Pressure Control Strategy and Simulation of High Pressure Common Rail System

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Abstract. Aiming at the control of the pressure wave of the high pressure tubing, considering the different design of the pressure reducing valve, two control strategies of the pressure reducing valve are discussed, and the problems to be noted when designing the control strategy are proposed. First, the actual high pressure common rail system is reasonably simplified as the “pump-nozzle-pipe” system, then based on the relevant theory (fluid flow formula, mass conservation law and relationship between fuel density and pressure), a mathematical model is established to determine the cam angular velocity, opening time and period and threshold of pressure reducing valve, and injection interval of nozzle, these parameters have a great influence on the pressure wave, and finally using the orthogonal test method to explore the influence of these parameters on the pressure wave by computer simulation. The research shows that only the injection interval of nozzle affects the size of the pressure wave range, other parameters only affects the pressure value at the time of stability. The larger the injection interval of nozzle, the smaller the size of pressure wave range. Therefore, when designing the control strategy, the injection interval of nozzle should be increased.

1. Introduction
High pressure tubing are widely used. They can be used as a medium for transporting submarine natural gas, petroleum and other energy sources and they can also be used as components and parts of construction machinery hydraulic tubing, diesel engine, direct injection gasoline engine and other machinery. The intermittent working process of fuel entering and ejecting the high pressure tubing will cause the pressure in the high pressure tubing to change, which will cause the deviation of the amount of fuel injected, thereby affecting the working efficiency of the engine. During the operation of the diesel engine, the low-pressure fuel is periodically pressurized by the high pressure fuel pump into the high pressure tubing, and then sprays out through the fuel injection nozzle to generate a suitable oil mist to chemically react to drive the engine.

Many scholars at home and abroad have studied the issue of high pressure common rail system. Wang Chenxin[1], Wang Xinjun[2] elaborated on the principle of high pressure common rail system and modelled the entire system based on knowledge related to fluid mechanics and used simulation software for computer simulation to explore the influence of various parameters on pressure waves. [1] further designed a new type of common rail pipe structure, but [1] did not test and verify the newly designed structure, and [2] did not consider the problem of bending of high pressure tubing to fuel resistance; Li Haiyang, Yang Haitao et al.[3] proposed the measurement methods of various physical parameters in the high pressure common rail system through experiments, but did not consider the effect of temperature on the results; Cai Liping[4] introduced several traditional calculation methods, and proposed an improved simulation calculation method in solving the high pressure tubing flow
equations, but the selected formulas are empirical, and the rationality of the selection has not been
discussed in depth; Li Pimao, Zhang Youtong et al.\cite{5} used the method which combined with
experiment and numerical calculation to study the influence of injection parameters on the amplitude
of pressure wave. In foreign countries, M F Russell and C S Bae\cite{6} studied the calculation method of
fluctuation interference and loss according to the laws of momentum conservation and mass
conservation and other laws, and proposed an improved method; A. Mittica\cite{7} used the finite element
method to establish the mathematical model of the diesel direct-acting piezoelectric injector. In many
studies, the study of the influence of various parameters of the high pressure common rail system on
the pressure wave gives a better answer to the selection of the optimal size components; the proposed
numerical method of computer simulation has greatly shortened the research cycle and saved the
experiment cost. However, few of these studies involve the study of the effect of pressure control
strategies on pressure waves.

Recently, two issues were discussed in\cite{8}: 1. How to adjust the check valve so that the pressure in the
high-pressure tubing is stable at the initial state (100MPa)? 2. How to adjust the check valve so that
the pressure in the high-pressure tubing is stable at 150MPa within 2 seconds?
The answer is given from the perspective of numerical calculation and mechanism analysis by\cite{8}. First,
the relationship between fuel density and pressure is determined by using the fourth-order Runge-
Kutta method. Secondly, the equations are listed according to the conservation of mass, thereby
obtaining the regulation law of the check valve.

