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Fuel engine injector control based on multiple search approach

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Abstract*—The main components of high-pressure common rail fuel injection system are high-pressure common rail fuel pump, common rail pipe and injector. The high-pressure fuel from the high-pressure oil pump is supplied to the injector through the high-pressure common rail oil pipe, which is precisely controlled by the electric control system and inject fuel into the cylinder. Generally, the power of the fuel engine is determined by the speed of the high pressure oil pump, and the key to improve the efficiency of it is to keep a constant pressure in oil pipe when high pressure oil pump is at a high speed. Therefore, methods need to be taken to control pressure in pipe when the cam rotates at a high speed, such as control speed of cam, adjust number of fuel injectors and the starting time of them in a cycle, with the aim of stabilizing internal pressure of oil pipe. In this study, we discretize the problem, establish the difference equation, and multiple searching method is adopted to find out the injector configuration mode which keep the pressure of the high-pressure oil pipe, with speed of, at 100MPa. Finally, we put forward a conjecture between the cam speed and the injector configuration, and carry out simulation verification.

***Keywords-* High pressure common rail fuel injection system; Dual injector system; Multiple-search method**

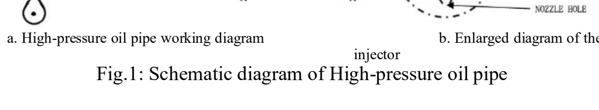
I. NTRODUCTION

With the improvement of social productivity and the increasing awareness of environmental protection and energy conservation, the reduction of automobile exhaust and the improvement of engine efficiency have become the development demand of the times. High pressure rail injection technology, as the main technology to reduce vehicle exhaust emissions, has been developed rapidly. The main components of high-pressure common rail fuel injection system are high-pressure common rail fuel pump, common rail pipe and injector. High-pressure fuel is supplied to the fuel nozzle through the common rail oil pipe and it comes from oil pump. Besides, electric control system precisely adjust oil pipe and oil pipe inject high-pressure fuel into the cylinder. The efficiency of fuel engine is mainly related to high pressure oil pump[1]. The main types of high pressure oil pumps in the market are produced by BOSCH company[1-5], Japan electric equipment company[6] and other companies[7-9], and its speed can reach $6000r/\min$ in general[10]. The work of fuel engine base on inflow and injection of fuel in oil pipe, but the intermittent working process will make the pressure in oil pipe under a unstable state and fuel injected would have deviation, thus affecting the engine's efficiency. Therefore, it is of great significance to establish a mathematical model to control the internal pressure of high pressure tubing for improving the engine efficiency.

Unfortunately, there would have accident if the fuel in oil pipe under high pressure, and if the fuel under a low pressure would cause extra loss of resources. So, it is necessary to maintain pressure in oil pipe near a certain value. For fuel engine, high-pressure oil pipe are generally equipped multiple injectors with the same working rule and a reducing valve is used to control pressure change in oil pipe. The working methods of the injector are divided into synchronous and asynchronous. When the injectors work synchronously, multiple search strategies can be used to find the optimal solution for the speed of oil pump that stabilizes pressure in oil pipe; When the fuel injectors work asynchronously, the multiple search strategy is used to find the speed of the oil pump and the time when the working time of fuel injectors separately to stabilize the pressure in the pipe. Because the asynchronous effect of the fuel injector is better and more complicated, we mainly study the problem of the asynchronous operation of the fuel injector.

In this paper, the differential equation model of the fuel engine system is first established to solve the configuration problem of the dual-injector nozzle system working asynchronously; however, when the high-pressure fuel pump rotates at high speed, it is difficult to maintain the pressure stability in the fuel pipe by only two injectors. In order to solve this problem, we use a Multiple search[11,12] strategy to find the number of injectors and their working methods that stabilize pressure in oil pipe at 100 MPa when high-pressure pump's speed reach $600 rad/s$, and we propose the conjecture of the regularity between the cam speed and the number of injectors and verified by simulation.

