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The Control Method Research for Piston Stop Position of the GDI Engine

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Abstract

The piston stop position is crucial for instantaneous reverse starting method in idle-stop system. It is very difficult to control the piston stop position. In this paper, the relationship between the piston stop position and the air inflow is set up. Then the piston stop position control problem is converted to the electronic throttle control problem. A feedforward-feedback controller for the piston stop position of the GDI engine is proposed. The feedforward controller is spring compensator and friction compensator, which can eliminate the nonlinearity error of throttle system. The feedback controller is backstepping controller which is adaptive for a class of nonlinear processes. The simulation results show that the composite control law enables the throttle opening to track input orders quickly and accurately. And there is no overshoot.

Keywords: *piston stop position, throttle control, nonlinear model, feedforward-feedback controller*

1. Introduction

Idle stop systems save fuel by shutting down engine automatically when the vehicle is idle. The engine restarts again quickly when the driver resumes driving information. Idle stop is one of key techniques to save fuel consumption and to reduce exhaust emissions in urban areas especially [1-3]. Now, both techniques of motors tarting and instantaneous reverse starting are used to restart engine usually in idle stop system. The motor starting technique restarts a vehicle's engine by the electric motor working. The high levels of starting number and reliability are required in this technique. The instantaneous reverse starting technique restart by injecting fuel directly into a cylinder while the engine is stopped, and igniting it to generate downward piston force. This technique not only saves fuel consumption, but also restarts the engine more quickly and quietly than a conventional idle-stop technique. The crucial point of the instantaneous reverse starting technique is the piston stop position when engine shut down. The condition is compression stroke piston stops in 60 °~ 80 ° CA before top dead center (BTDC), which ensures the air quantity in the compression stroke cylinder meeting up the combustion conditions for first time. It means the expansion stroke cylinder has more air volume, which makes the second combustion producing enough burning energy to make sure the engine starting quickly and reliably [4, 6-7]. There are more researches about starting motor starting technique. Literature [6] introduced the stop position control method by using the car alternating-current generator. The opening of electronic throttle is adjusted to control the piston stop position. The pulse width modulation (PWM) is used to

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control the opening of throttle, which is opened loop control method. The paper [8] introduce the feed-forward-feedback control method, which is based on PWM with proper period as control variable. The feedback controller is PID. In this paper, feed-forward and feedback composite method of the electronic throttle controller is designed. The feed-forward controller is spring compensator and friction compensator, which can eliminate the nonlinearity error of throttle system. The feedback controller is back-stepping controller which is adaptive for a class of nonlinear processes. The simulations show that the compound control method can control the electronic throttle opening to track the target quickly and accurately.

2. The Description for Piston Stop Position Control Problem

After engine shutting down, the crankshaft flywheel is still storing a certain momentum under the inertia effect. The kinetic energy in cylinder is consumed by drag torque and friction torque gradually. The crankshaft does decelerated motion and the piston is stop completely in the end. Under uncontrolled conditions, the bench experiment shows that the rate of compression piston stopping on 80 ° CA before top dead center is about 50%, and stopping on the 70 ° CA before top dead center is about 30%. The other stopping angle is about 20% [4].

The controlled physical variables are used to consume the kinetic energy of the crankshaft fly wheel regularly. The aim is to make the piston stop in 60°- 80°CA before top dead center. There are a lot of physical variables affecting the stop position of the piston, such as mechanical resistance, gas resistance, energy dissipation of external device, engine accessories work *etc*. The mechanical resistance of kinematic pair motion friction is not controllable. The gas resistance torque can be controlled by adjusting the size of the throttle opening. In addition, the drive loss power is controllable by engine accessories loss. However, when rotation speed is low, the method of drive loss power is invalid. In this paper, the control problem of gas resistance torque can be converted into the control problem of throttle opening. Thus the piston stop position is adjusted by opening of throttle. The aim of controlling piston stop position is achieved.

