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# Research on Rail Pressure Control of High-Pressure Common Rail System for Marine Diesel Engine Based on Controlled Object Model

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**ABSTRACT** The common rail pressure of a marine diesel engine high-pressure common rail system is typically controlled by a PID (Proportional Integral Differential) algorithm, which has problems with both overshoot and response delay. In this paper, the control method based on controlled object model for rail pressure is studied. The control model for a high-pressure common rail system is established within the controller. Through the developed control model and the current actual rail pressure obtained by real-time acquisition, the control variables issued by the controller to the controlled object are predicted and calculated. The fuel supply required in the whole high-pressure common rail system is calculated in advance, with the fuel supply then taken as the input value for the driving model in order to calculate the PWM control signal required by the high-pressure oil pump. A PWM control signal is used to control the proportional valve opening of the high-pressure oil pump, and then control the fuel pressure in the common rail pipe. In order to verify the correctness and control effects of the rail pressure control strategy based on controlled object model, a real-time simulation of the high-pressure common rail system has been developed by combining calculation equations with a MAP data query, which ensures real-time performance. At the same time, the real-time simulation of the high-pressure common rail system shows that the error between the simulation's calculation and the actual test data is less than 5%, and thus model accuracy is guaranteed. The control method based on controlled object model is combined with the real-time simulation model for the high-pressure common rail system to verify the control strategy function, and the control effect is then compared with that of the PID control. Comprehensive verification shows that the control strategy based on controlled object model can significantly improve the response delay and overshoot of PID control.

**INDEX TERMS** Marine diesel engines, model-based control, high-pressure common rail system.

## I. INTRODUCTION

Due to its high economy, reliability, and wide range of power performance, the traditional diesel engine has been used as the prime mover of the main propulsion device and generator set of ships [1]. With the increasing awareness of environmental protection, the emission regulations governing marine diesel engines are increasingly stringent. At the meeting of the Marine Environment Protection Committee, IMO adopted a two-stage target for NOx reduction. In 2011, it reached the Tier II limit, and in 2016, it reached the stricter Tier III limit in the emission control area [2]. Since 2016, the Chinese government has set emission limits and emission control zones for inland river basins and offshore areas. It has also introduced the "Emission Limits and Measurement Methods of Marine Engine Exhaust Pollutants (Phase I and II in China)" [3]. In the face of strict emission regulations, in-cylinder purification and exhaust post-treatment are used simultaneously as means of emission reduction. High-pressure common rail fuel injection technology is one of the main measures for

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in-cylinder purification. It can achieve good fuel atomization in the cylinder, flexibly control fuel injection under all working conditions for a diesel engine, optimize the combustion process, and reduce fuel consumption, noise, and emissions [4], [5].

A typical high-pressure common rail system is primarily composed of a control system and an actuator for the fuel system. The fuel system is comprises a high-pressure oil pump, common rail pipe, high-pressure oil pipe, and an electronically controlled injector. The high-pressure oil pump delivers fuel to the public supply tubing, with each injector connected to the public supply tubing to achieve accurate control of the oil pressure in the common rail. The high-pressure common rail system has been widely used in road diesel engines. However, due to the specific requirements for marine diesel engines in terms of speed, power, and operating environment, the high-pressure common rail systems utilized here have considerable differences in performance requirements, structural composition, and control mode. The single-cylinder power of the marine diesel engine is large and the fuel injection per cycle is large, which makes the fuel supply per cycle much larger than the common rail system [6]. Simultaneously, since marine diesel engines are generally used as the main propulsion device and generator set, they work in both propeller and generator condition, so the accuracy of diesel engine speed control is higher. At the same time, the structural size and mass inertia of marine diesel engines are much larger than those of vehicle diesel engines, which puts forward higher requirements for the control accuracy and dynamic response of the control system [7].

Since a single cycle of a marine diesel engine requires more fuel injection, and there is a certain hydraulic hysteresis in the high-pressure common rail system, when the injection is carried out with a large amount of fuel under a high common rail pressure, the pressure at the front end of the injector drops suddenly, with the pressure oscillation generated by the injection transmitted to the common rail pipe, thus affecting the stability of the injection pressure. In terms of the high-pressure common rail system structure, in order to reduce the rail pressure fluctuation and maintain a high common rail pressure, some of the marine high-pressure common rail systems use a multi-stage common rail structure to avoid the common use of the same rail cavity which makes the fluctuation affect multiple injectors or adds a pressure storage chamber before the injector to buffer and weaken the injection pressure instability caused by such a large injection volume [8], [9].

