# Research on Servo Oil Common Rail System for Marine Diesel Engine

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***Abstract* - Wartsila RT- flex 60C diesel engine is selected as the research object to analyze the characteristics of servo oil common rail system. Building the system simulation model, and design various mathematical model of the system. Based on the continuity equation and the motion equation of the fluid mechanics, the mathematical model with blocks is established on servo pumps, collector, common rail pipe of servo oil module. The volume of collector and servo oil common rail has greater effect on fluctuation of servo oil common rail system pressure. It is suggested that a greater collector volume or a segmented servo oil common rail tube be used to satisfying the performance of servo oil common rail system.**

***Keywords - servo oil common rail; simulation; model; diesel;***

## I. INTRODUCTION

To meet the higher economic and environmental demands of today's shipping industry, marine large low-speed electronically controlled common rail diesel system has been widely used. Over the past few years , researches on diesel common rail system mainly focused on the fuel common rail system, but less on the servo oil common rail system in our country, the literature [1] only simulated the exhaust valve in loop , the literature [2] only established a simplified model of the exhaust system . But they didn't establish a detailed servo oil common rail system .So, further researches need to be done. Based on the simulation model of the servo oil common rail system, this article simulated the system operating characteristics, analyzed the impact of relevant parameters on the servo oil common rail system, which has a certain significance for the independent research and development of electronically controlled servo oil common rail system [3-4].

## II. WORKING PRINCIPLE OF SERVO OIL COMMON RAIL SYSTEM

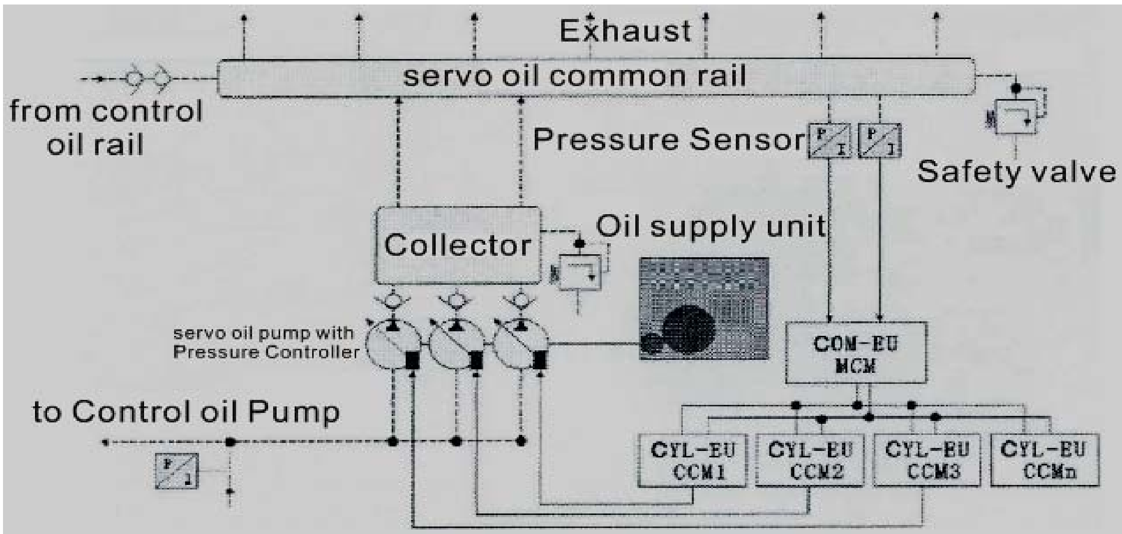


Fig.1. The principle of servo oil common rail system

The principle of the RT-flex60C's servo oil common rail system is shown in figure 1.The servo pump is driven directly by the crank shaft through the gear, pumping the servo oil to the collector. After the collector, the servo oil is fed into the servo oil common rail, preparing for the exhaust valve control. The whole system is divided into the servo pump, collector, servo oil common rail, exhaust control piston, VCU piston chamber, exhaust valve and other sub-models according to the position and hydraulic connection of each component, in the process of establishing Simulink simulation model of the servo oil common rail system of this type diesel [5-7].