The throttle opening and the amount of air flowing into cylinder is corresponding relationship. The more air flows into cylinder, the more inertial kinetic energy loss through gas working. The inertial kinetic energy drives the piston reciprocating motion. The piston stops when the inertial kinetic energy is consumed to zero. Actually, the study problem is tracking control problem. The throttle opening is varied according to different types of engine when piston stops in the some ideal position. The relationship of the throttle opening and stop position can be obtained by experiment of the test bench. The compensation quantity is designed according to water temperature, rotational speed and atmospheric pressure. The MAP of relationship of the throttle opening and stop position are made and is used to control the piston stop position.

Under normal conditions, the accelerator pedal position signal is processed in the ECU. This position signal translate into throttle opening signal, which sends to DC servo motor. The DC servo motor drives the throttle rotation. At the same time, the information of the opening angle of the throttle is detected by sensor. The output data of sensor are voltage signals which is send to ECU.

When engine works in shutting down condition, the power unit does not supply for engine electronic control system. The electronic throttle system does not work and the throttle goes back to limp home position. It is important to design idle stop control circuit which can control the throttle to work when the engine shut down. The idle stop control circuit is the control transfer circuit in essence. Thus the throttle opening is controlled to track the control goal when the engine is shut down.

3. Modeling of Electronic Throttle

A 4G15GDI electronic throttle is selected as throttle system. The electronic throttle structure is shown in Figure 1. The throttle system is composed with a DC servo motor, a set of reduction gears, a reset spring and a throttle plate. The throttle plate is driven by the DC servo motor through a set of reduction gears. The motor torque is counterbalanced by the dual return spring attached to the throttle plate's shaft. The throttle angle is detected by the throttle position sensors. The detected signal is converted to digital signals by the analog-to-digital converters, which is lumped together in the sensors section. The structure diagram is shown in Figure 1.

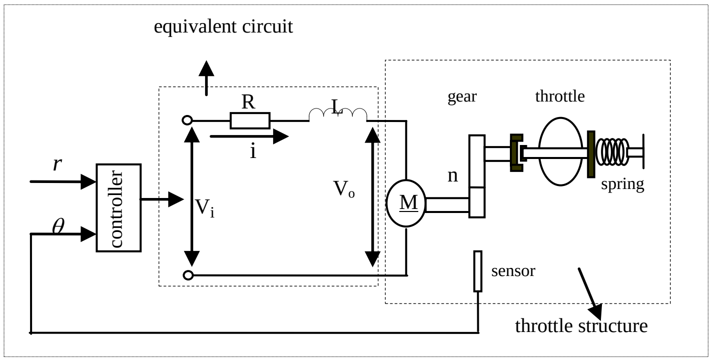

Figure 1. The Structure Diagram of Electronic Throttle

The equivalent circuit of DC servo motor is shown in left of Figure 1. The V_i is input voltage, V_o is back electromotive force, R is armature resistance, i is armature current, L is armature inductance, The Kirchhoff's law loop equation for the armature circuit is given by equation:

$$V_i = Ri + L\frac{di}{dt} + V_o \tag{1}$$

The back electromotive force of the armature V_o is directly proportional to motor angular velocity θ_0:

$$V_o = k_0 \frac{d\theta_0}{dt} \tag{2}$$

k_0 is motor torque constant.

Because electronic throttle is a low pass system and L value is very small, the influence of inductance for circuit loop is ignored. The following equation is obtained from equation(1):

$$i = \frac{V_i - V_o}{R} = \frac{V_i - k_0 \frac{d\theta_0}{dt}}{R} \tag{3}$$

According to the principle of dynamics, the torque of motor output T_0 is:

$$T_0 = Ki - J\frac{d^2\theta_0}{dt} \tag{4}$$

K is the motor counter electromotive force constant. J is lumped inertia of throttle plate, The gear ratio of reduction gear is n . Taking the equation(3)into(4), output torque of throttle T_j is obtained:

$$\begin{aligned} T_j &= nT_0 \\ &= n(K\frac{V_i - k_0\frac{d\theta_0}{dt}}{R} - T_f - J\frac{d^2\theta_0}{dt^2}) \end{aligned} \tag{5}$$

