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Dynamic Analysis of Common Rail Injection System for Diesel Engine based on AMESim/Simulink Co-Simulation

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ABSTRACT

In common rail diesel engine, the characteristics of fuel injection is concerned both the structure parameters of injector and control strategy of control system (ECU). Thus, fuel injection system, engine performance system and ECU constitute an interactive feedback system. In this paper, first of all, using the AMESim/Simulink co-model for the common rail diesel engine prepared in the previous work, simulation and verification are performed in the AMESim/Simulink co-model based on the initial values of the diesel engine used in the case. Then it is done Simulation and Validation on the AMESim/Simulink co-mode. The engine dynamometer and injection rate test rig makes it possible to take measurements of the engine performance and injection rate in order to compare with the modelling results. This allows the obtained co-model to be validated. Validated model is very effective in describing and analyzing the transient behavior connections among injectors, high-pressure pump, and controller and engine parameters. This co-model was then used to analyze the effects of various structural parameters of injector on the response characteristics of the fuel injection system and engine operation.

Keywords : Dynamic, analysis, AMESim/Simulink, co-simulation, common rail, diesel engine

I. INTRODUCTION

The high-pressure common rail system can improve the thermal efficiency of the engine by precisely controlling the injection time and injection pressure. Fuel injection is one of the most important processes in compression-ignition internal combustion engines because of its significant impact on exhaust gas emissions and thermal efficiency. The injection characteristics of a system can affect the engine's combustion and emission processes. From this, research on the dynamic analysis and experiments of the common pipeline fuel injection system is being intensified.

Above all, predecessors conducted research on the dynamic modeling of the common duct diesel engine fuel injection system. In literature [1], a dynamic modelling method for dynamic analysis of a common rail fuel injection system by Amesim/Simulink combination was mentioned. The simulation is based on a detailed geometrical description of the injection system and in modeling each subsystem as a separate control volume[2]. In literature [3], the total high-pressure part of the injection system, i.e. the fuel pump, rail and injector, has been modelled using the AMESim software. It was mentioned a model-based pressure waves reconstruction method, based on a control-oriented model of the high-pressure common rail injection system fuelled with gasoline [4]. The proposed mathematical model of high pressure common rail injection system which contains three sub-systems: high pressure pump sub-model, common rail sub-model and injector sub-model is a relative complicated nonlinear system, and the mathematical model is validated by the software Matlab and a virtual detailed simulation environment [5]. Com-bustion noise, fuel consumption, driving illumination and exhaust pollutants of a high-pressure system based on common rail architecture were analyzed using AMESim environment [6]. In literature [7], common Rail injection system model for diesel engines is derived by considering the components of the system as control volumes and applying elementary fluid dynamics and mechanics laws. In literature [8], a common rail fuel injection system of a single cylinder diesel engine has been proposed and the important parameters like injection pressure, energizing time and high pressure pipes diameter and length was analyzed such that to be compatible with the engine basic design in case of pressure waves and injected mass variations. Based on Kirchhoff's law, conservation equation, and Newton's motion degradation, mechanical, hydraulic, and electrical sub-models of diesel engine's high-pressure common rail fuel injection system were prepared[9]. After a complete characterization of the common-direct injection system, which involved mechanical and hydraulic characterization, a one-dimensional model has been obtained[10]. This model of common rail injection system has been previously obtained, including a complete characterization of the different components of the injector (mainly the nozzle, the injector holder and the electrovalve), and extensively validated by means of mass flow rate results under different conditions [11]. In literature [12], we presented a procedure for integrating different models and tools for a reliable design, optimization and analysis of a mechatronic system as a whole, encompassing the real process and the control system.

Next, through experiments, the accuracy of the dynamic model of the diesel engine common rail injection system was verified and analyzed. In literature [13], experiments were carried out to investigate the influence of injection pressure and injection timing on the temporal evolution of the injection rate and injection duration in a specially designed experiment rig equipped with a common rail injection system. A new experiment technique is described which permits visual display and easy evaluation of the discharge characteristics of any given fuel injection pump at random nozzle back-pressure[14]. In literature [15,16], a piezo-driven diesel injector, as a new method driven by piezoelectric energy, has been applied with a purpose to develop the analysis model of the piezo actuator to predict the dynamics characteristics of the hydraulic component (injector) by using the AMESim code and to evaluate the effect of this control capability on spray formation processes. The model of a dynamic common rail injection system model was developed by using a Simulink/Matlab code and are used an HCCI turbocharged diesel engine, 2776 cc, 4-stroke, and 4-cylinder[17-19].

In this way, the dynamic model of the common rail injection system was created using the AMESim code in the preceding literature, and its accuracy was verified through experiments.

However, in this paper, the dynamic model of the common rail injection system was created using Simulink/AMESim co-combination, and its accuracy was verified through experiments, and the effect of injector characteristic parameters on changes in engine performance and injection performance was analyzed.

II. EXPERIMENTAL METHOD

2.1. Engine test

A computer controlled engine dynamometer consisting of an automatic transmission, pre-transmission, rotating mass and electrical brake was used to simulate different driving conditions. The Alternating Current Dynamometer is an engine test facility that mimics the load placed on the engine while it is powering a vehicle. This allows testing of the engine over its entire range of operation in a controlled environment with additional measuring systems needed for research. This includes systems to measure emissions, temperatures and the pressure in each engine cylinder. This 300 HP AC dynamometer used in the experiment can measure two engines, but only one engine can be operated at a time. 128-Channel Data Acquisition System permits the acquisition of up to 64 analog inputs and 64 thermocouple inputs with sampling rates of up to 10 kHz. Laminar Flow Element mobile device provides precision measurement of air flow rate for accurate measurement of engine air flow