In terms of the control strategy for the common rail pressure, the traditional means is based on PID control [10]. Due to the large amount of circulating fuel injection and large mechanical structure size of marine diesel engines, the high-pressure common rail system will have more obvious rail pressure fluctuation and dynamic response delay. Moreover, the high-pressure common rail system of some diesel engines has multiple fuel injection functions, which requires a rail pressure control strategy with better control performance [11]–[13].

Targeting the problem of slow overshoot and response in traditional PID control methods, some scholars have put forward an optimized PID strategy for rail pressure control. Yue *et al.* proposed a PID control strategy by adding feedforward, which was used to determine the basic fuel quantity, with the PID feedback control used to adjust the target and the actual rail pressure difference [14]. Xiang *et al.* proposed a fuzzy PID control method, since idle state response speed and stability are better than the traditional PID control [15]. Li *et al.* proposed that adaptive PID control could be used in the DME engine to achieve rapid response and small overshoot [16]. Wang *et al.* proposed a multi-stage PID control. After reading MAP data in the ECU to obtain the target rail pressure value, the adjustment process was not undertaken directly, instead, it was divided into ten stages, with each stage having corresponding adjustment parameters, which had a better result than traditional PID control methods [17]. Although the optimized PID control method has a good control effect, the calibration of PID control parameters in this transient process is difficult due to working condition switching, and as a result, researchers began to propose new control methods.

Aiming at the problem of slow overshoot and response of PID control, a control strategy based on controlled object model is proposed. The simulation established in the control strategy can predict future changes of the operable variables and the influence they have on the controlled variables. When the input conditions change, the response can be predicted to better meet the control objectives. Some scholars have studied the control of common rail pressure using a model-based control strategy. For example, Liu *et al.* established a model for the diesel engine injection system and appropriately simplified it. Using this model, a track pressure controller, based on an active disturbance rejection control framework (ADRC) was designed, and the efficacy of the strategy verified [18]. Lino *et al.* proposed a rail cavity pressure controller based on the nonlinear common rail pressure model of fluid mechanics and verified the predictive ability of the control model and effectiveness of the control strategy through software in this loop [19]. Balluchi *et al.* proposed a multi-rate controller based on a hybrid common rail pressure model. The model in the controller can describe the continuous and discontinuous behaviors of the common rail system [20]. Hong *et al.* proposed a common rail pressure controller based on an empirical rail pressure model, which is effective at reducing steady-state pressure fluctuations through the pressure valve control valve and flow metering unit [21].

Through the research of these scholars, it can be seen that the model-based control strategy can improve the slow overshoot response found in traditional PID control, and also reduce a lot of work on the calibration of PID parameters. However, most of these studies considered automotive diesel or gasoline engines. The high-pressure common rail system of vehicle and marine high-power diesel engines

have different structures. The higher power of marine diesel engines requires more fuel injection, and the structural size and mass inertia of marine diesel engines are much larger than those of vehicle diesel engines. Therefore, the high-pressure common rail system of marine diesel engines needs higher control precision and a greater dynamic response. Therefore, traditional PID control has certain limitations in the control of marine diesel common rail systems. Therefore, it is meaningful to study model-based control strategies for the high-pressure common rail system of marine diesel engines.

In this paper, according to the characteristics of rail pressure control in the high-pressure common rail system of marine diesel engines, a model-based rail pressure control strategy is proposed. The purpose is to improve the common PID control strategy in rail pressure control, which is easy to produce control overshoot or response delay in the process of actual common rail pressure by following the target rail pressure, and resulting in a decreased fuel injection atomization performance, such that the economy of the whole diesel engine decreases and emissions increase. The numerical simulation of mechanisms such as the injector, common rail pipe, high-pressure oil pump, and the driving model for calculating PWM control signals are established. By inputting the target rail pressure, the actual rail pressure, and the current engine working conditions to the simulation, the quantity of fuel required by the whole high-pressure common rail system can be predicted. The driving model outputs the calculated fuel quantity as a PWM signal which can drive a high-pressure oil pump. The proportional valve opening of the high-pressure oil pump is controlled by a PWM control signal to realize the adjustment of the oil supply from the high-pressure oil pump, which then controls fuel pressure in the common rail pipe. In this paper, when the marine diesel engine is compared with road diesel engines, the fuel consumption per cycle is greater, the operational costs are higher, and there is a greater security risk in extreme tests, meaning that one cannot directly work on a marine diesel engine to research relevant control strategies. Many studies have shown that accurate real-time simulation model can provide conditions for the verification of control system [22], [23]. Therefore, this paper developed a real-time simulation of the high-pressure common rail system as the control object. The real-time simulation of the high-pressure common rail system includes the injector sub-model, common rail pipe sub-model, high-pressure oil pump sub-model, and fuel metering model. The developed model-based control strategy and high-pressure common rail real-time simulation are then combined to verify the functionality of the control strategy and compare it with traditional PID control strategy.

