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Introduction to the benchmark challenge on common rail pressure control of gasoline direct injection engines

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Abstract

As one of the most important actuators for gasoline direct injection technology, common rail systems provide the requested rail pressure for fuel injection. Special system characteristics, such as coupled discrete-continuous dynamic in the common rail system, limited measurable states, and time-varying engine operating conditions, impel the combination of advanced methods to obtain the desired injection pressure. Therefore, reducing the pressure fluctuation and satisfying engineering implementation have become noteworthy issues for rail pressure control (RPC) systems. In this study, the benchmark problem and the design specification of RPC proposed by 2018 IFAC E-CoSM Committee are introduced. Moreover, a common rail system model is provided to the challengers, and a traditional PI control is applied to show the problem behaviors. Finally, intermediate results of the challengers are summarized briefly.

Keywords: Common rail pressure control, gasoline direct injection engine, control system design, benchmark challenge

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1 Introduction

Direct injection technology is considered to have great advantage in improving engine combustion performance and has been recently introduced to gasoline engines [1–4]. The highlight of this technology is that the fuel can be injected directly into the combustion chamber of each cylinder [5, 6]. Common rail systems equipped with an electromagnetic drive injector as the key actuator of this technology establish injection pressure and realize the requirements of flexible cylinder injection. The combustion system of gasoline direct in-

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jection (GDI) engines is sensitive to the quality of fuel spray, such that even a droplet can deteriorate cylinder emission performance and reduce combustion efficiency significantly [7, 8]. The precise quantity of fuel injection depends on the stable pressure in fuel rail, and the engines have the high-speed work characteristics. Thus, the establishment time of rail pressure is expected to be as short as possible. Increasing the rail pressure can improve fuel atomization. Excessive high pressure often results in rebuffed fuel. Meanwhile, insufficient rail pressure reduces the spray penetration distance, which may ruin fuel mixing and directly break the interaction with airflow. Therefore, a reliable rail pressure control (RPC) is a challenging issue for ensuring combustion stability and output efficiency [9].

Common RPC has attracted increasing attention from researchers, who have mainly focused on the aspects of characteristic analysis, modeling, and control. In [10], the dynamic behavior of a fuel system that is influenced by injection valves and the common rail pressure is analyzed, and experimental data were used to validate a zero-dimensional model. Detailed common rail system models have been established, by which the important influence of rail pressure fluctuation on system performance has been analyzed [11–14]. These models are suitable for verification of system diagnostics and control performance. However, they usually have complex mathematical descriptions and require a large amount of experimental data, which nearly obscure the advantages of model accuracy. Thus, control-oriented models and model-based control methods have been discussed. In [15, 16], a mean model is proposed using an experimental identification method, which describes the average injection pressure in a cycle as a function of engine speed and pressure control valve current. The model is also used to design an injection pressure control strategy, and the proposed control architecture is composed of a model-based feedforward controller and an integral closed-loop action [17]. In [18], the coupled discrete and continuous interactions in the fuel injection system are considered. Moreover, a hybrid common rail model is developed, and a corresponding hybrid control approach was utilized for the RPC. In [19], a model reference adaptive control algorithm based on a common rail mean value model is designed to reduce the residual pressure in the rail. In [20, 21], a rail pressure controller is derived on the basis of a backstepping technique. Moreover, a triple-step method is designed in [22] to regulate the rail pressure under disturbance conditions, and the implementation of the controller is discussed in [23]. From the studies on RPC, model-based design methods have become the dominant design. An increasing number of nonlinear methods have also been developed to obtain an improved control performance. Most of these methods, however, depend on system states. The limited sensors constraint the engineering implementation of the proposed advanced nonlinear controllers. Hence, the design of RPC system and the estimation of certain system states are still problems worth discussing.

2 System principle and proposed model

In GDI engines, many types of common rail system structure are available, and the main differences among them are the structure of the high-pressure pump (HPP) and the installation position of the rail pressure sensor. Nevertheless, these common rail systems have similar working behaviors. For example, in a four-cylinder four-stroke GDI engine, the fuel RPC system is composed of a low-pressure circuit, the HPP, fuel rail, injectors, rail pressure sensor, and engine control unit (ECU), as shown in Fig. 1.