2.2 Injection test

Instantaneous injection-rate measurements from the Bosch type injection-rate meter [14] are used to characterize the injection systems experimentally. The injection systems are characterized based on the injector hydraulic characteristics, the injector response time and needle lift, the injection pressure, the injection rate and rate shape, the pressure drop in the injector, the effect of pilot injection on the main injection and the discharge coefficients. The experimental set-up consists of a high-pressure electronically controlled fuel injection system, an injection rate meter of the Bosch type and an electronic weight balance to analyze the injection rates and the injection quantities. Bosch tube method measures the injection rate through the pressure wave which is analog to flow flux caused by the injection event. The basic principle of the Bosch tube method is described as follows. The fuel is injected into a long tube of a known diameter filled with the fuel， as shown in fig.1. A pressure transducer is installed at the vicinity of the injector nozzle to record the pressure wave induced by the injection. The fuel discharge produces a pressure increase inside the measuring tube, which is proportional to the increase in the mass of fuel. The rate of this pressure increase corresponds to the injection rate. At this time, the pressure control device always controls the backpressure of check valve by the positioning motor, therefore accurate variation of pressure wave is delivered to the piezo pressure sensor. Also, the other end of the tube is an orifice (or a valve) which can controls the pressure of the tube and also reflects the pressure wave back to the transducer. The speed of sound of the fluid can be derived from the tube length and the time interval between the first and the reflected pressure wave. The final injection rate was derived using the acquired pressure wave signal, the flow area and the speed of sound of the fluid [17]. The experimental device consists of a electro-controlled high-pressure fuel injection system, a Bosch type injection meter, and an electronic mass balance, and is used to analyze the injection rate and injection quantity. The Bosch type injection meter acquires injection rate traces by measuring the pressure waves produced by an injection. The recorded injection rate measurements are further converted into instantaneous fuel mass to obtain the instantaneous discharge coefficients. Figure 1 shows the layout of the experimental set-up. A measuring tube with an inner diameter of 4.57mm and a length of 10,850mm was fabricated in the form of coil type with diameter of 250mm. The piezo-electric type pressure sensor was installed at the middle of the injector adaptor. A cylindrical accumulator with a volume of 700cc was installed in front of the back pressure regulator. This regulator was utilized to keep engine-like pressure condition to be required and can be adjusted up to the pressure range of 3MPa. The back pressure within the accumulator was measured by Bourdon tube pressure gauge. A needle lift sensor registers the needle lift. The Bosch type injection rate meter is used to investigate various injection modes including single and multiple injections. The injection parameters controlled are injection pressure, injection duration and the number of injections.

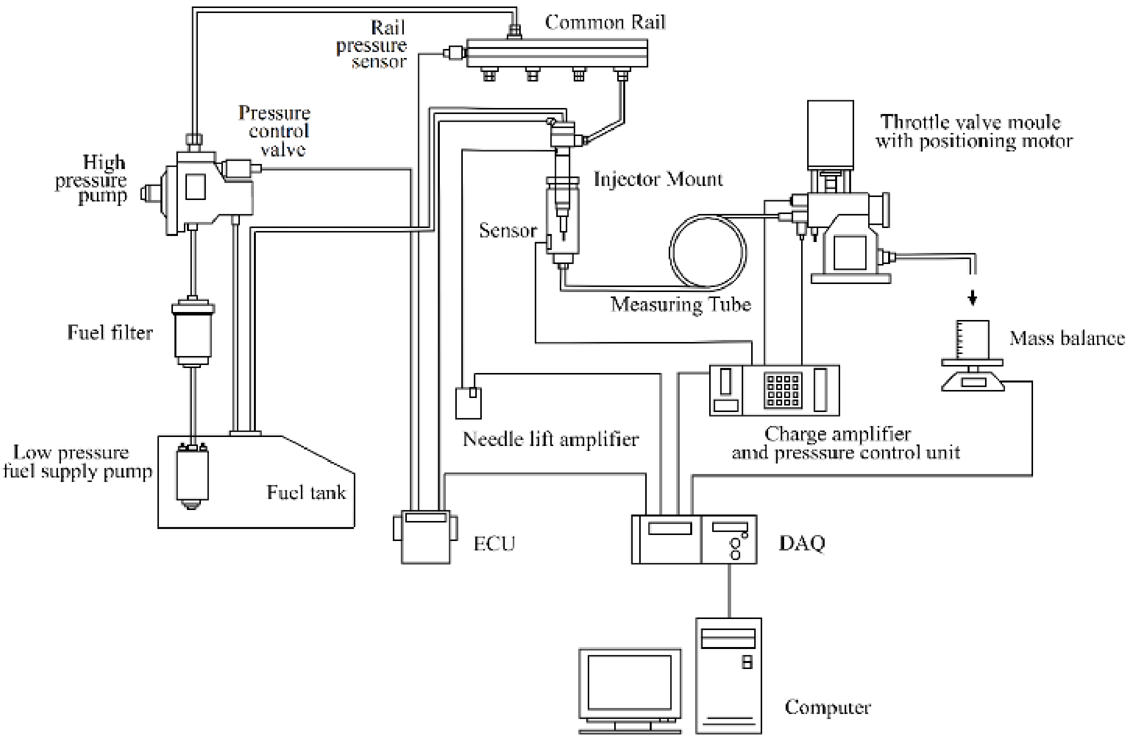

Fig.1 layout of the injection test rig

III. RESULTS AND DISCUSSION

3.1 Analysis on Diesel Engine Performance by AMESim/Simulink co-model

3.1.1 AMESim/Simulink co-model and initial parameter for Diesel Engine Performance Analysis

In order to study the influence of electro-controlled injector parameters on the variation of injection quantity per cycle in high pressure common rail system, an AMESim/Simulink co-model introduced in Literature [1] is employed. The basic specification of the engine used to simulate performance on the engine are as shown in Table 1. AMESim/Simulink co-model of corresponding engine is shown in Fig. 2.

Table 1. Engine specification

Parameters	Value
Engine type	4 cylinder diesel engine in line with turbo-charge system
Bore x Stroke	108 x 125 mm
Displacement	4.58 L
Length of connect road	250 mm
Compression ratio	18
Fuel system	FAW WFIERI CR
Rated power	162 kW
Rated speed	2300 r/min
Nozzle number	4
Nozzle diameter	0.3 mm

