Chapter

Pressure Fluctuation Characteristics of High-Pressure Common Rail Fuel Injection System

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Abstract

In high-pressure common rail fuel injection system, fuel pressure wave propagates back and forth in the system during fuel injection, and the cycle fuel injection volume is affected by the fluctuation of fuel injection pressure. Therefore, to reduce the influence of pressure fluctuation on the cycle fuel injection volume fluctuation, it is of great theoretical significance to analyze the mechanism of pressure fluctuation and its influence law. In this chapter, the dynamic pressure fluctuation characteristics of the high-pressure common rail fuel injection system are analyzed based on the injector inlet pressure, and experimental research and theoretical analysis are carried out for the time domain and frequency domain characteristics of injector inlet pressure fluctuation, aiming at revealing the pressure fluctuation mechanism and its influence law, and providing theoretical support for improving the control accuracy of multiple injection cycle fuel injection volume.

Keywords: diesel engine, high pressure common rail, fuel injection, pressure wave, time domain characteristics, frequency domain characteristics

1. Introduction

As the most advanced fuel system, the high-pressure common rail fuel injection system can realize the flexible, accurate and stable control of fuel injection pressure, fuel injection timing and cycle fuel injection volume, which can not only make the diesel engine power performance and economy best but also meet the increasingly strict requirements of emission regulations [1–5]. The existence of common rail separates the fuel supply process and fuel injection process of high-pressure common rail fuel injection system. The high pressure fuel pump only provides high pressure fuel to the common rail according to the working condition of the system. The electrical control unit (ECU) drives the high speed solenoid valve to control the injector to inject high pressure fuel into the cylinder. The two parts work independently. This is the main characteristic of high-pressure common rail fuel injection system different from traditional fuel injection systems [6–11]. The flow characteristics of high pressure fuel in high-pressure common rail fuel injection systems have an important effect on cycle fuel injection volume. The fuel injector is the main executing part of the fuel injection system. Due to the fuel inertia, fuel in the control chamber and nozzle volume does not immediately stop flowing when the control valve and needle of the injector are suddenly closed. The fuel kinetic energy near the control valve and needle is converted into local pressure gain, then this conversion propagates at the speed of sound to the control chamber and nozzle volume. Finally, the fuel compression wave or expansion wave is reflected back. Because of the energy imbalance, the dynamic pressure wave propagates and oscillates repeatedly in the system until the system reaches a stable state again due to the dissipative effect [12–18].

The pressure fluctuation has a significant effect on fuel injection rate, which affects the cycle fuel injection volume of the high-pressure common rail fuel injection system. Reference [19] proposed a simplified physical model to predict the fluctuation of fuel injection pressure. Reference [20] designed a fuel acceleration pipeline at the nozzle and two sets of control systems were added to control the fuel flow state in the pipeline. A numerical model was established to predict the fluctuation of fuel injection pressure in theory. To explore the relationship between pressure fluctuation frequency and system structure during fuel injection, Ref. [21] established an LC zero-dimensional equivalent model of common rail, high pressure fuel pipeline and injector. Aiming at the influence of pressure fluctuation of high-pressure common rail fuel injection system on cycle fuel injection volume characteristics, Ref. [22] studied the interrelationship between geometrical dimensions of high pressure fuel pipeline between common rail and injector and pressure fluctuation characteristics and cycle fuel injection volume. The research results show that the size change of high pressure pipeline has a significant impact on the characteristics of single injection cycle fuel injection volume. Reference [23] analyzed the characteristics of cycle fuel injection volume of high-pressure common rail fuel injection system under two working conditions. The research results show that the fuel pressure fluctuation in the injector internal pipeline has a more significant effect on the change of cycle fuel injection volume compared with the pressure fluctuation in the common rail. Reference [24] established a simulation model of high-pressure common rail fuel injection system. The simulation analysis shows that the pressure fluctuation in the fuel chamber during fuel injection is the main reason for the high frequency characteristics of fuel injection rate variation. However, the low frequency characteristics of fuel injection rate variation are determined by the fluctuation of fuel injection pressure.

Multiple injections is one of the main technical means for diesel engines to meet increasingly strict emission regulations. Many scholars have studied the influence of pressure fluctuation on multiple injection cycle fuel injection volume characteristics of high-pressure common rail fuel injection systems. Reference [25] studied the characteristics of cycle fuel injection volume under different injection modes of high-pressure common rail fuel injection systems. The research results show that the fuel pressure fluctuation has an important influence on cycle fuel injection volume because it affects the injection timing of pilot injection, main injection and post-injection. Pressure fluctuation generated after the main injection will cause the needle to be difficult to open during post-injection, which results in post-injection volume fluctuations. Reference [26] changed the injection interval between pilot injection and main injection of high-pressure common rail fuel injection system. It is found that the pulse width of the main injection fluctuates periodically with the increase of the injection interval between pilot injection and main injection when the actual pilot injection cycle fuel injection volume and the actual main injection cycle fuel injection volume are fixed. The pulse width fluctuation frequency only depends on the structural parameters and is independent of diesel engine speed, cycle fuel injection volume, fuel injection pulse width and fuel injection pressure. Reference [27] simulated and analyzed the influence of different fuel properties on the pressure wave and cycle fuel injection volume in high pressure fuel pipeline during three injection processes of high-pressure common rail fuel injection system. The results show that the post-injection fuel volume is affected by the pressure fluctuation caused by the main injection. The change of fuel properties leads to the different phases of pressure fluctuation, which affects the opening of the needle. Thus, the cycle fuel injection volume decreases with the increase of the bulk modulus of elasticity of the fuel. Reference [28] studied the influence of pilot injection timing and pilot injection fuel volume on soot, NOx, combustion noise and fuel consumption rate of diesel engines. The results show that the pressure wave caused by specific pilot injection timing in common rail and high pressure fuel pipeline leads to the dramatic change of main injection cycle fuel injection volume, especially when the injection interval between pilot injection and main injection changes. It has important influence on soot and NOx. In order to reduce the repeated reflection and propagation of fuel pressure fluctuation in high pressure fuel circuits, Ref. [29–31] designed a pressure storage chamber at the outlet end of high pressure fuel pump and developed a new type of high-pressure common rail fuel injection system. By studying the cycle fuel injection volume of the system and the conventional high-pressure common rail fuel injection system under different common rail pressures, it was found that the different arrangement of the two fuel injection systems leads to the difference in fuel pressure wave propagation and reflection, which leads to the variation of fuel pressure fluctuation characteristics and causes the cycle fuel injection volume to be different.

In this chapter, the pressure fluctuation of high-pressure common rail fuel injection system will be investigated theoretically. The dynamic pressure wave fluctuation mechanism of the system will be analyzed through the experiments. On this basis, the influence of different parameters on dynamic pressure wave during fuel injection will be studied. The results will provide the support for revealing the cycle fuel injection volume fluctuation and its generation mechanism of high-pressure common rail fuel injection system.

2. Composition and working principle of high-pressure common rail fuel injection system

High-pressure common rail fuel injection system is mainly composed of the low pressure fuel supply part, including tank, low pressure pump, the high pressure fuel injection part, including high pressure pump, common rail, electrical control injector, the fuel return circuit, which transfers excess fuel from each part back to the tank, the electrical control part, including ECU and various sensors, as shown in **Figure 1**. During the work process of high-pressure common rail fuel injection system, the plunger of high pressure pump moves downward with the rotation of the camshaft under the action of spring. Fuel is sucked into the high pressure pump plunger chamber from the tank by the low pressure pump through the fuel filter to complete the fuel absorption process. The plunger of the high pressure pump moves upward with the rotation of the camshaft driven by the cam. Fuel in the plunger chamber is compressed and the pressure increases. The pressurized fuel is pumped to the common rail through the high pressure pipeline to complete the process of fuel pressurization and fuel supply. The high pressure fuel in the common rail is distributed to the

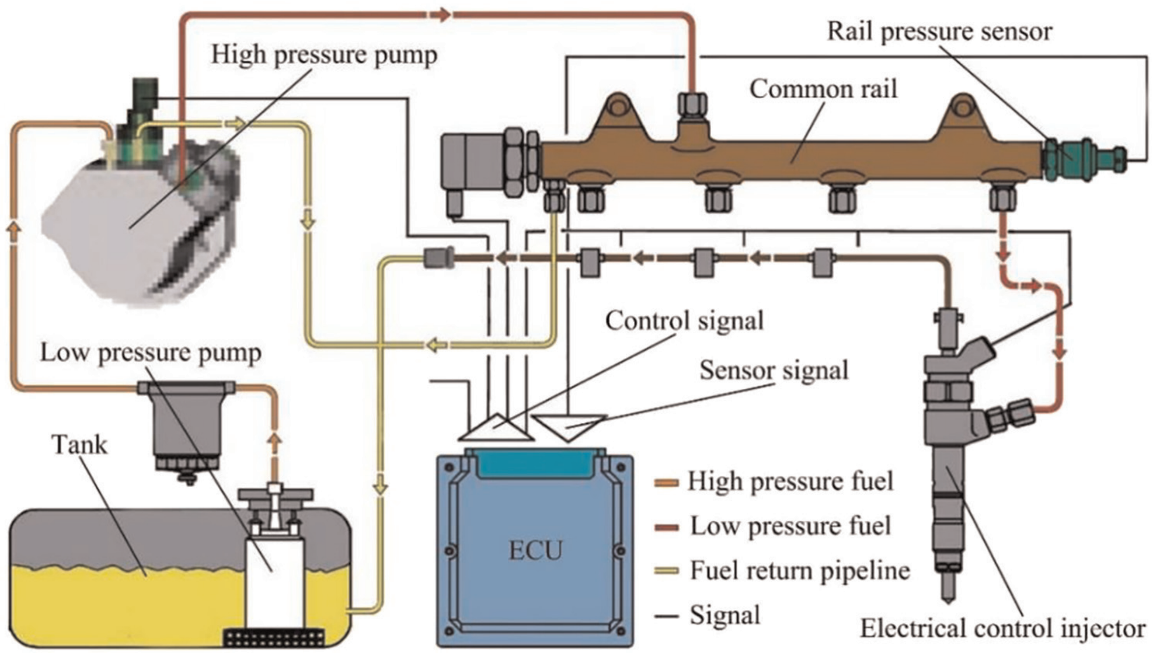

Figure 1.
Schematic diagram of the high-pressure common rail fuel injection system.