For the injection operation, the low-pressure circuit provides fuel from the tank to the HPP. A pressure control valve is installed in the HPP to allow control the fuel amount effectively. The driving signal of the pressure control valve is a square current with a variable duty cycle and period. A check valve and a limiting pressure are installed on the outlet of the HPP to avoid unwanted refluxes and prevents the rail from being damaged by excessive pressure, respectively. When the pressure is larger than the maximum setting pressure, it opens and the fuel flows back to the tank, and the pressure in the rail pipe gradually decreases. The fuel rail is connected to the HPP with the injectors to absorb 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 the fuel rail increases when the fuel is pumped into the rail pipe and decreases when injection occurs.

The main role of the HPP is to provide high-pressure fuel for the rail. A four-lobe cam-driven HPP, which is mounted on the engine camshaft, is a common structure. The piston motion is obtained on the basis of the eccentric profile of the camshaft. In one injection cycle, the operating process of the HPP consists of three phases, namely, the suck-, back-, and pump-flow phases (Fig. 2). In the suck-flow phase, the pressure control valve remains open, and the fuel flows into the HPP from the low-pressure circuit due to the pressure difference. With the presence of the check valve, back flow does not occur at the HPP outlet. In the back-flow phase, when the pressure control valve is still open, the fuel flows back to the low-pressure circuit, and the plunger moves from the bottom to the top dead center. The pump-flow phase also occurs when the cam runs from the bottom to the top dead center while the pressure control valve remains closed. In this phase, the fuel flows from the HPP to the common rail.

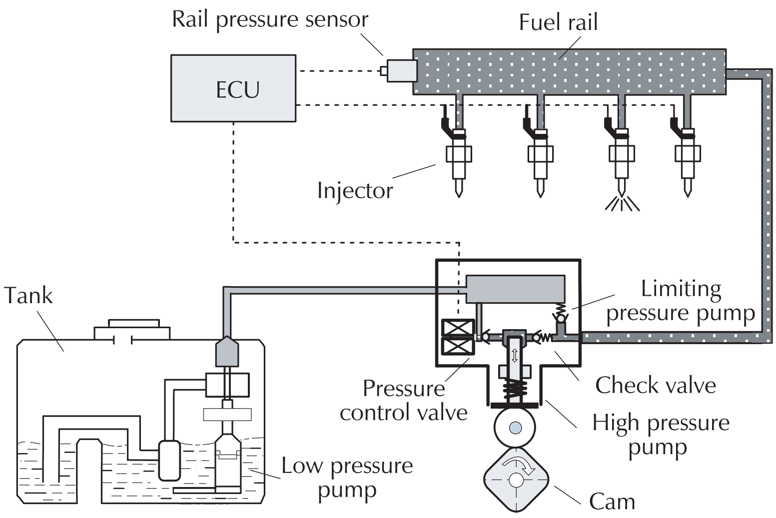

Fig. 1 The structure diagram of the common rail system in GDI engines.