In view of this, from the perspective of computer simulation and mathematical modeling, we conduct
research on pressure control strategies with the goal that reducing the size of pressure wave range.
Since the actual high pressure common rail system includes high pressure oil pumps, common rail
pipes, high pressure tubing, nozzles, and various sensors, which the most important parts are high
pressure oil pumps, high pressure tubing and nozzles, this system is simplified as a “pump-nozzle-
pipe” system. Based on related theories (fluid flow formula, mass conservation law, and the
relationship between fuel density and pressure), a mathematical model is established to determine
parameters that have a great influence on the pressure wave, and using MATLAB for computer
simulation to find more ideal parameter settings and the problems to be noted when designing the
control strategy are proposed. Different from the above research, we study the impact of pressure
control strategies on pressure waves from both fuel supply and fuel injection to provide a basis for
finding better pressure control strategies.

2. Model Building

2.1. Research Objects
The electronically controlled high pressure common rail system is relatively complex. It is mainly
composed of two major systems, fuel supply and electronic control, which jointly complete the supply
of diesel fuel. The basic composition structure is shown in Figure 1.

![Figure 1. Schematic diagram of high pressure common rail system](image-url)

Figure 1 shows the principle of the high pressure common rail system. Because the entire high
pressure common rail system involves the fields of mechanical engineering, fluid mechanics,
electromagnetics, Newtonian mechanics, chemistry, etc., it is very difficult to consider all factors and functional components. We only study three main parts: high pressure oil pump, high pressure tubing and nozzle. As the cam rotates to drive the plunger up and down, the pressure of the fuel in the plunger cavity is changed, so the pressurization function is realized. We consider the high pressure oil pump as a plunger cavity-cam system. So we will only study three parts: plunger cavity-cam system, high pressure tubing and nozzle.

**Figure 2.** Diagram of simplified high pressure common rail system

Figure 2 shows the main parts of our simulation.

2.2. Basic Knowledge

According to the principle of fluid mechanical energy conversion, the fluid flowing in a horizontal pipeline has dynamic pressure energy and static pressure energy (equal potential energy). Under certain conditions, these two forms of energy can be converted to each other, but the total energy does not change.\[9\]

Suppose the mass of the element is \( dm \), and the flow velocity is \( v \) when it passes through a hole of area \( S \), the pressure of high-pressure side is \( P_1 \), the pressure of low-pressure side is \( P_2 \), and \( dx \) is the displacement of the element under pressure \( F \). According to the kinetic energy theorem, we get:

\[
\frac{1}{2} m v^2 = F dx
\]

Taking “\( F=(P_1-P_2)S \)” into equation (1), we get:

\[
\frac{dm}{2} v^2 = (P_1 - P_2) S dx
\]

\[
\frac{1}{2} \frac{dm}{S dx} v^2 = P_1 - P_2
\]

Where \( \frac{dm}{S dx} \) is \( \rho_1 \), the density of the fuel in the high-pressure side. The results after finishing are as follows:

\[
v = \left( \frac{2(P_1 - P_2)}{\rho_1} \right)^{1/2}
\]

The rate of flow is defined by:

\[
Q = \mu S v = \mu S \left( \frac{2(P_1 - P_2)}{\rho_1} \right)^{1/2}
\]

Where \( Q \) represents the rate of flow (unit: mm$^3$/ms); \( \mu \) is the flow coefficient, which is dimensionless and represents the flow capacity of the hole.

When a fluid passes through a valve with an area \( S \), its flow rate \( Q \) satisfies equation (5).
According to \cite{1}, the relationship between fuel density and pressure under normal conditions is given by:

\[ \rho_{\text{oil}} = \rho_0 \left(1 + \frac{0.6 \times 10^2}{1 + 1.7 \times 10^{-3}} \frac{p_{\text{oil}}}{\rho_0} \right) \]  

(6)

From equation (5), we can get:

\[
p_{\text{oil}} = \left(1 - \frac{\rho_{\text{oil}}}{\rho_0} \right) \left[ 1.7 \times \left( \frac{\rho_{\text{oil}}}{\rho_0} - 1 \right) \times 10^{-3} - 0.6 \times 10^{-3} \right]^{-1} \]

(7)

Where \( \rho_{\text{oil}} \) represents the density of the fuel (unit: mg/mm\(^3\)), and \( p_{\text{oil}} \) represents the pressure of the fuel (unit: MPa). Under 100 MPa, the density of fuel is 0.85 mg/mm\(^3\), so \( \rho_0 = 0.80853816 \).