The structure of the article is as follows: In the section 2, a control model is established for the dual injectors of the fuel engine; The section 3 is the test results and discussion; conclusion is in Part 4.



a. High-pressure oil pipe working diagram b. Enlarged diagram of the injector

Fig.1: Schematic diagram of High-pressure oil pipe

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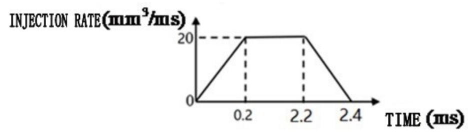

Fig.2: Schematic diagram of oil injection rate

II. MODELING OF DUAL FUEL INJECTOR STRATEGY FOR FUEL ENGINE

Taking a traditional double-injector high-pressure fuel system as an example, its working principle is shown in Fig. 1. Rotation of cam drives the plunger of oil pump move up and down to maintain pressure in high-pressure oil pipe. Because of it, taking the rotation axis of the cam as the origin and the vertical direction as the polar diameter, we established a polar coordinate system. When plunger hole under a higher pressure than it in oil pipe, plunger hole and oil pipe will be connected by opening check valve, and the oil enters oil pipe from A through injection of the plunger chamber of the high-pressure fuel pump. The main consist of injector is composed of a needle valve, a sealing retainer and a valve, and the injectors B and C start working and spraying oil outward when each working cycle reaches its own specific working time point. Besides, D is a one-way check valve and pressure in the high-pressure oil pipe is adjusted by it.

The known system parameters are as follows: the inner plunger hole's diameter is $D_0 = 5$ and plunger cavity's residual volume can be $V_0 = 20$ mm^3 when it reach top dead center'; as it moves to bottom dead center, plunger hole(including residual volume) will be filled with low-pressure oil whose pressure is $P_{\min} =$ 0.5 MPa and oil's mass is reset as $m_0 = 45.16$ g. Length of interior cavity of oil pipe is $L = 500$ mm, its radius $D = 10$ mm, and oil supply entrance A has a small hole whose diameter $D_1 = 1.4$ mm. As for needle valve, it has a injector whose diameter is $D_2 = 2.5$ mm, sealing retainer is a cone and its half angle of $\varphi = 9$, and the diameter of the lowermost injection hole is $D_3 = 1.4$ mm; the reducing valve at D closes when needle valve lift is 0, and it is opened and eject oil when needle valve lift is more than 0. The fuel injector works $n = 10$ times per second, and the duration of each work is $l = 2.4$ ms. The injector can only work ten times in 1 s. Let the injectors work periodically and the time of each work the interval is T. High-pressure own an initial pressure $P_0 = 100$ MPa.

In reality, the cam speed can reach up to 600 rad / s. In order to better stabilize the pressure in the high-pressure oil pipe and improve the working efficiency of the system, it often has multiple injectors. When the cam speed reaches the maximum value of 600 rad / s, how to set the number of injectors and the working time arrangement of the injectors to make the system work with the highest efficiency is the final problem to be solved urgently in this paper.

Note 1: The flow rate into and out oil pipe is $Q = CA\sqrt{2\Delta P/\rho}$ and Q is the amount of fuel flowing through hole per unit time (mm^3 / ms), $C = 0.85$ is the flow coefficient, A is the area of the small hole (mm2), the pressure difference between the small holes is ρ (mg / mm3).