injector of each cylinder by the high pressure pipeline. Fuel is injected into the cylinder through the nozzle when the solenoid valve coil of the injector is energized. The needle closes the nozzle hole to finish fuel injection when the coil is de-energized, which completes a fuel injection process. The fuel metering valve on high pressure pump connects the plunger chamber of the low pressure pump and high pressure pump, which is opened a triangle fuel metering hole. The ECU controls fuel supply volume by adjusting the opening of the fuel metering hole through outputs pulse width modulation signal, thus realizing the adjustment of common rail pressure. Through the feedback signals of various sensors, ECU outputs corresponding control signals according to the working state of the diesel engine and drives the high speed solenoid valve on the injector to realize the control of injection timing, injection duration and injection times, to complete the real-time control of the fuel injection system.

3. Theoretical study on pressure fluctuation of high-pressure common rail fuel injection system

The fluctuation characteristics of fuel pressure in the pipeline of high-pressure common rail fuel injection system can be represented as one-dimensional partial differential equations of unstable compressible flow, as shown in Eq. (1).

$$\begin{cases} \dfrac{\partial p}{\partial t} + u\dfrac{\partial p}{\partial x} + a^2\rho\dfrac{\partial u}{\partial x} = 0 \\ \dfrac{\partial p}{\partial x} + \rho\dfrac{\partial u}{\partial t} + \rho u\dfrac{\partial u}{\partial x} + 2\kappa\rho u = 0 \end{cases} \tag{1}$$

where u is the fuel flow velocity, a is the fuel pressure wave propagation velocity and κ is the fuel flow resistance coefficient.

The above-mentioned partial differential equations can be converted into ordinary differential equations as follows.

$$\begin{cases} \left(\frac{\mathrm{d}x}{\mathrm{d}t}\right)_{\mathrm{R}} = u + a, & \dfrac{\mathrm{d}p}{\mathrm{d}t} + a\rho\dfrac{\mathrm{d}u}{\mathrm{d}t} + 2a\kappa\rho u = 0 \\ \left(\frac{\mathrm{d}x}{\mathrm{d}t}\right)_{\mathrm{L}} = u - a, & \dfrac{\mathrm{d}p}{\mathrm{d}t} - a\rho\dfrac{\mathrm{d}u}{\mathrm{d}t} - 2a\kappa\rho u = 0 \end{cases} \tag{2}$$

where $(\mathrm{d}x/\mathrm{d}t)_{\mathrm{R}}$ and $(\mathrm{d}x/\mathrm{d}t)_{\mathrm{L}}$ are the trace lines of pressure waves propagating to the right and left inside the pipeline. The above equations directly reflect the propagation relationship of the pressure wave in the pipeline of high-pressure common rail fuel injection system. When the fuel pressure wave propagates to a specific position in the pipeline, the pressure at this position rises, that is, $\mathrm{d}p$ is positive, and the pressure wave is a compression wave. Conversely, the pressure drops. $\mathrm{d}p$ is negative, and it is an expansion wave. Assuming that the right direction of x axis is the positive direction, the fuel pressure wave propagating along the forward direction is called the right–traveling wave, while the fuel pressure wave propagating along the negative direction is called the left–traveling wave. The right–traveling wave is represented by $\mathrm{d}p_{\mathrm{R}}$ and $\mathrm{d}u_{\mathrm{R}}$, and it has the following relationship.

$$\mathrm{d}p_{\mathrm{R}} = a\rho\mathrm{d}u_{\mathrm{R}} \tag{3}$$

While the left–traveling wave is represented by $\mathrm{d}p_{\mathrm{L}}$ and $\mathrm{d}u_{\mathrm{L}}$, which has the following relationship.

$$\mathrm{d}p_{\mathrm{L}} = -a\rho\mathrm{d}u_{\mathrm{L}} \tag{4}$$

The pressure wave in the pipeline of high-pressure common rail fuel injection system is divided into left-traveling wave and right-traveling wave according to the direction of propagation and divided into compression wave and expansion wave according to the change of fuel pressure caused by propagation. Therefore, pressure waves in the pipeline can be divided into the following four types.

a. Right-traveling compression wave

   This kind of fuel pressure wave propagates along the positive direction of x-axis, where $\mathrm{d}p_{\mathrm{R}}$ is positive and $\mathrm{d}u_{\mathrm{R}}$ is also positive, that is, the fuel pressure and velocity along the x-forward propagating direction increase.

b. Right-traveling expansion wave

   This kind of fuel pressure wave propagates along the positive direction of the x axis, but $\mathrm{d}p_{\mathrm{R}}$ is negative and $\mathrm{d}u_{\mathrm{R}}$ is also negative, that is, the fuel pressure and velocity along the x-forward propagating direction decrease.

c. Left-traveling compression wave

   This kind of fuel pressure wave propagates backward along the x-axis, where $\mathrm{d}p_{\mathrm{L}}$ is positive and $\mathrm{d}u_{\mathrm{L}}$ is negative, that is, the fuel velocity decreases along the x-forward propagating direction. However, the fuel pressure and velocity of left-traveling compression wave along with the propagating direction increase.

d. Left-traveling expansion wave

This kind of fuel pressure wave propagates backward along the x-axis, where $\mathrm{d}p_L$ is negative and $\mathrm{d}u_L$ is positive, that is, the fuel velocity along the x-forward propagating direction increases.

Supposing the right-traveling wave arriving at x position of the fuel pipeline at moment t is $\mathrm{d}p_R$ and $\mathrm{d}u_R$, and the left-traveling wave is $\mathrm{d}p_L$ and $\mathrm{d}u_L$. According to the pressure wave synthesis theory, the total pressure variation $\mathrm{d}p$ and velocity variation $\mathrm{d}u$ are

$$\begin{cases} \mathrm{d}p = \mathrm{d}p_R + \mathrm{d}p_L \\ \mathrm{d}u = \mathrm{d}u_R + \mathrm{d}u_L \end{cases} \tag{5}$$

where $\mathrm{d}p$ and $\mathrm{d}u$ are called synthetic pressure waves at (x, t). In fact, after the fuel pressure wave is synthesized, each single-traveling wave continues to propagate along its determined direction. Therefore, the above equation can also be understood as the pressure fluctuation in a certain position in the pipeline can be decomposed into left-traveling waves and right-traveling waves.

When the pressure wave propagates to the boundary surface, another returned pressure wave can be obtained based on the pressure wave and the boundary condition at the moment, which is the reflected wave. Assuming that the right end of the high-pressure common rail fuel injection system is closed, the boundary condition is $u = 0$ and $\mathrm{d}u = 0$. At this time, if there is a right-traveling wave $\mathrm{d}p_R$ arriving on the left, there must be a left-traveling reflected pressure wave $\mathrm{d}p_L$, so that $\mathrm{d}u = \mathrm{d}u_R + \mathrm{d}u_L = 0$. It can be seen that the fuel reflected pressure wave from the pipeline end boundary surface is the result of coupling the boundary condition with the propagating fuel pressure wave.

There are three boundary conditions for pressure wave propagation and reflection in the pipeline of high-pressure common rail fuel injection system. The boundary type at common rail and high pressure fuel pipeline is outlet isobaric end (outlet opening end). The boundary type is the closed-end when the nozzle needle valve is closed. The boundary type is the orifice flow ends when the needle opening nozzle and injector injection.

a. Outlet isobaric end

There are the following equations when the right-traveling wave $\mathrm{d}p_R$ reaches the open end of the right end.

$$\begin{cases} \mathrm{d}p_R = a\rho \mathrm{d}u_R \\ \mathrm{d}p_L = -a\rho \mathrm{d}u_L \\ \mathrm{d}p = \mathrm{d}p_R + \mathrm{d}p_L = 0 \\ \mathrm{d}u = \mathrm{d}u_R + \mathrm{d}u_L \end{cases} \tag{6}$$

Thus, $\mathrm{d}p_L = -\mathrm{d}p_R$, $\mathrm{d}u_L = \mathrm{d}u_R$, $\mathrm{d}u = 2\mathrm{d}u_R$, $\mathrm{d}p = 0$.