*A. The Mathematical Model*

The mathematical model building of each component of diesel servo oil common rail system is based on the fluid dynamics and Newton's second law. In order to facilitate the calculation, in the modeling process, this article assuming the system is as follows:

ignore the local loss of the inlet and outlet and the system leakage;
ignore the frictional resistance;
the states of servo oil common rail and collector reach equilibrium instantaneously, and the state of each position is equal;
the temperature of the system fluid remain unchanged during the fluid flowing, and the flowing is unsteady incompressible flow.

In a certain container, the sum of the amount of compression due to pressure and the amount of liquid out flowing is equal to the amount of liquid entering. According to this principle, we can get the mathematical model of each sub system of the servo oil common rail system respectively[8].

*B. The Motion Equation of Single Plunger in Oil Pump*

The servo oil pump is swash plate plunger pump, whose motion equation is as follows:

$$H_{sz} = R_0 \sin\alpha \cos\varphi / \cos\gamma \tag{1}$$

Where, $H_{sz}$ is plunger displacement; $R_0$ is the average distance of the centerline of the pump cylinder to the centerline of the pump shaft; $\alpha$ is the swash plate tilt angle; $\beta$ is angle turned of the cylinder; $\gamma$ is the angle between the pump cylinder and the pump shaft.

The fluid flow continuity equation of single plunger in servo oil pump:

$$\frac{dP_{sz}}{dt} = \rho c^2 (A_{sz}\frac{dH_{sz}}{dt} + \varepsilon_1 \mu_{si} A_{si}\sqrt{\frac{2}{\rho}\left|P_{sz} - P_{si}\right|} - \frac{\pi d_{sz}(P_{sz} - P_{feed})\delta_{sz}^3}{12\eta L_{sz}} - Q_{ssi}) / V_{sz} \tag{2}$$

$$Q_{ssi} = \varepsilon_2 \mu_{ssi} A_{ssi}\sqrt{\frac{2}{\rho}\left|P_{sz} - P_{ssi}\right|} \tag{3}$$

Where, $P_{sz}$ is the plunger chamber pressure; $A_{sz}$ is the plunger area; $\mu_{si}$ is the inlet flow coefficient of servo pump; $V_{sz}$ is the plunger chamber volume; $A_{si}$ is the inlet area of the servo pump; $P_{si}$ is the inlet pressure of the servo pump; $d_{sz}$ is the plunger diameter of the servo pump ; $P_{feed}$ is the supply oil pressure of the servo pump; $\delta_{sz}$ is the plunger interval; $\eta$ is the dynamic viscosity; $L_{sz}$ is the plunger diameter; $Q_{ssi}$ is the flows entering the collector; $\mu_{ssi}$ is the inlet and outlet coefficient of the collector; $A_{ssi}$ is the inlet and outlet flow area; $P_{ssi}$ is the pressure of the collector;

$$\varepsilon_1、\varepsilon_2 = \begin{cases} 1, & \text{The plunger interlinked with the inlet} \\ 0, & \text{ot her} \end{cases}$$

*C. The Fluid Flow Continuity Equation of the Collector*

$$\frac{dP_{ssi}}{dt} = \rho c^2 (\sum Q_{ssi} - \sum Q_{sgi} - \varsigma_1 \mu_{ssi\_saf} A_{ssi\_saf}\sqrt{\frac{2}{\rho}P_{ssi}}) / V_{ss} \tag{4}$$

$$Q_{sgi} = \mu_{ssi} A_{sso}\sqrt{\frac{2}{\rho}\left|P_{ssi} - P_{sg}\right|} \tag{5}$$

Where, $Q_{sgi}$ is the flows entering the servo oil common rail; $\mu_{ssi\_saf}$ is the flow coefficient of the collector safety valve; $A_{ssi\_saf}$ is the flow area of the collector safety valve; $V_{ss}$ is the volume of the collector; $A_{sso}$ is the outlet area of the collector; $P_{sg}$ is the pressure of servo oil common rail;

$$\varsigma_1 = \begin{cases} 1, & \text{The relief valve opening} \\ 0, & \text{The relief valve closed} \end{cases}.$$

*D. The Fluid Flow Continuity Equation Of The Servo Oil Common Rail*

$$\frac{dP_{sg}}{dt} = \rho c^2 ((\tau_1 \mu_1 A_1 - \tau_2 \mu_2 A_2)\sqrt{\frac{2}{\rho}\left|P_{cg} - P_{sg}\right|} + \sum Q_{sgi} - \sum Q_{sgo}) / V_{sg} \tag{6}$$

