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Research on the Closed-Loop Control of Common Rail Pressure by Simulated Calculation based on Mathematical Algorithm

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Keywords ：amesim, closed-loop, rail pressure control, simulink

ABSTRACT

The control of common rail pressure influences the power, economy, and exhausted emission of the engine in the High Pressure Common Rail Injection System (HPCRIS), in which the fluctuation amplitude of steady state and the transient tracking speed are important indexes of controller's performance. To improve the performance of the controller, a HPCRIS control module is firstly developed based on the physical model, including rail pressure control state machine, demand flow calculation module and fuel metering control unit. Secondly, control strategy is validated by co-simulation between the HPCRIS control module built in the Matlab/Simulink environment and controlled object of the HPCRIS made in the AMESim environment. Finally, by analyzing the simulation results, the performance of HPCRIS control module is verified, which provides a basis for the transplantation of new HPCRIS.

INTRODUCTION

Nowadays, with the implementation of strict emission regulations, higher requirements for the performance of the electronic control system are put forward. The High Pressure Common Rail Injection System (HPCRIS) is the core component of the modern diesel engine, and has been active in the aspect of energy saving and emission reduction. The steady state performance is an important index of rail pressure control, which determines the precision of fuel injection control and affects various performance indicators such as the power, economy and exhaust emission. Thus, reducing the fluctuation of steady state railpressure and improving dynamic tracking of rail pressure have always been the focus and the hot point of the research.

Actually, controlling the rail pressure precisely is quite difficult because the inlet and outlet fuel flow of the common rail is discontinuous due to the discrete action of the high pressure pump (HPP) and injectors. To deal with the problem, a lot of work has been done to develop the commonrail system model. Mohamed H. Siliman et al. (Mohamed et al., 2017) and Balluchi et al. (Balluchi et al., 2006) have proposed the corresponding control model according to thephysical structure and the operating characteristics of HPCRIS. Also, Lino et al. (Linoet al., 2007) have designed a sliding mode controller based on physical model of HPCRIS to achieve the rail pressure tracking and reject disturbance. In addition, there are also previous studies focusing on the approach of PID or PID-plus (Wang et al., 2016) in order to overcome problems in the machine structure of HPCRIS and optimize the tracking and anti-disturbance of rail pressure. However, those proposed models do not sufficiently tune the Metering Unit (MeUn) to reduce the return fuel by the Pressure Release Valve (PRV). In addition, although the control rail pressure model using PID or PID-Plus control algorithm is easy to establish and develop, the tuning of the proportional, integral and differential coefficients in PID algorithm requires a lot of time and test resources, and the PID coefficient needs to be re-adjusted in the application of new models. Therefore, the control algorithm with PID or PID-Plus characteristics makes the portability and applicability of the control strategy worse.

A general pressure control architecture is proposed based on the physical model of the HPCRIS to reduce the transplanted problem of the controller on new HPCRIS, to decrease the calibrated workload and to possibly cut down the return fuel volume of the PRV and the energy cost of the HPP by using the MeUn alone. Its control performance is validated by co-simulation between the control models made in the Matlab/Simulink environment and the controlled models of the common rail system built in the AMESim (Advanced Modeling Environment for performing Simulation of engineering systems) environment. The proposed architecture can satisfy the

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demand of steady- state and dynamic rail pressure tracking totally.

This paper is organized by the following parts. Detail design of the rail pressure controller and controlled model of HPCRIS are presented in section 2. Then, section 3 shows the simulation results and the analysis of the results, which is used to verify the accuracy and applicability of the model. In the end, the contents and meanings are concluded in section 4.

METHOD DESCRIPTION

The proposed method is to deduce the relationship between derailing pressure and fuel flow according to relevant theoretical formulas, and then establish a general pressure control architecture. Finally, the proposed strategy is verified through the joint simulation of the control strategy model in MATLAB and the physical model in AMESim.

Derivation of Rail Pressure Control Equation

As shown in Fig. 1, HPCRIS has a complex structure, including high pressure pump, common rail, injector and electronic control unit (ECU) (Wintrich et al., 2017). Fuel in the tank is compressed to 4-6 bar through the low-pressure fuel pump and carried to high-pressure fuel pump. Under the control of the MeUn, fuel is carried to ram pump as required to control the pressure of common rail. Therefore, it is important to derive the relationship between rail pressure and MeUn flow for rail pressure control.

According to the relation between volume (V), density (ρ), and mass (m), if fuel density is changed and the volume of container remains constant, the fuel mass in the common rail will be changed as Equation (1).

$$dm = V d\rho \tag{1}$$

If fuel is regarded as compressed liquid, then the function of the relation between pressure and density is given by reference (Streeter et al., 1975).

$$k_f = \frac{dp}{d\rho/\rho} \tag{2}$$

where ρ is the fuel density and p is the rail pressure. k_f related to the function of pressure is the bulk modulus of elasticity，it can be given by reference (Hountalas and Kouremenos, 1998).

$$k_f = 1.2 \cdot 10^4 \cdot (1 + 0.6 \frac{p}{600}) \tag{3}$$

After differentiation through time t, Equation (2) can be rewritten as Equation (4).

$$\frac{dp}{dt} = \frac{K_f}{\rho} \cdot \frac{d\rho}{dt} = \frac{K_f}{\rho V} \cdot \frac{dm}{dt} \tag{4}$$

where dm/dt is the variant fuel quantity in high pressure section (high pressure section (HPS) means that the pressure is equal to all the volume of rail pressure), while V is the volume of all the fuel stored in high pressure section.

