Rail Pressure Control of Common Rail System for Gasoline Direct Injection Engines

I. Introduction

In order to reduce pollutant emissions required by more and more strictly emission legislations, and further improve engine performance perceived by customers. The gasoline direct injection (GDI) has been applied to the gasoline engine system for improving the engine combustion. To achieve a better air-fuel mixture, the fuel is injected directly into the combustion chamber of each cylinder in GDI engines, leading to economic fuel consumption, powerful torque output and efficient emission reduction. As one of the most important parts in GDI engine fuel-path system, the fuel Rail Pressure Control (RPC) system provides the requested rail pressure for injectors to realize gasoline direct injection. Hence, the rail pressure control for obtaining desired injection pressure and reducing effectively the pressure fluctuation becomes one of the fundamental control tasks in GDI engines.

II. Plant and Control Problem Statement

Taking a four-cylinder four-stroke GDI engine as an instance, the fuel Rail Pressure Control system, shown in Fig. 1, is composed mainly of the low pressure circuit, the high pressure pump (HPP), the fuel rail, the injectors, the rail pressure sensor and the electronic control unit (ECU).

The few details of the injection operation are illustrated as follows. The low pressure circuit provides low pressure fuel coming from the tank to the high pressure pump. The pressure control valve is installed in high pressure pump, whose driving signal is square current with a variable duty cycle and period. It allows for effective control to the amount of fuel. The check valve installed in outlet of high pressure pump is to avoid unwanted refluxes, and the limiting pressure valve installed on outlet of HPP prevents the rail from the damage by excessive pressure, when the pressure is larger than the setting maximum pressure. It is opened and the fuel flows back to the tank, the pressure in rail pipe gradually decreases. The fuel rail connects the high pressure pump with the injectors and absorbs the pressure pulsation. High pressure fuel is injected directly into the combustion chamber by electro-injectors, and the fuel injection pulse width commands are given by the ECU. The pressure in fuel rail rises when the fuel is pumped into the rail pipe and the pressure drops when the injection occurs. The main role of the high pressure pump is to provide high

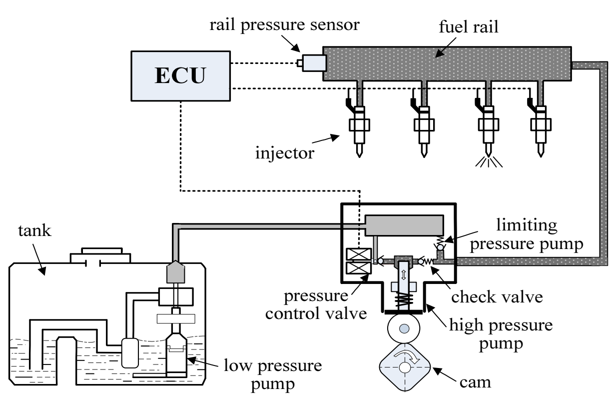

Fig. 1. The structure diagram of the common rail system of GDI engines

pressure fuel for fuel rail. One of the common structures is four-lobes cam-driven which mounted on the engine camshaft, and the piston motion is obtained according to the eccentric profile of the camshaft. In one injection cycle, the operating process of HPP, shown in Fig. 2, consists of three phases: the suck flow process, the back flow process and the pump flow process. (1) In the suck flow process, the pressure control valve remains open, the fuel flows into the high pressure pump from the low pressure circuit due to the pressure difference. As the check valve exists, there is no back-flow at the outlet of the HPP. (2) In the back flow process, when the pressure control valve is still open, the fuel flow back to the low pressure circuit alone with plunger moves from the pump bottom dead center from the pump top dead center. (3) The pump flow process happens when the cam runs to the pump top dead center from the pump bottom dead center and the pressure control valve keeps close. At this time, the fuel flows to the fuel rail.