There are two injectors with the same working rules in the dual-injector fuel engine. The working methods of the injectors are divided into two types: synchronous and asynchronous. This paper mainly focuses on the asynchronous working mode of the injectors. If the residual volume in the plunger hole of the pressure oil pump is V_0, the volume of the plunger hole at time t is as follows:

$$V_1(t,w) = V_0 + \pi D^2/4 \cdot (h_{\max} - h(t,w)) \tag{1}$$

Among them, the maximum height that the plunger hole can rise is $h_{\max}$, the height of the plunger rise at time t is $h(t,w)$, the cam speed is w, and, hereinafter, $V_1(t,w)$ is denoted to $V_1(t)$, Let the mass of oil in the plunger hole at time t is as follows:

$$V_1(t,w) = V_0 + \pi D^2/4 \cdot (h_{\max} - h(t,w)) \tag{2}$$

$$L(t) = \begin{cases} 1, \operatorname{mod}(w \cdot t, 2\cdot\pi) \neq 0 \\ 0, \operatorname{mod}(w \cdot t, 2\cdot\pi) = 0 \end{cases}$$

Among them, $m_{in}(t)$ is mass of fuel injected into oil pipe at t in Δt time, and m_0 is the total mass of fuel in cavity as plunger reaches the lowest point and refilled. Let the cross-sectional area of A is $S_A = \pi D_1^2/4$, mass of fuel in oil pipe is $m(t)$, volume of oil pipe is $V = \pi \cdot L \cdot D^2/4$. Thus, the density of the oil in the plunger hole is $\rho_1(t) = m_1(t)/V_1(t)$, density in oil pipe is $\rho(t) = \frac{m(t)}{V}$, pressure of oil inlet is $P_1(t) = G(\rho_1(t))$, pressure in oil pipe is $P(t) = G(\rho(t))$. Introduce 0-1 variable:

$$I(t) = \begin{cases} 1, P_1(t) > P(t) \\ 0, P_1(t) < P(t) \end{cases} \tag{3}$$

Available, the oil flow into oil pipe at the time t:

$$Q_{in}(t) = I(t) \cdot C \cdot S_A \cdot [2(P_1(t) - P(t))/\rho_1]^{0.5} \tag{4}$$

Thus, the mass of the oil flow into in oil pipe per unit time is:

$$m_{in}(t) = Q_{in}(t) \cdot \rho_1(t) \cdot \Delta t \tag{5}$$

For oil pipe, when fuel injector is opened, the oil output area is determined by the lifting height of the needle valve and the area of the injector. Let the half angle of the cone be α, so the oil output area restricted by the needle valve is the side area of the circular table:

$$S_B(t) = \pi H(t) \cdot \sin\alpha \cdot [D_2 + H(t) \cdot \sin\alpha \cdot \cos\alpha]$$

Among them, the lift height of needle valve is $H(t)$ and pole diameter of cam is r. Because the size of the injector also affects the oil injection rate, and the area of injector of the injector is $S_0 = \pi D_3^2/4$, and the effective oil injection area is $S_2(t) = \min(S_0, S_B(t))$.

There are i injectors in fuel pipe, let start working time of each injector be $t_{i,work}$. Through the relationship between pressure and density, the oil injection rate of injector i is:

$$Q_{out}(t, t_{i,work}) = F(t, t_{i,work}) \cdot C \cdot S_2(t - t_{i,work}) \cdot (2P(t) \cdot \rho(t))^{0.5} \tag{6}$$

Among them, $F(t,t_i)=\begin{cases}1, & t_i<\mathrm{mod}(t,T)<t_i+l\\ 0, & else\end{cases}$. Hereinafter, oil injection rate of the injector i at time t is $Q_{i,out}(t)$. The mass of fuel injected by the injector i per unit time is:

$$m_{i,out}=Q_{i,out}(t)\cdot\rho(r)\cdot\Delta t$$

Similarly, the injection flow of the reducing valve at any time is:

$$Q_{jian}(t)=J(P(t))\cdot C\cdot\pi\cdot D_4^{\ 2}/4\cdot(2P(t)/\rho(t))^{0.5} \quad (7)$$

The mass of fuel sprayed by the pressure reducing valve per unit time is:

$$m_{jian}=Q_{jian}(t)\cdot\rho(r)\cdot\Delta t$$

Among them, $J(P)=\begin{cases}1, & P>P_0\\ 0, & P<P_0\end{cases}$, the threshold for the reducing valve to open P_0.