In the high pressure section, dm/dt should be equal to the difference of fuel flow rate from the HPP and injection mass from the injector. It can be expressed as Equation (5).

$$\frac{dm}{dt} = q_{m\_i} - q_{m\_o} \tag{5}$$

where q_{m_i} is the mass flow rate from the HPP, and q_{m_o} is the mass flow rate from the injector which includes fuel lost from injector and fuel discharged from PRV. Since the optimization of control strategy focuses on the control of MeUn, the former factor is primarily considered. Normally, a map of q_{m_i} value is related to both duty ratio of MeUn (d_r) and HPP speed (n), and can be acquired by a calibration test. Order $q_{m\_i} = f(d_r, n, t)$ .The q_{m_o} is the sum of injected fuel quantity in each cylinder and returned fuel from an injector, which is a nonlinear function involving the pressure and time. Let $q_{m\_o} = g(p, t)$ and the Equation (4) can be rewritten as Equation (6).

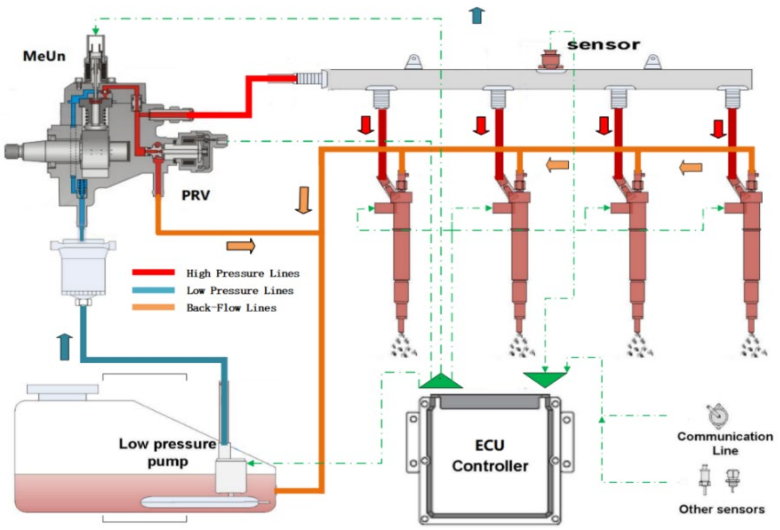

Figure 1. Common Rail Fuel Injection System

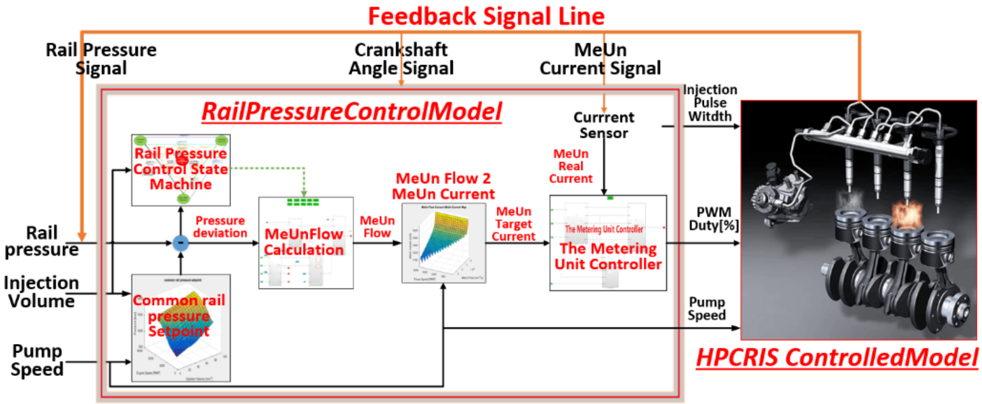

Figure 2. Rail Pressure Control Structure

$$\frac{dp}{dt} = \frac{K_f}{\rho}\cdot\frac{dp}{dt} = \frac{K_f}{\rho V}\cdot\left[f\left(d_r, n, t-\frac{\tau}{\omega}\right) - \delta(t)\cdot g(p,t)\right] \quad (6)$$

where t denotes the time, $\delta(t) = \begin{cases} 0, injector\ is\ closed \\ 1, injector\ is\ openned \end{cases}$ and ω is angular speed of HPP. The τ is the lag angle of MeUn control and d_r is duty ratio of MeUn. As an important control unit of rail pressure, MeUn's physical lag characteristic must be considered. Since the fuel absorption and compression process of the plunger account for 180 degrees each, lag angle of MeUn control is equal to 180 degrees, which means $\tau = 180^{\circ}$. The Equation (6) can be rewritten as Equation (7).