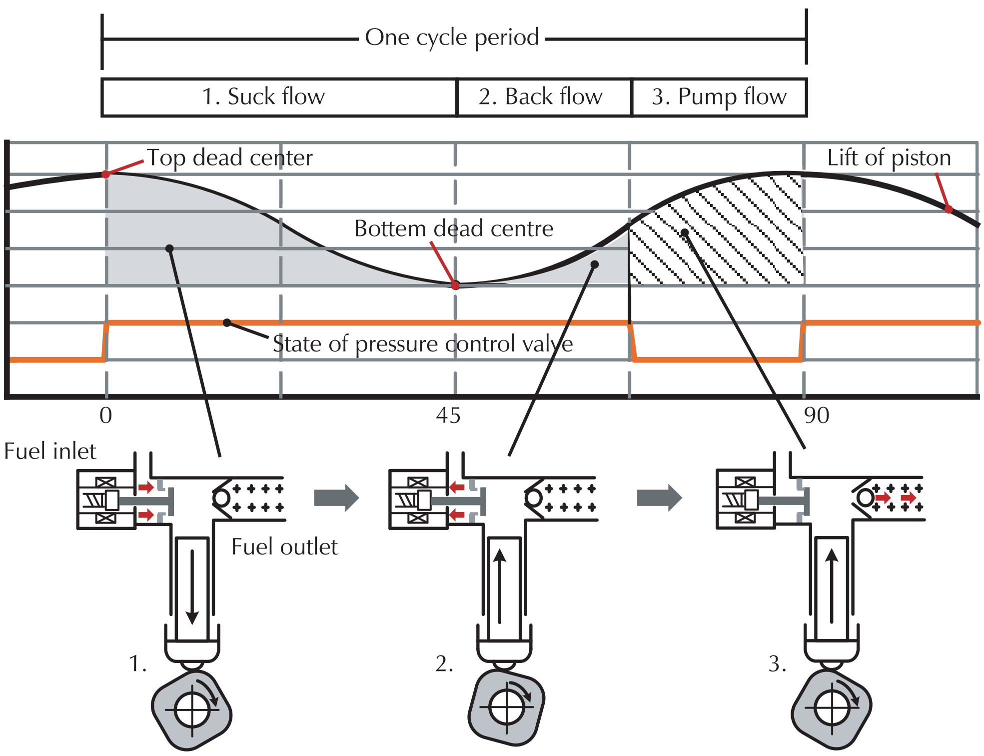

Fig. 2 The operating principle of the cam-driven high pressure pump.

Assume that the shadow area formed by the piston lift curve (Fig. 2) represents the fuel volume at each phase. Then, the fuel volume in the pump-flow phase is determined by the closing duration of the pressure control valve (expressed by duty cycle). Thus, the rail pressure can be regulated by pumping the fuel volume to the common rail pipe with a fixed volume. The pump-flow phase only occurs during the movement of the plunger from the bottom to the top dead center.

On the basis of the structure and working principle of the common rail system, the signal flow of each element on the forward control channel is shown in Fig. 3.

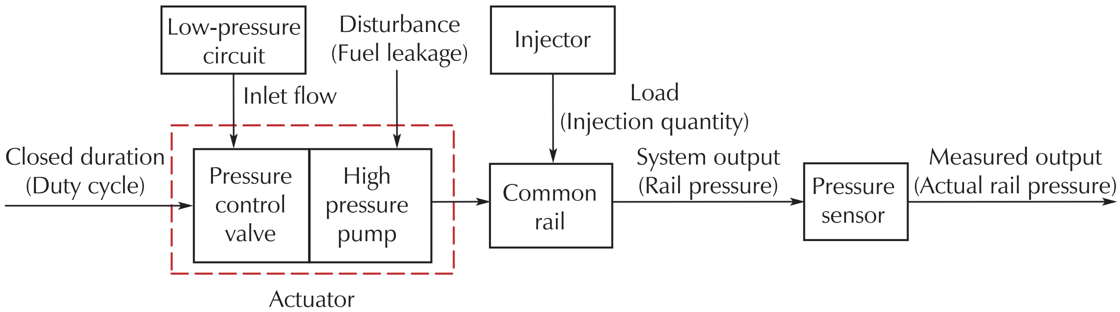

Fig. 3 Control signal flow diagram of the common rail system.

A dynamics model of the fuel injection system contains three parts [24, 25], namely, the HPP, the common rail, and injectors. In the following modeling process, the dynamic characteristics of the pressure control valve, the limiting pressure valve, and the check valve are ignored. Table 1 lists the description of the model parameters used.

Table 1 Description of model parameters.

Symbols	Description
K_f	Bulk modulus of elasticity
p_p	Pump pressure
q_u	Inlet flow of HPP
q_{pr}	Flow supplied to fuel rail
q_0	Fuel leakage
V_p	Volume of the HPP
θ	Camshaft angle
A_p	Piston bore
h_p	Instantaneous axial displacement of piston
ω_{cam}	Camshaft speed
A_{tp}	Cross-sectional area of the inlet ports of HPP
A_{pr}	Cross-sectional area of the outlet ports of HPP
c_{tp}	Flow coefficient in the inlet port
c_{pr}	Flow coefficient in the outlet port
ρ	Fuel density
p_t	Fuel pressure in low-pressure circuit
p_r	Fuel pressure in fuel rail
V_r	Liquid volume of fuel rail

A) High pressure pump

In this study, the influences of temperature variation on the volumes of the HPP and fuel rail pipe are ignored, and less gas content in hydraulic oil is assumed. On the basis of the fluid mass conservation principle and effective liquid elastic modulus calculation formula, the fuel flow pressure equation in the HPP can be expressed as

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

The pump pressure change is caused by the inflow, outflow of fuel and volume change. The volume of the HPP $V_p(\theta)$ is related to the camshaft angle θ and can be calculated as

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

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

where V_{pmax} is the maximum volume of the HPP chamber, and $\frac{dh_p}{d\theta}$ is a nonlinear function depending on the angle and profile of the camshaft.