Where, $\mu_1$ is the flow coefficient of the reducing valve between the control oil rail and the servo oil rail; $A_1$ is the flow area of the reducing valve; $\mu_2$ is the flow coefficient of the check valve from the servo oil rail to the control oil rail; $A_2$ is the flow area of the check valve; $P_{cg}$ is the pressure of the control oil rail; $P_{sg}$ is the pressure of the servo oil rail; $Q_{sgi}$ is the flow entering the servo oil rail; $Q_{sgo}$ is the flow out flowing the servo oil rail; $V_{sg}$ is the volume of the servo oil common rail ; $\tau_1 = \begin{cases} 1, & P_{sg} < 8MPa \\ 0, & P_{sg} \geq 8MPa \end{cases}$; $\tau_2 = \begin{cases} -1, & P_{cg} \leq P_{sg} \\ 0, & P_{cg} > P_{sg} \end{cases}$.

*E. The Motion Equation of the Exhaust Valve*

$$\frac{d^2 h_{pg}}{dt^2} = (A_{ph} P_{ks} - A_{pq} \frac{P_{pq0} V_{pq0}}{V_{pq}}) / m_{pg} \qquad (7)$$

Where, $h_{pg}$ is the exhaust valve lift; $A_{pq}$ is the air active area of the exhaust valve; $P_{pq0}$ is the initial pressure of the control air in the exhaust valve; $V_{pq}$ is the volume of the control air chamber in the exhaust valve; $V_{pq0}$ is the initial volume of the control air chamber in the exhaust valve; $m_{pg}$ is the mass of the exhaust valve stem.

*F. The Motion Equation of the VCU Exhaust Control Piston*

$$\frac{d^2 h_{ks}}{dt^2} = A_{ks}(P_{sg} - P_{ks}) / m_{ks} \qquad (8)$$

Where, $h_{ks}$ is the displacement of the VCU control piston ; $A_{ks}$ is the area of the VCU control piston; $P_{ks}$ is the pressure of the VCU control piston chamber; $m_{ks}$ is the mass of the of the VCU control piston.

*G. The Fluid Continuity Motion Equation of the Exhaust Control Piston*

$$\frac{dP_{ks}}{dt} = \rho c^2 (A_{ks} \frac{dh_{ks}}{dt} + \eta_1 \mu_3 A_3 \sqrt{\frac{2}{\rho} |P_{mb} - P_{ks}|} - \eta_2 \mu_4 A_4 \sqrt{\frac{2}{\rho} P_{ks}} - A_{ph} \frac{dh_{pg}}{dt}) / V_{ks} \qquad (9)$$

Where, $P_{ks}$ is the pressure of the VCU control piston chamber; $A_{ks}$ is the area of the VCU control piston ; $h_{ks}$ is the displacement of the VCU control piston; $\mu_3$ is the flow coefficient of the check valve from the main bearing to the VCU; $A_3$ is the flow area of the check valve; $P_{mb}$ is pressure of the main bearing lubricating oil; $\mu_4$ is the flow coefficient of the safety valve before exhaust valve; $A_{ph}$ is the flow area of the safety valve; $A_{ph}$ is the lubricating oil active area of the exhaust valve; $h_{pg}$ is the lift of the exhaust valve stem; $V_{ks}$ is the volume of the VCU control piston chamber; $\eta_1 = \begin{cases} 1, & P_{mb} > P_{ks} \\ 0, & P_{mb} \leq P_{ks} \end{cases}$; $\eta_2 = \begin{cases} 1, & \text{safety valve open} \\ 0, & \text{safety valve closed} \end{cases}$.

## III. SETTING OF BOUNDARY PARAMETERS IN THE SIMULATION

In this system, the initial pressure of the servo oil pump is the main bearing lubricating oil pressure. The oil pressure of the servo oil common rail should be slightly lower than that of the collector. In the Matlab/Simulink simulation, used Fourth-order Runge-Kutta method, the step is 0.00001. The load of the diesel engine is setting to be 99.9%, the tilt angle of servo pump swash plate is 16.5°,the volume of the servo oil common rail and the collector is respectively 0.5m$^3$and 0.25 m$^3$.