$$\int\frac{dp}{dt} = \frac{K_f}{\rho V}\cdot\int\left[f\left(d_r, n, t-\frac{\tau}{\omega}\right) - \delta(t)\cdot g(p,t)\right]dt \quad (7)$$

Based on the conservation of mass law, the lost fuel quantity of common rail in one engine work cycle should be equal to the sum of injection fuel quantity.

$$M_{m\_o} = \int \delta(t)\cdot g(p,t)\, d \quad (8)$$

According to the physical structure of the HPP, the range of integration $t \in \left[t_c - \frac{\tau}{\omega}, t_c\right]$ is selected to compute M_{m_o}, where t_c is the current moment. After integrated, Equation (7) can be reduced to Equation (9).

$$P_s - P_r = \frac{K_{fc}}{\rho_c V}\cdot\left[\int_{t_c-\frac{\tau}{\omega}}^{t_c} f\left(d_r, n, t-\frac{\tau}{\omega}\right)dt - \frac{M_{m\_o}}{2}\right] \quad (9)$$

where P_s is the target rail pressure, P_r is the actual rail pressure, K_{f_c} and ρ_c are the bulk modulus of elasticity and density under current rail pressure, respectively. In order to satisfy the demand of digital calculation, Equation (9) must be disposed by discretization. One can assume that the period of control is T, then discrete Equation (9) will be presented as Equation (10).

$$P_s - P_r = \frac{K_{fc}}{\rho_c V}\cdot\left(\sum_{i=0,\ldots}^{\varphi}\left[f(d_r, n, t_c - i\cdot T)\right]\cdot T - \frac{M_{m\_o}}{2}\right) \quad (10)$$

where $\varphi = \frac{\tau}{\omega\cdot T}$. By reorganizing Equation (10), the flow rate of the rail pressure deviation can be given as Equation (11).

$$f_{err}(d_r, n, t_c) = (P_s - P_r)\cdot\frac{\rho_c V}{T\cdot K_{f_c}} + \frac{M_{m\_o}}{2\cdot T} - \sum_{i=1,\ldots}^{\varphi}\left[f(d_r, n, t_c - i\cdot T)\right] \quad (11)$$

where $f_{err}(d_r, n, t_c)$ is the flow rate of rail pressure deviation. As injection fuel is not continuous, disturbance must be handled by mean way for the MeUn. The flow rate can be given as Equation (12).

$$f_{Inj}(d_r, n, t_c) = \frac{n\cdot M_{m\_o}}{60} \quad (12)$$

where $f_{Inj}(d_r, n, t_c)$ is the flow rate of injection fuel and n is the speed of HPP. Thus, the demand calculation flow rate of MeUn can be gained as

$$f(d_r, n, t) = f_{err}(d_r, n, t_c) + f_{Inj}(d_r, n, t_c) \quad (13)$$

Modelling of Rail Pressure Controller

According to the relationship between rail pressure and MeUn, the rail pressure control model includes two modules: rail pressure control state machine and MeUn demand flow calculation. The architecture and principle of controller are shown in Fig. 2(Hong et al., 2012). First, according to the track pressure control state machine, the current state of the control system is determined and the corresponding value is output. Then MeUn demand flow calculation module selects corresponding algorithm in terms of the output value of rail pressure control state. Finally, the MeUn demand flow is converted into MeUn's opening current according to the Map table.

Rail Pressure State Machine Module

According to the engine operating state and the rail pressure deviation, optimal control mode is selected by the rail pressure state machine.

Under the normal condition – no mechanical and electrical problems, three control modes can be outputted by rail pressure state machine, which includes opened-loop control mode, closed-loop control mode, and Pre-closed-loop control mode.

Control mode of rail pressure state machine is judged by engine speed, real rail pressure, rail

