Research on Pressure Controls of High Pressure Tubing Based on Simulation Model

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Abstract. High pressure oil pipeline is the basic component of many fuel engines, and it has great application and development value in transportation, aerospace and other national defense military. This paper examines ways to control the pressure change in the high pressure tubing under the condition of intermittent fuel entry and ejection. On the basis of determining the balance equation of each part of the injection system, this paper presents different design schemes of the high pressure fuel system under different operating parameters of the high pressure tubing by establishing the simulation model.

1. Introduction
As the most important component of fuel engine, high pressure tubing determines the working efficiency and service life of fuel engine. Due to the intermittent work of fuel entering and ejecting, the pressure changes in the high-pressure tubing, resulting in the deviation of the amount of fuel ejecting, thus affecting the working efficiency of the engine. Therefore, by establishing a mathematical model, this paper makes a detailed analysis of how to stabilize the pressure change in the high-pressure tubing and improve the working efficiency of the engine. Further, to raise the pressure inside the high-pressure tubing, how to adjust the opening time of the one-way valve to maintain balance. The second is the way to control the angular velocity of cam-driven plunger movement in the high-pressure oil pump to make the pressure in the high-pressure oil pipe as stable as possible at 100MPa. In order to facilitate the model, we do not consider the temperature change caused by the pressure change in the high-pressure tubing and the energy loss caused by the high-pressure pump fuel entering the high-pressure tubing and spouting fuel.

2. Modeling
The balance of the pressure in the high-pressure tubing is through fuel into and out of intermittent work, reduced pressure in fuel jet caused by high pressure pipe, again by adding fuel to the compensation, the process can use the energy conservation equation, the following for each process modeling, we, in turn, determine the changes of each part and the corresponding boundary conditions of model is given finally.
2.1. High Pressure Tubing Pressure Regulation Model

Taking a certain type of high pressure tubing as an example, the length of its inner cavity is 500 mm, the inner diameter is 10 mm, the diameter of the oil supply inlet hole is 1.4 mm, through the one-way valve switch to control the length of oil supply time, and the one-way valve A will be closed every time after opening 10 ms. The nozzle B works 10 times per second, and the injection time is 2.4 ms. The pressure supplied by the high pressure oil pump at the entrance A is always 160 MPa. It is studied how to set the duration of each opening of the one-way valve so that its pressure is stable at 100 MPa; At the same time, the equilibrium problem of increasing internal pressure of high pressure tubing at a different time is discussed.

(1) Energy Conservation Model.

In the sealed high-pressure oil pipe, the amount of fuel entering the high-pressure oil pipe through the high-pressure oil pump per unit time is $Q_A$:

$$ Q_A = CA \sqrt{\frac{2\Delta P}{\rho}} $$

The single amount of fuel ejected by fuel injection nozzle B is $Q_B$:

$$ Q_B = \int_0^t v_B dt $$

The entry and ejection of fuel oil in the high-pressure oil pipe satisfy the equilibrium relationship, thus the fuel continuity equation in the high-pressure oil pipe can be listed as follows:

$$ \frac{Q_A}{1} m = Q_B $$

Where, $\frac{Q_A}{1}$ is the rate of the high-pressure oil pump entering the high-pressure oil pipe, $t$ is the opening time of the one-way valve, $n$ is the number of the high-pressure pump, and the number of the oil $n = \frac{100}{10+t}$.

(2) Hydrodynamic Model.

The fluid model of high pressure common rail fuel injection system is mainly calculated based on the solution of Navier-Stokes equation, namely, the conservation of continuity, the conservation of capacity and the conservation of momentum. The formula is as follows:

Continuity conservation equation:

$$ \frac{\partial m}{\partial t} = \sum_{\text{boundaries}} m $$

Energy conservation equation:

$$ \frac{\partial}{\partial t} \left( \text{me} \right) = -P \frac{\partial V}{\partial t} + \sum_{\text{boundaries}} (mH) - hA_s (T_{\text{fuel}} - T_{\text{wall}}) $$

Momentum conservation equation:

$$ \frac{\partial m}{\partial t} + \sum_{\text{boundaries}} (m u) - 4C_d \frac{\rho u |u|}{2} \frac{dA}{D} - C_p \left( \frac{\rho u |u|}{2} \right) A $$

2.2. Characteristic Analysis of Oil Injection System is Based on Simulation Calculation

The high pressure tubing is the intermediate link connecting the high pressure oil pump and each injector, which are equivalent to the accumulator and plays a role in suppressing the pressure fluctuation. High pressure oil pump CAM drive plunger up and down movement, so that the fuel into the high pressure tubing. How to determine the CAM angular velocity, so that the pressure in the high-
pressure tubing as far as possible to stabilize at about 100 MPa, in order to improve the work efficiency of the fuel engine is particularly important.