The T_f (N · m) is equivalent friction loss torque of gear set. The relationship of motor rotation θ_0 and throttle angle θ is:

$$\theta = \frac{1}{n}\theta_0 \tag{6}$$

The dynamics of the throttle shaft derived from the relationship between the throttle angle θ and torques. There are three torques acted upon the throttle shaft: the reset spring torque, the viscous and try friction torques, output torque of throttle.

$$J_{th}\frac{d^2\theta}{dt^2} = T_j - T_r - T_{tf} \tag{7}$$

Where, J_{th} is the moment of inertia of throttle($\mathrm{kg \cdot m^2}$). T_r is reset spring torque, which is the relevant with the elastic coefficient of spring and the initial throttle opening θ_i and rotation angle:

$$T_r = K_r(\theta - \theta_i) \tag{8}$$

T_{tf} is frictional resistance torque, which includes try friction and viscous friction.

$$T_{tf} = K_v\frac{d\theta}{dt} + K_c \operatorname{sgn}\frac{d\theta}{dt} \tag{9}$$

K_v is coefficient of viscous friction, K_c is coefficient of try friction. Taking the equations (5), (6), (8) and (9) into (7), the throttle plate dynamics can now be expressed by following formula:

$$(J_{th} + Jn^2)\frac{d^2\theta}{dt^2} = nK\frac{V_i - k_0\frac{d\theta_0}{dt}}{R} - K_r(\theta - \theta_i) - K_v\frac{d\theta}{dt} - K_c \operatorname{sgn}\frac{d\theta}{dt} \tag{10}$$

For the convenience of design control system, the state variables of piston position control system is defined: $x = [\theta \quad \dot{\theta}]^T$. V_i is input control variable and the throttle angle θ is output control variable. The state model is given:

$$\begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ \dfrac{-K_r}{J_{th} + n^2 J} & \dfrac{-K_v - K_c \operatorname{sgn}}{J_{th} + n^2 J} - \dfrac{n^2 K_0 K}{(J_{th} + n^2 J)R} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \dfrac{nK}{(J_{th} + n^2 J)R} \end{bmatrix} V_i + \begin{bmatrix} 0 \\ K_r \end{bmatrix} \theta_i \tag{11}$$

$$y = \theta$$

4. A Feed Forward - Feedback Composite Controller Design

Electronic throttle has the nonlinear characteristics, time-varying and uncertainty. They are caused by friction or a mechanical back lash which comes from the mechanical conditions and motion. Discontinuous or “hard” nonlinearity that cannot be locally approximated with a linear function are present in the throttle system. Examples are back lash, dead-zones and relays. For those nonlinear and uncertainty, a single control strategy is difficult to meet the requirements of control accuracy and response characteristics. A feedforward - feedback composite controlling method is used to compensate the error caused by nonlinear. The control system structure is shown in Figure 2. The feedforward controller is spring compensator and friction compensator, which can eliminate the nonlinearity error of throttle system. The feedback controller is backstepping controller, which is adaptive for a class of nonlinear processes. The backstepping controller designing is a recursive procedure for systematically selecting the control Lyapunov functions that guarantees closed-loop system be stable. The control method is virtual control. The virtual control variable is substituted into the real input control equation conversely. Then the control law is obtained. In Figure 2, θ_r is opening to track, and θ is the actual opening. The θ_r is decided by the piston stop position, that is different according to different engine. The opening θ_r is derived by bench test. The electronic throttle is described as SISO system. The input is normalized voltage u . The output is throttle opening angle θ , which is described in measured voltage.