By using the flow calculation formula of the throttle orifice, the fuel flow q_u and q_{pr} are respectively calculated as follows:

$$q_u = \mathrm{sgn}(p_t - p_p)c_{tp}UA_{tp}\sqrt{\frac{2|p_t - p_p|}{\rho}}, \tag{3}$$

$$q_{\mathrm{pr}} = \begin{cases} c_{\mathrm{pr}} A_{\mathrm{pr}} \sqrt{\dfrac{2|p_{\mathrm{p}} - p_{\mathrm{r}}|}{\rho}}, & p_{\mathrm{p}} > p_{\mathrm{r}}, \\ 0, & p_{\mathrm{p}} \leqslant p_{\mathrm{r}}. \end{cases} \tag{4}$$

The dynamics of q_{pr} is a piecewise function because the check valve (one-way valve) is equipped between the HPP and the fuel rail. U is the action status of the pressure control valve. If $U = 0$, then the valve is closed, whereas $U = 1$ indicates that the valve is opened.

B) Common 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 a fuel container with a certain volume, that is, the fuel volume change is solely caused by inflow and outflow. Hence,

$$\dot{p}_{\mathrm{r}} = \frac{K_{\mathrm{f}}(p_{\mathrm{r}})}{V_{\mathrm{r}}}(q_{\mathrm{pr}} - q_{\mathrm{ri}}), \tag{5}$$

where q_{ri} is the sum of injection flows (i.e., $q_{\mathrm{ri}} = \sum\limits_{k=1}^{4} q_{\mathrm{ri}k}$ with $k = 1, 2, 3, 4$) for the four injectors; and $q_{\mathrm{ri}k}$ is the injection flow of the kth injector, which can be expressed as follows:

$$q_{\mathrm{ri}k} = \mathrm{sgn}(p_{\mathrm{r}} - p_{\mathrm{i}k}) c_{\mathrm{ri}k} A_{\mathrm{ri}k} \sqrt{\frac{2|p_{\mathrm{r}} - p_{\mathrm{i}k}|}{\rho}}, \tag{6}$$

where $p_{\mathrm{i}k}$ is the injection pressure of the kth injector; and $A_{\mathrm{ri}k}$ and $c_{\mathrm{ri}k}$ are the cross-sectional area and the flow coefficient of the kth injector inlet, respectively.

C) Injectors

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

$$\begin{aligned} \dot{p}_{\mathrm{i}k} =& \frac{K_{\mathrm{f}}(p_{\mathrm{i}k})}{V_{\mathrm{i}k}}(\mathrm{sgn}(p_{\mathrm{r}} - p_{\mathrm{i}k}) c_{\mathrm{ri}k} A_{\mathrm{ri}k} \sqrt{\frac{2|p_{\mathrm{r}} - p_{\mathrm{i}k}|}{\rho}} \\ &- \mathrm{sgn}(p_{\mathrm{i}k} - p_{\mathrm{cyl}k}) E_{\mathrm{T}k} c_{\mathrm{i}k} A_{\mathrm{i}k} \sqrt{\frac{2|p_{\mathrm{i}k} - p_{\mathrm{cyl}k}|}{\rho}}), \end{aligned} \tag{7}$$

where $p_{\mathrm{cyl}k}$ is the cylinder pressure of the kth chamber; $A_{\mathrm{i}k}$ and $c_{\mathrm{i}k}$ are the cross-sectional area and the flow coefficient of the kth injector nozzle, respectively; and $E_{\mathrm{T}k}$ is the square signal, which is determined by the injection pulse width.