It can be seen that the signs of the incident pressure wave and reflected pressure wave are opposite when the pressure wave propagates to the common rail. However, the absolute value of the amplitude of the pressure wave is the same and the wave velocity is the same. The changing of the pressure at the pipeline end is zero and the changing of the velocity is twice that of the incident pressure wave. The reflection with the property of incident wave and reflected wave is opposite (an expansion wave, the other is compression wave) and the absolute value of the amplitude of the pressure wave is the same is called complete negative reflection. Therefore, the pressure wave reflection at common rail and high pressure fuel pipelines is a complete negative reflection.

b. Closed-end

There are the following equations when the right-traveling wave $\mathrm{d}p_\mathrm{R}$ reaches the closed end of the right end.

$$\begin{cases} \mathrm{d}p_\mathrm{R} = a\rho \mathrm{d}u_\mathrm{R} \\ \mathrm{d}p_\mathrm{L} = -a\rho \mathrm{d}u_\mathrm{L} \\ \mathrm{d}p = \mathrm{d}p_\mathrm{R} + \mathrm{d}p_\mathrm{L} \\ \mathrm{d}u = \mathrm{d}u_\mathrm{R} + \mathrm{d}u_\mathrm{L} = 0 \end{cases} \tag{7}$$

Thus, $\mathrm{d}p_\mathrm{L} = \mathrm{d}p_\mathrm{R}$, $\mathrm{d}u_\mathrm{L} = -\mathrm{d}u_\mathrm{R}$, $\mathrm{d}p = 2\mathrm{d}p_\mathrm{R}$, $\mathrm{d}v = 0$.

It can be seen that the amplitude of the incident pressure wave and reflected pressure wave is the same after the pressure wave propagates to the nozzle when the needle is closed. However, the velocity disturbance value is the opposite. The velocity change at the nozzle is zero and the pressure variation is twice the amplitude of the incident pressure wave. The reflection with the property of incident wave and reflected wave is the same, that is, an expansion wave or a compression wave at the same time, and the amplitude of pressure disturbance is the same is called complete positive reflection. Therefore, the pressure wave reflection when the needle valve is closed is a complete positive reflection.

c. Orifice flow end

As shown in **Figure 2**, the fuel flow from the C-C boundary surface of the pipeline with section F through the orifice of section F_t to the space where the backpressure is p_c. Assuming the initial state of fuel is $p_0 = p_c$ and $u_0 = 0$, the Bernoulli equation and continuity equation from C-C boundary surface to the minimum throat section t-t can be established as follows.

$$\begin{cases} p + \frac{1}{2}\rho u^2 = p_c + \frac{1}{2}\rho u_t^2 \\ Fu = \alpha_F F_t u_t \end{cases} \tag{8}$$

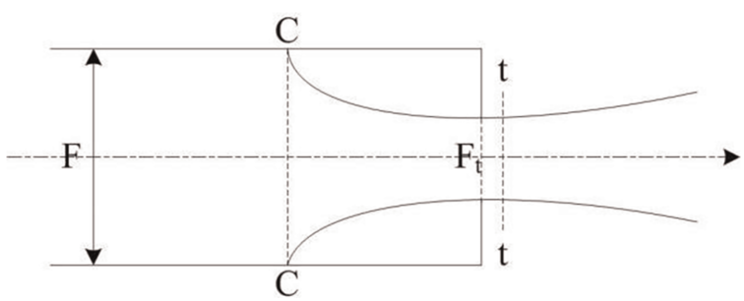

Figure 2.
Schematic of the orifice flow end.

where p and u are the fuel pressure and flow rate at C-C boundary surface, α_F is the orifice flow coefficient, and u_t is the fuel flow velocity at t-t of minimum throat section.

Thus,

$$p = \frac{1}{2}\rho u^2 \frac{1-\phi_F^2}{\phi_F^2} + p_c \tag{9}$$

where, $\varphi_F = \alpha_F F_t / F$, which is the effective flow section ratio at the orifice end.

The above equation is the boundary condition equation of the orifice outflow of the high-pressure common rail fuel injection system. It can be seen from the above equation that when $\varphi_F = 0$, $F_t = 0$, is a closed-end, and when $\varphi_F = 1$, $F_t = F$, is a outlet flow opening end. However, the reflection at the boundary surface transitions from complete positive reflection to complete negative reflection when φ_F changes from 0 to 1.

4. Test bench of dynamic pressure fluctuation for high-pressure common rail fuel injection system

The test bench mainly includes the high pressure common rail injection system test stand, the operation stand and the water-cooling unit. It can measure the dynamic injection characteristics of a six-cylinder engine, such as the fuel injection rate, the fuel injection volume and the fuel injection duration for each cycle at most. The main functions of the high-pressure common rail fuel injection system test bench are to drive high pressure fuel pump, supply fuel to high pressure fuel pump at specified temperature and pressure, drive fuel injection system and real-time measure the dynamic injection characteristics of the system. The operation stand includes monitoring system, electronic control device for single injection instrument, driving equipment of fuel injector, programmable synchronous timing pulse generator, common rail pressure regulator, torque analyzer and DC power supply, etc. It can realize the real-time control and monitor of the experimental process. All of the experimental data results and the environmental parameters of each injector can be recorded after the experiment.

The high-pressure common rail fuel injection system test bench is shown in **Figure 3**. It consists of a driving motor, high pressure fuel pump, common rail, common rail pressure sensor, oscilloscope, single injection instrument, mechanical part of injection flow and rate (IFR), injector, pressure sensor, high pressure pipeline, electronic part of IFR, ECU, computer terminal and tank. The high pressure fuel pump is driven by the driving motor, which can provide a stable speed input for the system. The common rail pressure sensor installed on the common rail measures the common rail pressure in real-time and feeds the common rail pressure signal back to the ECU. The ECU adjusts the fuel flow into the common rail by adjusting the flow control valve on the high pressure fuel pump to stabilize the common rail pressure. In order to measure the dynamic pressure fluctuation at the injector inlet, a piezoresistive high pressure sensor is installed near the injector end on the high pressure pipeline between the common rail and the injector. The oscilloscope receives the pressure signal at the injector inlet measured by the pressure sensor and the common rail pressure signal measured by the common rail pressure sensor and stores

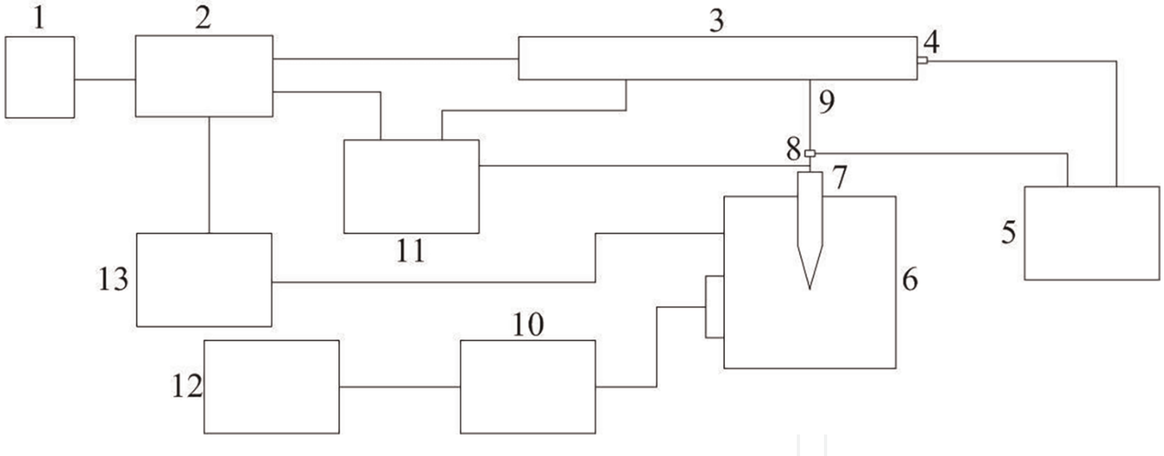

Figure 3.
Schematic diagram of the high-pressure common rail fuel injection system test bench. 1. Drive motor 2. High pressure fuel pump 3. Common rail 4. Common rail pressure sensor 5. Oscilloscope 6. Mechanical part of IFR 7. Injector 8. Pressure sensor 9. High pressure pipeline 10. Electronic part of IFR 11. ECU 12. Computer terminal 13. Tank.

the signal data. The ECU provides control current to the injector according to experiment conditions to complete fuel injection under different conditions.

5. Fluctuation mechanism of dynamic pressure wave for high-pressure common rail fuel injection system

The fuel pressure wave reciprocating propagates within high-pressure common rail fuel injection system during fuel injection. The cycle fuel injection volume is affected by the fuel injection pressure. Therefore, it has important theoretical and practical significance on system optimization design and taking effective method to reduce the adverse impact of pressure fluctuations on cycle fuel injection volume characteristics by thorough analysis on the fluctuation mechanism and influence rule of dynamic pressure wave for high-pressure common rail fuel injection system. Theoretically, the fuel injection pressure refers to the fuel pressure near the nozzle hole. However, due to the fuel pressure in the nozzle is high and the size of the fuel cavity in the nozzle is small, it is difficult to install the pressure sensor near the nozzle hole. More importantly, the installation of a pressure sensor close to the nozzle hole will cause the change of flow field distribution in the nozzle, thus affecting the dynamic pressure fluctuation characteristics of the system. Since the fuel pressure wave propagation in the system with a limited speed, the fuel pressure of the injector inlet only slightly lags behind the nozzle volume pressure at time sequence (fuel injection pressure). It is easily measured and can actually represent the dynamic pressure fluctuation characteristics of the system. In this chapter, the fuel pressure of the injector inlet is used instead of injection pressure to analyze the dynamic pressure wave of the system [32].