*A. Simulation Experiment*

Figure 2 is the general block diagram of the servo oil common rail system. The input of servo oil common rail system is the servo oil pump speed after the transmission of the growth gear. Finally, obtained the displacement of the VCU control piston and the open speed of the exhaust valve.

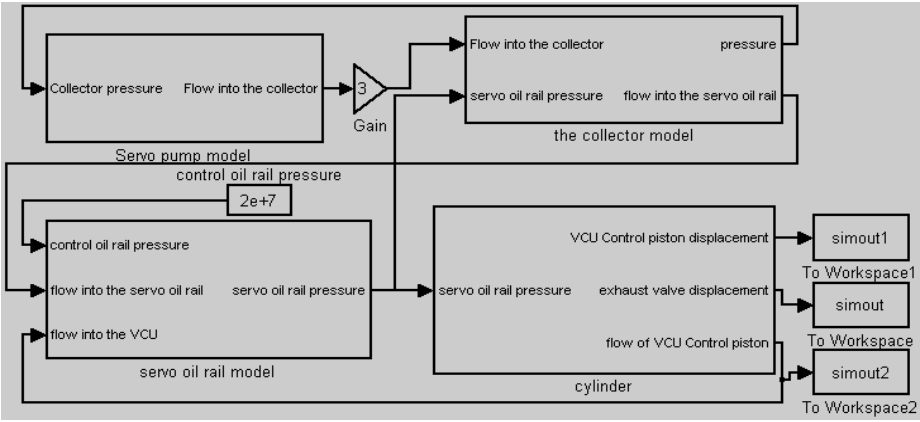


Fig.2: the simulation block diagram of the servo oil common rail system.

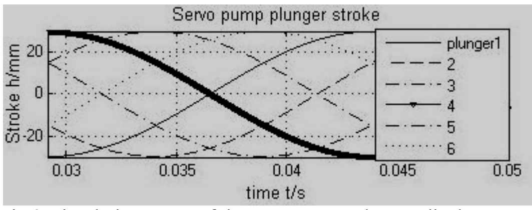


Fig.3: simulating curve of the servo pump plunger displacement.

Figure 3 is the plunger lift simulating curve of the servo pump. It can be seen plunger lift curve soft, output stable, and the pressure curves intertwined. The pressure wave generated by the each plunger superimposed on each other, reducing the servo oil rail pressure fluctuations. It's very conducive to the stability of rail pressure in the servo oil common rail system.

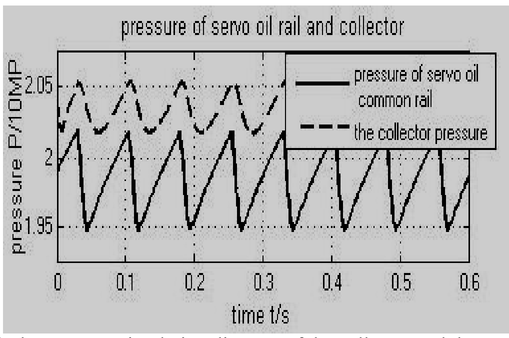


Fig.4: the pressure simulating diagram of the collector and the servo oil common rail.

Just as shown in figure 4, the pressure of the collector is slightly higher than that of servo oil common rail due to that the oil in collector is directly from oil pump, with less pressure drop. In this figure, the pressure of the collector experienced eight times rise corresponding to eight times servo oil pump supplying oil to the collector. And there are eight times pressure drop of servo oil common rail respectively caused by eight times exhaust; Rail pressure fluctuations less than 0.05MPa, the simulating results is very good, achieving the purpose of the simulation. Figure 5 is the exhaust valve lift cure. Valve open and close response rapidly, meeting the actual exhaust valve operating characteristics.

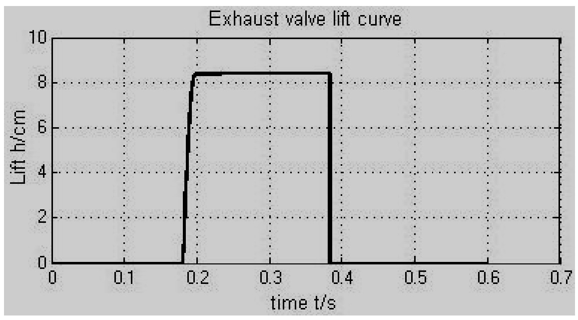


Fig.5:the exhaust valve lift simulating cure.