In summary, let the mass of oil in the high-pressure oil pipe at the initial moment be $m(0)$, the mass of oil in the high-pressure oil pipe at time $t+\Delta t$ is:

$$m(t+\Delta t)=m(t)+m_{in}(t)-\sum_{i=1}^{n}m_{i,out}(t)-m_{jian}(t)$$

Since the pressure in the pipe is mainly affected by the injection cycle and the oiling cycle, let the injection cycle be T and the rotation cycle of cam be $T'=2\pi/w$.

$$\min Z(t_{work})=\max\left(\left|\frac{P(t)-100}{100}\right|\right),t\in[(N-1)\cdot T\cdot T',N\cdot T\cdot T']$$

$$s.t\begin{cases}P(t+dt)=G(\rho(t+dt))\\ P_1(t+dt)=G(\rho_1(t+dt))\\ \rho(t+dt)=\dfrac{m(t+dt)}{V}\\ \rho_1(t+dt)=\dfrac{m_1(t+dt)}{V_1(t)}\\ m(t+dt)=m(t)+dm\\ m_1(t+dt)=L(t)\cdot((m_1(t)-m_{in}(t))+(L(t)-1)\cdot m_0\\ dm=m_{in}(t)-\sum_{i=1}^{n}m_{out}(t)-m_{jian}(t)\\ m_{in}(t)=Q_{in}(t)\cdot\rho_1(t)\cdot dt\\ m_{i,out}(t)=Q_{i,out}(t)\cdot\rho(t)\cdot dt\\ m_{jian}(t)=Q_{jian}(t)\cdot\rho(t)\cdot dt\end{cases} \quad (8)$$

Next consider T_1T_2 as a system cycle to study, take a positive integer N, and simulate the entire system in the time $NT\cdot T'$. We build the optimal model with the minimum maximum relative error limit of the pressure fluctuation in the tube in the last system cycle can be established, and it is shown in formula (8). Among the equations above, t_{work} is an array composed of the working time points of each injector in a cycle; N is a sufficiently large positive integer.

Through simulation, in the working mode of single injector and dual injector, the optimal solution speed w is proportional to the number of injectors n, and the pressure P_0 is maintained at around101 MPa, and the working time of the fuel injector is nearly equidistant in a cycle. Therefore, it is only necessary to search for the optimal working time of multiple injectors at the maximum speed. We use multiple search strategies; the algorithm iteration process is as follows:

Step1: Let the initial solution for the working time of the injectors be $t_{work}=[t_{1,work},t_{2,work},\cdots,t_{n,work}]$.

Step2: Select a larger step size Δt_1. For each injector i, let $t_{i,work0}=t_{i,work}-\Delta t_1$, $t_{i,work0}=t_{i,work}+\Delta t_1$ or $t_{i,work0}=t_{i,work}$, and generate a new solution t_{work0} from t_{work}. Iterate over all possible t_{work0}, and choose the t_{work0} that can minimize $\min Z(t_{work0})$ as the new solution t_{work}.

Step3: Select a smaller step size Δt_i and perform Step 2 until t_{work} is fixed.

III. Test Results and Discussion

The equation between fuel pressure P and fuel density ρ is $r(t)=-2.413\sin(w\cdot t+7.854)+4.8236$; The equation between polar diameter and polar angle of cam is $r(t)=-2.413\sin(w\cdot t+7.854)+4.8236$; The equation between the height of the plunger and the rotation angle φ of the cam is

$$h(t)=\max_{\varphi}((-2.413\sin(w\cdot t+7.854)+4.8236)\cdot\cos(\varphi)) \quad \varphi\in[0,2\pi]$$

Through the simulation, the optimal situation of the pressure change in pipe under working mode of double injector is shown in Fig. 3.