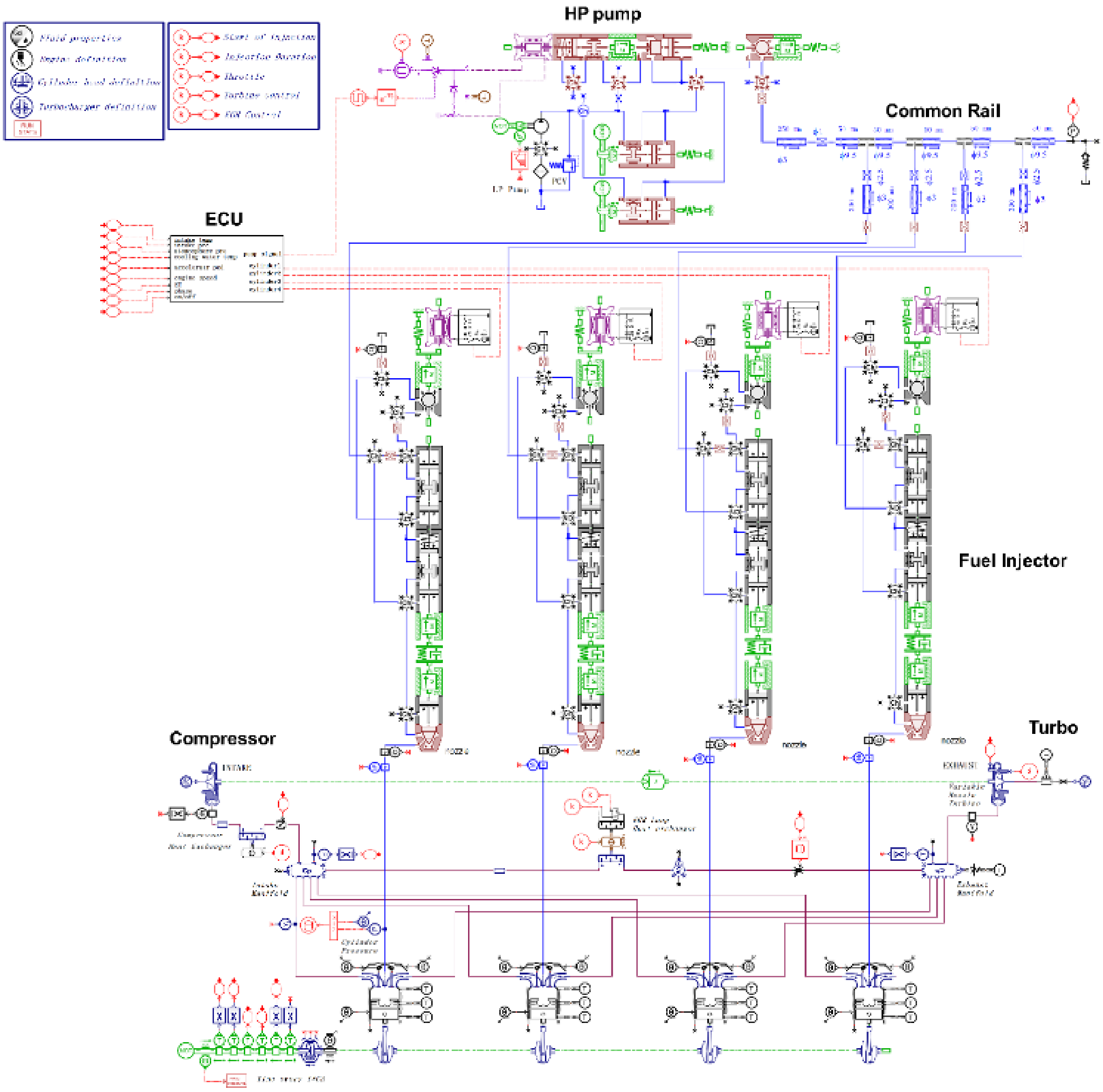

Fig 2. AMESim/Simulink co-model of corresponding engine [1]

When combustion in the cylinder is simulated, the fuel injection flow rate, which is obtained from experiment in different control pulse duration (Fig. 3), is an important input data. In this co-simulation model, the fuel injection flow rate is also calculated from the sub-model for the common rail injection system.

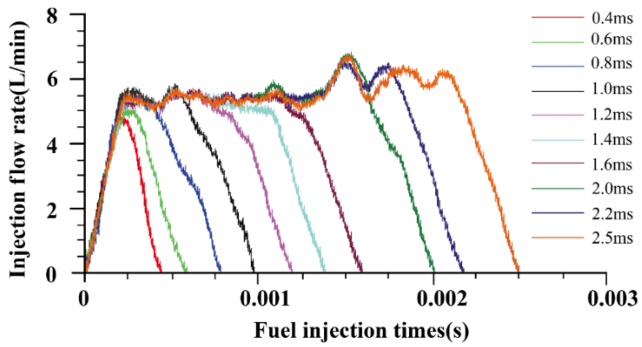

Fig. 3 Fuel injection flow rate

3.1.2 Analysis result on Diesel Engine Performance by AMESim/Simulink co-model

Fig.4 is the results calculated according to the co-model in Fig.2, in which the history of engine speed and rail pressure varies with the opening degree of engine accelerator pedal is shown. In the Fig.4, the initial opening degree of engine accelerator pedal is 50 % and the load is 600 N.m. Then, the opening degree of engine accelerator pedal is increased to 100 %. Again, the opening degree of engine accelerator pedal is decreased to 50 %. According to results in Fig.4, the engine speed and the rail pressure can vary consistent with the opening degree of engine accelerator pedal, which indicates the co-simulation model can work properly.

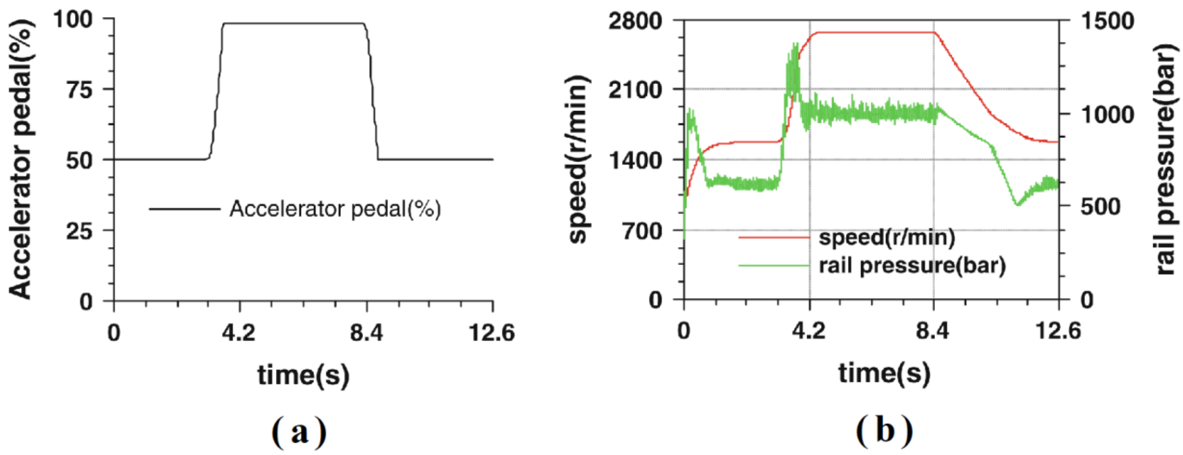

Fig. 4 Engine speed and rail pressure vs. opening degree of engine accelerator pedal.
(a) Opening degree of engine accelerator pedal (b) Engine speed and Rail pressure

3.2 Engine Performance Characteristics Test

Table2-table12 shows experimental dataset measured through Engine test and injection test.