The change rate of the amount of compression caused by the pressure change of the plunger chamber $Q_{cv}$, and the amount of oil from the plunger chamber to the oil outlet valve $Q_{cd}$ satisfies the equilibrium relationship. Thus, the fuel continuity equation in the plunger chamber of the high pressure oil pump can be listed as follows:

$$Q_c = Q_{cv} + Q_{cd}$$

Instantaneous oil input in the plug chamber:

$$Q_c = S_c \frac{\partial h_c}{\partial t}$$

Rate of compression caused by pressure change in plunger chamber:

$$Q_{cv} = \frac{V_c}{E} \frac{\partial P_c}{\partial t}$$

Amount of oil from plunger chamber to outlet valve:

$$Q_{cd} = C \times A \times \sqrt{\frac{2(P_c - P_d)}{\rho}}$$

The piston motion equation is given below. The CAM makes the displacement of the plunger calculated in time $\Delta t$ as:

$$h_c = (r \cos \sigma - e \cos \theta) - (r - e)$$

Where, $r$ is the camshaft radius (mm); $e$ is the eccentricity (mm).

As in the time $t$ to $t + \Delta t$, $\sigma$ is very small, $\cos \sigma \rightarrow 1$, so:

$$h_c = e(1 - \cos \theta)$$

Where $\theta$ is the function of CAM speed $\omega$, $\theta = \omega t = \pi n / 30$. According to this, we get the movement process of oil pump plunger:

$$h_{c1} = e(1 - \cos(\pi n / 30))$$

$$h_{c2} = e[1 - \cos(\pi n / 30)(t + \frac{n}{30})]$$

Next, we establish the model of high pressure tubing.

The amount of fuel flowing into the high pressure tubing $Q_{de}$, the amount of fuel flowing into the control chamber of the injection tubing $Q_{ek}$, and the fuel compression ratio within the rail shall satisfy the following relationships:

$$\sum Q_{de} = \frac{V}{E} \frac{\partial P}{\partial t} + \sum Q_{ek}$$

The flow rate from the oil outlet valve of the high pressure oil pump to the high pressure oil pipe $Q_{de}$ can be expressed as follows:

$$Q_{de} = S_{de} C \sqrt{\frac{2(P_f - P_s)}{\rho}}$$

Fuel quantity from high pressure tubing to injector control chamber:

$$Q_{ek} = S_{ek} C \sqrt{\frac{2(P_e - P_d)}{\rho}}$$

Fuel inflow, outflow and fuel compression ratio of injector needle seat chamber shall meet the following requirements:
When studying the moving model of the injector needle valve, we assume that the needle valve parts do not fall off each other and do not leak. The needle valve assembly is subjected to downward high pressure tubing pressure $F_3$ and spring force $F_4$, to upward pressure chamber pressure $F_1$, and to hydraulic pressure of the needle valve seal seat $F_2$. The force analysis of needle valve is as follows:

$$\frac{\partial V_y}{\partial t} + V_y \frac{\partial P_y}{\partial t} = Q_m - Q_a$$

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![Figure 1. Force Analysis of Needle Valve.](image)

When $F_3 + F_4 + F_5 > F_1 + F_2$, the needle valve is in the closed state (or in the process of descending); When $F_3 + F_4 + F_5 < F_1 + F_2$, the spray begins.