Figure 4 shows the characteristics of solenoid valve drive current, fuel injection rate and injector inlet pressure before and after fuel injection of high-pressure common rail fuel injection system when the cycle fuel injection volume is 30 mm^3. As shown in the figure, there is a delay characteristic between solenoid valve energized and fuel injection due to the hydraulic delay of the system. In addition, the fuel injection duration is longer than the solenoid valve energized time. The opening of the

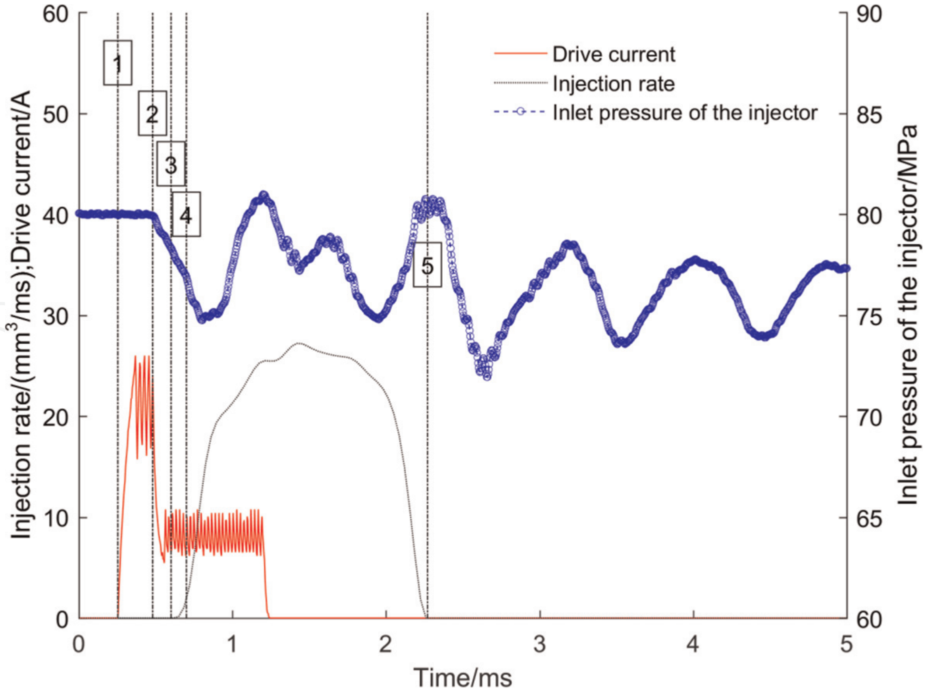

Figure 4.
Drive current of the solenoid valve, injection rate and inlet pressure of the injector at a fuel injection volume of 30 mm^3.

solenoid valve and needle causes the pressure drop at the injector inlet. The fuel injection duration is consistent with the opening time of the needle due to the dynamic response characteristics of the needle.

According to the various characteristics of driving current of the solenoid valve, fuel injection rate and injector inlet pressure at different moments, as shown in **Figure 4**, the curves can be divided into five different stages as follows.

1–2 stages. The solenoid valve coil of the injector is energized, the current gradually increases and the electromagnetic force increases. But the electromagnetic force of the solenoid valve is less than the pretightening force of the control valve reset spring. The control valve is pressed against the control seat and the outlet orifice is not open. The control chamber is filled with high pressure fuel. The resultant of fuel hydraulic pressure on the upper end surface of the needle and pretightening force of reset spring of the needle is larger than the fuel hydraulic pressure on the lower end surface of the needle. The needle is pressed against the needle seat and the injector does not injection fuel. Thus, the injector inlet pressure remains unchanged at 80 MPa.

2–3 stages. The injector inlet pressure drops. The reason is analyzed as follows. The control valve overcomes the pretightening force of the control valve reset spring and moves upward under the action of electromagnetic force of the solenoid valve, and the outlet orifice is opened. The high pressure fuel in the control chamber is discharged to the tank through the low pressure return fuel circuit. The fuel pressure in the control chamber drops rapidly. However, the resultant of fuel hydraulic pressure on the upper end surface of the needle and pretightening force of reset spring of the needle is still larger than the fuel hydraulic pressure on the lower end surface of the needle. The needle is still pressed against the needle seat. The nozzle hole is closed and the injector does not inject fuel. The control valve opens the outlet orifice suddenly arousing an instantaneous expansion wave, which starts between the control valve and the control valve seat.

3–4 stages. The injector inlet pressure continues to drop, but the pressure drop gradient increases. The reason is that the resultant of fuel hydraulic pressure on the upper end surface of the needle and pretightening force of reset spring of the needle is less than the fuel hydraulic pressure on the lower end surface of the needle as the decreasing of the control chamber fuel pressure. The needle moves upward and opens the nozzle hole. The injector starts fuel injection and the fuel injection rate appears. The sudden opening of the needle also arouses an instantaneous expansion wave, which starts between the needle and the needle seat and propagates upward. Due to the existence of the needle channel orifice on the injector body, the pressure drop at the injector inlet is not significant, however, the pressure drop gradient is larger than that when the control valve is opened alone.

4–5 stages. Complete fuel injection process. The expansion wave aroused by the moving parts working processes of the system propagates upward along the fuel circuit in the injector. When it propagates to the common rail, reflecting back a compression wave. This compression wave attempts to recover the fuel pressure in the fuel circuit to the initial value. When it propagates to the injector inlet causes the inlet pressure increasing, as shown in **Figure 4**. In fact, there is no expansion wave generated between the needle and the needle seat when the needle reaches its maximum lift, and the size of the nozzle hole becomes the main factor limiting fuel injection.

Stage after 5. The nozzle hole is closed by the needle and fuel injection is stopped. The closing of the needle will cause a water hammer effect in the system, a compression wave in the nozzle aroused and propagates upward along the fuel circuit in the injector. The inlet pressure increases when it propagates to the injector inlet, as shown in **Figure 4**. Since then, the needle and control valve shut down completely. The pressure wave propagates repeatedly in the system. Because the hydraulic shear resistance restrains the pressure wave oscillation, the amplitude of the fuel pressure wave decreases gradually, and the pressure at the injector inlet shows an attenuation oscillation characteristic.

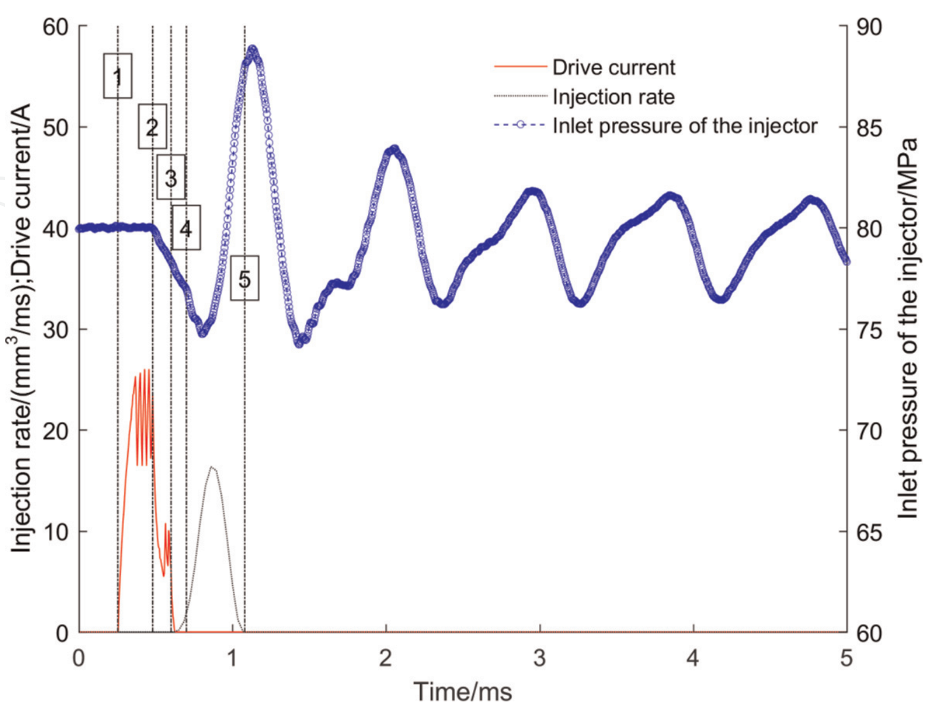

Figure 5.
Drive current of the solenoid valve, injection rate and inlet pressure of the injector at a fuel injection volume of 3 mm^3.