TABLE 1 THE COMPARISON OF EXPERIMENTAL AND SIMULATED DATA.

| diesel engine rotational speed /(r/min) | diesel engine load /% | Exhaust duration /s | experimental values of Servo oil rail pressure /MP | Simulation values of Servo oil rail pressure /MP | Rail pressure fluctuations /MP |
|---|---|---|---|---|---|
| 99.6 | 75 | 0.2012 | 16.40 | 16.536 | 0. 02 |
| 108.7 | 89.9 | 0.2024 | 18.00 | 18.273 | 0. 04 |
| 114 | 99.9 | 0.2035 | 19.90 | 20.254 | 0. 04 |

Comparison to table 1 shows that, in a typical condition simulation experiment, the error of the simulation results is small, reflecting the system simulation model is very consistent with the actual situation.

*B. The Effect of the Servo Common Rail System Parameters on the System*

The effect of the pump speed on the oil supply rate.

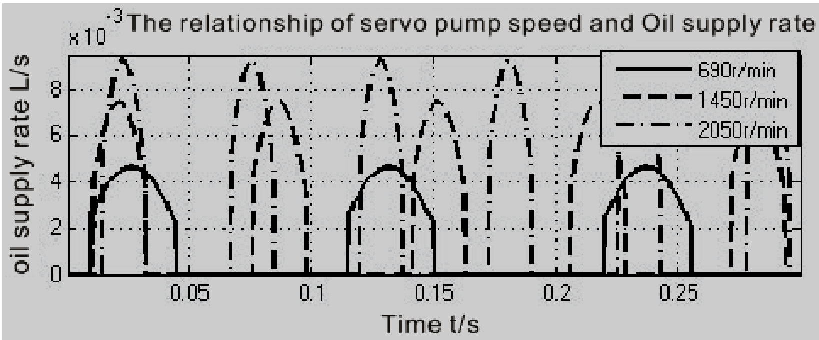


Fig.6:the relationship between servo oil pump speed and the oil supply rate

What figure 6 showing is, under a circumstance of the servo oil common rail system volume is a constant, the frequency and amount of oil changes with the servo pump speed increasing. From the preceding formula (2), we know oil supply rate depends on the flow coefficient, flow area and pressure difference of the valve. For a certain valve, the former two parameters are constant. So the pressure difference before and after the valve decides the oil supply rate. When the oil pump speed increases, the increased pressure within unit time increases, causing pressure difference between the oil supply pump and the collector becomes larger. So the oil supply rate increases. At the same time, due to the increasing of oil pump speed, the oil supply rate to the collector increases, slowing the pressure drop rate of plunger chamber after supplying oil to the collector comparing to under lower oil pump speed. At high oil pump speed, the oil supply rate increases and decline rapidly. Secondly, as the servo oil pump is swash plate pump, the angle between the oil inlet and the oil outlet is larger than the angle between the oil outlet and the oil inlet. So the beginning oil supply rate is different from the ending oil supply rate.

*C. The Effect of Collector and Servo Oil Common Rail Volume on the Pressure of Servo Oil Common Rail*

Figure 7 and figure 8 are respectively the impact of various collector and servo oil common rail volumes on the pressure fluctuations of servo oil common rail under the circumstance of constant diesel engine load and servo pump speed, rail pressure of servo system is 20MPa. From the figures, we know the smaller the volume, the greater the pressure fluctuations. Theoretically, greater collector and servo rail volume are more conducive to stable pressure. But large volumes will slow the pressure building rate in the engine starting process, and will make the pressure following property under various load poor. So under the premise of meeting the requirements of diesel starting, we should better to select larger buffer volume. In comprehensive consideration, the volumes of common rail and collector are respectively 0.5 and 0.25 cubic meters. Another good solution is dividing the rail into several segments, union each segment by valves. In the working process, open each segment progressively, thus can build pressure very soon. After building pressure, the volume can be expanded.