## II. CONTROLLED OBJECT MODEL-BASED RAIL PRESSURE CONTROL STRATEGY

The traditional rail pressure control of a high-pressure common rail system is based on PID regulation, as shown in Figure 1. This method takes the difference between the target rail pressure and the actual rail pressure as the control target, and

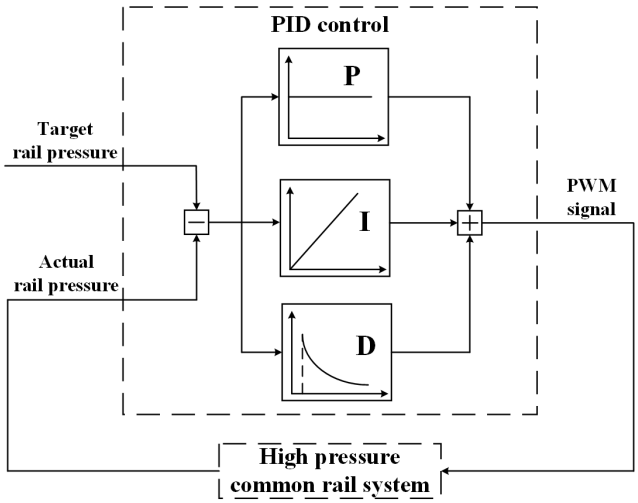


**FIGURE 1.** Traditional PID control framework.

completes the control of a high-pressure oil pump through P, I, D's three links. The selection of the three P, I, D parameters can be determined using fixed parameters, fuzzy algorithms, sliding mode variable structures, or other methods.

In the traditional PID control method, the target rail pressure changes under variable working conditions. When the actual rail pressure is adjusted to the target rail pressure, a rail pressure control overshoot and response delay will occur, which affects the injection pressure of the common rail system and leads to the deterioration of combustion in the diesel engine under variable working conditions, which is not conducive to its economy or emission levels.

In order to tackle the above problems, this paper studies the controlled object model-based rail pressure control strategy and proposes an overall framework for the control model as shown in Figure 2. The developed control model receives the engine speed signal, fuel injection pulse width signal, target rail pressure value, and actual rail pressure value, and obtains the PWM control signal through calculation of each control model. The PWM signal is used to control the proportional valve opening and the fuel supply of the high-pressure oil pump, so as to control the fuel pressure in the common rail pipe.

The established control models include the injector, common rail pipe, high-pressure oil pump, and drive control models. The injector control model receives the pulse width signal and the actual rail pressure signal and calculates the predicted fuel consumption. According to the difference between the target and actual rail pressure, the common rail control model uses the rail pressure increment equation to calculate the fuel storage. In other words, the pressure change in the common rail is taken as the known amount to calculate the quantity of fuel required in the common rail. The high-pressure oil pump control model calculates the required proportional valve opening according to the fuel storage provided by the common rail model, the fuel consumption

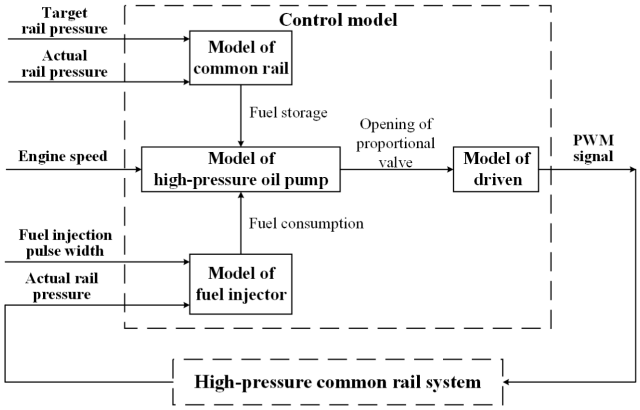


**FIGURE 2.** Overall framework based on model control.

calculated from the injector model, and the actual engine speed signal. The driving model receives the proportional valve opening signal from the high-pressure oil pump control model to calculate the PWM control signal needed by the actual high-pressure oil pump. The PWM signal is used to control the proportional valve opening of the high-pressure oil pump and control the oil supply of the high-pressure oil pump. The change of rail pressure in the common rail pipe caused by the change of fuel supply in the high-pressure oil pump can be used as the actual rail pressure input value for the control model, so as to effectively achieve closed-loop control.

