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Research Article

High pressure common rail injection system modeling and control

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ABSTRACT

In this paper modeling and common-rail pressure control of high pressure common rail injection system (HPCRIS) is presented. The proposed mathematical model of high pressure common rail injection system which contains three sub-systems: high pressure pump sub-model, common rail sub-model and injector sub-model is a relative complicated nonlinear system. The mathematical model is validated by the software Matlab and a virtual detailed simulation environment. For the considered HPCRIS, an effective model free controller which is called Extended State Observer – based intelligent Proportional Integral (ESO-based iPI) controller is designed. And this proposed method is composed mainly of the referred ESO observer, and a time delay estimation based iPI controller. Finally, to demonstrate the performances of the proposed controller, the proposed ESO-based iPI controller is compared with a conventional PID controller and ADRC.

1. Introduction

In recent years, with the increase of diesel engine cars, the pollution caused by diesel engine has become a serious issue and stringent emission legislation has been issued. It is necessary to improve diesel engine's performance to meet the emission legislation's demands. High pressure common rail injection system (HPCRIS) plays a vital role in reducing the emission of the diesel engine [1].

The modeling of HPCRIS provides a powerful tool to evaluate the suitability of the high pressure common rail system for diesel engine applications. Many high pressure common rail system models have been previously proposed, based on the equations of the physics underlying the process or alternatively developed through simulation packages like Matlab/Simulink, AMESim, Hydsim etc.

In high pressure common rail system, common-rail pressure not only determines the injection-fuel pressure, but also is the main measurement parameter of fuel-injection. The stabilization and excessive response of common-rail pressure will directly affect starting, idling and acceleration mode of diesel engine's dynamic performance. So common-rail pressure control is a key to improve diesel engine's performance. The main objective of the rail pressure control is to achieve stable and high fuel pressure in the common rail.

Lino et al. [2] provided a model of common rail injection system for diesel engine and designed a sliding mode control for common rail pressure control by using this model to track the reference pressure and reject disturbances. To overcome the shortage of traditional PID controller, many self-tuning PID controllers have been proposed, such as an adaptive fuzzy sliding mode control which based on sliding mode variable structure control and fuzzy logic [3], genetic algorithm nonlinear PID controller [4] and RBF neural network adaptive PID controller [5].

Hong et al. [6] proposed a coordinated control strategy for the common rail pressure using a metering unit and a pressure control valve to assure the tracking performance of the rail pressure as well as to handle the discontinuous fuel flows by the high pressure pump and the fuel injectors. Chatlatanagulchai et al. [7] designed a QFT-based controller to control common rail pressure, the controller performs well in tacking, stability and disturbance rejection specification. A controller based on active disturbance rejection control for common-rail pressure control is proposed in [8], the proposed controller tracks the reference pressure well in spite of the system uncertainties and appearance of the external disturbance.

HPCRIS is a complicated system including nonlinear, uncertainties and external disturbance such as measurement noise. In this paper, an ESO-based iPI controller based on model-free control is designed to overcome the un-modeled dynamics and uncertainties of the system [9]. Model-free control is based on

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new results on fast derivatives estimation of noisy signals, its main advantages are: its simplicity and robustness [10].

The iPI controller uses the ultra-local model which is continuously updated according to input and output to modelize the nonlinear complex HPCRIS. An extended state observer (ESO) [11,12] and time-delay estimation method [13] which replaced the algebraic parameter identification [14] are used to estimate F to replace the method which is obtained via the knowledge of the control and output variables u and y.

The organization of this paper is as follows. A detailed nonlinear model of HPCRIS is presented in next section. An overview of ESO-based iPI controller is described in Section 3. Section 4 gives the simulation results to validate the model and to prove the performance of ESO-based iPI controller. The ESO-based iPI controller is implemented in both Matlab/Simulink and AEMSim environment compared with classic PID controller and ADRC. In Section 5, the conclusion and future work are summarized.

2. Modeling of high pressure common rail injection system (HPCRIS)

In this section, the working principle of the HPCRIS is firstly presented; then based on the continuity equation, the momentum equation and Newton's motion law, the model of HPCRIS which contains three sub-models is built.

2.1. Description and modeling of HPCRIS

The major function of HPCRIS is to deliver high pressure fuel based on the working condition of engine. Its corresponding architecture is shown in Fig. 1. The main elements of the system are a high pressure pump, a common rail, the injectors and the electronic control unit (ECU). The tank supplies the low pressure fuel to the high pressure pump, where the pressure of the fuel is raised to the desired level. The high pressure pump sends high pressure fuel to the common rail, the common rail distributes the fuel to each injector and the injectors inject high pressure fuel to the cylinder.