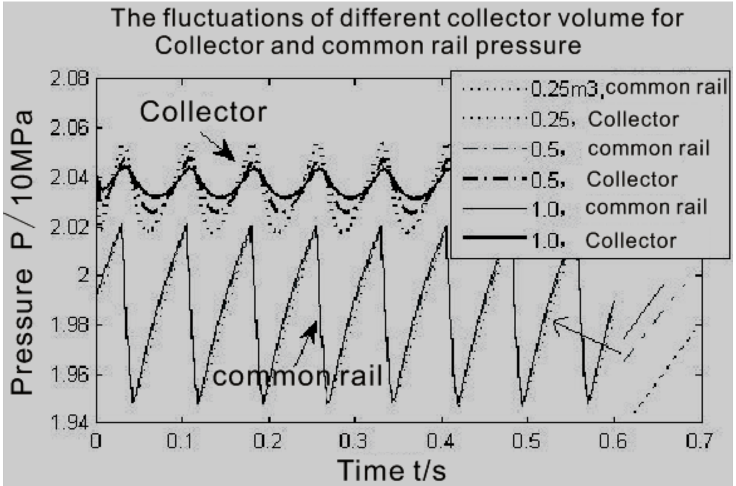


Fig.7: the impact of collector volume on the pressure stability

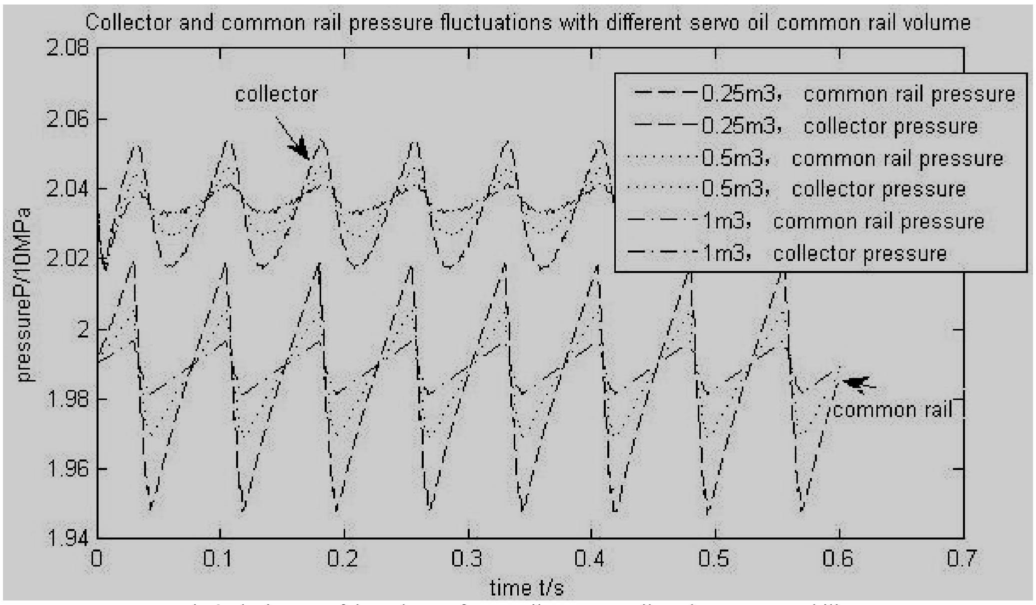


Fig.8: the impact of the volume of servo oil common rail on the pressure stability

Figure 9 is the impact of various diesel loads on the pressure fluctuations servo oil common rail with the volumes of servo oil common rail and collector constant. Every fall and rise of the curve represents the exhaust and servo pump supplying oil. It can be seen the greater the load, the greater the rail pressure fluctuation caused by the exhaust. This may be because with the load increasing, the servo pump speed increases, thus cause the amount and frequency of oil supply increase (seeing figure 6). The larger the single oil supply, the greater the effect on pressure fluctuation. The rail pressure fluctuations of this system are less than 5%, meeting the requirements of actual common rail system.