Table 2. Measurement power accounting on the engine speed

engine speed(r/min)	800	1200	1400	1600	1900	2100	2400
power(kW)	56	135	166	198	209	215	213

Table 3. Measurement BSFC accounting on the engine speed

engine speed(r/min)	800	1200	1400	1600	1900	2100	2400
BSFC (g/(kW·h))	237	203	206	209	217	220	218

Table 4. Measurement air fuel ratio accounting on the engine speed

engine speed(r/min)	800	1200	1400	1600	1900	2100	2400
air fuel ratio	15.4	18.4	21.3	20.8	24	23.7	24.8

Table 5. Measurement boost pressure ratio accounting on the engine speed

engine speed(r/min)	800	1200	1400	1600	1900	2100	2400
boost pressure ratio	1.3	1.95	2.44	2.47	2.51	2.49	2.45

Table 6. Measurement cylinder pressure accounting on the crank angle (1390r/min)

crank angle(deg)	270	293	315	338	360	383	405	428	450
cylinder pressure(Mpa)	0.61	1.05	1.83	4.92	11.84	13.96	4.27	2.67	1.45

Table 7. Measurement cylinder pressure accounting on the crank angle (2110r/min)

crank angle(deg)	270	293	315	338	360	383	405	428	450
cylinder pressure(Mpa)	0.63	1.14	1.83	5.16	12.63	14.02	4.63	2.69	1.47

Table 8. Measurement injection quantity per cycle accounting on the control pulse duration (60Mpa)

control pulse duration(ms)	0.38	0.65	0.94	1.47	2.00
injection quantity per cycle(mm^3)	6.9	15.3	59.3	103.6	146.8

Table 9. Measurement injection quantity per cycle accounting on the control pulse duration (120Mpa)

control pulse duration(ms)	0.38	0.65	0.94	1.47	2.00
injection quantity per cycle(mm^3)	7.2	54.7	89.2	152.6	196.5

Table 11. Measurement injection rate accounting on the fuel injection time (60 MPa)

fuel injection time(ms)	0	0.3	0.6	0.9	1.2	1.5	1.8	2.1	2.4	2.7
injection rate(L/min)	0.21	2.78	3.26	3.35	3.43	3.84	3.07	0	0	0

Table 12. Measurement injection rate accounting on the fuel injection time (120 MPa)

fuel injection time(ms)	0	0.3	0.6	0.9	1.2	1.5	1.8	2.1	2.4	2.7
injection rate(L/min)	0.24	3.03	3.26	3.48	3.46	3.52	3.74	4.36	2.56	0

3.3 Accuracy verification on the AMESim/Simulink co-model

Accuracy verification on the co-model can be done through the comparative analysis on the results of the co-model simulation and the engine performance characteristic experiments. Fig5-Fig.8 shows the experimental and simulation comparison curve which measured at same sample point under the same condition. When the rail pressure is 60 and 120MPa, the calculated and measured fuel injection quantity at different injection pulse duration are shown in Fig. 5, and the calculated and measured injection flow rate is shown in Fig. 6. It is found that the calculated and measured results are in good agreement. The measured and simulated diesel engine performance is shown in Fig. 7, and the cylinder pressure is shown in Fig. 8.

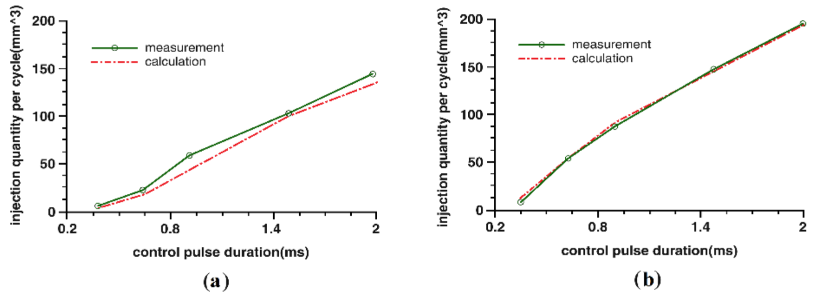

Fig. 5 The calculated and measured fuel injection quantity per cycle.

(a) Rail pressure 60 MPa (b) Rail pressure 120 MPa

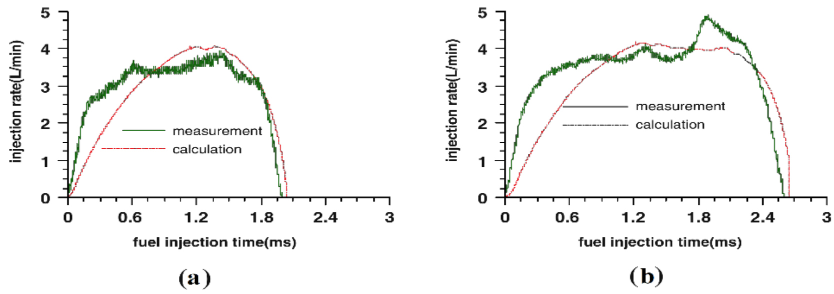

Fig. 6 The calculated and measured fuel injection flow rate.