3. Simulation Algorithm

3.1. Theoretical Model

Based on the contents above, we model the following three parts in order to provide theoretical basis for computer simulation.

- Plunger cavity-cam system modeling:

When the plunger is at bottom dead center, the mass of low-pressure fuel entering the plunger cavity is \( M_{\text{total}} \), which satisfies:

\[
M_{\text{total}} = \rho_{\text{oil}}(p_{\text{low}})V_{\text{cavityMax}}
\]

(8)

Where \( p_{\text{low}} \) is the pressure of low-pressure fuel and \( V_{\text{cavityMax}} \) is the maximum volume of the plunger cavity.

From time \( t \) to time \( t + dt \), the mass of fuel entering the high pressure tubing from the plunger cavity, \( \Delta m_{\text{inPipe}}(t) \), satisfies:

\[
\Delta m_{\text{inPipe}}(t) = \int_{t}^{t+dt} Q dt
\]

(9)

\[
Q = \begin{cases} 
\mu S \left( \frac{2[p_{\text{cavity}}(t) - p_{\text{pipe}}(t)]}{p_{\text{cavity}}(t)} \right)^{\frac{1}{2}}, & p_{\text{cavity}}(t) > p_{\text{pipe}}(t) \\
0, & p_{\text{cavity}}(t) \leq p_{\text{pipe}}(t)
\end{cases}
\]

(10)

Where \( p_{\text{cavity}}(t) \), \( p_{\text{pipe}}(t) \), and \( p_{\text{cavity}}(t) \) are respectively the fuel density in the plunger cavity, the fuel pressure in the high pressure tubing, and the fuel pressure in the plunger cavity at time \( t \), and \( S \) is the area of the high pressure tubing inlet.

For the movement of the cam, we take the center of rotation of the cam as the origin of the coordinate and establish a plane rectangular coordinate system and let the state of plunger at the bottom dead center as the state of time 0 and evenly select \( n \) (more than enough) points on the cam, which are \((x_i, y_i)\), \( i = 1, 2, \ldots, n \). Figure 3 shows the model of the cam.

![Figure 3. Cam schematic](image-url)
At time \( t \), the plunger height \( h_{\text{plunger}}(t) \) satisfies:

\[
h_{\text{plunger}}(t) = \max_{1 \leq i \leq n} | y_i(t) |
\]

(11)

The relationship of time step \( dt \) and the angular velocity of cam \( \omega \) is given by:

\[
\begin{bmatrix}
x_i(t) \\
y_i(t)
\end{bmatrix} =
\begin{bmatrix}
\cos \omega dt & -\sin \omega dt \\
\sin \omega dt & \cos \omega dt
\end{bmatrix}
\begin{bmatrix}
x_i(t - dt) \\
y_i(t - dt)
\end{bmatrix}
\]

(12)

In order to improve the efficiency of the program, the height of the plunger in each state can be stored in an array.

The fuel pressure \( p_{\text{cavity}}(t) \) in the plunger cavity at time \( t \) satisfies:

\[
p_{\text{cavity}} = p_{\text{oil}}(\rho_{\text{cavity}}(t))
\]

(13)

\[
\rho_{\text{cavity}}(t) = \frac{m_{\text{cavity}}(t)}{V_{\text{cavity}}(t)}
\]

(14)

\[
m_{\text{cavity}}(t) = \begin{cases} 
M_{\text{total}} \omega = 2\pi n (n = 0,1,2,...) \\
m_{\text{cavity}}(t - dt) - \Delta m_{\text{inPipe}}(t - dt), \text{else}
\end{cases}
\]

(15)

\[
V_{\text{cavity}}(t) = \pi h_{\text{cavity}}(t) \left( \frac{D_{\text{cavity}}}{2} \right)^2
\]

(16)

\[
h_{\text{cavity}}(t) = H - h_{\text{plunger}}(t)
\]

(17)

Where \( m_{\text{cavity}}(t) \), \( V_{\text{cavity}}(t) \), \( D_{\text{cavity}} \) and \( h_{\text{cavity}}(t) \) are respectively the mass of the fuel in the plunger cavity, the volume of the plunger cavity, the diameter of the plunger cavity, and the height of the plunger cavity at time \( t \), \( H \) is the vertical coordinate corresponding to the highest point of the plunger cavity.