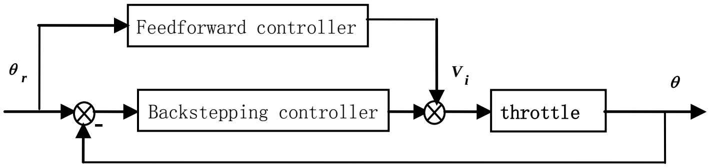

Figure 2. The Control System Structure

4.1. Feedforward Controller Design: The feedforward controller is spring compensator and friction compensator. The spring force and friction force always act on the throttle system at the same time, so real parameters are changed with the different throttle opening. In this paper, the feedforward controller design is finding the friction modeling and the spring nonlinear characteristic by experiment or physical derivation. There are many researchers about the friction and spring problem [9, 10] studies. Based on the quasi-static experiment, spring and friction characteristic can be obtained by voltage input signal and the output signal of angle and armature current. With the input voltage increasing or decreasing with very small rate, the armature current of throttle is varied slowly. In this case, the armature current is regarded as constant which is important. The characteristic is shown in Figure 3.The output of the throttle is intermittent changing caused by the nonlinear of throttle system. A series of critical point marked A, B, C and D points are produced. Those points contain information of throttle position, the input voltage and the friction force direction corresponding. In the critical point of the system, *i.e.*, $x = [\theta \quad \dot{\theta}]^T = [0,0]$, the law of the friction torque varying is obtained by equation (10), and the nonlinear spring characteristic is plotted shown in Figure 4. The mathematical relationship is defined as:

$$V = \begin{cases} k_2(\theta - \theta 3) & V_1 < V < V_2 \\ V_2 + k_3(\theta - \theta_3) & V > V_2 \\ k_1(\theta - \theta_3) - V_1 & V < V_1 \end{cases} \qquad (12)$$

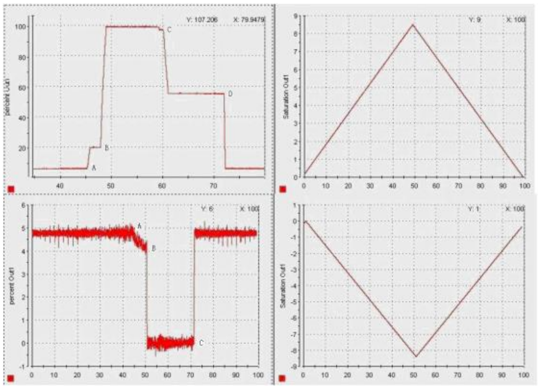

Figure 3. The Input and Output of Throttle

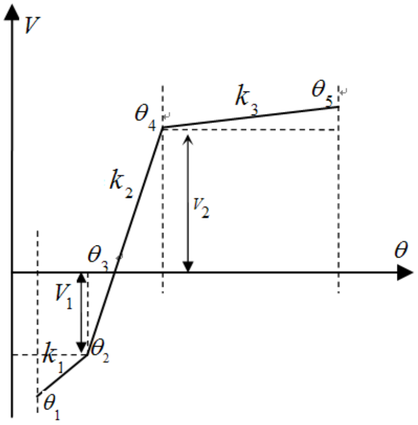

Figure 4. The Nonlinear Spring Characteristic

The spring and friction characteristic are used to compensate the nonlinear error.

4.2. Feedback Controller Design: The feedback controller is designed by using backstepping method that is a recursive procedure. The virtual control variable is derived by selecting the control Lyapunov functions that guarantees closed-loop system be stable. Then we deduce the control law by substituting the virtual control variable into the real input control equation conversely.

The $e_1 = \theta_r - \theta$ is defined, and Lyapunov function $V_1 = \frac{1}{2}e_1^2$ is selected, then we have:

$$\dot{V}_1 = e_1 \cdot \dot{e}_1 = e_1 \cdot (\dot{\theta}_r - x_2)$$

Defining x_2 is virtual control variable, and let $k_1 > 0$. Supposing virtual control variable is $x_2^* = \theta_r + k_1 e_1$. If $x_2 = x_2^*$, then $\dot{V}_1 = -k_1 e_1^2 < 0$. Then the system is asymptotically stable, and the output θ tracks input θ_r gradually. In fact $x_2 \neq x_2^*$, so we need further to design. The second error variables is defined as $e_2 = x_2^* - x_2 = \theta_r + k_1 e_1 - x_2$, and select Lyapunov function $V_2 = \frac{1}{2}(e_1^2 + e_2^2)$, then:

$$\dot{V}_2 = e_1 \cdot \dot{e}_1 + e_2 \cdot \dot{e}_2 = e_1 \cdot (\dot{\theta}_r - x_2) + e_2 \cdot \dot{e}_2 = -k_1 e_1^2 + e_2 (e_1 + \dot{e}_2)$$