Combined with the above analysis, the movement equation of needle valve is:

$$m_{zf} \frac{\partial^2 h_{zf}}{\partial t^2} = F_1 + F_2 - F_3 - F_4 - F_5$$

$$F_1 = \begin{cases} \pi d_2^2 P_y / 4, & h_{zf} = 0 \\ \pi d_2^2 P_a / 4, & h_{zf} > 0 \end{cases}$$

$$F_2 = P_{xfq} \frac{\pi (d_e^2 - d_2^2)}{4}$$

$$F_3 = P_h \frac{\pi d_2^2}{4}$$

$$F_4 = k_{zf} (h_{zh0} - h_{zh})$$

$$F_5 = C_{zf} \frac{dh_{zf}}{dt}$$

Where, $F_1$ is the pressure chamber fuel on the needle valve (N); $F_2$ is the force of fuel on the needle valve at the needle seat (N), $d_2$ is the projection diameter below the needle valve sealing bag (mm); $F_3$ is the force of the control chamber fuel on the control piston (N); $F_4$ is the spring force (N), $h_{zh0}$ is the decrease in volume of the spring (mm), $k_{zf}$ is the spring degree of steel; $F_5$ is the
damping force received during the movement of the needle valve \((N)\), \(C_{df}\) is the needle valve damping coefficient.

3. Empirical Analysis

3.1. Analysis of Pressure Stabilizing Model of High Pressure Tubing

Check data to get the elastic modulus \(E\) under high pressure oil of 0-200 MPa, combined with the modulus of elasticity, the definition, the fuel oil pressure variation is proportional to the density variation, proportion coefficient is \(E/\rho\), the elastic modulus and the relationship between the pressure and per unit time into the flow of high pressure oil pipe can be concluded that jet velocity image along with the change of the pressure is as follows:

![Image of Jet Velocity with Pressure](image)

Figure 2. Image of Jet Velocity with Pressure.

After further determining the pressure \(P_{\theta}\) of the high pressure tubing after one injection of oil from the nozzle, the initial velocity of oil injection from the high pressure oil pump \(v_1\) and the rate of oil injection \(v_2\) when it is stable at 100 MPa can be obtained.

\[ t = 0.2917 \]

Change the pressure in the high pressure tubing to 150 MPa, change the time parameter required to reach the equilibrium state, and obtain the initial opening time of the one-way valve at 150 conditions after the adjustment process of about 2 s, 5 s and 10 s:

\[ t_2 = 0.7107, t_5 = 0.7012, t_{10} = 0.6997 \]

3.2. Characteristics Analysis of Oil Injection System

The simulation calculation of plunger injection system can be used to analyze the injection characteristics of the injection system, study the influence of the geometric parameters of the oil pump, oil pipe and injector on the injection characteristics, and investigate various dynamic transition processes, internal flow conditions and the motion characteristics of the needle valve. The mass conservation equation, the motion equation of plunger cavity and the needle valve and the pressure wave equation in the high-pressure oil pump - high-pressure oil pipe - nozzle injection system is solved simultaneously. MATLAB operation can be obtained in the high pressure oil pump, oil outlet valve cavity, high pressure tubing and needle valve cavity pressure curve and the law of fuel flow.

In the solution of the pump and boundary equations, the four unified equations can be respectively compiled into a functional M file. As the following equation:
Among them, $\alpha$ is the fuel compress coefficient; $V_y$ is the pressure chamber volume of the nozzle; $P_y$ is the nozzle chamber pressure; $f_y$ is the cross-sectional area of the nozzle flange, 

$$f_y = \pi \left( \frac{d_e^2 - d_f^2}{4} \right);$$

$y$ is the needle valve lift; $C$ is the fuel flow coefficient; $f_q$ is the cross-sectional area of the channel at the seal seat, 

$$f_q = \frac{d_f^2}{4};$$

$P_d$ is the high pressure oil pump internal fuel pressure; $P_t$ is the pressure at the nozzle; 

$$f_t = \frac{d_f^2}{4};$$

$u_0$ is the high pressure oil pump end tube fuel speed; 

$\eta$ depends on the movement of the needle valve, 

$$\eta = \begin{cases} 
0, & (\text{static}) \\
1, & (\text{move}) 
\end{cases}$$

$\delta$ is the direction of movement of fuel through the needle valve, 

$$\delta = \begin{cases} 
1, & (P_d - P_y \geq 0) \\
-1, & (P_d - P_y < 0) 
\end{cases}$$

To sum up, it can be concluded that when the CAM rotation angular velocity is $27.53 \text{rad/s}$, the pressure in the high pressure tubing should be as stable as possible at about 100MPa with the minimum deviation.

4. Conclusion
The pressure control scheme proposed in this paper can meet the demand of different parameters of different high-pressure tubing and has strong universality and extensibility. Although the scheme presented in this paper has some limitations and shortcomings, it can provide some guidance and Suggestions for the design of the high-pressure tubing control scheme of the fuel engine, so as to make the production of the internal combustion engine more efficient.

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