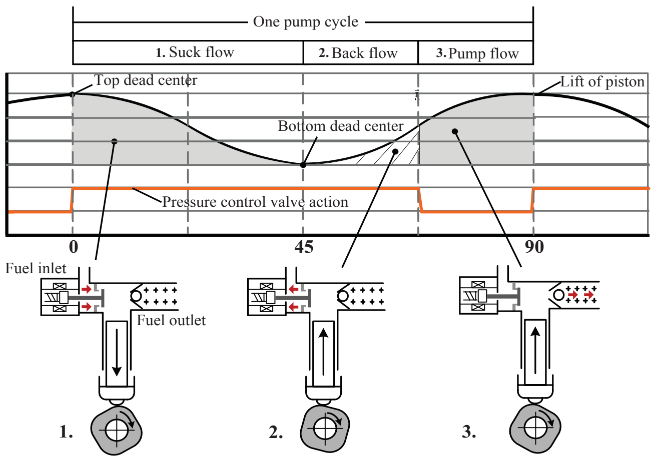

Fig. 2. The operating principle of the cam-driven high pressure pump

If the shadow area formed by the piston lift curve represents the fuel volume at each phase, then the fuel volume in the pump flow stage, which is pumped into common rail pipe, is determined by the closing duration of the pressure control valve (expressed by duty cycle), which means the rail pressure can be regulated by the pumping fuel for the common rail pipe with fixed volume. It is worth pointing out that the pump flow phase only occurs during the movement of the plunger from the bottom dead center to the top dead center.

The task of the common rail system is to establish the desired rail pressure swiftly and avoid large rail pressure fluctuations. The volume of fuel pumped into the common rail can be controlled by adjusting the closing time (duty cycle) of the pressure control valve. The difficulty of control lies in: the common rail system is a integrated mechanical-electric-liquid nonlinear system, pressure propagation in pipe has characteristic of distribution parameters, the action of high-pressure pump and injector are discrete, and the system has less measurable information (only common rail pressure sensors are available). In this scenario, the rail pressure control for obtaining desired injection pressure and effectively reducing the pressure fluctuation becomes one of fundamental control tasks in GDI engines.

III. Simulation model

As a reference, a dynamics model form of the fuel injection system is provided in the following. According to the structure and working principle of the common rail system, the signal flow of each element on the control forward channel is described in Fig. 3.

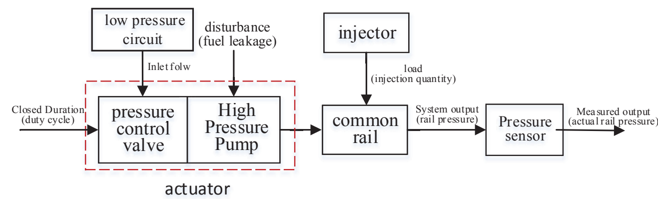

Fig. 3. Control signal flow diagram of the common rail system

In the following modelling process, the dynamic characteristics of the pressure control valve, the limiting pressure valve and the check valve are ignored. System modeling mainly considers three parts: high pressure pump, common rail and injectors.

- High Pressure Pump (HPP)

Here, ignoring the impact of temperature variation on the volume of the high-pressure pump and fuel rail pipe, and supposing less gas content in hydraulic oil. According to the fluid mass conservation principle and effective liquid elastic modulus calculation formula, the fuel flow pressure equation in the high pressure pump is shown as follows:

$$\dot{p}_p = \frac{K_f}{V_p(\theta)}\left(-\frac{dV_p(\theta)}{dt} + q_u - q_{pr} - q_0\right), \tag{1}$$

The pump pressure change is caused by fuel inflow-outflow and volume change. In the equation, K_f is the bulk modulus of elasticity associating with the pump pressure p_p, q_u is the inlet flow of the HPP. q_{pr} is flow supplied to the fuel rail, q_0 is the fuel leakage. The volume of the high pressure pump $V_p(\theta)$ is related to the camshaft angle θ and can be calculated as

$$V_p(\theta) = V_{\text{pmax}} - A_p h_p(\theta), \tag{2a}$$

$$\frac{dV_p(\theta)}{dt} = -A_p\frac{dh_p}{dt} = -A_p\omega_{\text{cam}}\frac{dh_p}{d\theta}, \tag{2b}$$

where V_{pmax} is the maximum volume of the high pressure pump chamber, A_p is the piston bore, h_p is the piston instantaneous axial displacement. ω_{cam} is the camshaft speed, and $\frac{dh_p}{d\theta}$ is a nonlinear function depending on the angle and the profile of camshaft.

Using flow calculation formula of throttle orifice, the fuel flow q_u and q_{pr} are computed by

$$q_u = \text{sgn}(p_t - p_p) c_{tp} U A_{tp} \sqrt{\frac{2|p_t - p_p|}{\rho}}, \tag{3a}$$

$$q_{pr} = \begin{cases} c_{pr} A_{pr} \sqrt{\frac{2|p_p - p_r|}{\rho}}, & p_p > p_r \\ 0 & , \; p_p \leq p_r \end{cases} \tag{3b}$$

Because a check valve (one-way valve) is equipped between high pressure pump and fuel rail, the dynamics of q_{pr} is a piecewise function. p_t and p_r are the fuel pressure in low pressure circuit and the fuel rail, A_{tp} and A_{pr} are the cross-sectional area of inlet and outlet of HPP, c_{tp} and c_{pr} are flow coefficient in inlet port and outlet port respectively, ρ is fuel density. Moreover, U is the action status of the pressure control valve. If $U = 0$, the valve is closed, while $U = 1$ indicates the valve is on opened.