Letting $k_2 > 0$, if $e_1 + \dot{e}_2 = -k_2 e_2$ is set up, then $\dot{V}_2 = -k_1 e_1^2 - k_2 e_2^2 < 0$,

According to Lyapunov stability theory, the first-order no difference tracking of system would realize if the error e_1 , e_2 are asymptotically stable.

The function $e_1 + \dot{e}_2 = -k_2 e_2$ is no difference tracking condition. Taking e_1 and e_2 into $e_1 + \dot{e}_2 = -k_2 e_2$, the following equation is derived:

$$\dot{x}_2 = (1 + k_1 k_2)\theta_r + (k_1 + k_2)\dot{\theta}_r + \ddot{\theta}_r - (1 + k_1 k_2)\theta - (k_1 + k_2) x_2 \tag{13}$$

Taking the equation (13) into (10), the control law is obtained:

$$u = \frac{J'(1 + k_1 k_2)}{nk}\theta_r + \frac{J'(k_1 + k_2)}{nk}\dot{\theta}_r + \frac{J'}{nk}\ddot{\theta}_r + (\frac{nk_0}{R} - \frac{J' k_1}{nk} - \frac{J' k_2}{nk} + \frac{k_c}{nk}\mathrm{sgn} + \frac{k_v}{nk}) x_2$$
$$+ (\frac{kk_0}{nk} - \frac{J'}{nk} - \frac{J' k_1 k_2}{nk}) x_1 + \frac{k_v}{nk}\theta_i$$

Where $J' = J_{th} + Jn^2$.

The $\mathrm{sgn}(x_2)$ existed in control law is a serious nonlinear item, that causes oscillation. In order to eliminate the oscillation, the $\mathrm{sgn}(x_2)$ is transformed to $sat(\Delta x))$, where

$$sat(\Delta x) = \begin{cases} 1 & \Delta x > 1 \\ \Delta x & -1 \le \Delta x \le 1 \\ -1 & \Delta x < -1 \end{cases} .$$

5. Simulation Analysis

After the design of the feedforward-feedback composite controller, the implementation was made in the real-time environment Matlab/Simulink. The simulation model of the closed-loop system is shown as Figure 5. The control system is composed of control object and controller. The physical parameters of electronic throttle are shown in Table 1.