## III. ESTABLISHMENT OF CONTROL MODEL

In the controlled object model-based control strategy developed herein, the control model is divided into four parts—the injector, rail, high-pressure oil pump, and drive control models. The process of establishing each model is shown below.

### A. CONTROL MODEL OF INJECTOR

In order to meet the real-time control requirements, the injector model uses the Bernoulli equation combined with the nozzle flow and the throttling effect at the throat of the needle valve seat. This equation considers the throttling benefit at the nozzle and the throat of the needle valve seat in the working process of the injector. The equation is shown as follows:

$$\frac{dm_{inj}}{dt} = \sqrt{2 \cdot \rho_{fuel} \cdot (p_{zpipe} - p_{cyl}) \cdot \frac{1}{\left(\frac{\xi_{NS}}{A_{NS}{}^2} + \frac{\xi_{NH}}{A_{NH}{}^2}\right)}}, \quad (1)$$

The formula, $\rho_{fuel}$ is fuel density (kg/m$^3$). $P_{pipe}$ is the fuel pressure at the inlet of the electronically controlled injector ($MPa$). $P_{cyl}$ for cylinder pressure (MPa). $\zeta_{NS}$ is the flow coefficient at the throat of the needle valve seat. $\zeta_{NH}$ is the flow coefficient of the nozzle. $A_{NH}$ is the cross-sectional area of the nozzle $(m^3)$.

### B. COMMON RAIL PIPE CONTROL MODEL

The common rail pipe is simplified to a zero-dimensional cavity. The common rail pressure in the common rail pipe is modeled by mathematical function calculation, mainly considering the compressibility of fuel oil. According to the hydraulic performance of the fuel oil, the formula for the fuel pressure change in the common rail pipe can be obtained as follows:

$$\frac{dP_{rail}}{dt} = \frac{E_d}{V_{rail}} \cdot \frac{1}{\rho_{rail}} \cdot \left(\frac{dm_{pump}}{dt} + \frac{dm_{inj}}{dt}\right), \quad (2)$$

The fuel in the common rail pipe is treated as a constant volume, and the incremental equation describing the common rail pressure can be obtained from the compressibility of the fuel itself. In turn, the pressure change in the common rail can be taken as a known amount so as to be able to calculate the fuel required in the common rail. The simplified equation is as follows:

$$\Delta V_f = \frac{\Delta p_r V_r}{E_d}, \quad (3)$$

In the equation, $\Delta p_r$ is the difference between the current and target rail pressures. $E_d$ is the elastic modulus of the fuel used. $\Delta V_f$ is the fuel supply required by the common rail. $V_r$ is the volume of the common rail cavity.

### C. CONTROL MODEL OF HIGH-PRESSURE OIL PUMP

The working model for the high-pressure oil pump is mainly used to convert the oil supply required by the common rail system to the opening of the proportional valve. In the actual working process, the opening of the high-pressure oil pump proportional valve is related to the volumetric oil supply rate of the high-pressure oil pump. The oil supply volume flow of a high-pressure oil pump can be calculated by the oil supply required by the high-pressure oil pump, and thus the opening requirement for the high-pressure oil pump proportional valve can be inversely calculated.

Because the high pressure oil pump of the common rail fuel system used in this paper is a three cam plunger, and the medium-speed marine diesel engine studied in this paper is a six-cylinder medium-speed engine, according to the speed ratio of crankshaft and camshaft, the primary plunger fuel supply process of the high pressure oil pump is exactly the same as the primary fuel injection process for the electronic control injector in the diesel engine, and thus the corresponding fuel supply flow under different fuel supply quantities of the high-pressure oil pump can be readily obtained. The equation is as follows:

$$\dot{V}_p = \max\left[\Delta V_p \frac{n}{60iZ}, \dot{V}_{p\max}\right] \quad (4)$$

In this equation: $n$ is diesel engine speed ($r/\min$). $i = 0.5$, which corresponds to a four-stroke diesel engine. $Z$ is the number of diesel cylinders. $\dot{V}_{p\max}$ is the maximum flow limit in the high pressure oil pump proportional valve, which is related to its mechanical structure.