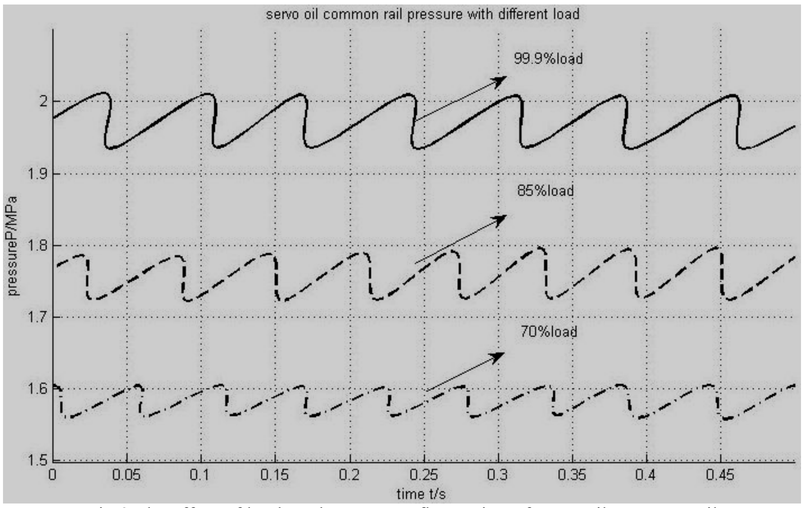


Fig.9: the effect of load on the pressure fluctuation of servo oil common rail

*D. The Effect of Various Diesel Loads on the Exhaust Valve*

The exhaust valve is an important component of a diesel engine. If the exhaust valve opens too early, it will reduce the power performance, poorer the economic property.

If it opens too late, it will reduce the amount of gas exhaust and the amount of air entering, thus will affect the engine performance. Seeing from figure 10, after the servo oil rail pressure decline due to engine load reducing, the exhaust valve from beginning open to fully open will delay, close will also delay. So the diesel engine performance will be impacted. To improve the diesel engine economic property, should advance the conduction time of common rail electromagnetic valve.

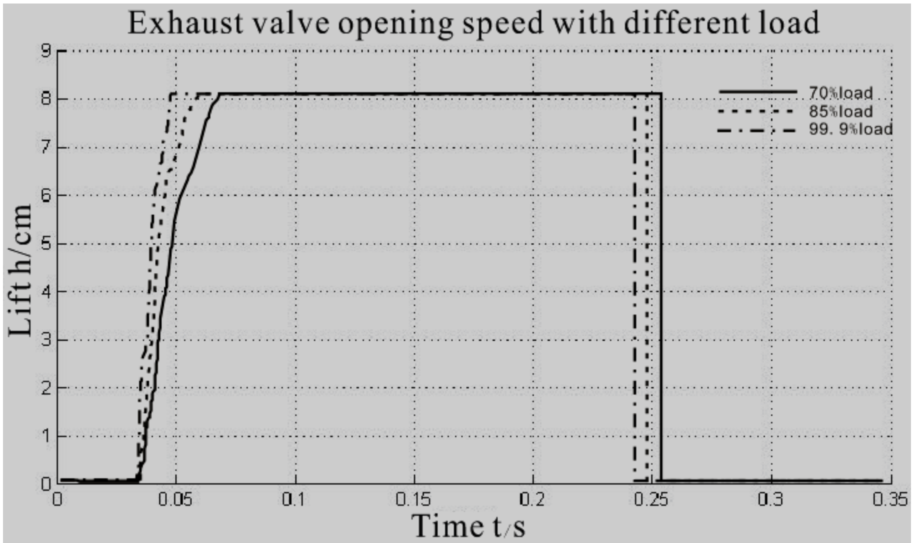


Fig.10: the effect of various diesel loads on the open speed of exhaust valve

IV. CONCLUSION

The servo oil common rail is the important component of an electronically controlled diesel engine. It has significant reference value for electronically controlled common rail servo system independent research to simulation model and simulation analysis. This paper's research work can be summarized as follows: if the servo pump rotational speed increases, the frequency and amount of oil supply will also increase; if the volumes of collector and servo oil common rail increase, the servo oil common rail pressure fluctuations will reduce, but to meet the diesel engine starting requirement of building pressure quickly, in comprehensive consideration, the volumes of common rail and collector are respectively 0.5 and 0.25 cubic meters, rail pressure fluctuations are less than 0.1MPa, meeting the system requirement; if the engine load increases, the rail pressure fluctuation will increase; if the engine load reduces, the exhaust valve from beginning open to fully open will delay, close will also delay; To improve the diesel engine economic property, should advance the conduction time of common rail electromagnetic valve.

## ACKNOWLEDGMENT

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