D) Pressure control valve action

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

$$T_{\mathrm{hpp}} = \frac{60 \times 2}{4 n_{\mathrm{e}}}, \tag{8}$$

where n_{e} is the engine speed (rev/min), and $n_{\mathrm{e}} = \dfrac{60}{\pi}\omega_{\mathrm{cam}}$. A complete injection cycle takes place in a 360° camshaft angular interval and consists of four injectors that start every 90°. The timing relationship between the duty cycle and the closed angle is shown in Fig. 2. The pressure control valve is fully opened during the suck-flow phase; thus, the range of duty cycle is 0–0.5.

E) Model verification

On the basis of preceding equations, the simulation model is established in SIMULINK, as shown in Fig. 4. The setting engine speed is $n_{\mathrm{e}} = 3000$ r/min, the injection pulse width is 2.2 ms, and the duty cycle of the pressure control valve is set as 0.4. The simulation result of the common rail pressure is shown in Fig. 5. From the figure, the pressure in the common rail pipe decreases gradually, that is, the pump oil volume is less than the injection volume. The fuel volume in the HPP chamber is decreased on the basis of the duty cycle, thereby decreasing the pressure in the HPP. Thus, the simulation model can adequately reflect the main characteristics of the common rail system.

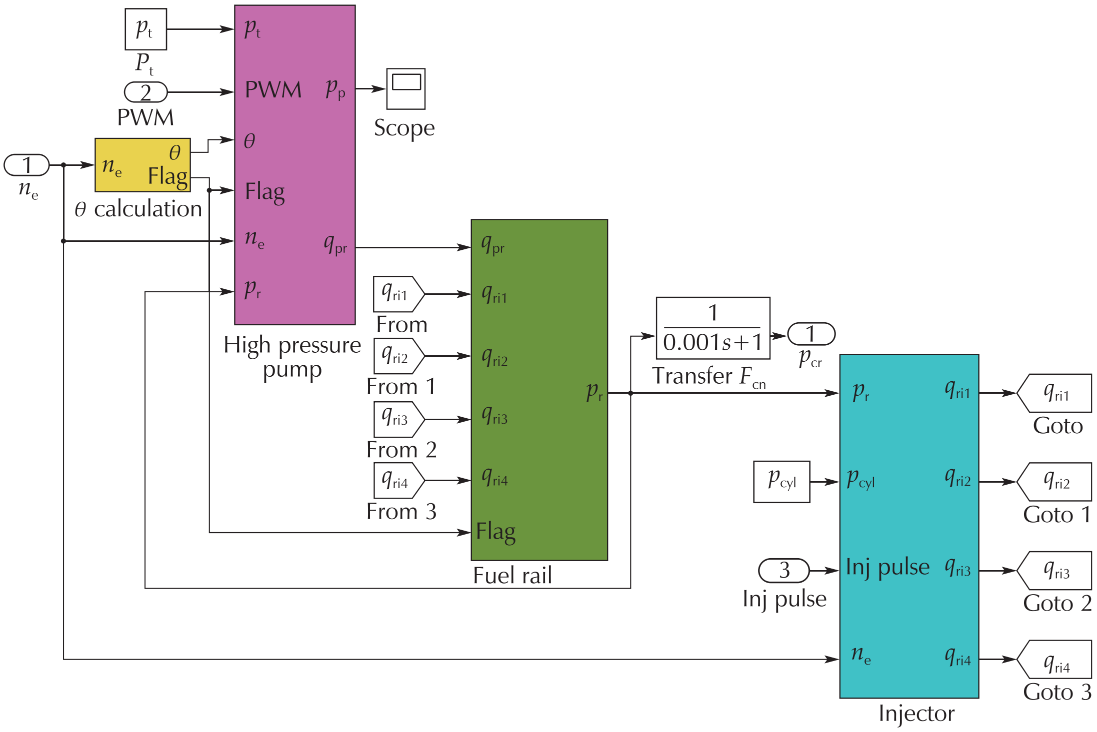

Fig. 4 SIMULINK model.