- Fuel rail

As a storage component, the main effect of the rail is to absorb pressure waves and provide the desired injection pressure. The fuel rail can be considered as a fuel container with a certain volume, that means the fuel volume change is solely caused by inflow and outflow. Hence, we have

$$\dot{p_r} = \frac{K_f(p_r)}{V_r}(q_{pr} - q_{ri}), \tag{4}$$

where V_r is the liquid volume of the fuel rail, q_{ri} is the sum of injection flows, *i.e.* $q_{ri} = \sum_{k=1}^{4} q_{rik}$ with $k = 1, 2, 3, 4$ for the four injectors and q_{rik} is the injection flow of the kth injector, which can be expressed as follows

$$q_{rik} = \text{sgn}(p_r - p_{ik}) c_{rik} A_{rik} \sqrt{\frac{2|p_r - p_{ik}|}{\rho}}, \tag{5}$$

where p_{ik} is the injection pressure of the kth injector, A_{rik} and c_{rik} are the cross-sectional area and the flow coefficient of the inlet of the kth injector respectively.

- Injectors

As the actuator of the fuel injection system, the electro-injectors can guarantee fast response and high fuel injection precision. The injectors can be considered as valve driven by ECU. Because the volume change of the injector is very small and can be neglected, the pressure change depends only on the inflow and outflow here. Hence, the model of the injector is derived as follows

$$\begin{aligned} \dot{p}_{ik} = \frac{K_f(p_{ik})}{V_{ik}} \Bigg(& \text{sgn}(p_r - p_{ik}) c_{rik} A_{rik} \sqrt{\frac{2|p_r - p_{ik}|}{\rho}} \\ & - \text{sgn}(p_{ik} - p_{cylk}) E_{Tk} c_{ik} A_{ik} \sqrt{\frac{2|p_{ik} - p_{cylk}|}{\rho}} \Bigg), \end{aligned} \tag{6}$$

where p_{cylk} is the cylinder pressure of the kth chamber, A_{ik} and c_{ik} are the cross-sectional and the flow coefficient of the kth injector nozzle respectively, E_{Tk} is the square signal, which is determined by injection pulse width.

- Pressure control valve action

The rail pressure can be adjusted by controlling the closing duration of pressure control valve. However, because the injection cycle is related to engine speed, the closing duration is a variable-cycle duty ratio signal. Take the four-leaf cam as an example, the cycle of the pressure control valve is calculate by

$$T_{hpp} = \frac{60 * 2}{4 \cdot n_e}, \tag{7}$$

where n_e is engine speed(rev/min), and $n_e = \frac{60}{\pi}\omega_{\text{cam}}$. A complete injection cycle takes place in a 360^{o} camshaft angular interval and consists of four injectors starting every 90^{o}. The timing relationship between duty cycle and closed angle is shown in Fig. 4. The pressure control valve is fully opened during the suck flow phase, so the range of duty cycle is $0 \sim 0.5$. Conversion relationship between duty cycle and output signal of the valve can be found in Fig. 5.

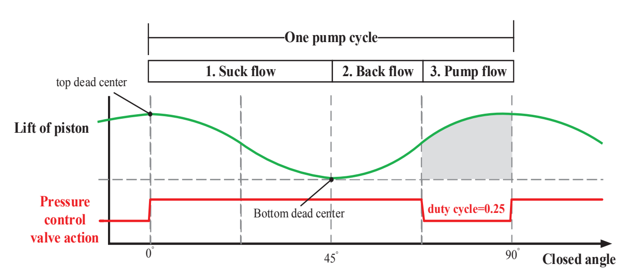

Fig. 4. timing relationship between duty cycle and closed angle

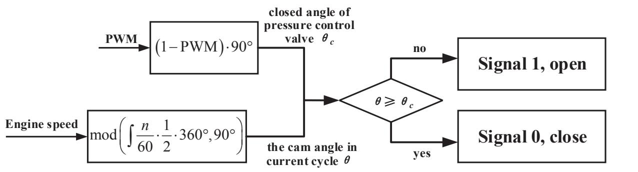

Fig. 5. Conversion relationship between duty cycle and valve output signal

- Simulation model and example

Based on the mathematical equations described above, the simulation model is established in SIMULINK as shown in Fig. 6. Setting engine speed $n_e = 3000\ rpm$ and injection pulse width is 2.2 ms, and the duty cycle of the pressure control valve is changed as shown in Fig. 7. The simulation result of common rail pressure is shown in Fig. 8. It can be seen that rail pressure rises to maximum value under small injection pulse width and large duty cycle, in contrast, the pressure in the common rail pipe gradually decreases gradually when the duty cycle is small enough, *i.e.*, the pump oil volume is less than the injection volume. It indicates that the simulation model can adequately reflect the characteristics of the rail pressure control system of the GDI engine.