It can be seen from the above analysis that the pressure fluctuation characteristic of the injector inlet before the opening of the needle is independent of the energized time of the solenoid valve coil or the fuel injection duration since this phenomenon will be caused whenever the control valve or the needle starts to move. The pressure fluctuation characteristic of the injector inlet caused by the water hammer effect is obviously dependent on the energized time of the solenoid valve coil since it is generated after the needle valve is closed. Therefore, when the energized time of the solenoid valve coil is shorter, the time interval between the two pressure peaks in **Figure 4** is small, the third pressure peak and the subsequent pressure oscillation peak depend on the energized time of the solenoid valve coil due to the pressure wave interaction. As shown in **Figure 5**, the pressure fluctuation amplitude of the injector inlet is significant when the cycle fuel injection volume of the system is 3 mm^3. The fusion of the two pressure peaks is called hydraulic resonance as shown in **Figure 4**.

6. Study on the influence factors of dynamic pressure wave in high-pressure common rail fuel injection system

According to the wave mechanism of dynamic pressure wave for high-pressure common rail fuel injection system, the fuel injection rate and fuel injection duration are different with different fuel injection pulse widths, which results in different cycle fuel injection volumes. The change of injection pulse width has a different influence on pressure fluctuation characteristics in the system when the fuel is injected. In addition, the cycle fuel injection volume is different even if the injection pulse width is the same when the high-pressure common rail fuel injection system is under different common rail pressures. Therefore, this section mainly analyzes the influence rule of two key control parameters of the system, namely injection pulse width, and common rail pressure, on the dynamic pressure wave in the system, which provides support for the study of the fluctuation characteristics of cycle fuel injection volume of the system.

Figure 6 shows the pressure fluctuation characteristics of injector inlet during fuel injection of high-pressure common rail fuel injection system with injection pulse width of 400, 600, 800 and 1000 μs, respectively. It can be seen from the figure that under the same high pressure pipeline size and common rail pressure, the injector inlet pressure with different injection pulse widths shows attenuation fluctuation characteristics. The smaller the injection pulse width, the larger the pressure fluctuation amplitude during the injection duration. With the increase of injection pulse width from 400 μs to 1000 μs, the change rate of pressure fluctuation amplitude at injector inlet decreases, and the average injector inlet pressure increases after injection. This is because the system circulates less fuel injection with small pulse width under the same common rail pressure. After the needle is seated and the nozzle is closed, the high pressure fuel in common rail immediately flows through the high pressure pipeline to replenish that injected in the injector. The larger the injection pulse width, the more fuel needed to replenish and the longer the time required. In addition, with the increase of injection pulse width, the needle gradually reaches its maximum lift. At this time, the injection pulse width only affects the moment when the needle closes the nozzle but has no influence on the needle from opening to reaching its maximum lift. Therefore, as shown in the figure, there is no significant difference between the injector inlet pressure from the first trough to the first crest when the injection pulse width is 800 μs and 1000 μs.

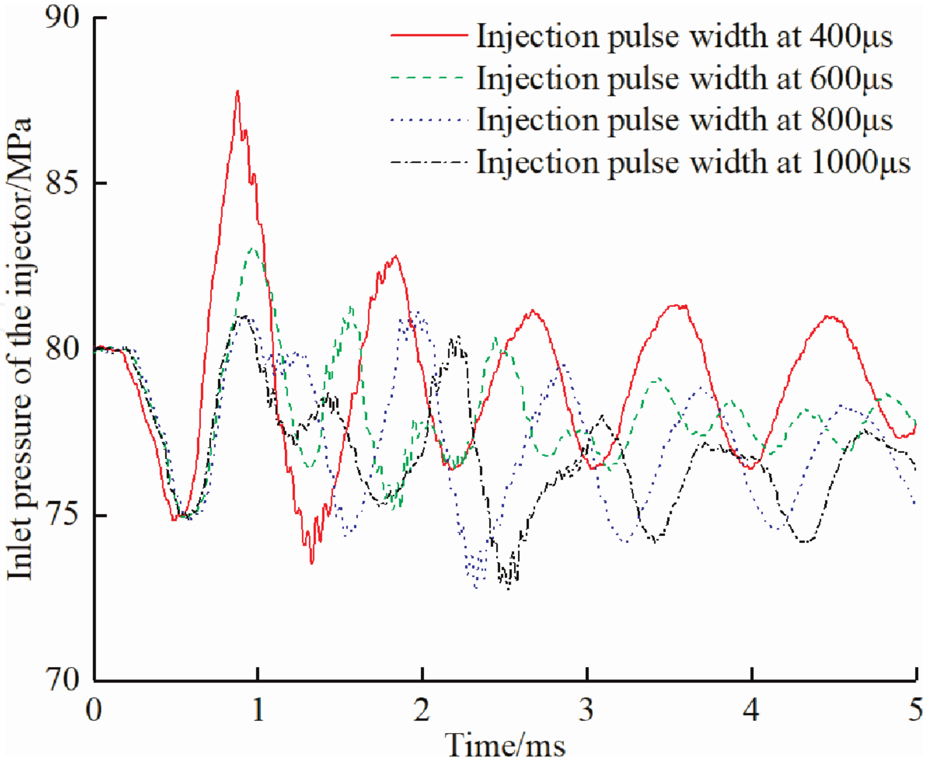

Figure 6.
Inlet pressure of the injector at different injection pulse widths.

Figure 7 shows the pressure fluctuation characteristics of injector inlet during fuel injection of high-pressure common rail fuel injection system with rail pressures of 40, 80, 120 and 160 MPa, respectively. Each point in the figure is the difference between the injector inlet pressure at specific common rail pressure and the set common rail pressure when the injection pulse width is 800 μs, which reflects the pressure fluctuation characteristics of the system under different common rail pressures more intuitively. As shown in the figure, the inlet pressure of the injector decreases to a certain extent under different common rail pressures when the size of the high pressure pipeline and injection pulse width is constant. The inlet pressure fluctuation of the injector under the common rail pressure of 40 MPa is obviously different from that under the other three common rail pressures. The average pressure fluctuation of the injector inlet under this common rail pressure is higher than that under the other three common rail pressures. The inlet pressure fluctuation rules are consistent when the common rail pressure increases from 80 MPa to 160 MPa. The higher the common rail pressure, the larger the inlet pressure drop amplitude and the lower the average value of pressure fluctuation. When the size of the high pressure pipeline and injection pulse width is the same, the injection pressure increases with the increase of common rail pressure, and the injection pulse width required by the needle to reach the maximum lift decreases. When the common rail pressure is 40 MPa, due to the low common rail pressure, the moment when the control valve fully opens outlet orifice lags behind, and the moment when the control valve closes outlet orifice is advanced. The fuel pressure relief time in the control chamber is shortened and the needle does not reach its maximum lift. The nozzle is closed again before it is fully opened. At this time, the maximum fuel injection rate and the fuel injection duration of the system are small. Therefore, the average value of pressure fluctuation at the injector inlet is high. With the increase of common rail pressure, the difference of control valve opening outlet orifice decreases. But the pressure difference between the control chamber and low

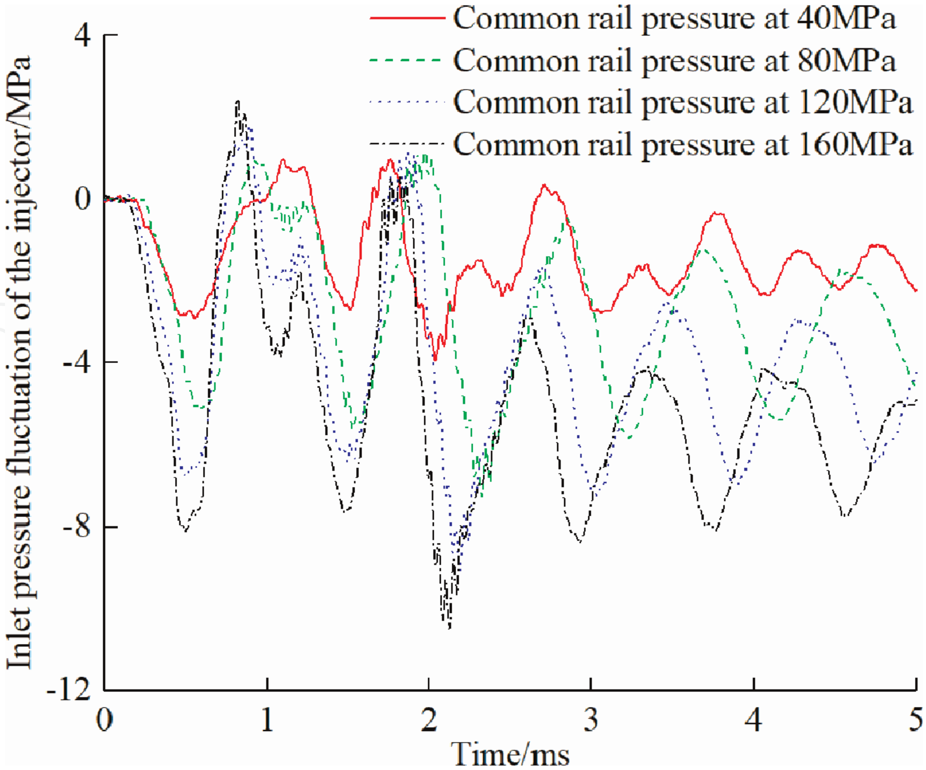

Figure 7.
Inlet pressure fluctuation of the injector at different common rail pressures. (a) Common rail pressure at 40 MPa. (b) Common rail pressure at 80 MPa. (c) Common rail pressure at 120 MPa. (d) Common rail pressure at 160 MPa.

pressure fuel circuit and nozzle volume and cylinder is larger when the common rail pressure is high. The fuel discharge rate of the control chamber and the fuel injection rate of the nozzle hole are accelerated, which results in the system pressure drop gradient increases. In addition, the higher the common rail pressure, the longer the needle is maintained at the maximum lift position. This is the main reason why the higher the common rail pressure, the larger the injector inlet pressure drop amplitude, the lower the average pressure fluctuation.