### D. DRIVE MODEL

The operation of the high-pressure oil pump proportional valve is driven by the solenoid valve, and its opening is related to the current in the solenoid valve. The current of the solenoid valve is determined by the duty cycle of the driving voltage. The control strategy can be described by the following equation for the relationship governing the duty cycle of the driving voltage of the solenoid valve.

$$V(t) = (R + R_1) \bullet i(t) + \frac{d\lambda}{dt} \tag{5}$$

$$\lambda = L(s) \bullet N \bullet i(t) \tag{6}$$

In the equation, $V(t)$ is the output voltage of the control strategy ($V$). $i(t)$ is the drive current ($A$). $R$ is the resistance of the solenoid valve coil ($\Omega$). $R_1$ is equivalent to the internal resistance of the control strategy drive chip ($\Omega$). $\lambda$ is the magnetic chain value of the magnetic circuit. $L(s)$ is the equivalent inductance of the solenoid valve coil.

## IV. ESTABLISHMENT OF REAL-TIME SIMULATION MODEL FOR HIGH-PRESSURE COMMON RAIL

The marine diesel engine's fuel consumption per cycle is higher than the road diesel engine. Thus, the economic cost is higher, and the test data for transient conditions are harder to obtain, as the common rail pressure maybe be as high as 160 MPa, meaning there are great security risks involved in certain extreme test conditions. Therefore, in order to verify the developed control strategy and ensure the safety of the test, this paper presents real-time simulation of the high-pressure common rail system as the controlled object. The high-pressure common rail real-time simulation is created using Simulink software. The overall framework for the high-pressure common rail real-time simulation is shown in Figure 3. The method of surrogate model is used to establish the real-time simulation model of the high-pressure common rail system. The high-precision hydraulic system model of the high-pressure common rail system provides accurate characteristic data for the real-time simulation model. The real-time simulation model is established by using map data and formulas. The real-time simulation model has the high precision of performance simulation model and real-time performance.

The real-time simulation of the high pressure common rail system is used as the controlled object of the control strategy for functional verification of the aforementioned control strategy.

In the process of establishing the real-time simulation of the high-pressure common rail fuel system, the real-time simulation of the whole high-pressure common rail fuel system is built from the following sub-models—high-pressure fuel pump, common rail pipe, injector, and mass conversion. The real-time simulation can emulate and calculate the pressure in the common rail pipe and the amount and rate of fuel injection according to the input pulse width and PWM signal. The output injection quantity and injection rate can be compared with experimental values to verify the accuracy of the model.

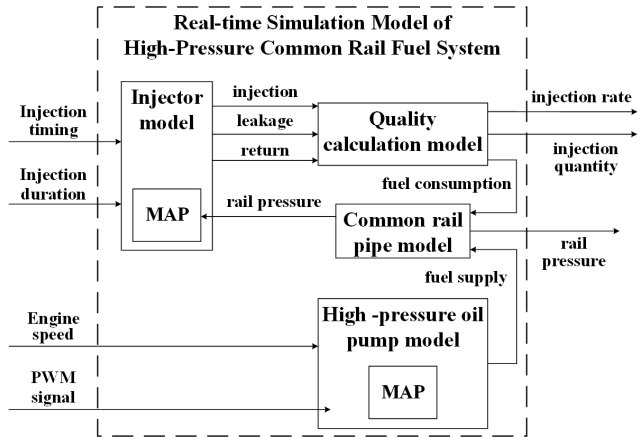


**FIGURE 3.** Overall framework of the high-pressure common rail real-time simulation.

Under the premise of ensuring accuracy, the rail pressure output from the real-time simulation of the high voltage common rail system can be used as the actual rail pressure input value for the control model in the control strategy. The PWM signal needed in the real-time simulation of the high voltage common rail system can be retrieved from the control model in the control strategy. Therefore, it is feasible to jointly evaluate the control strategy and real-time simulation of the high-pressure common rail.

In the real-time simulation of the high-pressure common rail system, the MAP from the actual test data exists in the established process for the high-pressure oil pump sub-model. The high-pressure oil pump sub-model can find MAP data utilizing the PWM control and diesel engine speed signals to obtain the oil supply signal transmitted to the common rail pipe sub-model. The common rail pipe model receives the fuel supply signal from the high-pressure oil pump sub-model and the consumption signal from the mass transfer sub-model. The common rail pressure at the current time is then calculated formula. MAP data, such as fuel consumption and fuel leakage, are stored when generating the injector sub-model. The injector sub-model can find the corresponding MAP data according to common rail pressure and injection pulse width to obtain fuel consumption values that are then passed to the mass transfer sub-model. The mass conversion model simplifies the fuel consumption per cycle from the injector sub-model to a piecewise function, calculates the injection rate, and then outputs the fuel consumption at different times in the common rail pipe to the common rail pipe model so as to allow pressure calculation. The common rail pipe model calculates the current rail pressure according to the fuel consumption and supply.

