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Modeling and Analysis of Servo Oil Common Rail System of Marine Intelligent Diesel Engine

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Abstract: In order to observe the dynamic performance of servo oil Common Rail System of marine intelligent diesel engine, 7RT-flex 60C diesel engine is selected as the research object to analyze the characteristics of common rail system. Based on the continuity equation and the motion equation of the fluid mechanics, the mathematical models with blocks are established for servo pumps, collectors, common rail of servo oil and vent control module, etc. All the mathematical models are calculated by MATLAB/Simulink for the rail pressure of collector and servo oil common rail, servo pump plunger stroke, plunger cavity pressure and flow, front chamber pressure and the displacement of piston, exhaust valve lift, etc. to achieve the dynamic simulation and parameter analysis. Finally, the simulation results of curve and data are compared with the basic characteristics of actual servo oil common rail system to verify the correctness of the model.

Key words: Marine intelligent diesel engine, servo oil, common rail system, modeling

INTRODUCTION

The most potential marine electronic control common rail diesel engine today, can not only realize accurate control of fuel injection timing and exhaust timing, but also can determine the required and the best injection pressure according to different condition which can meet the requirements of economy and emissions (Li *et al*., 2004) and it's the reason for modeling and simulation of each system on diesel engine. At present, the modeling and simulation of ship diesel engine common rail system are mainly concentrated on the fuel common rail system and less on the research of the servo oil common rail system and control oil. The pressure parameter changes of the parts of the servo oil system can be displayed more intuitively by simulation of the servo oil system which can provide necessary support for the research of the common rail system. Based on the analysis of the 7RT-flex 60C diesel engine common rail system from the Wartsila Company, this article builds a model and study the simulation of the servo oil common rail system. By analyzing the simulation data, the designers can observe the diesel engine's dynamic performances, influenced by system's conditions, structure and control parameters, so as to optimize the designated parameters (Yun and Jiang, 2004).

THE WORKING PRINCIPLE OF SERVO OIL COMMON RAIL SYSTEM

The servo oil common rail system is shown in Fig. 1, mainly includes control system, cylinder control components (Cylinder Control Module, CCM), the servo oil pump with controller, oil collector, servo common rail pipe, pressure sensor, safety valves and exhaust valves and other parts. The entry of the servo oil common rail pipe is connected to the collector by two vertical tubes, while the exits are connected to the inlet of the common rail solenoid valves of the exhaust control unit (Vent Control Unit, VCU) located at the top of them. The lubricating oil in the servo oil pump entry is from the main bearing lubricating oil, it wills inflow into the collector after being charged. The collector is equipped with safety relief valve, when the oil pressure is more than 23 MPa, the safety valve will open and the oil will be discharged into the crankcase.

Each servo pump is equipped with a pressure controller, the cylinder control module CCM controls the pressure controller according to the rail pressure measured by the pressure sensor and the setting servo oil rail pressure of the control system based on the diesel engine load, so as to control the servo oil pump to take a variable displacement adjustment, then keep the rail pressure around the set value (10-20 MPa) (Yuan *et al*., 2012; Teng and McCandless, 2005).

Figure 2 is the exhaust control principle diagram of the servo oil system, the traditional diesel engine opens the exhaust valve through the hydraulic oil driven by cam shaft, it can't control the opening time and closing speed of the exhaust valve accurately when the exhaust valve opens and closes, while the servo system can make an accurate exhaust timing control by controlling the moment of electricity and the electricity pulse width of the common rail solenoid valve. Accurate timing can affect

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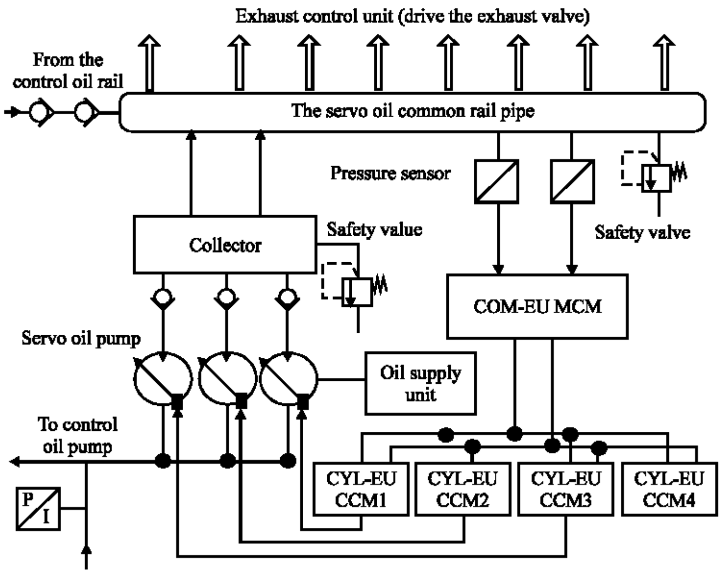