It can be seen from the above analysis that the dynamic pressure wave of the system shows different fluctuation characteristics under different injection pulse widths and common rail pressures when the size of the high pressure pipeline is constant. Therefore, the dynamic pressure fluctuation frequency and amplitude of the system are further analyzed with the injection pulse width of 400, 600, 800 and 1000 μs and common rail pressure of 40, 80, 120 and 160 MPa, respectively, to reveal the dynamic pressure wave variation rule of the system under different injection pulse width and common rail pressure.

The area enclosed below the power spectrum density curve represents the amount of energy generated by the fluctuation in the frequency range [33]. **Figure 8** shows the power spectrum density obtained by the fast Fourier transform of injector inlet pressure fluctuation under different injector pulse widths and common rail pressures. As shown in **Figure 8(a)**, when the common rail pressure is 40 MPa and the injection pulse width is 400, 600 and 1000 μs, the dynamic pressure wave energy of the system is mainly in the frequency band of 799 Hz–1199 Hz, and the crest characteristics of power spectrum density are significant, and all reach the main crest at the frequency of 999 Hz. At this time, the dynamic pressure wave in the system shows obvious periodic fluctuation characteristics, which mainly fluctuates in the frequency of the

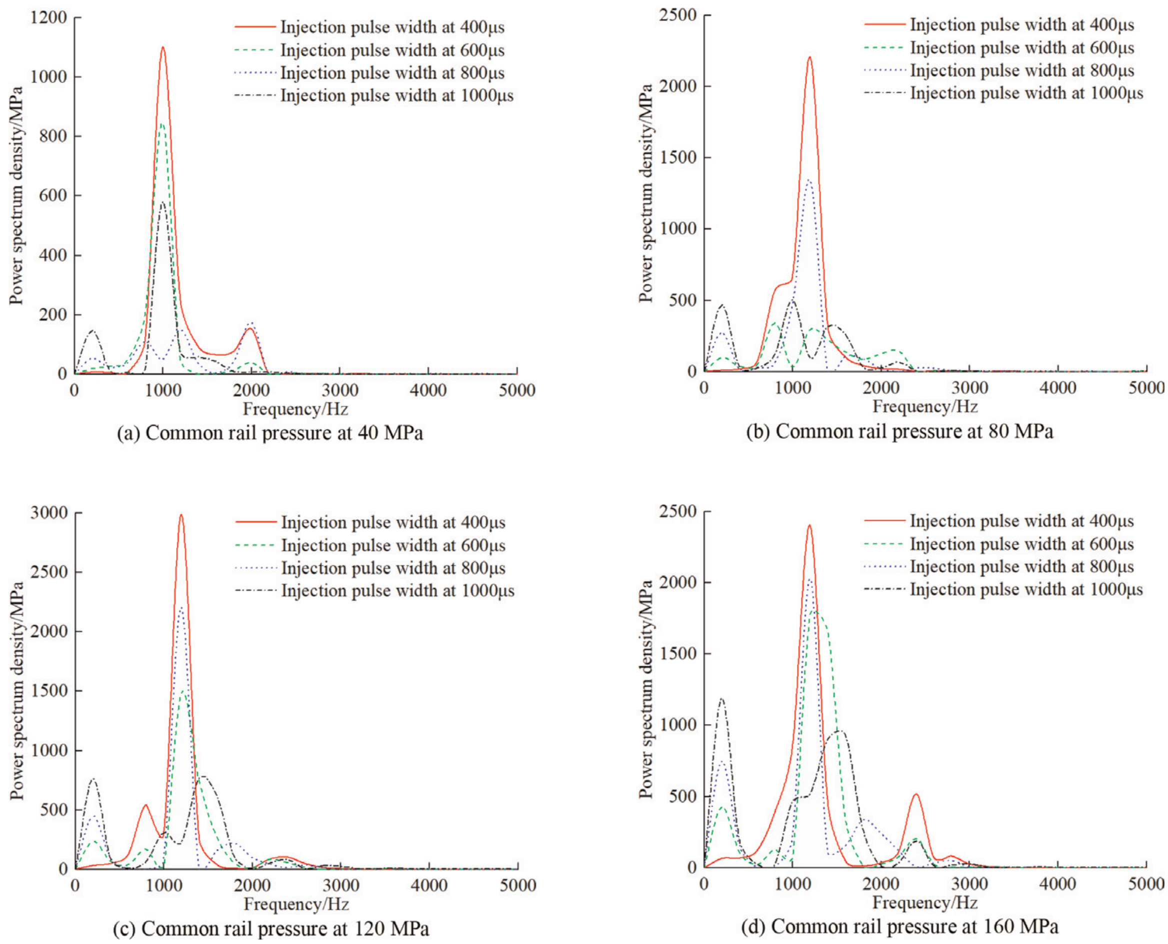

Figure 8.
Power spectrum density of the injector inlet pressure at different common rail pressures and injection pulse widths.

main crest. When the fuel injection pulse width is 800 μs, the crest characteristics of the dynamic pressure wave power spectrum density are not obvious, and the fuel pressure fluctuation does not show significant periodic fluctuation characteristics. The pressure wave mainly fluctuates at the frequency of 799, 1199 and 1998 Hz. This may be because the pressure wave frequency aroused by the control valve and needle movement in the system reaches the resonance frequency under this injection pulse width, and all kinds of pressure waves propagate repeatedly and superimpose in the system, which changes the pressure wave frequency characteristics. At this time, the fluctuation characteristic of the system is the most complex, and the influence on the dynamic injection characteristic of the system is the most serious.

The injector inlet pressure power spectrum density differs greatly under the four injector pulse widths when the common rail pressure is 80 MPa. The dynamic pressure wave energy of the system is between the frequency band of 599 Hz to 1398 Hz and 799 Hz to 1398 Hz, respectively when the injection pulse width is 400 μs and 800 μs. The power spectrum densities of pressure waves at the injector inlet under the two injection pulse widths have significant crest characteristics, both of which show obvious periodic fluctuation characteristics at the main crest frequency of 1199 Hz. The dynamic pressure wave of the system does not show periodic fluctuation when the injection pulse width is 600 μs and 1000 μs, which shows multi-frequency characteristics. As shown in **Figure 8(b)**, the pressure wave at the injector inlet mainly fluctuates at the frequency of 799 Hz and 1199 Hz when the injection pulse width is

600 μs. The dynamic pressure wave mainly fluctuates at the frequency of 199 Hz and 999 Hz when the injection pulse width is 1000 μs.

As shown in **Figure 8(c)**, the power spectrum density of pressure wave at the injector inlet shows obvious crest characteristics when the common rail pressure is 120 MPa and the injection pulse width is 400, 600 and 800 μs, respectively. The main crest frequency of the pressure wave power spectrum density is 1199 Hz under three injection pulse widths. However, the energy frequency bands are different. The dynamic pressure wave energy is mainly between the frequency band of 799 Hz–1398 Hz when the injection pulse width is 400 μs and 800 μs. The main frequency band of the dynamic pressure wave becomes wider when the injection pulse width is 600 μs, ranges from 799 Hz to 1798 Hz. At the same time, the dynamic pressure wave of the system shows the characteristics of multi-frequency fluctuation when the injection pulse width is 1000 μs, which mainly fluctuates at the frequency of 199 Hz and 1398 Hz.

As shown in **Figure 8(d)**, the variation rule of pressure wave power spectrum density at the injector inlet is similar to that of common rail pressure is 120 MPa when the common rail pressure is 160 MPa and the injection pulse width is 400, 600 and 800 μs, respectively. The dynamic pressure wave mainly fluctuates periodically at the main crest frequency of 1199 Hz. But the energy bands of dynamic pressure wave are different under three injection pulse widths, and the energy band of dynamic pressure wave becomes smaller with the increase of injection pulse width. The main energy bands of the dynamic pressure wave are 599 Hz–1598 Hz, 999 Hz–1798 Hz and 999 Hz–1398 Hz, respectively when the injection pulse width increases from 400 μs to 800 μs. The dynamic pressure wave of the system also shows the multi-frequency fluctuation characteristics when the injection pulse width is 1000 μs, which mainly fluctuates at the frequency of 199 Hz and 1598 Hz.

The fuel density increases with the increase of pressure, which results in the acceleration of pressure wave propagation in the system. Comparing the pressure wave power spectrum density of injector inlet at different common rail pressures under the same injection pulse width in **Figure 8**, it can be seen that the crest characteristics of pressure wave power spectrum density at injector inlet under different common rail pressures are significant, except for the pressure wave multi-frequency fluctuation operating points. The main crest frequency of the injector inlet pressure wave power spectrum density is the lowest when the common rail pressure is 40 MPa, that is, the pressure wave frequency in the system is low when the common rail pressure is low.