- High pressure tubing modeling:

At time \( t \), the pressure of fuel in the high pressure tubing, \( p_{\text{pipe}}(t) \), satisfies:

\[
p_{\text{pipe}}(t) = p_{\text{oil}}(\rho_{\text{pipe}}(t))
\]

(18)

\[
\rho_{\text{pipe}}(t) = \frac{m_{\text{pipe}}(t)}{V_{\text{pipe}}}
\]

(19)

\[
m_{\text{pipe}}(t) = m_{\text{pipe}}(t - dt) + \Delta m_{\text{inPipe}}(t - dt) - \Delta m_{\text{outPipe}}(t - dt)
\]

(20)

\[
\Delta m_{\text{outPipe}}(t) \approx \rho_{\text{pipe}}(t) \mu S_{\text{access}}(t) \left\{ \frac{2(p_{\text{pipe}}(t) - p_{\text{air}})}{\rho_{\text{pipe}}(t)} \right\}^{1/2} dt
\]

(21)

Where \( \rho_{\text{pipe}}(t) \), \( m_{\text{pipe}}(t) \), and \( V_{\text{pipe}} \) are respectively the density and mass of fuel in high pressure tubing at time \( t \), and the volume of high pressure tubing. \( \Delta m_{\text{outPipe}}(t) \) is the mass of fuel spewing out form high pressure tubing from time \( t \) to time \( t + dt \), \( p_{\text{air}} \) is the external air pressure, and \( S_{\text{access}}(t) \) is the effective flow area of the fuel injection nozzle.

- Nozzle modeling:

We attribute the change in the nozzle to the change in the lift of the needle valve which affects the effective flow area \( S_{\text{access}}(t) \). \( S_{\text{access}}(t) \) can be expressed by:

\[
S_{\text{access}}(t) = \min \{ S_1(t), S_2 \}
\]

(22)

\[
S_1(t) = \pi \left( \frac{D(t)}{2} \right)^2 - \pi \left( \frac{D_{\text{needle}}}{2} \right)^2
\]

(23)
\[ S_2 = \pi \left( \frac{D_{\text{orifice}}}{2} \right)^2 \]  

(24)

\[ D(t) = 2 \tan \theta \left[ h_{\text{needle}}(t) + \frac{D_{\text{needle}}}{2 \tan \theta} \right] \]  

(25)

Where \( D_{\text{needle}} \) and \( D_{\text{orifice}} \) are respectively the diameter of the needle valve and the diameter of the nozzle top, \( S_2 \) is the area of the nozzle top, \( h_{\text{needle}}(t) \) is the lift of needle valve at time \( t \), \( \theta \) is the half angle of the seal seat. The schematic diagram of the nozzle is shown below.

![Nozzle schematic](image_url)

**Figure 4. Nozzle schematic**

### 3.2. Algorithm Design

Based on the theoretical model above, we design simulation algorithms, and use MATLAB to perform simulation experiments. The input of the algorithm are the cam angular velocity and the simulation time, and the output is the graph “the change trend of the pressure wave in the high pressure tubing”. The specific steps of the simulation algorithm are as follows:

- **Step1**: Enter cam angular velocity \( \omega \) and simulation time \( t_{\text{simulation}} \), set the current time \( t = 0 \), \( m_{\text{cavity}}(t) = M_{\text{total}}, p_{\text{pipe}}(t) = 100 \text{ MPa}, m_{\text{pipe}}(t) = \rho_{\text{oil}}(100 \text{ MPa})V_{\text{pipe}} \) and then go to Step2.
- **Step2**: Determine whether the plunger reaches the bottom dead center according to whether \( \omega t \) is equal to \( 2\pi n (n = 1, 2, \ldots) \). If the plunger reaches the bottom dead center, then according to formula (8), set \( m_{\text{cavity}}(t) \) to \( M_{\text{total}} \) and go to Step3; otherwise, \( m_{\text{cavity}}(t) \) remains unchanged and also goes to Step3.
- **Step3**: According to formula (13) \~ (25), calculate \( p_{\text{cavity}}(t) \) and \( p_{\text{pipe}}(t) \) and determine whether \( p_{\text{cavity}}(t) \) is greater than \( p_{\text{pipe}}(t) \). If so, use formula (9) \~ (10) to calculate \( \Delta m_{\text{inPipe}}(t) \) and set \( m_{\text{cavity}}(t + dt) = m_{\text{cavity}}(t) - \Delta m_{\text{inPipe}}(t) \) , go to Step4; otherwise set \( m_{\text{cavity}}(t + dt) = m_{\text{cavity}}(t) \), also go to Step4.
- **Step4**: Use the motion data of needle valve to calculate the position of needle valve \( h_{\text{needle}}(t) \) and determine whether the needle valve reaches bottom dead center according to whether the \( h_{\text{needle}}(t) \) is 0. If so, update \( \Delta m_{\text{outPipe}}(t) \) according to formula (2) and go to Step5; otherwise, set \( \Delta m_{\text{outPipe}}(t) = 0 \) and update \( m_{\text{pipe}}(t + dt) \) according to formula (20), and also go to Step5.
- **Step5**: Set \( t = t + dt \), if \( t = t_{\text{simulation}} \) go to Step6, otherwise go to Step2.
- **Step6**: Use time \( t \ (0 \leq t \leq t_{\text{simulation}}) \) as the independent variable, and the generated \( p_{\text{pipe}}(t) \) as the dependent variable, use the function “plot” to print the graph “the change trend of the pressure wave in the high pressure tubing”.

### 4. Simulation Results

#### 4.1. Case of Single Nozzle

The data used in this simulation are as table 1. The edge curve of cam and movement law of needle valve are as Figure 5 and Figure 6.
The motion cycle of the needle valve is 100 ms. In each cycle, the movement law in the first 2.5 ms is shown in Figure 6, and the lift of needle valve in the latter 97.5 ms is always 0.

### Table 1. Data

| Parameter                                      | Value     |
|------------------------------------------------|-----------|
| Length of high pressure tubing                | 500 mm    |
| Diameter of high pressure tubing              | 10 mm     |
| Initial pressure of high pressure tubing      | 100 MPa   |
| Diameter of oil supply port                   | 1.4 mm    |
| Diameter of fuel injection port               | 1.4 mm    |
| Diameter of needle valve                      | 2.5 mm    |
| Half angle of seal seat                       | 9°        |
| Diameter of plunger cavity                    | 5 mm      |
| Maximum height of plunger cavity              | 5.8446 mm |
| Pressure of low-pressure fuel                 | 0.5 MPa   |
| Spraying cycle of nozzle                      | 100 ms    |
| Diameter of pressure reducing valve outlet    | 1.4 mm    |

The above data are derived from the question A of the China Undergraduate Mathematical Contest in Modeling (CUMCM) in 2019. In order to determine at what angular velocity of cam, the pressure in the high pressure tubing can be stabilized in the initial state, we first determine a rough range of angular velocity of cam, and then continuously dichotomize to obtain more accurate angular velocity of cam. Through simulation experiments, under the given data, let the time step is 0.01 ms, when the high pressure tubing has only one nozzle, to keep the high pressure tubing in its original state, the angular velocity of the cam should be 0.0285 rad/ms. Figure 7 shows the change of pressure wave when the angular velocity of cam is 0.0285 rad/ms. The figure 7 shows that the pressure wave is between 97 MPa and 103 MPa, indicating that the value of the parameter has reached the goal.