Fig. 1: Working principle of common rail servo system

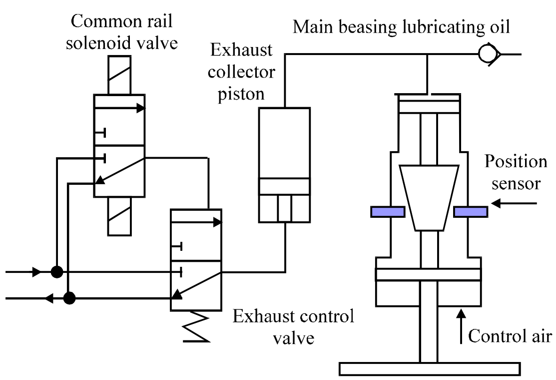

Fig. 2: Working principle diagram of exhaust control

the temperature and quantities of the exhaust gas accessed into the supercharger and have a positive impact on the performance of the diesel engine (Yang *et al*., 2012).

MATHEMATICAL MODEL OF THE SERVO OIL COMMON RAIL SYSTEM

The servo oil common rail system simulation conducts a research based on the modeling of the liquid flow mathematical equations, including the mass conservation equation, momentum conservation equation and equation of state, etc. In order to adopt practical solution to simulate and solve the servo oil common rail system, in this article, the modeling of the system made the following assumptions (Yu *et al*., 2002; Seykens *et al*., 2005):

- The lubricating oil keeps a constant temperature in the working process of the whole system, not considering the influence of the heat change on system
- The lubricating oil in the working process of the whole system is considered as incompressible meson and makes a one dimensional unsteady flow
- The chamber volume is centralized, without considering the pressure propagation time, everywhere has an equal pressure under the same moment
- Not considering the resistance and cavitation of the lubricating oil flowing in the pipeline
- Not considering the leakage and elastic deformation of various components

Servo oil pump: The servo oil pump, also known as "the servo oil booster pump" (7RT-flex 60C diesel engine has three servo oil pumps), is an axis syncline disc variable piston pump which is driven by the crankshaft via two intermediate gears to provide power, meet its rotation speed and delivery requirements, at the same time the adjustment of variable displacement can be done by changing the angle of the inclined plate.

- Single servo pumps plunger displacement equation:

$$S_p = R_0 \sin\alpha \cos\beta / \cos\gamma \tag{1}$$

where, S_p is plunger displacement; R_0 is centerline of the shaft to the pump cylinder centerline average distance; α is swash plate angle of inclination; β is cylinder corner; γ is an angle between pump cylinder and shaft.

- The flow of lubrication oil within single plunger cavity meets the requirement of continuity equation (Fan *et al*., 2011):

Eq. Missing (2)

$$Q_{c\_in} = \varepsilon_2 \mu_{c\_in} \sqrt{\frac{2}{\rho}|P_1 - P_c|} \quad (3)$$

Eq. Missing (4)

$$\rho = -0.001201p^2 + 0.6002p + 822.1 \quad (5)$$

$$\varepsilon_1, \varepsilon_2 = \begin{cases} 1, \text{ plunger open} \\ 0, \text{ other condition} \end{cases} \quad (6)$$

where V_1 is a single plunger cavity volume; E is lubricating oil elastic modulus which is a certain value (Zhang *et al*., 2005) taken by the empirical formula; ñ is lubricating oil density; P_1 is piston cavity pressure; A_1 is piston area; μ_s, A_s is servo pump inlet flow coefficient and area; P_s is servo pump into the oil pressure; Q_{c_in} is into the collector flow; μ_{c_in}, A_{c_in} is collector inlet flow coefficient and area; P_c is collector pressure; ε_1, ε_2 are step functions.