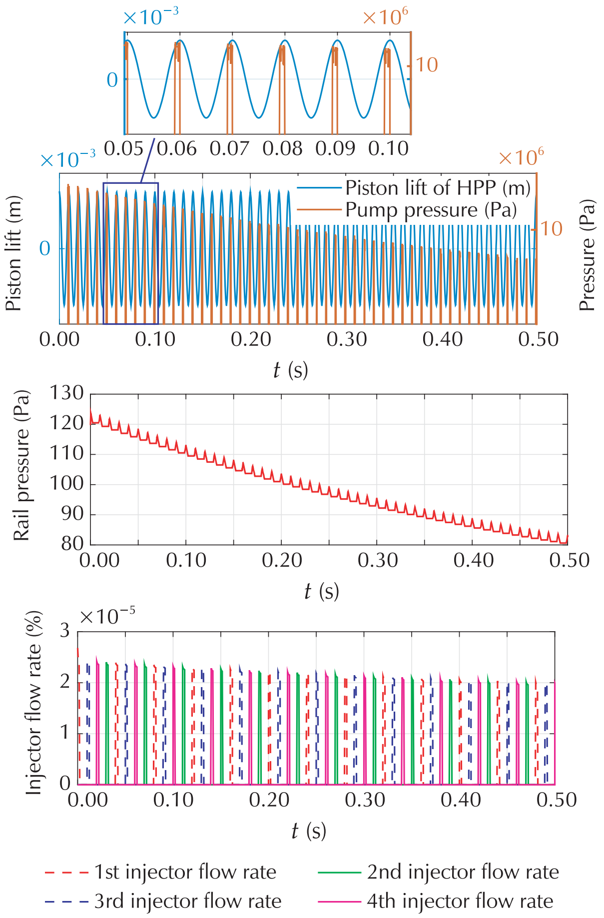

Fig. 5 Model verification curves.

3 Benchmark problem

The benchmark problem is to start the RPC model and regulate the rail pressure at the desired pressure immediately.

The requirements for the steady state are as follows:

a) the pressure regulation time should be less than 150 ms.

b) the steady-state error of the pressure tracking should be within ±0.3 MPa.

c) the range of the common rail pressure is 5–20 MPa, and the controller test results should cover the range.

d) the closed loop is stable.

The requirements for the transient condition are as follows:

e) the overshoot must be sufficiently suppressed.

f) the tracking error of the pressure tracking should be within ±0.3 MPa.

g) the control variable must not be a large oscillation.

A well-turned PI controller is used in the case of Fig. 6. It fulfils nearly all the requirements; however, the performance for requirement g) should be improved. Evaluation results are also shown in Fig. 7, where the engine speed and the fuel injection volume can be changed, and only requirements d) and e) are fulfilled. Thus, the requirement for the steady and the transient conditions must be fulfilled with variations of the engine speed and the fuel injection volume.

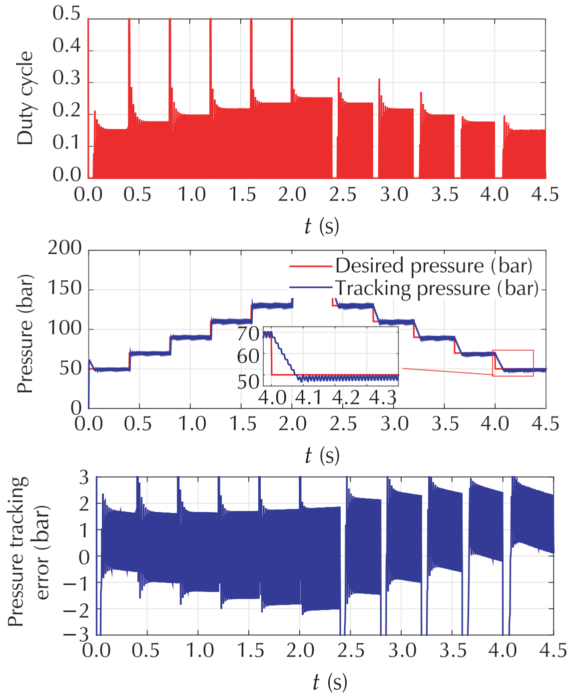

Fig. 6 A fixed PI controller for rail pressure system.

The model has the state transition between "pump" and "no pump", and between "injection" and "no injection" (Fig. 5). Thus, stabilizing the rail pressure is difficult. The entire control process has discrete behaviors, and these characteristics cannot be ignored. Hence, suppressing the overshoot is also difficult. Moreover, we have to relay on the feedforward for the purpose. Other important features are that the control must be designed as discrete time systems. They are actuated every engine stroke and it is related to engine operating conditions, as proven by equations (2) and (7). In addition, the system has limited measurable information (that is, only common rail pressure sensors are available). Thus, this problem is considered complex, which indicates that the combination of methods is required.