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**Figure 5.** Edge curve of cam (unit of polar diameter: mm)  **Figure 6.** Movement law of needle valve.
4.2. Case of Double Nozzles with Pressure Reducing Valve

The dual-nozzle high pressure tubing system with pressure reducing valve is essentially the same as the previous case, the mathematical model is basically unchanged, but more things are considered (including the synchronization of the two nozzles, the control strategy of the pressure reducing valve), the parameters opening time and cycle and threshold of pressure reducing valve, and injection interval of nozzle are added. In addition, following formulas are added.

\[
\Delta m_{\text{reducingValve}}(t) = \int_{t-dt}^{t} \rho_{\text{pipe}}(t) Q_r dt
\]  

(26)

\[
Q_r = \begin{cases} 
\mu S_r \left( \frac{2[p_{\text{pipe}}(t) - p_{\text{low}}]}{p_{\text{pipe}}(t)} \right)^{1/2}, & p_{\text{pipe}}(t) > p_{\text{threshold}} \\
0, & p_{\text{pipe}}(t) \leq p_{\text{threshold}}
\end{cases}
\]  

(27)

Or

\[
Q_r = \begin{cases} 
\mu S_r \left( \frac{2[p_{\text{pipe}}(t) - p_{\text{low}}]}{p_{\text{pipe}}(t)} \right)^{1/2}, & nU \leq t < nU + \Delta u \\
0, & nU + \Delta u \leq t < (n+1)U
\end{cases}
\]  

(28)

Where \( \Delta m_{\text{reducingValve}}(t) \) is the mass of fuel flowing from pressure reducing valve at time \( t \) to time \( t - dt \), \( Q_r \) is the rate of flow of pressure reducing valve under different strategies, \( S_r \) is the effective flow area of pressure reducing valve, \( p_{\text{threshold}} \) is the threshold of pressure reducing valve, and \( U \) is the opening cycle of pressure reducing valve, \( \Delta u \) is the opening time of pressure reducing valve.

The high pressure tubing system after adding an nozzle and a pressure reducing valve is shown in Figure 8.

Assuming that the physical properties of the two nozzles, the injection pulse width (injection duration) and the injection cycle are the same, and only that the movement of the needle valve is different. The schematic diagram of the needle valve movement law of nozzle A and nozzle B is shown in Figure 9. Horizontal axis represents time and Vertical axis represents the lift of needle valve, \( T \) is the injection cycle of the nozzle A and nozzle B, and \( \Delta t (0 \leq \Delta t < T) \) is the injection interval of the nozzle A and nozzle B in one cycle. With the goal of keeping the pressure in the high pressure tubing at 100 MPa and the size of pressure wave range as small as possible, we discuss the control strategy of the pressure reducing valve from two directions:

1. Control the pressure reducing valve to open periodically, each time opened it lasts for a period of time, to reduce the pressure in the high pressure tubing.
2. Set a threshold for the pressure reducing valve, as long as this value is exceeded, the valve will open to reduce the pressure in the high pressure tubing.
First, qualitative analysis. The longer the opening time of the pressure reducing valve and the lower the angular velocity of cam, then the lower the pressure in high pressure tubing. Therefore, first consider the value of $\omega$, $U$, $\Delta u$, and then consider the value of $\Delta t$.

Then quantitative analysis. In the range of $0.05 \text{ rad/ms} \sim 0.06 \text{ rad/ms}$, the value of $\omega$ is taken every 0.001 units; in the range of $1000 \text{ ms} \sim 1500 \text{ ms}$, the value of $U$ is taken every 100 units; in the range of $1 \text{ ms} \sim 5 \text{ ms}$ the value of $\Delta u$ is taken every 1 unit. After determining $\omega$, $U$, $\Delta u$, in the range of $0 \text{ ms} \sim 90 \text{ ms}$, the value of $\Delta t$ is taken every 10 units.

According to the combination of values of $\omega$, $U$, $\Delta u$ and $\Delta t$, orthogonal test was conducted. By results of orthogonal test, we found that when $\Delta t$ is about half of $T$, the size of pressure range in the high pressure tubing after stabilization is the smallest, but $\Delta t$ or small or large will cause the size of pressure range in the high pressure tubing to increase. Figure 10 and Figure 11 shows the difference in pressure waves in high pressure tubing when the cam angular velocity is 0.058 rad/ms, opening cycle of pressure reducing valve is 1.2 s, opening time of pressure reducing valve is 1 ms and $\Delta t$ is respectively 50 ms and 0 ms. After screening all parameter combinations, the first control strategy is that:

Fuel supply strategy: The angular velocity of cam is 0.058 rad/ms, $\Delta t$ is 50 ms.