To sum up, the dynamic pressure wave of the system has different frequency characteristics under different injection pulse widths and common rail pressure when the size of the high pressure pipeline is constant. It either fluctuates at the main crest frequency or shows the characteristics of multi-frequency fluctuation. While the dynamic pressure wave of the system shows low frequency fluctuation under a low common rail pressure at the same injection pulse width.

As shown in **Figures 6** and **7**, the average injector inlet pressure varies with different common rail pressure and injection pulse width. The difference between average injector inlet pressure and setted common rail pressure not only reflects the decreased amplitude of injection pressure in the fuel injection process but also reflects the average amplitude of pressure fluctuation in the system after fuel injection. Therefore, the average pressure drop is defined in this chapter as the difference between the setted common rail pressure and the average injector inlet pressure which locating the moment of injector solenoid valve energized time coordinates from 0 to

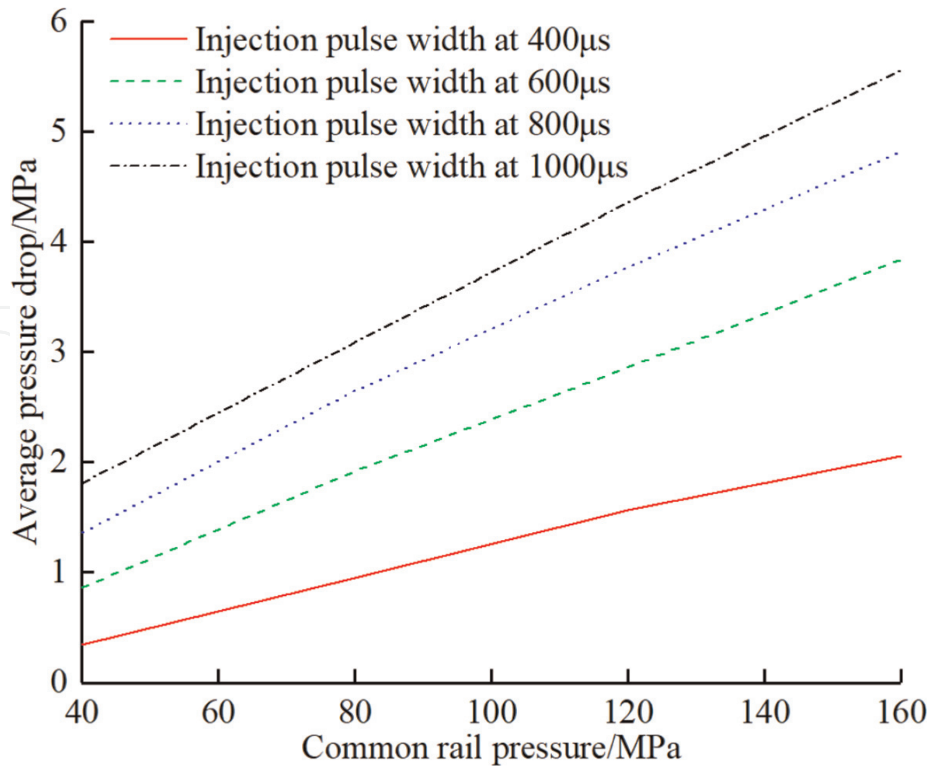

Figure 9.
Variation characteristics of the average pressure drop caused by common rail pressure at different injection pulse widths.

10 ms. **Figure 9** shows the average pressure drop with common rail pressure under different injection pulse widths. As shown in the figure, the average pressure drop increases linearly with the increase of common rail pressure from 40 MPa to 160 MPa when the size of the high pressure fuel pipeline and injection pulse width is constant. The larger the injection pulse width, the faster the average pressure drop increase rate. In addition, the average pressure drop increases with the increase of injection pulse width from 400 μs to 1000 μs when the size of high pressure fuel pipeline and common rail pressure are constant, and the average pressure drop amplitude increases with the increase of common rail pressure at the two same injection pulse widths. It can be seen that the higher the common rail pressure and the injection pulse width, the larger the average pressure drop.

The first trough of the pressure wave at the injector inlet is the minimum pressure of the system after the fuel injection when the nozzle hole is opened, which reflects the maximum pressure drop after stable fuel injection. The first crest of injector inlet pressure is the maximum compression wave returned in the system when the injection pulse width is large. It reflects the pressure fluctuation amplitude of hydraulic resonance when the injection pulse width is small. Therefore, the first trough and the first crest of the injector inlet pressure are the characteristic parameters reflecting the dynamic pressure wave characteristics of the system. **Tables 1** and **2** show the trough and crest values of injector inlet pressure fluctuation when the common rail pressure is 40, 80, 120 and 160 MPa and the injection pulse width is 400, 600, 800 and 1000 μs, respectively. For comparative analysis, all values in **Table 1** are the difference between the setted common rail pressure and the injector inlet pressure wave trough, and all values in **Table 2** are the difference between injector inlet pressure wave crest and setted common rail pressure.

Common rail pressure/MPa	Injection pulse width/μs			
	400	600	800	1000
40	2.91	2.89	2.91	2.84
80	5.17	5.15	5.14	5.15
120	6.77	6.74	6.75	6.79
160	7.98	8.14	8.13	8.05

Table 1.
Trough of the injector inlet pressure fluctuation at different common rail pressures and injection pulse widths.

Common rail pressure/MPa	Injection pulse width/μs			
	400	600	800	1000
40	5.34	2.32	0.98	0.10
80	7.82	3.06	0.96	1.04
120	6.12	3.77	1.83	1.87
160	6.75	3.84	2.46	2.50

Table 2.
Crest of the injector inlet pressure fluctuation at different common rail pressures and injection pulse widths.

As shown in **Table 1**, the troughs of injector inlet pressure fluctuation increase approximately linearly with the increase of common rail pressure under different injection pulse widths when the high pressure fuel pipeline size is constant. The change of injector pulse width has little effect on the trough of injector inlet pressure fluctuation under the same common rail pressure. The trough of injector inlet pressure fluctuation between four injector pulse widths has a maximum difference of 0.16 MPa under the same common rail pressure. It can be seen that the trough of dynamic pressure fluctuation is independent of injection pulse width, but increases with the increase of common rail pressure. The reasons are as follows. The higher the common rail pressure, the faster the opening response of the needle. The fuel injection rate increases after the needle is opened during the same time, the effective flow area at the nozzle hole increases and the fuel in nozzle volume injects through the nozzle hole more quickly. The pressure drop of the system increases and the trough of the pressure wave increases. The increase of injection pulse width under the same common rail pressure does not affect the early opening of the needle. That is, the needle motion state is the same before the injection pulse width is 400 μs. Therefore, the change of injection pulse width does not affect the trough of system pressure fluctuation.

As shown in **Table 2**, the crests of injector inlet pressure fluctuation increase with the increase of common rail pressure under the same injection pulse width when the size of high pressure fuel pipeline is constant. The smaller the injection pulse width, the higher the crest of injector inlet pressure fluctuation under the same common rail pressure. This is because the higher the common rail pressure with the same injection pulse width, the faster the fuel pressure wave propagates in the system. The superposition time of the large amplitude compression wave reflected from the common rail and the expansion wave aroused by the opening of the needle is advanced, which leads to the increase of the crest of pressure fluctuation at the injector inlet. The smaller the injection pulse width, the shorter the injection duration when the common rail pressure is the same. The short of opening and closing time of the needle will lead to a decrease in the encounter time of the first pressure crest and the third pressure crest as shown in **Figure 4**. The hydraulic resonance effect of the fuel pressure wave is more significant. Therefore, the crest of injector inlet pressure fluctuation decreases with the increase of injection pulse width at the same common rail pressure.

7. Conclusions

In this chapter, the pressure fluctuation of high-pressure common rail fuel injection system is studied theoretically. On the basis of revealing the wave mechanism of dynamic pressure wave, the influence of different parameters on the dynamic pressure wave of the fuel injection is investigated. The conclusions are as follows.

1. The theoretical study of pressure fluctuation for high-pressure common rail fuel injection systems shows that the reflected fuel pressure wave returned from the boundary surface of the pipeline end is the result of the coupling of boundary conditions and propagated pressure wave. The reflection of the pressure wave at common rail and high pressure fuel pipeline is a complete negative reflection. The reflection of the pressure wave when the needle closing is a complete positive reflection. The boundary condition type of needle opening nozzle hole is an orifice flow outlet end.

2. The dynamic pressure wave mechanism in high-pressure common rail fuel injection system is revealed. The results show that there is a delay characteristic from the solenoid valve energizing to fuel injection, and the fuel injection duration is longer than the solenoid valve energized time. The opening of the solenoid valve and needle causes the drop of injector inlet pressure. The fuel injection duration is consistent with the opening time of the needle. The fluctuation characteristics of injector inlet pressure before the opening of the needle are independent of the solenoid valve energized time or fuel injection duration. However, the fluctuation characteristics of injector inlet pressure caused by the water hammer effect when the needle closing obviously dependent on the solenoid valve energized time.