(a) Rail pressure 60MPa and injection pulse duration 2ms

(b) Rail pressure 120Mpa and injection pulse duration 2ms

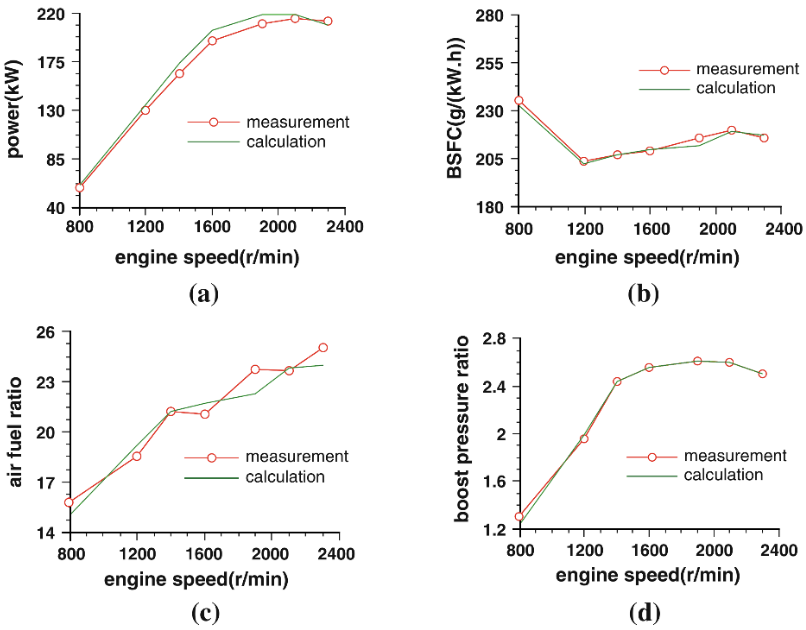

Fig. 7 Comparison of diesel engine performance

(a) Power curve accounting on the engine speed

(b) Brake Specific Fuel Consumption curve accounting on the engine speed

(c) Air fuel ratio curve accounting on the engine speed

(d) Boost pressure ratio curve accounting on the engine speed

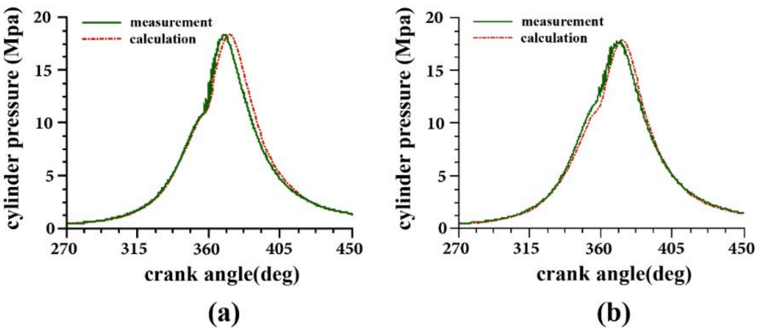

Fig. 8 The comparison of cylinder pressure
(a) 1390 r/min and 100 % load (b) 2110 r/min and100 % load

It can be seen from the figure that the simulation prediction results are in good agreement with the measured values. A good agreement is achieved between them, which indicates the sub-model of engine performance simulation has high accuracy. From the above comparison results, it can be seen that the injection quantity per cycle, fuel injection rate, and engine characteristic parameters can be accurately predicted by using the simulation model in both numerical and time sequence, which proves the accuracy of the simulation model. Then on this basis, the cycle fuel injection quantity fluctuation (CFIQF) is analyzed to reveal the law of the effects of injector parameters on CFIQF.

3.4 Influence analysis of the injector characteristic parameters on the engine performance and fuel injection performance using a verified dynamic model

Common rail injector is the key component of the high pressure common rail fuel injection system. Electronically controlled injector uses solenoid valve for electromagnetic conversion, generating electromagnetic force, to control the sealing ball valve action and control chamber oil pressure, to achieve the control of needle valve action, and then achieve the purpose of flexible adjustment of the injection process. Due to the complex structure of the components involved in electric, magnetic, mechanical and hydraulic functions integrated in the electronic fuel injector, the parameters of each part of the injector will directly or indirectly affect the injection quantity per cycle of the system. The change of parameters will also cause the fluctuation of injection quantity per cycle, resulting in the deterioration of the working stability of the injection system and its matching diesel engine [8]. The armature residual clearance is the minimum distance between the electromagnet and armature when they are absorbed. The control valve lift is the maximum motion displacement of the control valve and armature when the solenoid valve is from closed to fully opened. The needle valve preload force is the spring force on the needle valve when the solenoid valve is closed due to the pre-compression of the solenoid valve return spring. The needle valve preload is determined by the reset spring of the needle valve in the fuel injector. Its change lead to changes the duration of the injection by changing the speed of the opening and reset of the needle valve, and causing the change of the injection quantity per cycle. Needle valve lift is the maximum upward displacement of the needle valve, that is, the lift of the needle valve, determines the movement time of the needle valve and the throttling characteristics between the needle valve and the nozzle hole. Therefore, its change causes to change of the injection quantity per cycle. The change of armature residual clearance, control valve lift and needle valve preload will all cause the change of solenoid valve response characteristics, and indirectly lead to the fluctuation of injection quantity per cycle by affecting the release and establishment process of fuel pressure in the control chamber. Therefore, the armature residual clearance, needle valve lift and needle valve preload are the main parameters that affect the circulating fuel injection rate of high pressure common rail injection system. Because the change of these parameters will make the circulating injection amount of the fuel injection system fluctuate obviously. Therefore, the influence of four parameters of armature residual clearance, control valve lift, needle valve lift and needle spring preload must mainly be considered in the analysis of circulating fuel injection fluctuation.

3.4.1 Influence of Control Valve Lift on the engine performance and fuel injection performance

The variability of injector control-valve-lift has great influence on engine performance variability. Normally the injector control-valve-lift is 0.07 mm. In the idle speed condition (650 r/min), shown in Fig.9, when injector control valve lift increases, the needle lift, fuel injection flow rate and cylinder pressure have greater change in sequence four cycles, and the fluctuation amplitude of engine speed and rail pressure increases (Fig.10), which will result in the increase of vibration in the engine performance.