After the establishment of the real-time simulation of the high-pressure common rail fuel system, static and dynamic model accuracy verification was carried out. Since the real-time simulation of the high pressure common rail fuel system is mainly aimed at the analysis of control strategy, verification of the real-time simulation of the high pressure

common rail fuel system aims to confirm both the fuel injection quantity and the common rail pressure. The fuel injection quantity verification analysis of the injector (Figure 4) was obtained from calculations in the simulation, with the results showing that the simulation error is not more than 5 %.

Real-time simulation of the high-pressure common rail fuel system was verified dynamically. The established real-time simulation is controlled by adding a simple PID algorithm, with the initial pressure in the common rail set to 140 MPa, to reduce the pressure balancing time in the common rail. The pressure fluctuation of the common rail in the simulation is below 140 MPa, as shown in Figure 5, which proves that the established real-time simulation of the high-pressure common rail can represent the decrease in rail pressure caused by the fuel injection per cycle and the increase of rail pressure under the action of the high-pressure oil pump. The established real-time simulation can demonstrate the rail pressure fluctuation in the common rail pipe during fuel injection, which can be used as the controlled object of the control strategy to verify the control effect of the control strategy on the rail pressure.

## V. VERIFICATION OF RAIL PRESSURE CONTROL STRATEGY BASED ON MODEL

The controlled object model-based rail pressure control strategy replaces the PID control algorithm and the high-pressure common rail real-time simulation to jointly verify the control strategy function. The overall framework of the co-simulation of the control model and the controlled model is shown in Figure 6. The fuel injection pulse width, diesel engine speed, target rail pressure, and the actual rail pressure are input into the control model. The PWM signal of the high-pressure oil pump is then calculated by the control model. The real-time simulation of the controlled model receives PWM signals from the control model and the injection pulse width and timing signals are set according to the operational status of the diesel engine. The rail pressure value under the current control signal is calculated through the real-time simulation. The current rail pressure value is again input into the control model as the actual rail pressure signal, and the next operation is carried out to form a closed-loop control effect.

The controlled object model-based control strategy is combined with the real-time simulation of the high voltage common rail system to verify the control strategy, and the PID control effect is compared and analyzed. Since the common rail pressure control mainly includes the steady-state rail pressure control and the transient rail pressure control, the verification also includes confirmation of the rail pressure establishment process from zero accumulators up to a certain target value. The process of verifying that the target rail pressure is constant and maintaining this during the cyclic injection is conducted by verifying the rail pressure adjustment process when the target rail pressure suddenly changes.