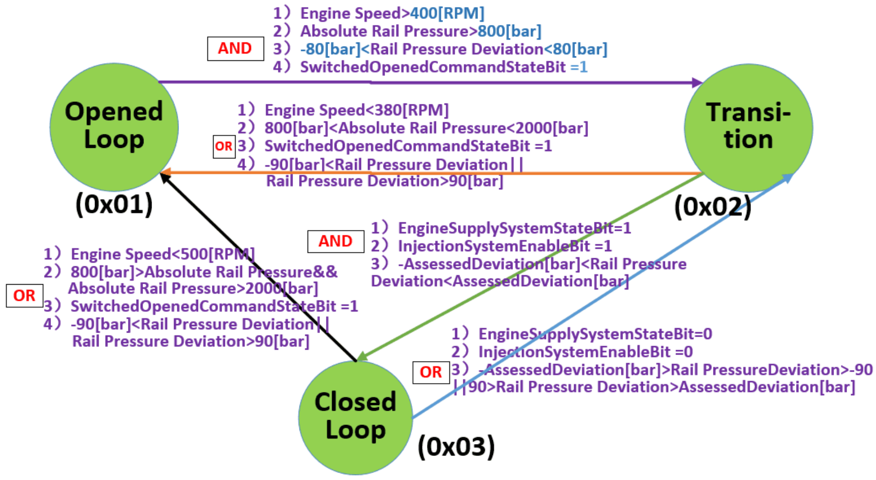

Figure 3. Conversion Machine of Rail Pressure Control Mode

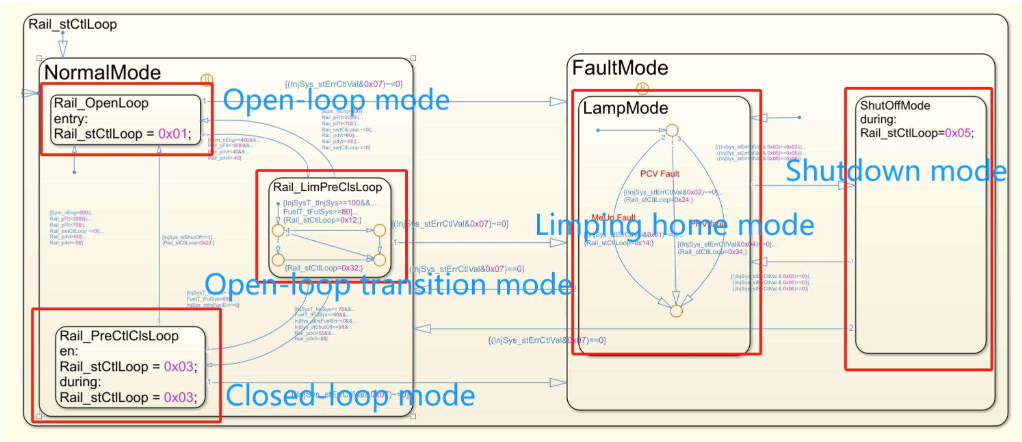

Figure 4. Modelling of Rail Pressure Control State Machine

pressure deviation, engine fuel supply system mark and injection enable mark, etc. The conversion relationship and Simulink module of the three control modes is shown in Fig. 3 and Fig. 4, respectively. These values of the rail pressure control mode will be computed during the different operating condition and they are demonstrated on the Tab. 1.

Table 1. Output results and meanings under different control modes

Control Mode	Output Value
OpenedLoop Mode	0x01
Transitional Mode	0x02
ClosedLoop Mode	0x03

MeUn Flow-Rate Calculation

In the opened-loop mode, the output value of the rail pressure state machine is 0x01, which means that MeUn demand flow is confirmed by the rail pressure deviation $(P_s - P_r)$. If the rail pressure deviation is positive, the high pressure pump will be set to the maximum fuel supply under current speed, otherwise it will be zero. Its Simulink module is shown in Fig. 5.

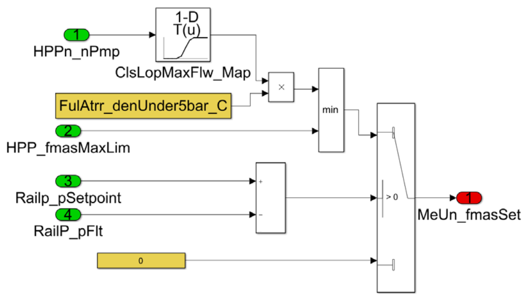

Figure 5. MeUn Flow Calculation Under Opened-loop Mode

In the transitional mode, the output value of the rail pressure state machine is 0x02. The MeUn demand flow is calculated on the basis of rail pressure deviation $(P_s - P_r)$ and deviated demand flow $(f_{err}(d_r, n, t_c))$ value, and the relationship between them is presented in Tab. 2. According to the logical relationship shown in Tab.2, MeUn demand flow calculation under the transitional mode can be structured as shown in Fig. 6.

In the closed-loop control mode, the rail pressure state machine outputs 0x03, and rail pressure control is in steady-state tracking operation. The difficulties of rail pressure control under this condition lie in the problems of rail pressure sensor filtering and control lag of high-pressure oil pump. The sliding average filter is used to eliminate the influence of fuel injection process on rail pressure in steady state. In order to eliminate the control lag of high pressure oil pump (when the crankshaft angle is 180 degrees, the oil absorption and compression process causes a delay in entering the common rail tube), MeUn flow calculation formula (13) is used to solve the problem of control lag in terms of fully considering the fuel quality stored in the pressure plunger. Fig. 7 is a model expression of formula (13).

Controlled Model of HPCRIS

According to the physical structure of HPCRIS in Fig. 1, hydraulic circuit model of HPCRIS is built by using fluid library, electric library, and signal library of AMESim software as shown in Fig. 8. Its structure is composed of one low pressure supply fuel pump, one MeUn common rail, four injectors, one controller, and three high pressure ram pumps with a 120-degree angle with each other, etc. In addition, the speed ratio of engine to pump is 2:1.

PWM drive circuit is connected to MeUn solenoid and receives the duty ratio signal from the output PWM value of Simulink. After measured by the current sensor, the current in the MeUn solenoid is fed into control model in the Simulink environment as input parameter for controller. The common rail tube is provided with a PCV valve and a pressure sensor. The PCV valve is a mechanical pressure relief valve used to prevent rail pressure from being excessively high and the rail pressure information is collected by the pressure sensor. The control interface of four injectors are connected to the controller, and the controller provides the injection pulse width to control the fuel injection quantity. The controller in Fig. 8 is the interface of the control model built by Simulink, responsible for sensor information collection and control signal output. The meaning of the port and the direction of data transmission are shown in Tab. 3 below.