Pressure reducing valve control strategy: Opening cycle of pressure reducing valve is 1.2 s, opening time of pressure reducing valve is 1 ms.

Figure 10. Pressure wave diagram when $\Delta t$ is 50 ms

It can be seen from the figure that the reduction of the former pressure wave range ($96.0139 \text{ MPa} \sim 102.6308 \text{ MPa}$) compared to the latter pressure wave range ($93.9826 \text{ MPa} \sim 103.7715 \text{ MPa}$) is 3.172 MPa. It shows that the injection interval of nozzle has a certain effect on the adjustment of the pressure wave range. This is because when $\Delta t$ is not 0, the two nozzles in one injection cycle do not spray at the same time, dispersing the injection amount, so the fluctuation range is small; and when $\Delta t$ is 0, two nozzles in one injection cycle spray at the same time, the injection is concentrated, so the fluctuation range is large.

According to the above analysis, when $\Delta t$ is 50 ms, the pressure wave range is the smallest, so only need to determine the value of $\omega$ and $p_{\text{threshold}}$. In the range of $0.05 \text{ rad/ms} \sim 0.06 \text{ rad/ms}$, the value of $\omega$ is taken every 0.001 units; In the range of $100 \text{ MPa} \sim 103 \text{ MPa}$, the value of $p_{\text{threshold}}$ is taken every 0.5 units.

According to the value combination of $\omega$ and $p_{\text{threshold}}$, an orthogonal test is carried out. After testing all the parameter combinations, the second control strategy is obtained as:

Fuel supply strategy: The angular velocity of cam is 0.057 rad/ms, $\Delta t$ is 50 ms.

Pressure reducing valve control strategy: $p_{\text{threshold}}$ is 102.5 MPa.

Figure 12 and Figure 13 shows the difference in pressure waves in high pressure tubing when the cam angular velocity is 0.057 rad/ms, $\Delta t$ is 0ms, and $p_{\text{threshold}}$ is respectively 103 MPa and 102.5 MPa. Combining the above two pressure control strategies, the pressure wave range controlled by the first strategy is: $96.0139 \text{ MPa} \sim 102.6308 \text{ MPa}$. the pressure wave range controlled by the second strategy is: $97.0617 \text{ MPa} \sim 102.5009 \text{ MPa}$. The pressure wave controlled by the second strategy is more stable than the pressure wave controlled by the first strategy. The first strategy only needs to set the opening
rule of the pressure reducing valve in advance, while the second strategy needs to add pressure sensors to control the opening of the pressure reducing valve in real time.

Figure 12. Pressure wave change when the threshold is 103 MPa
Figure 13. Pressure wave change when the threshold is 102.5 MPa

5. Conclusion
In this paper, the high pressure common rail system was reasonably simplified, and the model was established according to the relevant theory for computer simulation. Different from the above research, we studied the impact of pressure control strategies on pressure waves from both fuel supply and fuel injection and proposed the factors to be considered when designing strategies. We have discussed two control strategies. The advantage of the first strategy is that only the opening parameters of the pressure reducing valve need to be set in advance, and the control is simple; the disadvantage is that the parameter design is different in different situations, and the flexibility is poor. The advantage of the second strategy is that the peak value of the pressure wave can be effectively controlled, and the pressure wave changes smoothly; the disadvantage is that additional pressure sensors are required, and the timeliness of pressure feedback is demanding. The first control strategy is suitable for the case where the pressure wave range is not strict, the second control strategy is suitable for the case where the pressure wave range is strict. In addition, the parameter values of the above strategies can be flexibly changed according to the actual situation, but multiple nozzles should be work at intervals, so that the range of pressure wave can be narrowed. The simulation algorithm mentioned in this paper can be extended to systems with multiple fuel supply ports and multiple fuel injection nozzles, and the shape data of the cam can also be changed according to the actual situation.

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