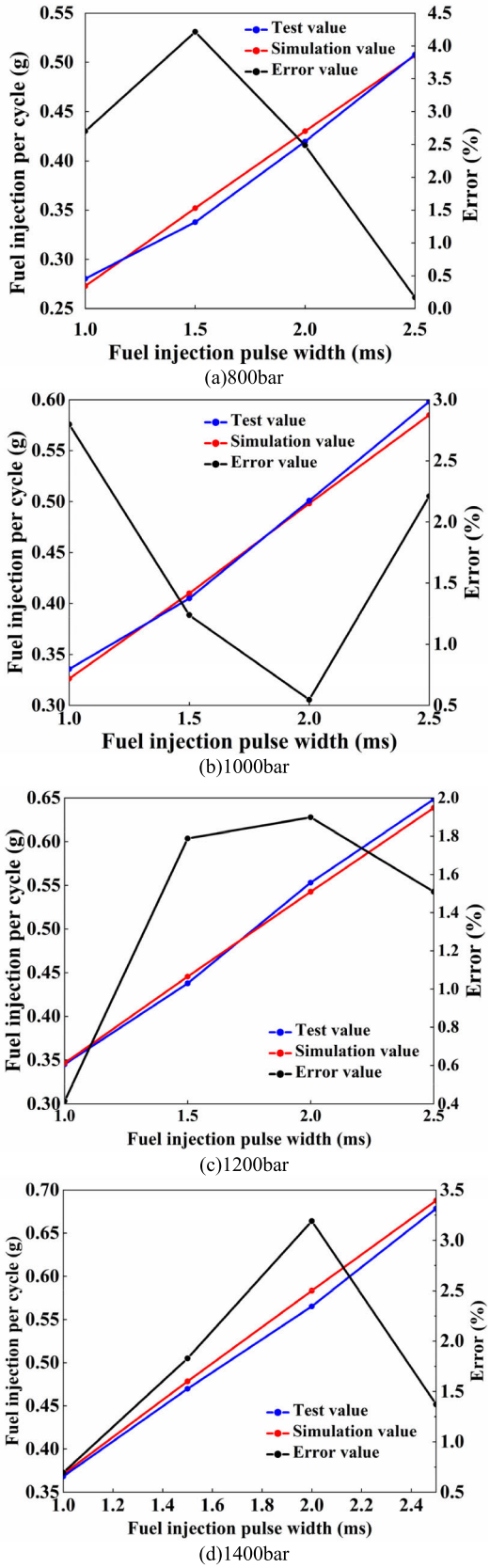


**FIGURE 4.** Error comparison.

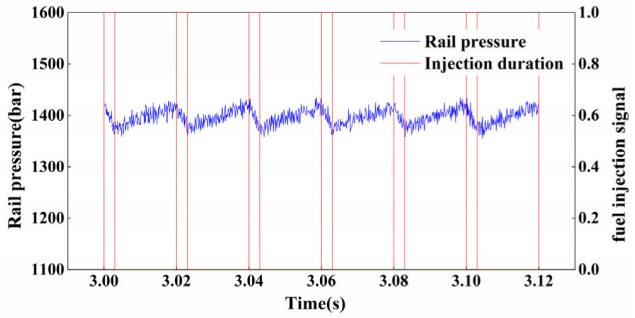


**FIGURE 5.** Rail pressure fluctuation verification.

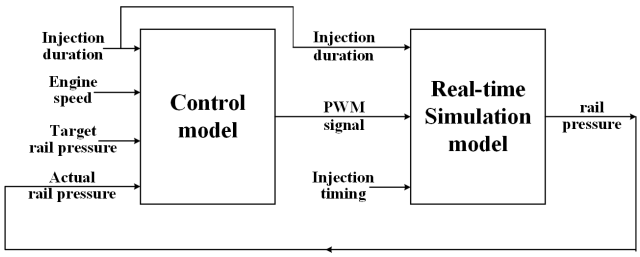


**FIGURE 6.** The overall framework of the joint simulation of control and real-time models.

### A. 0-700BAR RAIL PRESSURE STORAGE PROCESS

As shown in Figure 7, taking 700bar as the target rail pressure, the process of establishing rail pressure during the starting of a diesel engine is simulated. In the process of raising the rail pressure from 0 to 700 bar, in order to quickly establish the common rail pressure within 0.25 s, the open-loop control method is adopted. At this time, the effect of PID control is the same as that of controlled object model-based control. When the common rail pressure reaches 500 bar, in order to make the common rail pressure smoothly reach the expected value, the open-loop control is switched to closed-loop control. Due to the overshoot problem of PID itself, after switching to closed-loop control, the PID control needs to be stable for a certain time. The controlled object model-based control method is used to predict the oil supply for the model. Therefore, when the expected common rail pressure is reached, the controlled object model-based control method can be stabilised by calculating the oil supply. Therefore, compared with PID control, the common rail pressure fluctuation is more balanced after switching the controlled object model-based control strategy to the closed-loop control at 0.25 s, once the starting rail pressure is established.

### B. RAIL PRESSURE ADJUSTMENT PROCESS FROM 700BAR TO 1400BAR

As shown in Figure 8, the target rail pressure changes suddenly. When the working conditions of the diesel engine change, the target rail pressure changes, and the control strategy establishes a new rail pressure value. When the target rail pressure is adjusted from 700 bar to 1400 bar, as a means of establishing a new target common rail pressure, the trends