Table 2. The logical relationship between $P_s - P_r$, $f_{err}(d_r, n, t_c)$ and MeUn demand flow under the transitional mode

Rail pressure deviation	Flow of rail pressure deviation	MeUn demand flow
$P_s - P_r > 0$	$f_{err}(d_r, n, t_c) > 0$	MeUn opening is 100%
$P_s - P_r > 0$	$f_{err}(d_r, n, t_c) < 0$	$f(d_r, n, t_c)$
$P_s - P_r < 0$	$f_{err}(d_r, n, t_c) > 0$	$f_{err}(d_r, n, t_c)$
$P_s - P_r < 0$	$f_{err}(d_r, n, t_c) < 0$	MeUn opening is 0

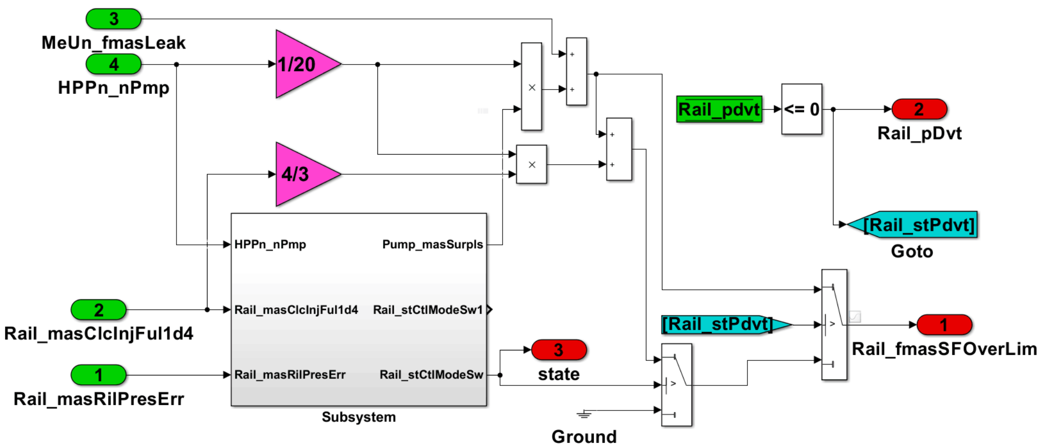

Figure 6. MeUn Flow Calculation Under Transitional Mode

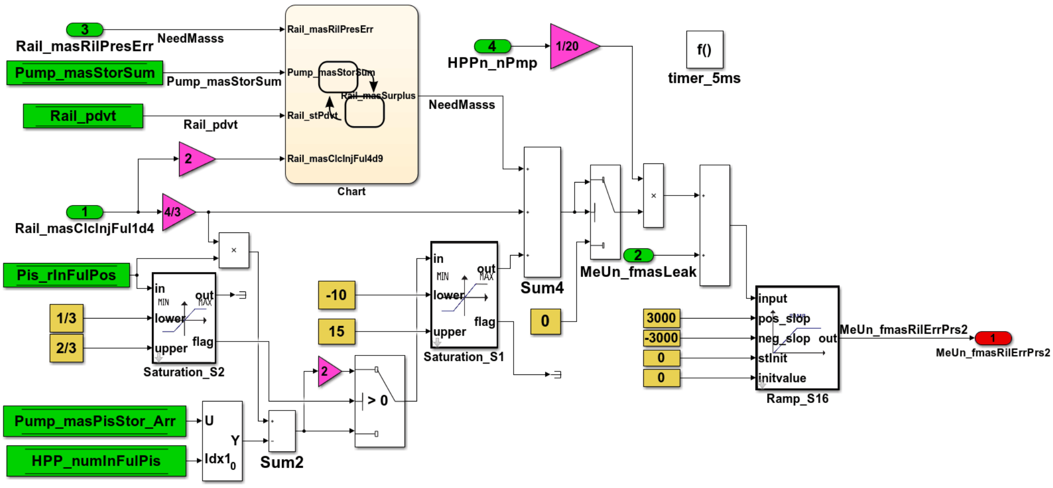

Figure 7. MeUn Flow Calculation Under Closed-loop Mode

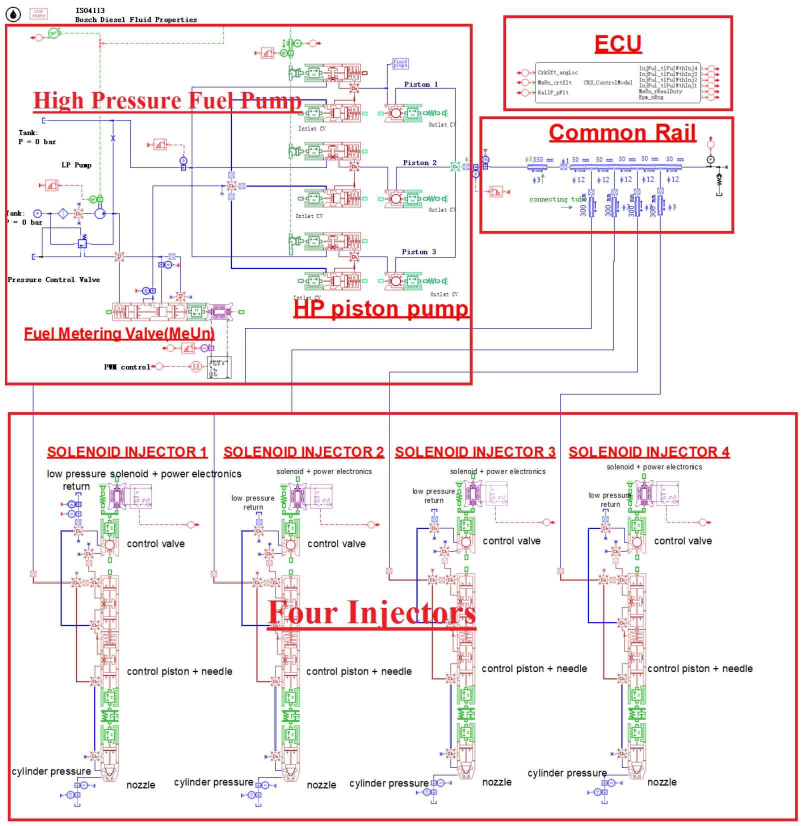

Figure 8. Controlled Physics Model of HPCRIS

Table 3. Input and output signal direction and meaning

Symbol	Direction	Meaning
CrkSft_angLoc	A->S	Location of CrankShaft Angle
MeUn_crtReal	A->S	Sensor Current in the MeUn
RailP_pReal	A->S	Sensor Rail Pressure
InjFul_tiPulWthInj1~4	S->A	Injection Pulse Width
MeUn_rRealDuty	S->A	MeUn Real Duty
Epm_n	S->A	Engine Speeds

SIMULATION RESULTS AND ANALYSIS

Simulink and AMESim are the basis of simulation, which are used to respectively build control model of rail pressure and controlled model of high pressure common rail fuel injection system. The co-simulation of model-in-loop is completed through the interface of the data interaction between the two software.

In the co-simulation model, Simulink is used as a master and AMESim is used as a slaver. The step size of discrete solver is set to 1×10^{-6} s on the basis of the required precision related to fuel injector pulse width control. The model of the co-simulation is shown in Fig. 9. It is divided into three parts in the model, namely the test case, the rail pressure control model and the physical controlled model of HPCRIS. Test cases are used to input engine operation data, mainly including fuel injection quantity and engine speed. The rail pressure control model built by Simulink represents rail pressure control strategy in the previous section and the physical controlled model of HPCRIS is the interface of physical model of HPCRIS presented in Fig. 8.

For validating the control strategy, different engine speed and fuel injection cases are assigned to test controller performance of rail pressure tracking and anti-disturbance.

Three test cases are designed to estimate the controller's performance: 1) under the condition of the same engine speed with different load, 2) under the condition of the different engine speed with the same fuel injection, 3) under the condition that engine speed and fuel injection change simultaneously. And they are utilized to cover typical engine operating condition.