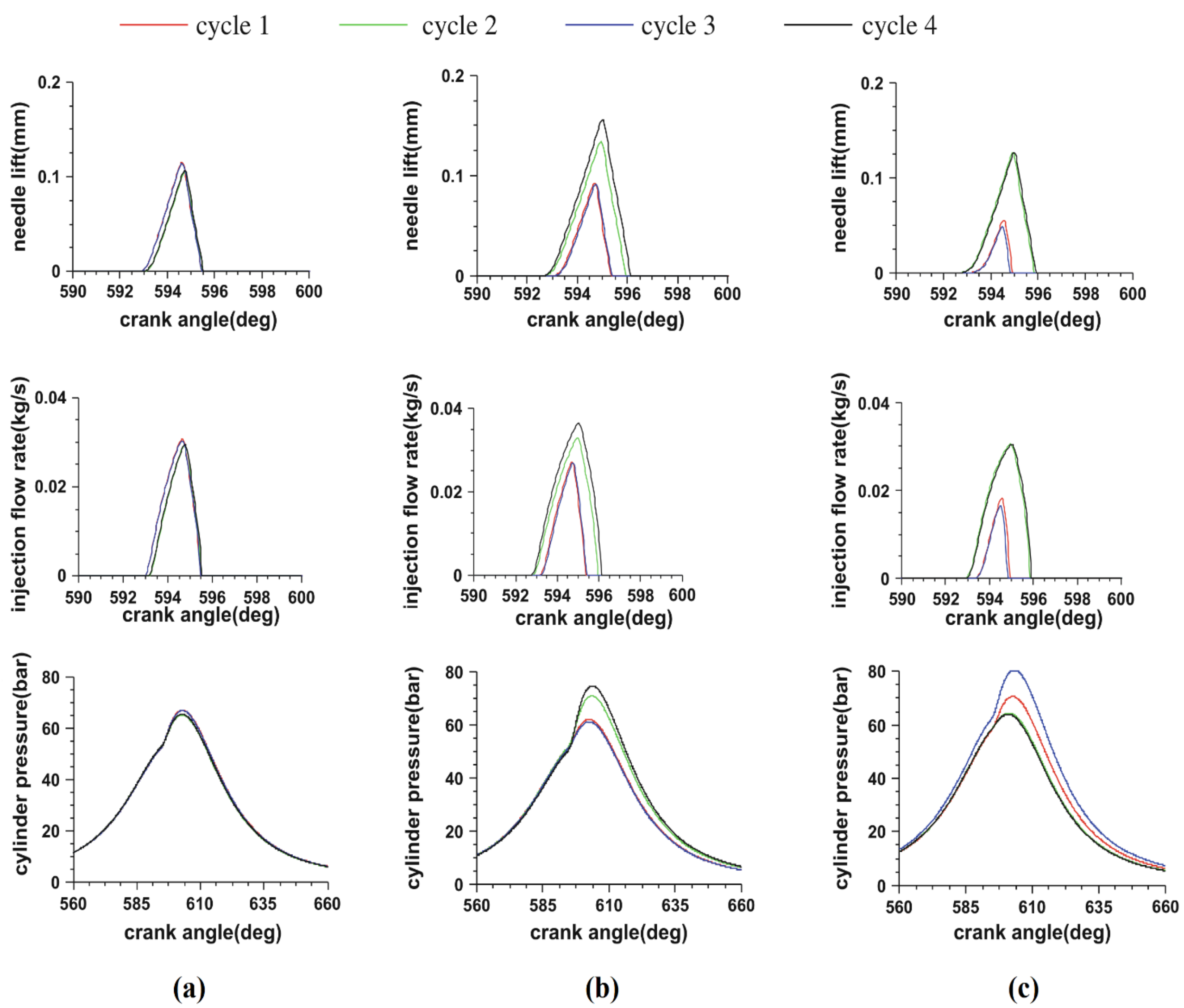

Fig. 9 The influence of control valve lift in idle speed condition.
(a) control valve lift 0.07mm (b) control valve lift 0.10mm (c) control valve lift 0.12mm

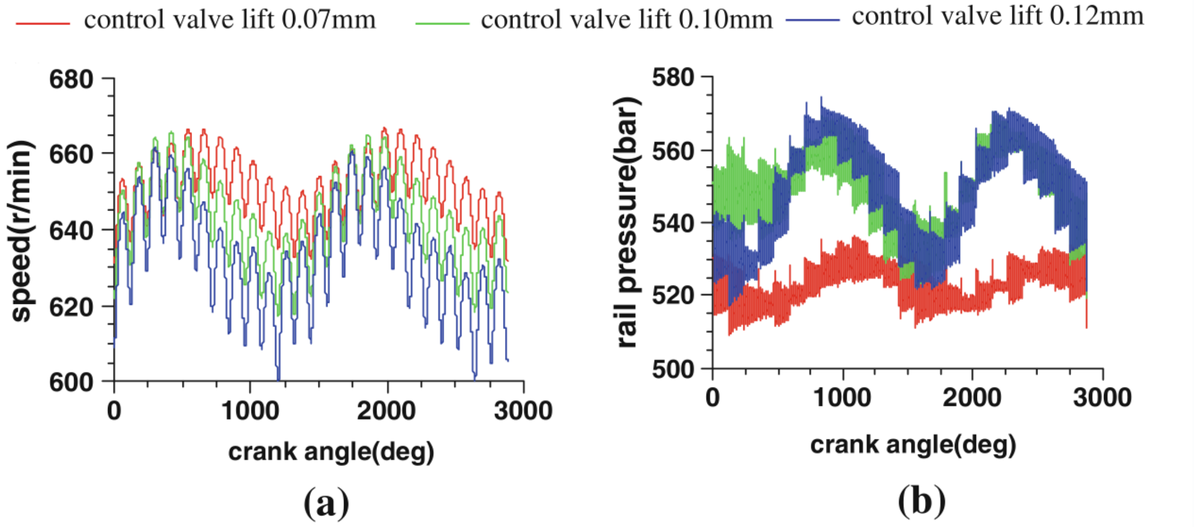

Fig. 10 The influence of control valve lift on engine speed and rail pressure in the idle speed condition.
(a) engine speed (b) rail pressure

When the injector control valve lift is 0.07, 0.10, 0.12mm respectively, the engine speed is 650 ± 15 r/min, 640 ± 20 r/min, 635 ± 27r/min, which indicates that the engine speed decreases from 650 to 635 r/min and speed fluctuation amplitude increases from ± 15 r/min to ±27r/min. In the maximum torque condition (1150N.m), when injector control valve lift increases, the needle lift, fuel injection flow rate and cylinder pressure have also some changes in different cycle, but their fluctuation amplitude is not as big as that in idle speed condition.

When the injector control-valve-lift is 0.07, 0.10, 0.12 mm, the engine speed is 1838, 1820, 1748 r/min respectively (Fig.11).The decrease of engine speed will result in decrease of the power output in the engine. Meantime the rail pressure decrease slightly, but the fluctuation amplitude of engine speed and rail pressure is not big. Under the condition of a constant rail pressure, it was considered the influence of the control valve lifting on the injection performance when injection pulse width is 1.5ms.

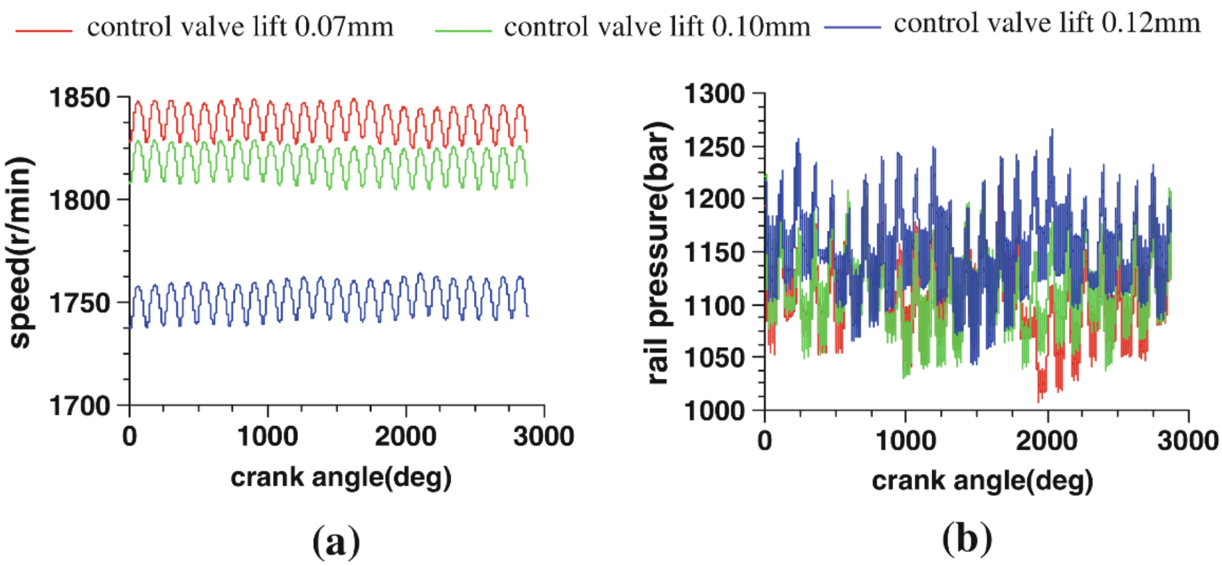

Fig. 11 The influence of control valve lift on engine speed and rail pressure in the maximum torque condition.
(a) engine speed (b) rail pressure