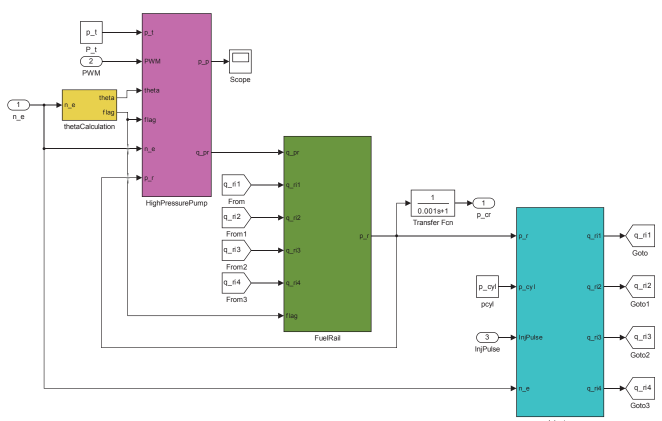

Fig. 6. SIMULINK model

IV. Performance Evaluation of Control System

The control performance will be evaluated from three aspects: adjustment time, overshoot and steady-state error, and under three working conditions:

1. Step signal tracking. Step signal tracking is usually a dynamometer mode test condition, by stabilizing tracking the

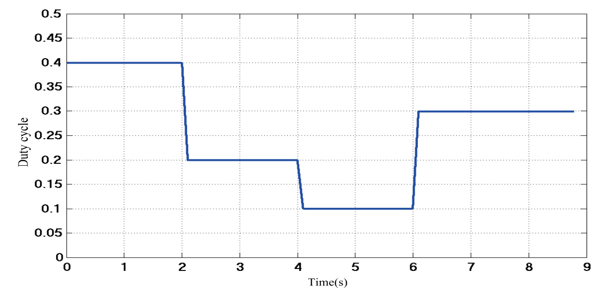

Fig. 7. Duty cycle curve

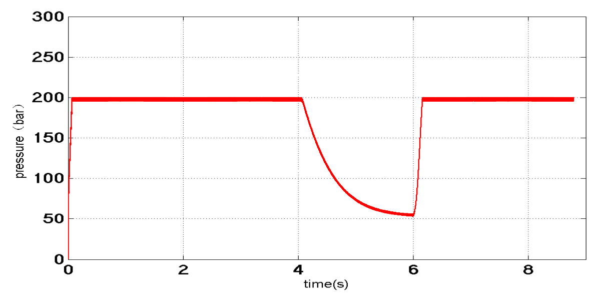

Fig. 8. Rail pressure curve

different desired pressure to evaluate the control performance. This tracking performance accounts for 60% of the total score, wherein adjustment time, overshoot and steady-state error account for 20, respectively.

(Note: (1) The minimum adjustment time should be less than 150ms, the steady-state error of the pressure tracking should be within $\pm$0.3MPa; (2) The range of the common rail pressure is 5MPa 15MPa, or even higher, in this benchmark, the maximum common rail pressure is 20MPa; (3) Practically, the change of the common rail pressure is not greater than 2MPa, that is, the step value should be within 2MPa; (4) Because of the discrete behavior of fuel injection and pump oil, there may exist fluctuation of the common rail pressure, for the convenience, the tracking error is integrated when the tracking error is stabilized, and the steady-state error will be evaluated by the deviation of the ratio of integral value to integral time and the desired value.)

2. Random time-varying tracking. Mainly test the transient tracking performance of the controller, such as sine and slope tracking, the evaluation factor is tracking error, the tracking performance under this condition accounts for 20% of the total score.

3. Conditions with time-varying disturbances. Suppose the step signal tracking condition, change the cam speed or the fuel injection volume of the fuel injector, so as to validate the robustness of the controller. This condition accounts for 20% of the total score and the specific evaluation factors are the same as step signal tracking.

V. Statement and acknowledgement

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VI. Reference

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