Load change with the same engine speed is a common condition of internal-combustion engine. For instance, fuel injection is changed to reduce or increase the engine speed fluctuation in fixed speed cruise or electric power generation. In the test case, fuel injection volume is assigned to 20 mm^3/per, 50 mm^3/per and 85 mm^3/per under the 2000 r/min. The simulated results are shown in Fig. 10. The rail pressure fluctuation can be controlled in the range of one time fuel injection quantity. Obviously, the fluctuation of 20 mm^3 fuel injection is smaller than the condition of 85 mm^3, it can explain that the rail pressure fluctuation under steady condition is related to fuel injection volume. In addition, transitional condition can be well handled by control strategy without rail pressure overshot and excessive delay in the tracking process. The current of MeUn solenoid and the flow rate of MeUn are displayed in Fig. 10(b) and Fig. 10(c) respectively. MeUn Precision Position Control module is able to accomplish the tracking of target Current and flow-rate. However, there is a certain deviation in some operating condition which is affected by MeUn's calibration and software timer precision.

Rail pressure tracking under the operating condition that Engine speed changes with the same load is presented in Fig. 11. In the test case, the setting of single injection volume is 60 mm^3/per, and engine speed is ranged from 800 r/min, to 1800 r/min, then to 2500 r/min and to 3600 r/min at the end respectively and ramp transition in the process of engine speed change internal is adopted.

It can be seen from Fig. 11 (a) that the upper phase fluctuation value of steady-state rail pressure can be basically controlled within the single cylinder cycle injection rail pressure drop value, and does not change with the change of speed. Moreover, the center line of rail pressure fluctuation is basically consistent with the target rail pressure. Secondly, with the increase of the diesel engine speed, the number of fuel supply by the three plunger pumps and the number of fuel injection by the injector in the unit time of the high-pressure oil pump will gradually increase, which will lead to the increase of the frequency of the rail pressure fluctuation, in line with the objective law of the frequency increase of the rail pressure fluctuation with the increase of the diesel engine speed. When the speed is switched with different working conditions, the track pressure tracking process is slightly delayed, but the delay time is less than 0.2 s, which can meet the control requirements of different speed.