Servo oil collector: For the collector, the inflows and outflows of the lubricating oil meet continuity equation as follows:

$$\frac{V_c}{E}\frac{dP_c}{dt} = \sum Q_{c\_in} - Q_{r\_in} - \varsigma_1 \mu_{saf} \sqrt{\frac{2}{\rho}P_c} \quad (7)$$

$$Q_{r\_in} = \mu_{c\_out} A_{c\_out} \sqrt{\frac{2}{\rho}|P_c - P_r|} \quad (8)$$

$$\varsigma_1 = \begin{cases} 1, \text{ Safety valve open} \\ 0, \text{ Safety valve close} \end{cases} \quad (9)$$

where V_c is the collector volume; E is lubricating oil elastic modulus; Q_{r_in} is the flow into the servo oil rail; μ_{saf}, A_{saf} is flow coefficient and flow area of safety valve of collector; æ$_1$ is step function; μ_{c_out}, A_{c_out} is discharge coefficient and area of safety valve of collector; P_r is servo oil rail pressure.

Servo oil common rail pipe: In view of the servo oil common rail pipe, the inflows and outflows of the lubricating oil satisfy the continuity equation:

$$\frac{V_r}{E}\frac{dP_r}{dt} = (\tau_1\mu_1 A_1 - \tau_2\mu_2 A_2)\sqrt{\frac{2}{\rho}|P_k - P_r|} + Q_{r\_in} - Q_{r\_out} \quad (10)$$

$$\tau_1 = \begin{cases} 1, P_r < 8\text{MPa} \\ 1, P_r < 8\text{MPa} \end{cases} \quad \tau_1 = \begin{cases} -1, P_k \le P_r \\ 1, P_r > P_r \end{cases} \quad (11)$$

where V_r is the volume of servo oil common rail pipe; E is lubricating oil elastic modulus; Q_{r_out} is the servo fuel common rail discharge flow; μ_1, A_1 is flow coefficient and flow area of reducing valve between control the fuel rail and servo oil rail; P_k is control oil rail pressure μ_2, A_2 is flow coefficient and flow area of check valve between control the fuel rail and servo oil rail; τ_1, τ_2 are step functions.

Exhaust control module: The exhaust control module in the servo oil common rail system, as shown in Fig. 2, including exhaust control unit VCU and exhaust valve (Coppo *et al*., 2004):

- The VCU control piston equation of motion:

$$m_v \frac{d^2 h_v}{dt^2} = A_v (P_r - P_v) \quad (12)$$

- The continuity equation of the piston chamber of the exhaust control unit:

$$\frac{V_v}{E}\frac{dP_v}{dt} = A_v \frac{dh_v}{dt} + \eta_1 \mu_m A_m \sqrt{\frac{2}{\rho}|P_m - P_v|} - \eta_2 \mu_{v\_saf} A_{v\_saf} \sqrt{\frac{2}{\rho}P_v} - A_e \frac{dh_e}{dt} \quad (13)$$

$$\eta_1 = \begin{cases} 1, P_m > P_v \\ 0, P_m \le P_v \end{cases} \quad \varsigma_1 = \begin{cases} 1, \text{ Safety valve open} \\ 0, \text{ Safety valve close} \end{cases} \quad (14)$$

where, V_v is VCU control piston cavity volume; E is lubricating oil elastic modulus; μ_m, A_m is flow coefficient and flow area from main bearings to VCU's one-way valve; P_m is the main bearing oil pressure; μ_{v_saf}, A_{v_saf} is flow coefficient and flow area before the exhaust valve safety valve; A_e is exhaust valve oil in the role of area; h_e is the stem lift of exhaust valve; ç$_1$, ç$_2$ are step functions.

- The equations of motion of the exhaust valve:

$$m_e \frac{d^2 h_e}{dt^2} = A_e P_v - A_{as} P_{as} \quad (15)$$

$$P_{as} = \frac{P_{as} V_{as0}}{V_{as}} \quad (16)$$

where m_e is the quality of the exhaust valve stem; A_{as} is air action area of exhaust valve; P_{as} is control air pressure of exhaust valve; P_{as0} is control air initial pressure of exhaust valve; V_{as} is control the air chamber volume of exhaust valve; V_{as0} is the initial volume of the exhaust valve air chamber.

CALCULATION OF SIMULATION

Model building: The simulate calculation is based on the 7RT-flex 60C marine type electronically controlled common rail diesel engine of the Wartsila company. Build a servo oil common rail system simulation model in MATLAB/Simulink, as shown in Fig. 3, 4 and 5.

The model is divided into four subsystems, including the servo pump, collector, servo oil common rail and exhaust control. Among them, the exhaust control subsystem includes the VCU model and the exhaust valve model. The speed of the servo oil pump after being driven by the growth speed gear acts as the input of the servo oil common rail system, finally get the servo oil pump pressure, servo oil common rail pipe pressure, VCU control piston displacement, open speed of the exhaust valve and some other parameters by calculation.

