.