3. The influence rules of injection pulse width and common rail pressure on dynamic pressure wave of high-pressure common rail fuel injection system are analyzed. The results show that the inlet pressure of the injector fluctuates in attenuation mode when the injection pulse width is different. The smaller the injection pulse width, the larger the amplitude of pressure fluctuation during fuel injection duration. The change rate of inlet pressure fluctuation amplitude decreases with the increase of injection pulse width. The average injector inlet pressure increases after fuel injection. The inlet pressure of the injector decreases to some extent under different common rail pressure. The average inlet pressure fluctuation of the injector is higher than that of the other three common rail pressures when the common rail pressure is 40 MPa. The inlet pressure fluctuation rules are consistent when the common rail pressure increases from 80 MPa to 160 MPa. The higher the common rail pressure, the larger the injector inlet pressure drop amplitude and the lower the average pressure fluctuation. The dynamic pressure wave of the system has different frequency characteristics

under different injection pulse widths and common rail pressure. It either fluctuates at the main crest frequency or shows the characteristics of multi-frequency fluctuation. However, the dynamic pressure wave of the system shows low frequency fluctuation characteristics under a low common rail pressure at the same injection pulse width. The average pressure drop increases linearly with the increase of common rail pressure, and the increase rate of average pressure drop is faster with the increase of injection pulse width. In addition, it increases with the increase of fuel injection pulse width, and the higher the common rail pressure between the two same fuel injection pulse widths, the larger the increased amplitude of the average pressure drop. Both the trough and crest of dynamic pressure waves increase with the increase of common rail pressure. The smaller the fuel injection pulse width, the higher the crest of the pressure wave at the injector inlet.

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Conflict of interest

The authors declare no conflict of interest.

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References

[1] Kim K, Si W, Jin D, Kim J-H, Cho J, Baek S, et al. Characterization of engine oil additive packages on diesel particulate emissions. Journal of Mechanical Science and Technology. 2020;**34**(2):931-939

[2] Wang TJ. Effects of insulation on exhaust temperature and subsequent SCR efficiency of a heavy-duty diesel engine. Journal of Mechanical Science and Technology. 2019;**33**(2):923-929

[3] Ismael MA, Heikal MR, Aziz ARA, Syah F, Zainal EZA, Crua C. The effect of fuel injection equipment on the dispersed phase of water-in-diesel emulsions. Applied Energy. 2018; **15**(222):762-771

[4] Balz R, von Rotz B, Sedarsky D. In-nozzle flow and spray characteristics of large two-stroke marine diesel fuel injectors. Applied Thermal Engineering. 2020;**180**:115809

[5] Yu H, Goldsworthy L, Brandner PA, Li J, Garaniya V. Modelling thermal effects in cavitating high-pressure diesel sprays using an improved compressible multiphase approach. Fuel. 2018; **15**(222):125-145

[6] Ghiji M, Goldsworthy L, Brandner PA, Garaniya V, Hield P. End of injection process in a single-hole diesel injector. At Sprays. 2018;**28**(1): 23-45

[7] Wu X, Deng J, Cui H, Xue F, Zhou L, Luo F. Numerical simulation of injection rate of each nozzle hole of multi-hole diesel injector. Applied Thermal Engineering. 2016;**108**:793-797

[8] Qiu T, Song X, Lei Y, Liu X, An X, Lai M. Influence of inlet pressure on cavitation flow in diesel nozzle. Applied Thermal Engineering. 2016;**25**(109): 364-372

[9] Soriano JA, Mata C, Armas O, Ávila C. A zero-dimensional model to simulate injection rate from first generation common rail diesel injectors under thermodynamic diagnosis. Energy. 2018;**1**(158):845-858

[10] Oerley F, Hickel S, Schmidt SJ, Adams NA. Large-Eddy simulation of turbulent, cavitating fuel flow inside a 9-hole diesel injector including needle movement. International Journal of Engine Research. 2017;**18**(3):195-211

[11] Ghiji M, Goldsworthy L, Garaniya V, Brandner PA, Hield P, Novozhilov V, et al. Effect of residual air bubbles on diesel spray structure at the start of injection. Fuel. 2019;**1**(241):25-32

[12] Ferrari A, Paolicelli F. Modal analysis of fuel injection systems and the determination of a transfer function between rail pressure and injection rate. Journal of Engineering for Gas Turbines and Power. 2018;**140**(11): 112808-112808-112808-112811

[13] Luo T, Jiang S, Moro A, Wang C, Zhou L, Luo F. Measurement and validation of hole-to-hole fuel injection rate from a diesel injector. Flow Measurement and Instrumentation. 2018;**61**:66-78

[14] Rehman KU, Liu X, Wang H, Zheng L, Rehman RU, Cheng X, et al. Effects of black soldier fly biodiesel blended with diesel fuel on combustion, performance and emission characteristics of diesel engine. Energy Conversion and Management. 2018;**173**:489-498

[15] Lee Y, Lee CH. An uncertainty analysis of the time-resolved fuel

injection pressure wave based on BOSCH method for a common rail diesel injector with a varying current wave pattern. Journal of Mechanical Science and Technology. 2018;**32**(12):5937-5945

[16] Piano A, Boccardo G, Millo F, Cavicchi A, Postrioti L, Pesce FC. Experimental and numerical assessment of multi-event injection strategies in a solenoid common-rail injector. SAE International Journal of Engines. 2017 [cited 2017 Sep 18];**10**(4):2129-2140. Available from: http://papers.sae.org/2017-24-0012/

[17] Yu H, Goldsworthy L, Brandner PA, Garaniya V. Development of a compressible multiphase cavitation approach for diesel spray modelling. Applied Mathematical Modelling. 2017; **45**:705-727

[18] Rounthwaite NJ, Williams R, McGivery C, Jiang J, Giulliani F, Britton B. A chemical and morphological study of diesel injector nozzle deposits - insights into their formation and growth mechanisms. SAE International Journal of Fuels and Lubricants. 2017;**10**(1): 106-114

[19] Ubertini S. Injection pressure fluctuations model applied to a multidimensional code for diesel engines simulation. Journal of Engineering for Gas Turbines and Power. 2006;**128**(3):694

[20] Catalano LA, Tondolo VA, Dadone A. Dynamic Rise of Pressure in the Common-Rail Fuel Injection System. In 2002 [cited 2017 Aug 10]. Available from: http://papers.sae.org/2002-01-0210/

[21] Catania AE, Ferrari A, Manno M, Spessa E. Experimental investigation of dynamics effects on multiple-injection common rail system performance. Journal of Engineering for Gas Turbines and Power. 2008;**130**(3):032806

[22] Beierer P, Huhtala K, Vilenius M. Experimental Study of the Hydraulic Circuit of a Commercial Common Rail Diesel Fuel Injection System. In 2007 [cited 2017 Aug 12]. Available from: http://papers.sae.org/2007-01-0487/

[23] Bianchi GM, Falfari S, Brusiani F, Pelloni P, Osbat G, Parotto M. Numerical Investigation of Critical Issues in Multiple-Injection Strategy Operated by a New C.R. Fast-Actuation Solenoid Injector. In 2005 [cited 2019 Apr 20]. p. 2005-01–1236. Available from: http://papers.sae.org/2005-01-1236/

[24] Seykens XLJ, Somers LMT, Baert RSG. Modelling of common rail fuel injection system and influence of fluid properties on injection process. [cited 2017 Aug 9]; Available from: http://citeseerx.ist.psu.edu/viewdoc/summary?doi=10.1.1.475.2416

[25] Henein NA, Lai M-C, Singh IP, Zhong L, Han J. Characteristics of a Common Rail Diesel Injection System under Pilot and Post Injection Modes. In 2002 [cited 2019 Apr 19]. p. 2002-01–0218. Available from: http://papers.sae.org/2002-01-0218/

[26] Herfatmanesh MR, Peng Z, Ihracska A, Lin Y, Lu L, Zhang C. Characteristics of pressure wave in common rail fuel injection system of high-speed direct injection diesel engines. Advances in Mechanical Engineering. 2016;**8**(5):168781401664824

[27] Boudy F, Seers P. Impact of physical properties of biodiesel on the injection process in a common-rail direct injection system. Energy Conversion and Management. 2009;**50**(12):2905-2912

[28] Badami M, Millo F, D'Amato DD. Experimental Investigation on Soot and NOx Formation in a DI Common Rail Diesel Engine with Pilot Injection. In

2001 [cited 2019 Apr 20]. p. 2001-01–0657. Available from: http://papers.sae.org/2001-01-0657/

[29] Ferrari A, Paolicelli F, Pizzo P. Hydraulic performance comparison between the newly designed common feeding and standard common rail injection Systems for Diesel Engines. J Eng Gas Turbines Power-Trans Asme. 2016;**138**(9):092801

[30] Ferrari A, Mittica A. Response of different injector typologies to dwell time variations and a hydraulic analysis of closely-coupled and continuous rate shaping injection schedules. Applied Energy. 2016;**169**:899-911

[31] Catania AE, Ferrari A. Development and performance assessment of the new-generation CF fuel injection system for diesel passenger cars. Applied Energy. 2012;**91**(1):483-495

[32] Li P, Zhang Y, Li T, Xie L. Elimination of fuel pressure fluctuation and multi-injection fuel mass deviation of high pressure common-rail fuel injection system. Chin J Mech Eng. 2015;**28**(2):294-306

[33] Ismail MY, Mamat R, Ali O, Aziz A, Mohd A, Kamarulzaman M, et al. The combustion of N-butanol-diesel fuel blends and its cycle to cycle variability in a modern common-rail diesel engine. Journal of Engineering and Applied Science. 2016;**1**(11):2297-2301