Parameter initialization: Before simulation, the parameters need to be initialized. For the servo oil pump, the initial pressure of the plunger chamber is the pressure of the main bearing lubricating oil, a plunger located in the initial position when oil is pressed, the position of other plungers will have a corresponding angle lagging. Assumed that there is a certain initial oil pressure in the collector and the servo oil common rail pipe and the pressure in oil collector is slightly higher than the servo oil common rail pipe. For VCU control piston, assumed that it is located in the situation when the common rail electromagnetic valve is closed. For exhaust valve, assumed that the upper part of it suffers the pressure from the main bearing lubricating oil and closed. In the process of calculation, the MATLAB/Simulink use an ode4 algorithm and set the calculation step length to be 0.00001, the load of the diesel engine to be 99.97%, the rotate speed of the servo oil pump to be 2052 rpm, the angle between oil cylinder and pump shaft to be 10°, the

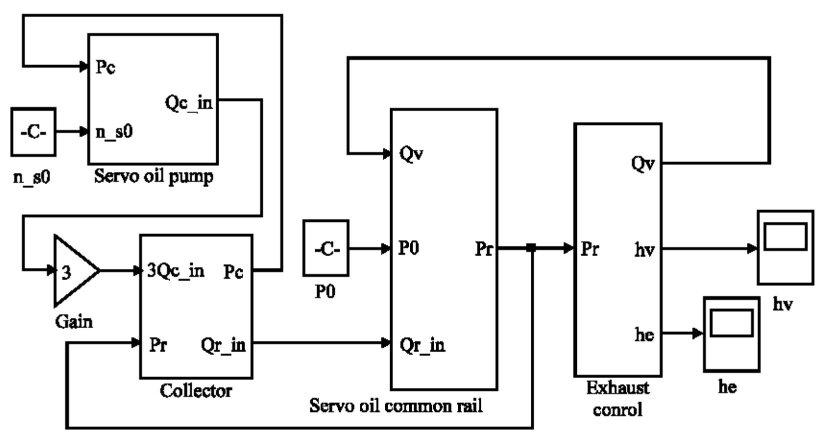

Fig. 3: Simulation diagram of servo rail

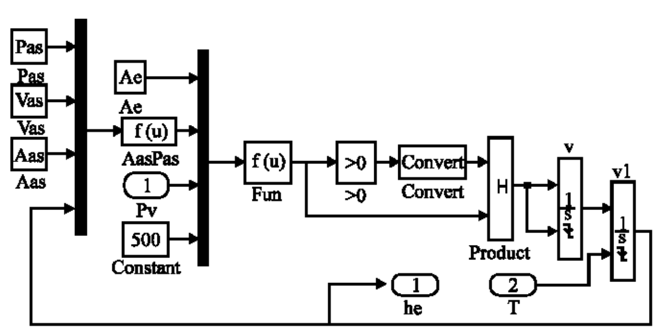

Fig. 4: Simulation diagram of exhaust

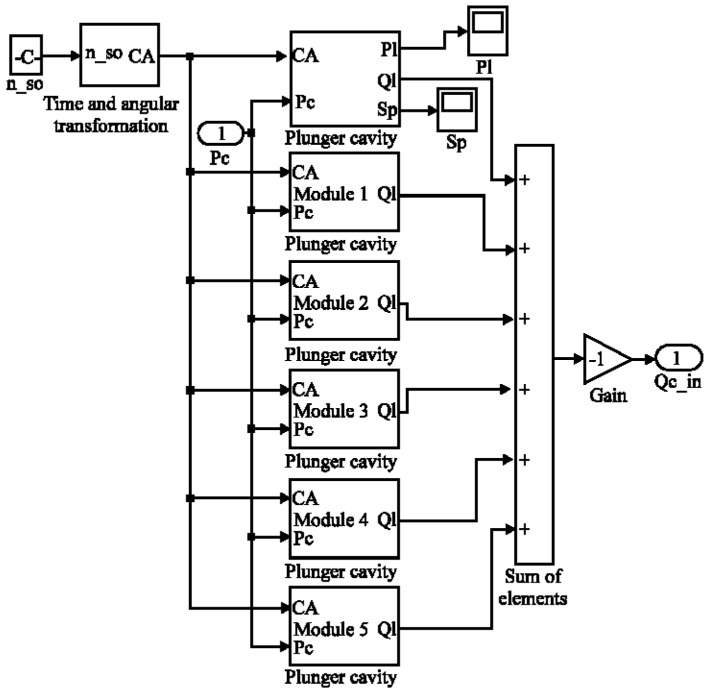

Fig. 5: Simulation diagram of servo oil pump

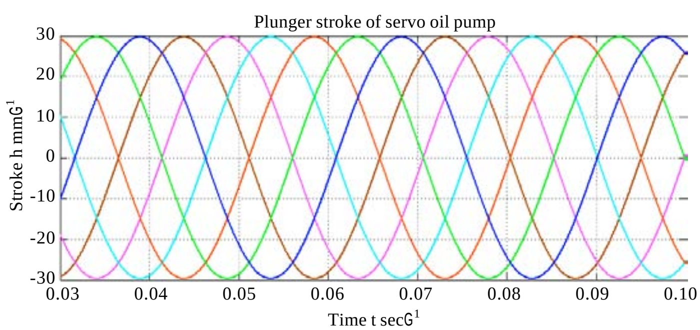

Fig. 6: Plunger stroke of servo oil pump

angle of swash plate to be 17°, the volume of the collector and the servo oil common rail pipe to be 0.5m3 and 3m3 respectively.

Analysis of simulation results: The simulation calculation can analyze and judge that if each parameter change trend of the diesel engine is consistent with theoretical analysis in a working cycle from the servo oil pump plunger displacement and piston chamber pressure, the pressure of the collector and the servo oil common rail pipe, the exhaust control piston displacement and the cavity pressure, exhaust valve lift and so on. The specific results of the analysis described as below.