ECU is the core of HPCRIS, it determines the pressure of fuel based on the work condition of the engine by the acquisition of various vehicle sensor signals.

High pressure common rail injection system has a complex injection process, there are many factors which affect the performance of fuel injection, such as fuel leakage between various components, the elastic deformation of high pressure fuel pipe, fuel compressibility under high temperature and high pressure and flow rate loss when the fuel flows through variable cross-section etc.

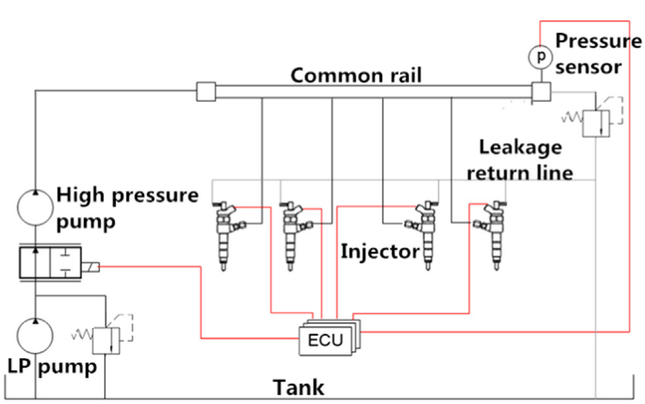

Fig. 1. High pressure common rail system.

Based on the system characteristics, some assumptions are given: the fuel temperature variation is not considered during operations to make sure that the system state can be represented by the pressure; the fluid dynamic phenomena connected to the flows through the pipes is neglected; the volume of low pressure pump and cylinders is set as infinite and the pressure is set as constant; the electro hydraulic valve is considered as a variable section [15].

In the model, the bulk modulus of elasticity E which expresses the compressibility of the fuel:

$$E = -\frac{dP}{dV/V} = \frac{dP}{d\rho/\rho} \tag{1}$$

where the increase of dP causes the volume decrease of a unit volume of liquid $-dV$. dV/V is dimensionless.

From (1), the time derivative of the fuel pressure can be obtained [16]:

$$\frac{dP}{dt} = -\frac{E}{V}\cdot\frac{dV}{dt} \tag{2}$$

where V is the instantaneous volume of the chamber, dV/dt expresses the volume changes caused by mechanical parts piston, the intake and the outtake flows. Considering that the factors affect the volume change of the fuel, (2) can be rewritten as:

$$\frac{dP}{dt} = -\frac{E}{V}\cdot\left(\frac{dV_0}{dt} - Q_{in} + Q_{out}\right) \tag{3}$$

which is the basic pressure dynamics equation in each control volume. In (3), Q_{in} is the intake flow and Q_{out} is outtake flow, dV_0/dt is the volume change rate caused by the mechanical piston, which will be specified for the high pressure pump, the common rail pipe and the injector, respectively. Except the high pressure pump, all the elements in HPCRIS have a constant volume.

Based on the energy conservation law, intake and outtake flows Q_{in} and Q_{out} can be expressed as

$$Q = sign(\Delta P)\cdot\mu\cdot S_0\cdot\sqrt{\frac{2|\Delta P|}{\rho}} \tag{4}$$

where $sign(\Delta P)$ is the sign function affecting the flow direction, μ is the discharge coefficient, S_0 is the orifice section, ρ is the dual density and ΔP is the fuel pressure difference across the orifice.

2.2. High pressure pump subsystem

The high pressure pump consists of three identical hydraulic rams mounted on the same shaft with a relative phase of 120°. Since the pump is powered by the camshaft, its evolution depends on the engine speed. It is connected by a small orifice to the low pressure circuit and by a delivery valve with a conical seat to the high pressure circuit.

According to (3), the pressure dynamics in the high pressure pump can be expressed as:

$$\frac{dP_p}{dt} = -\frac{E}{V_p}\cdot\left(\frac{dV_p}{dt} - Q_p + Q_{pcr} + Q_{pl}\right) \tag{5}$$

where Q_p is the inlet fuel flow, Q_{pcr} is the outlet fuel flow to the common rail, Q_{pl} is the fuel leakage flow and P_p is the fuel pressure of high pressure pump. The volume of the high pressure pump V_p changes according to the camshaft motion expressed by

$$\frac{dV_p}{dt} = -S_p\omega_{rpm}\frac{dh_p}{d\theta} \tag{6}$$

where S_p is sectional area of plunger, $S_p = \pi {d_p}^2/4$, d_p is the diameter of plunger, h_p is the plunger lift, ω_{rpm} is the camshaft rotational speed, and θ is the camshaft angle.