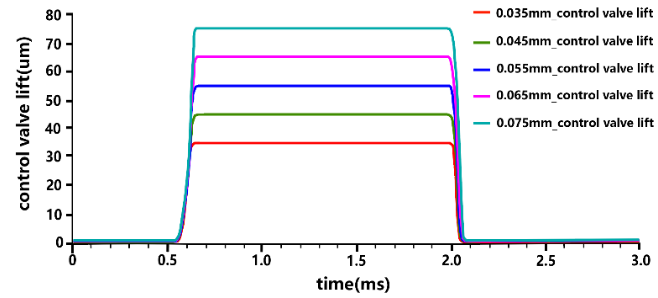

Fig.12 lifting of the control valve

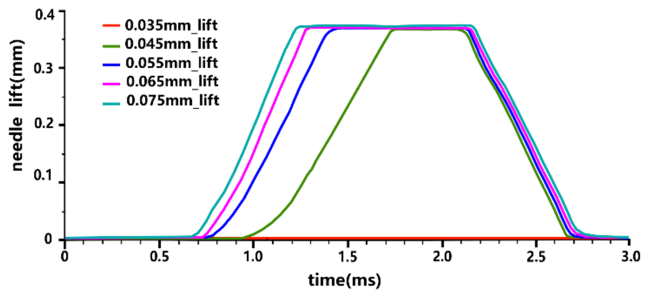

Fig.13 Change of needle valve lifting according to lifting of the control valve

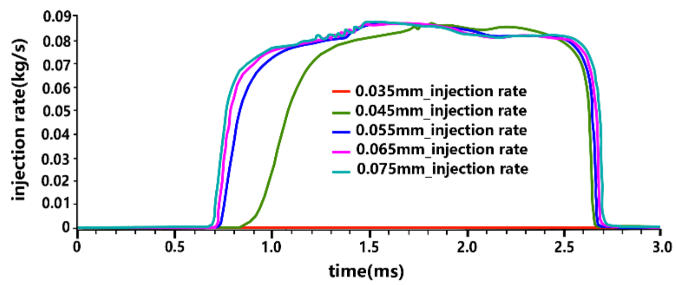

Fig.14 Change of injection rate according to lifting of the control valve

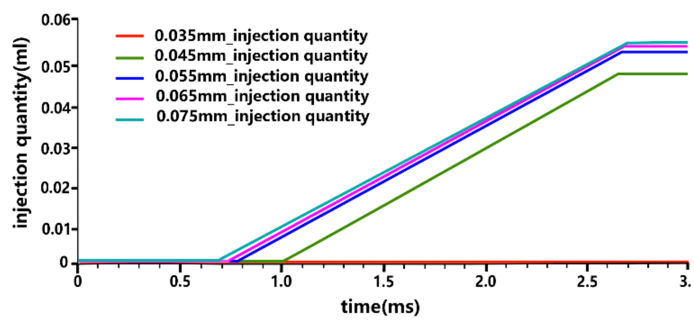

Fig.15 Change of injection quantity according to lifting of the control valve

As shown in Fig. 13 under maximum moment conditions, when the maximum lift of the control valve is 0.035mm, the needle valve of the injector cannot be opened for normal injection. With the increase of the maximum lift of the control valve, the opening response speed of the needle valve is significantly increased, and when the maximum lift of the control valve is increased to 0.065mm, the opening response characteristics of the needle valve are not significantly changed. The maximum lift of the control valve has little effect on the closing response characteristics of the needle valve. It can be seen from Fig. 14 and Fig. 15 that with the increase of the maximum lift value of the control valve, the fuel injection volume of the injector also increases. The reasons for the above phenomenon are as follows: before the maximum lift of the control valve increases to 0.065mm, the spacing between the ball and the seat of the control valve is very small, forming a throttle hole, which limits the fuel flow rate of the control chamber from the outlet hole, resulting in the fuel pressure of the control chamber is difficult to reduce rapidly, so the response speed of the needle valve opening is slow; After the maximum lift of the control valve is increased to 0.065mm, the throttle hole formed by the distance between the ball and the seat of the control valve no longer plays a major throttling role, but is determined by the outlet hole aperture connected with the control chamber. Therefore, the maximum lift value of the control valve must be greater than 0.065mm to get rid of the limitation of the needle valve opening response characteristics. The displacement and motion time of the control valve will be increased with the increase of the control valve lifting, which will indirectly affect the motion response of needle valve through the change of the fuel pressure in the control chamber and cause the change of the injection quantity per cycle.

3.3.2 Influence of Preload in Needle Spring on the engine performance and fuel injection performance

The variability of preload in needle spring has also important influence on engine performance variability, especially in idle speed condition (650 r/min). When the preload in needle spring is 55N, 43N, 33N respectively, the needle lift and fuel injection flow rate and the cylinder pressure for sequence four cycles are shown in Fig. 16. With the decreasing of preload of needle spring, the fluctuation amplitude of needle lift, fuel injection flow rate and cylinder pressure increases slightly, which results in the fluctuation amplitude of engine speed and rail pressure increases slightly (Fig.17). In the maximum torque condition (1150N·m), with the decreasing of preload in needle spring the engine speed increases, but the fluctuation amplitude of the engine speed has no distinct change, and the rail pressure and its fluctuation amplitude have also no distinct change. When the preload of needle spring is 55N, 43N and 33N, the average engine speed is 1838, 1847 and 1856 r/min respectively (Fig. 18).