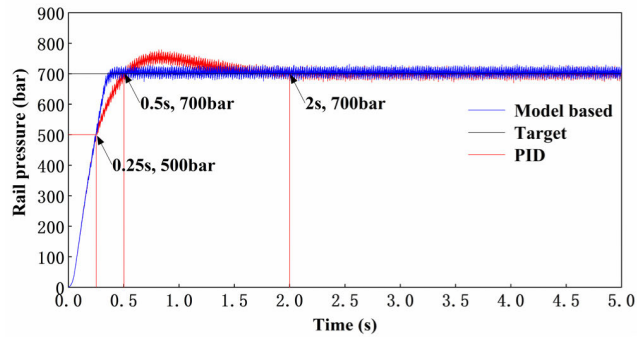


**FIGURE 7.** 0-700bar rail pressure change.

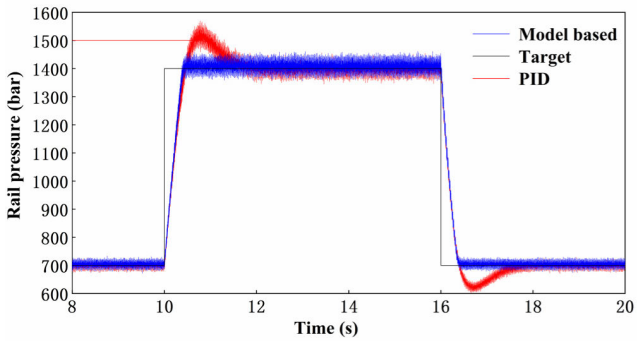


**FIGURE 8.** 700-1400bar rail pressure change.

using controlled object model-based rail pressure and PID control are similar. However, when the common rail pressure reaches the expected pressure, the PID control will have an overshoot of about 100 bar, and then a certain response time is needed to completely stabilize the common rail pressure. When the common rail pressure is close to the expected value, according to the difference between the expected common rail pressure and the actual common rail pressure, the amount of oil supplied by the high-pressure oil pump to the common rail pipe is gradually adjusted, so as to provide a smooth transition to the common rail pressure.

### C. THE PROCESS OF MAINTAINING RAIL PRESSURE AT 1600BAR

As shown in Figure 9, the target rail pressure is constant, and the rail pressure fluctuation caused by fuel injection in each cycle under the stable working condition of the diesel engine is simulated. When the target common rail pressure is 1600bar, the diesel engine speed is 1000r / min, such that the injection frequency is 0.12s, and the injection pulse width is 3ms, hence the corresponding PID and controlled object model-based rail pressure control effects are comparable. The figure shows that both the PID control algorithm and the controlled object model-based control strategy can stabilize the common rail pressure at around 1600 bar, and that the fluctuation range of the common rail pressure is within 5 %. In terms of the common rail pressure fluctuation, although the common rail pressure fluctuation of the controlled object

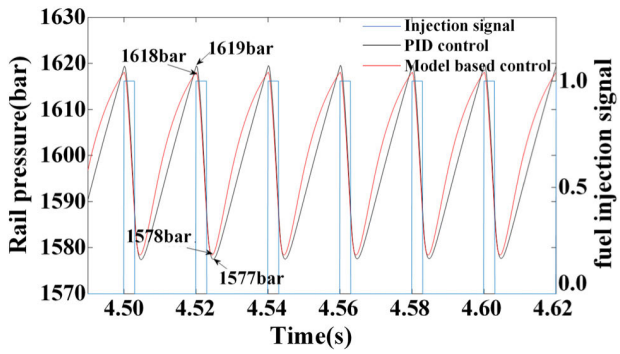


**FIGURE 9.** Maintain rail pressure at 1600 bar.

model-based control strategy is only 2 bar less than that of the PID control algorithm, the change rate of the controlled object model-based control in the rail pressure rise stage after the injection is more stable than that of the PID control algorithm.

## VI. CONCLUSION

Since common rail pressure control typically includes both steady and transient rail pressure control, this paper verifies the establishment of rail pressure in the starting process, the adjustment of rail pressure when working conditions change, and the constant target rail pressure in the steady process.

When the target rail pressure is constant, through the verification of 100 % working condition and 1600 bar target common rail pressure in the diesel engine, although the common rail pressure fluctuation of controlled object model-based control is only 2 bar smaller than that of PID control, the change in speed of the controlled object model-based control in the rail pressure rise stage after injection is more stable than that when using PID control.

In the stage of rail pressure establishment from 0 bar to 700 bar, the control strategy based on the control model can completely establish the target common rail pressure within 0.5 s and achieve stability. Compared with the PID control algorithm, it takes 2 s to completely stabilize the common rail pressure, which significantly improves the response problems associated with PID control.

When the target rail pressure changes, the target common rail pressure changes from 700 bar to 1400 bar. The controlled object model-based rail pressure control and PID control show similar trends, but when the common rail pressure reaches the expected pressure, the controlled object model-based control strategy can reduce the overshoot of 100 bar compared with PID control.

Comprehensive verification shows that the controlled object model-based control strategy can significantly improve the response delay and overshoot of PID control. Because of the good regulation ability, the controlled object model-based control strategy has further exploration value. The model-based control strategy is based on the control model, so if the control model of the marine diesel engine can be established, the controlled object model-based control method can also be explored in the performance optimization control of the whole engine. This control method has a certain promotion ability. The control model is simulated and calculated in the control strategy, and the real controlled object runs in practice. The control strategy is controlled by the target value calculated by the model and the real value of the actual mechanism, which is a form of a digital twin. Therefore, the research on the control strategy based on the model also provides a research basis for the application of digital twin technology in the field of marine diesel engine control.

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