In Fig. 11 (b), the average current value in the MeUn solenoid coil decreases as the starting speed increases, which indicates that the opening of the MeUn increases as the speed increases for a normally open flowmeter valve. With the increase of diesel engine speed, the amount of fuel injection per unit time will increase, and the oil supply of high pressure oil pump will inevitably increase, so the increase of MeUn opening is in line with the objective fact. The MeUn flow rate in Fig. 11 (c) is based on the current in Fig. 11 (b) and the flow rate of the high-pressure oil pump in the table to obtain different current currents, so the current and flow rate of MeUn have the same fluctuation trend and similar fluctuation frequency.

At present, the current control in MeUn electromagnetic coil can basically meet the rail pressure control needs, if the current tracking control of MeUn electromagnetic coil can be optimized, the overall performance of the control system will be further improved.

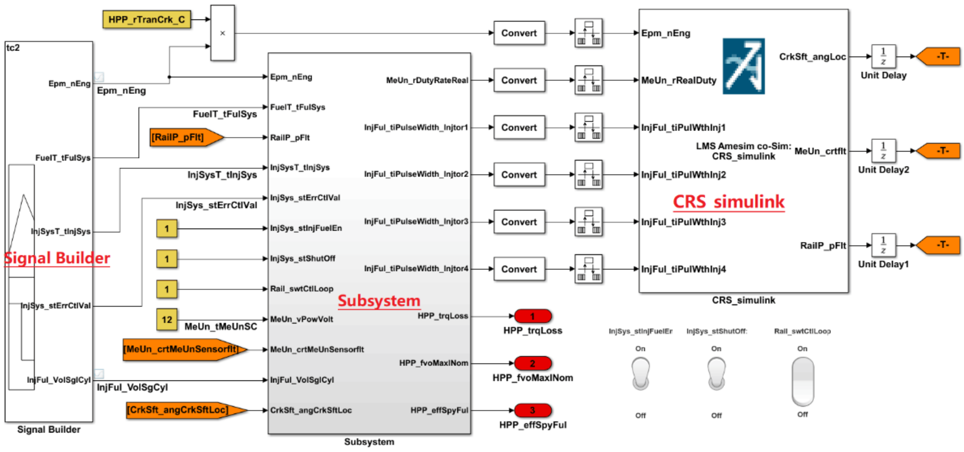

Figure 9. Co-simulation Model between physical model and control model

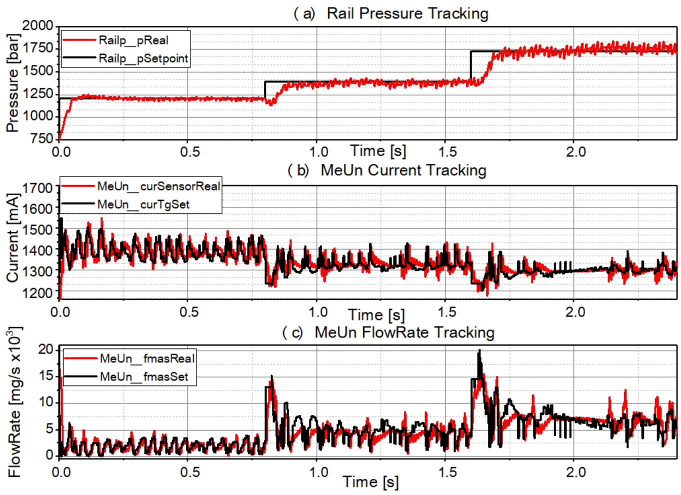

Figure 10. The results of simulation with a fixed engine speed (2000 r/min) and various loads (fuel injection volume is assigned to 20 mm^3/per, 50 mm^3/per and 85 mm^3/per): (a) Rail Pressure Tracking; (b) MeUn Current Tracking; (c) MeUn FlowRate Tracking.

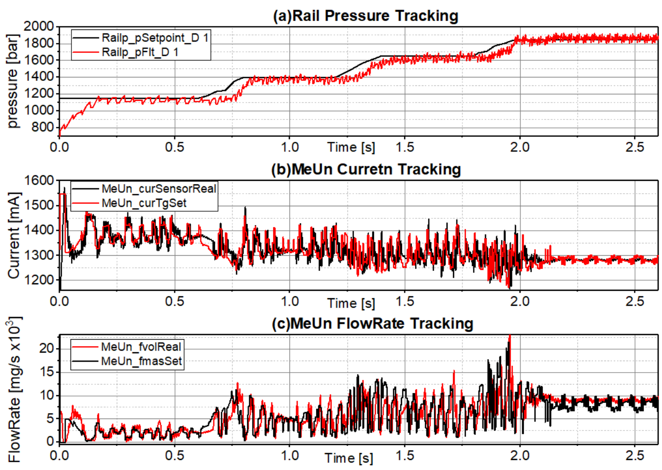

Figure 11. The results of simulation with a fixed load (single injection volume is 60 mm^3/per) and various engine speeds (engine speed is ranged from 800 r/min, to 1800 r/min, then to 2500 r/min and to 3600 r/min): (a) Rail Pressure Tracking; (b) MeUn Current Tracking; (c) MeUn FlowRate Tracking.