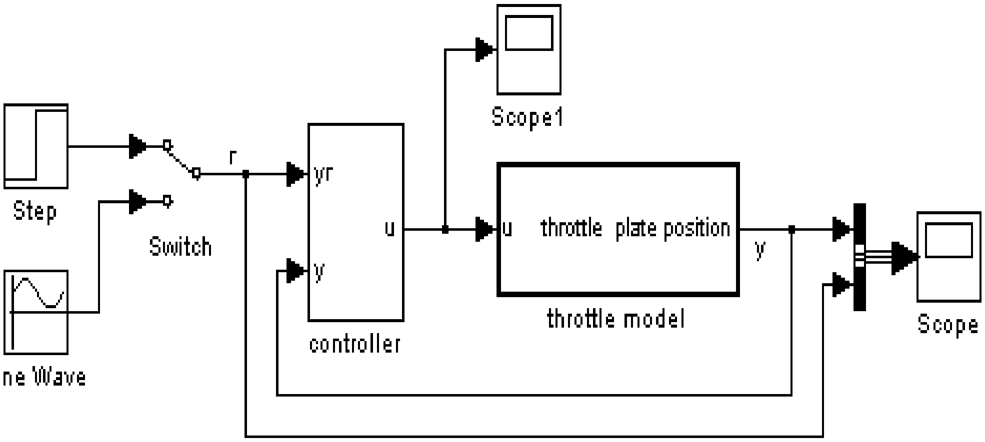

Figure 5. Closed-loop System Simulation Model

Table 1. The Plant Parameter Values

v_i	n	θ_i	R	δ	k_{r1}	k_{r2}
12 V	22.08	0.116 rad	2.8 Ω	63.662 %/rad	3.89×10^{-4} N·m/rad	0.0033 N·m/rad
k_v	k_c	k	k_0	J	B	
0.1393 N·m	0.0048 N·m	0.016 N·m/A	0.016 V/(rad/s)	4.0×10^{-6} $Kg\cdot m^2$	0 N·m/rad	

The δ is throttle opening coefficient of radians converting percentage.

The selection of controller parameters is presented following the previous section given design method. The parameter k1 is selected by the requirement of adjust time. The parameter k2 is selected by the requirement of static error. The selection of controller parameters is presented following the previous section given design method.

The parameter k1 is selected by the requirement of adjust time. The parameter k2 is selected by the requirement of static error. In this paper, the adjust time is less than 120ms and the static error is less than 15%. According to selecting adjust time law [11], we select controller parameters $k_1 = k_2 = 30$ and control period 4ms. The step with 10-60 is selected to be input. The step response figure is showed in Figure 6. There is a 10% overshoot and closed-loop regulating time is 278 ms when the input jumps from 10 to 60. The response time is more than 100 ms that does not meet the control requirement. Second step, we select controller parameters $k_1 = k_2 = 60$ and control period 2ms, the other parameters are same as the first step. The step response figure is showed in Figure 7. The Figure 7 shows that there is no overshoot, and the regulating time is 85 ms, which meets the control requirements.

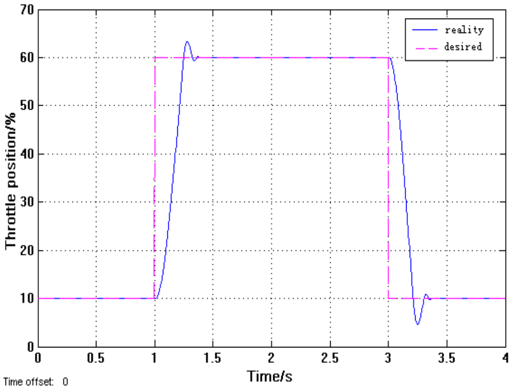

Figure 6. Step Response ($k_1 = k_2 = 30$)

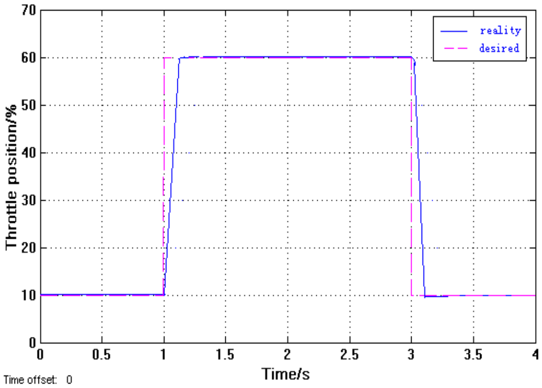

Figure 7. Step Response($k_1 = k_2 = 60$)

6. Conclusion

In this paper, the piston stop position control problem is converted to the electronic throttle control problem. According to the working principle of electronic throttle, the control model of electronic throttle is derived. The composite controller is designed in view of the nonlinear features of throttle model. The feedforward controller can eliminate the nonlinearity error of throttle system. The feedback control law is deduced by substituting the virtual control variable into the real input control equation conversely, and the feedback controller is adaptive. The simulation results show that the composite control law enables the throttle opening to track input orders quickly and accurately. And there is no overshoot.

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