Overall, 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 (expressed as duty cycle) of the pressure control valve. In the system, the rail pressure is the only state measured directly. Additional system states must be observed when the controller design based on states is used.

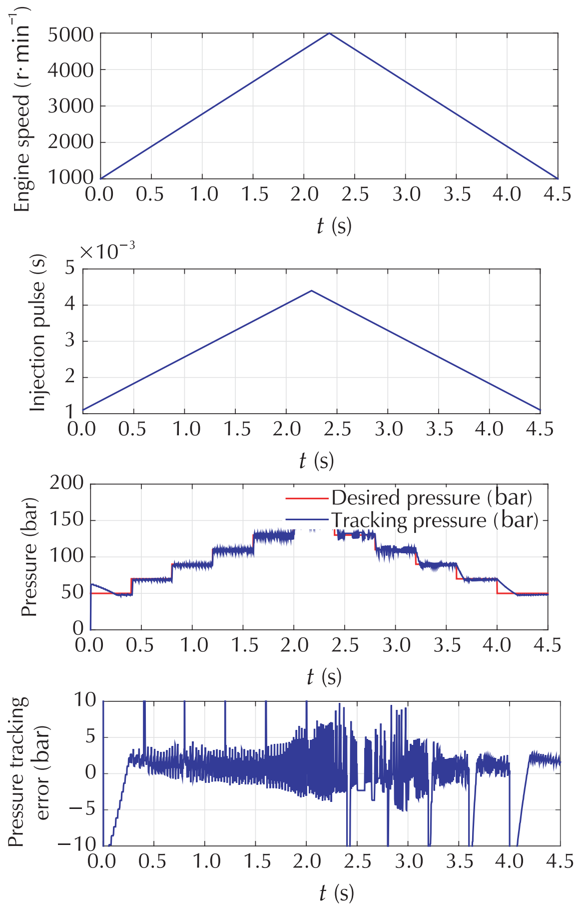

Fig. 7 A fixed PI controller for rail pressure system with varying conditions.

4 Approaches and results of the benchmark

Four challengers reported their intermediate results in the 5th IFAC Conference on Engine and Powertrain Control, Simulation, and Modeling at Changchun, China on September 20–22, 2018. Among them, three succeeded in adjusting the rail pressure, although some levels of error robustness problem were still observed. The challengers'works can be summarized as follows.

Challenger A analyzed the rail pressure characteristics of a common rail system. They reported that the system has discrete event features and strong nonlinearity. To address this problem, they introduced a method that relies less on model accuracy and proposed a linear active disturbance rejection controller. To simplify the parameter tuning process, Challenger A used a genetic algorithm to optimize the controller and they focused on improving the controller performance further.

Challenger B proposed a rail pressure controller based on predictive functional control (PFC) to handle the nonlinearity and discontinuity caused by the common rail pressure system (CRPS). A control-oriented piecewise linear model was presented to simplify the CRPS. The simulation results on a benchmark showed that the rail pressure tracked the setpoint accurately even with some perturbations. Given the conciseness of the PFC algorithm, the controller could calculate the online solution in a short period of time, thereby realizing the strategy on a fast response system.

Challenger C provided a reasonable simplified model and proposed a terminal sliding mode control strategy to design the rail pressure controller with Lyapunov stability. Simulations were performed to test the validity of the designed controller by tracking different reference rail pressures.

Challenger D established the state space equation of the common rail system firstly, then, the linearization method was used for dynamics equations based on the rail pressure setpoint, and the LQT algorithm was applied for tracking control.

5 Conclusions

The program committee of 2018 IFAC E-CoSM and the Technical Committee on Vehicle Control and Intelligence provided a common rail system model and the benchmark problem of RPC. Four challengers reported their intermediate results on September 20–22, 2018 at Changchun, China. We appreciate the efforts of and enthusiasm of the challengers. We feel that this program is very useful for academic parties and the industry, and we are not only concerned about the control results but also the control design process. Although various approaches have been used by researchers to handle RPC, control input oscillation and robustness remain a problem. We expect further works on the state observation of common rail systems.

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