For automotive engine, most operating condition of internal-combustion engine is in the process of frequent acceleration and deceleration, so the target rail pressure is bound to change frequently. To satisfy the demand of the power, economy and exhaust emission, the actual rail pressure is required to be able to track the target rail pressure quickly when the speed and load acutely change at the same time. The third test case is designed based on the above condition, as shown in Fig. 12.

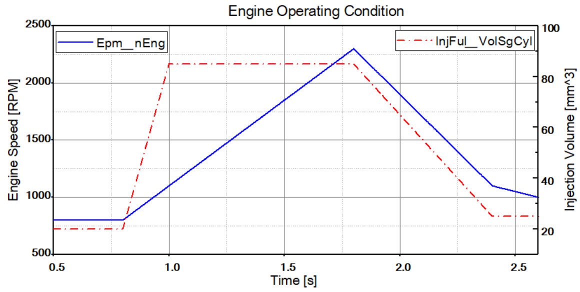

Figure 12. Engine operating condition with various loads and engine speeds

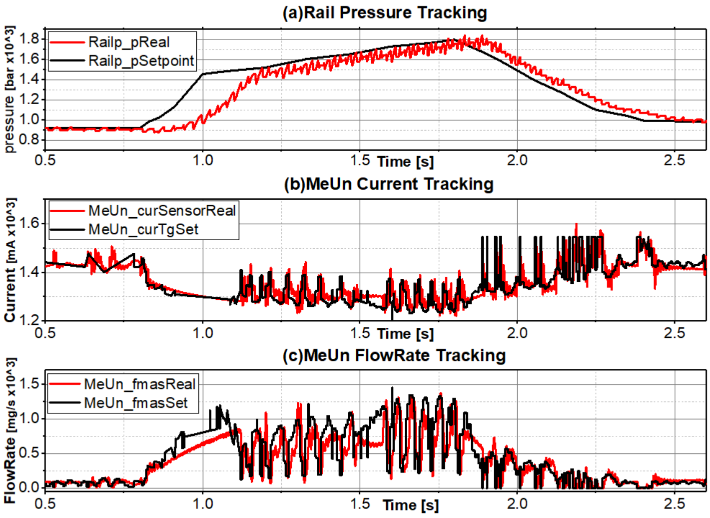

Figure 13. The results of simulation with various loads and engine speeds: (a) Rail Pressure Tracking; (b) MeUn Current Tracking; (c) MeUn FlowRate Tracking.

Rail pressure tracking under engine operating condition in Fig. 12 is demonstrated in Fig. 13. When engine operating condition changes, excessive rail pressure deviation is exhibited because of lower engine speed and rapidly increasing fuel injection quantity. Actually, it is essentially caused by the structure limit of HPP. The above factors lead to a slow increase in the rail pressure. Such phenomenon is exhibited in the Fig. 13(a). When the rail pressure demand decreases, fuel delivered to common rail will not be reduced immediately, and the delivering process must be continued for some time as the physical and mechanical structure's hysteresis of HPP. The phenomenon can be observed. For example, the actual rail pressure is slightly higher than target rail pressure when the simulation time is greater than 1.8 s ($t > 1.8$ s). In a word, actual rail pressure follows the target rail pressure well via the application of the control strategy.

CONCLUSIONS

In this study, a new rail pressure control algorithm based on the physical model of common rail system is proposed. By calculating the MeUn demand flow rate, the corresponding PWM duty cycle of the MeUn solenoid valve is determined. To verify the controller's performance, three test cases were designed to represent the different typical operating conditions of the engine. The simulation results show that the controller has good performance of track pressure tracking and interference suppression. In the process of rapid rise of rail pressure, there is no overshoot problem in actual rail pressure tracking, and it can be transferred from one working condition to another.

The control strategy proposed in this paper can not only be applied to the control of common rail system structure, but also eliminate the application of PID or PID-Plus algorithm which increases the workload of calibration. This means that it reduces the calibration effort, facilitates the transfer of control strategies to other new common rail systems or diesel engines, and provides a basis for early verification of rail pressure controller performance in new common rail systems or new engine applications.

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基於數學算法的共軌壓力閉環控制的模擬計算研究

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摘要

控制共軌壓力是高壓共軌噴射（High Pressure Common Rail Injection System，HPCRIS）系統的核心，直接影響發動機的動力性、經濟性和排放性。其中，穩態波動幅值和瞬態跟蹤速度是衡量控制器性能的重要指標。爲了提高控制器的性能，首先在物理模型的基礎上開發了HPCRIS控制模塊，包括軌壓控制狀態機、需求流量計算模塊和燃油計量控制單元。其次，通過在Matlab/Simulink環境中構建的HPCRIS控制模塊與在AMESim環境中製作的HPCRIS被控對象的聯合仿真，驗證了控制策略。最後，通過分析仿真結果，對HPCRIS控制模塊的性能進行了驗證，爲新型HPCRIS的移植提供了依據。