Because there are six cylinders in the servo oil pump, oil cylinder arranged along the direction of pump shaft circumferential uniformly, we can see that the servo oil pump plunger displacement changes similar to the sine function from Fig. 6 and the output is stable, the kind of waveform makes the back pressure fluctuation be reduced after an overlap generated by the pump outlet pressure waves and is good for the servo oil common rail pressure being stable.

Figure 7 is the result of simulating the servo oil pump plunger and plunger chamber pressure, simulation begins when the plunger is at the starting of the oil compression stroke, in the diagram we can see that the plunger

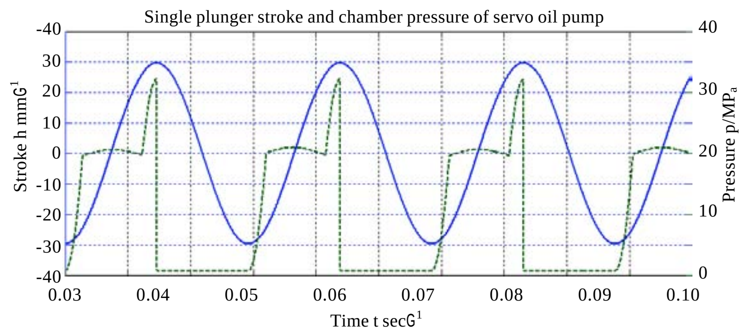

Fig. 7: Single plunger stroke and chamber pressure of servo oil pump

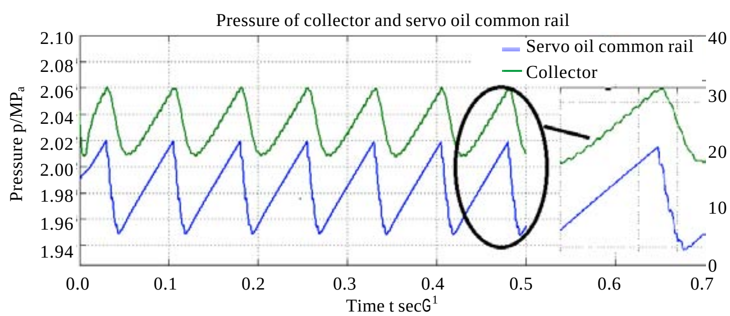

Fig. 8: Pressure of collector and servo rail

chamber pressure increases quickly with the plunger motions until the piston chamber and oil discharge outlet are interlinked, at this time the pressure of the plunger chamber rises slowly. The plunger flow will be less than the oil discharge outlet flow with the movement of the plunger and the plunger chamber pressure decreases. When the plunger turns around the oil discharge outlet, the plunger is still in the stroke of compression, the pressure rises quickly. When the plunger turn around the compression stroke, linked with oil suction port, the pressure drops rapidly. The curve of simulation accords with the actual working process of the plunger pump.