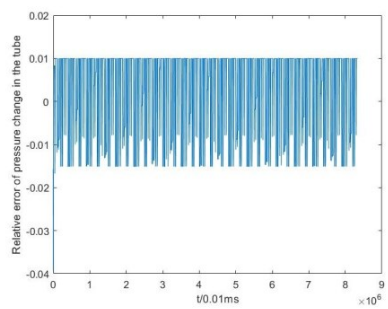

Fig. 3: Optimal solution of double injectors for pressure changes in the lower tube

When the threshold of the pressure reducing valve $P_0=101MPa$, the cam rotates at an angular speed of $150\,rad/s$, the time interval is $t_0=41ms$, and the pressure fluctuation in the tube is within 1%.

When rotation speed of oil pump reaches $600rad/S$, multiple-searching method are performed with the step size of $\Delta t=3ms$ and $\Delta t=1ms$, respectively, and The optimal situation of the pressure change in the tube is shown in Fig. 4.

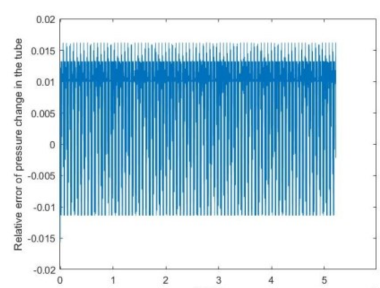

Fig. 4: Pressure change in high-pressure oil pipe at 600rad/S

It can be seen from Fig.4 that pressure fluctuation in oil pipe is around 1.5%. In addition, in the two cases of double injectors and the maximum speed of the high-pressure fuel pump, the number of injectors is proportional to the speed of the fuel pump and the working time of the injector is uniform and equidistant within a working cycle.

This paper conjectures that the speed of the fuel engine high-pressure fuel pump is proportional to the number of injectors, and the working time point at which the injector starts to work should be uniform and equidistant within a working cycle. In order to verify this conjecture, 12 and 16 working time point are set at the high-pressure pump speed of $w = 900 rad / S$ and $w = 1200 rad / S$, respectively. The simulation experiments are performed when the time point is equidistant and uniform in a cycle. The results are shown in Fig. 5 and Fig.6.

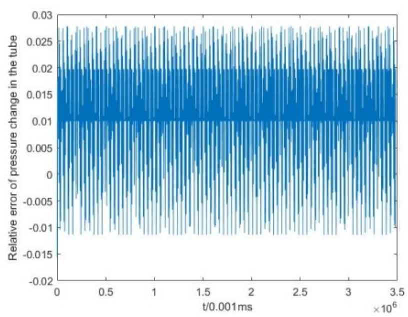

Fig. 5: Pressure change in high-pressure oil pipe at $900 rad / S$

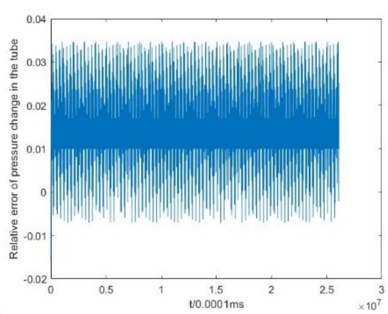

Fig. 6: Pressure change in high-pressure oil pipe at $1200 rad / S$

It can be seen from the above figure that the pressure in the high-pressure oil pipe is basically stable in both cases, so this conjecture is true.

IV. CONCLUSION

In this paper, the multiple-searching method is adopted to solve the configuration of the double injectors system and the maximum rotation speed of cam to keep the pressure in the high-pressure oil pipe basically stable, and the conjecture of the law between the cam speed and the injector is proposed and verified. In today's increasing awareness of energy saving and emission reduction, with the improvement of the performance of high-pressure fuel pumps, the conjecture in this paper provides a strong application foundation for the development of high-pressure common-rail injection injector systems. However, the multiple-searching method used in this paper tends to fall into the local optimal solution, which requires a large amount of computation to traverse all the solutions as far as possible. It is a problem that has not been solved in this paper.

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