Figure 8 is the curve for the collector and the servo oil common rail pipe pressure simulation, as can be seen from the figure, the servo oil pump supply the servo oil continuously, the pressure of the collector and servo oil pipe increases gradually and the pressure increase rate of the common rail pipe is less than the collector because of the greater common rail pipe volume. The common rail pipe pressure begins to decline when the first pulse width is arrived, with the reduction of rail pressure, the servo oil flow into the collector and guard the pressure against being depressed in the common rail pipe. When the exhaust valve opens fully, the pressure is no longer falling and increase again with the servo oil flowing into the collector and the common rail pipe, preparing for the next work. In a circulation, it is necessary to drive exhaust valve for seven times, therefore seven times of pressure fluctuations take place.

In the simulation initialization, assumed that the first cylinder is located in the injection starting point, we can calculate that the power time of the forth cylinder exhaust common rail solenoid valve is about 0.17s from the angle of the fuel injection moment and exhaust moment and rotating speed. Figure 9 shows a curve of the VCU control piston chamber pressure of the forth cylinder. Before the arrival of pulse width, the control piston chamber suffer a stable pressure which is 5.1MPa, from the main bearing lubricating oil, when the pulse width arrives, the servo oil rail pressure have an effect on the bottom of the control piston suddenly and bring about a sharp fluctuations. With the movement of the control piston, the pressure fluctuation in the upper chamber decreases and tends to be stable finally. Because the simulation considering the gravity influence of the control piston, results in that the pressure of the double piston chamber is slightly lower than the servo oil rail pressure.

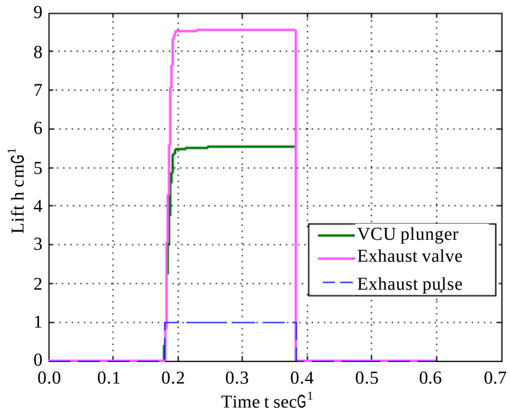

Fig. 9: Displacement of single VCU plunger and exhaust valve

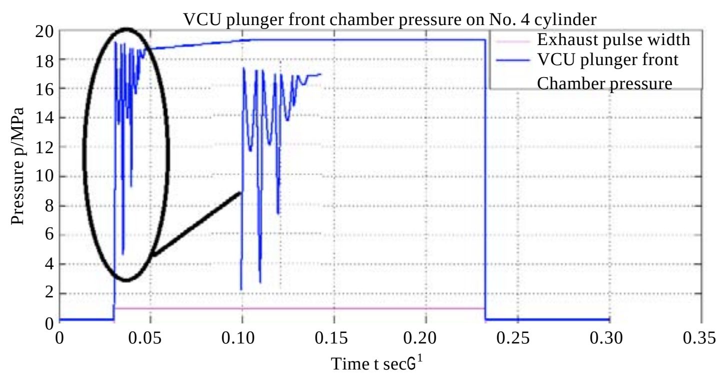

Fig. 10: Pressure simulation diagram of VCU plunger front chamber

Figure 10 shows a simulation curve of the control piston and the exhaust valve displacement, from the diagram, we can see that the control piston and the exhaust valve begin to action when the arrival of the exhaust pulse width, the upper chamber of the control piston is effected a pressure of 5.1 bar from the main bearing lubricating oil and the bottom of the exhaust valve is effected with control air of 7 bar, but due to the compressibility of the liquid is much less than the gas, the movement speed of the exhaust valve is greater than the control piston. When the exhaust pulse ended, the exhaust valve and control piston is reset rapidly, the result of the lift changes by simulating is reasonable.

CONCLUSIONS

This study takes a research on the working principle of the servo oil common rail system of the 7RT-flex 60C diesel engine and adopts the block modeling method to establish simulation model of the servo oil common rail system, including the servo pump of common rail system, collector, servo oil common rail pipe, VCU control piston and exhaust valve. Simulation results show that the model can reflect the working process of the common rail system very well, simulation of the rail pressure, the flow in the system, the exhaust valve lift and the change of other parameters accord with actual work process, not only meet the simulation requirements, but also establish the theoretical basis for the further design and parameters research of the servo oil common rail system.

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