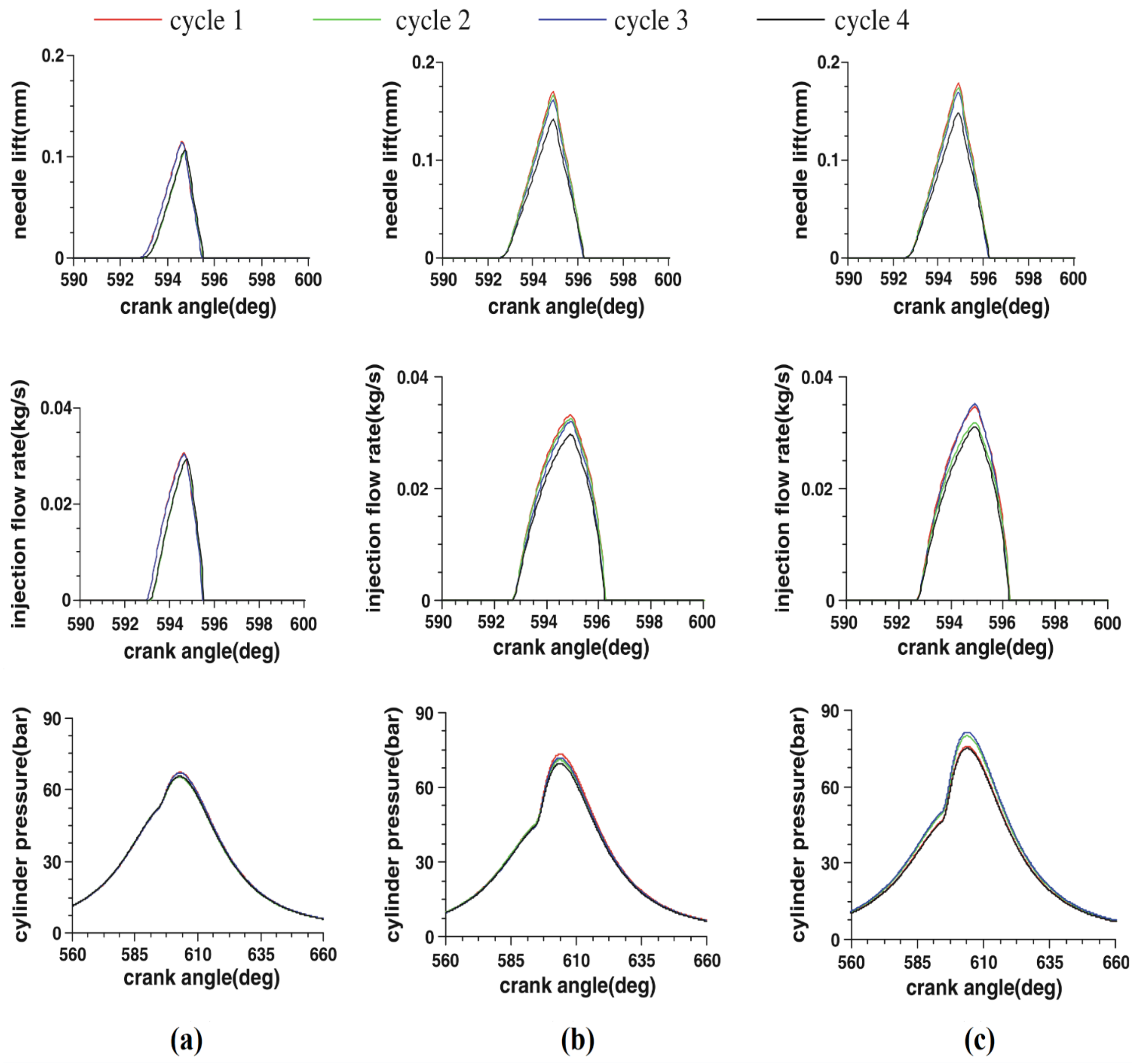

Fig. 16 The influence of preload of needle spring in idle speed condition.

(a) preload of needle spring 55 N

(b) preload of needle spring 43N

(c) preload of needle spring 33 N

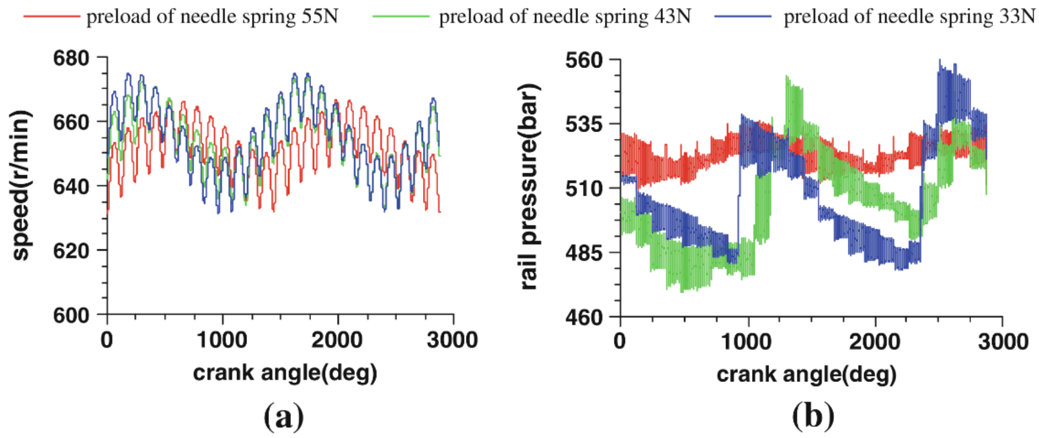

Fig. 17 The influence of preload of needle spring on engine speed and rail pressure in idle speed working condition.

(a) engine speed (b) rail pressure

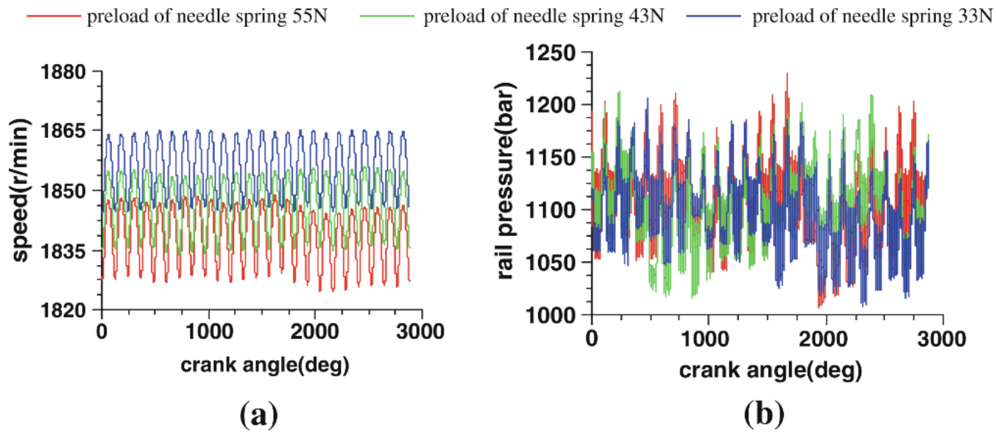

Fig. 18 The influence of preload of needle spring on engine speed and rail pressure in maximum torque working condition.

(a) engine speed (b) rail pressure

Under the condition of a constant rail pressure, it was considered the influence of preload of needle spring on the injection performance when injection pulse width is 1.5ms.

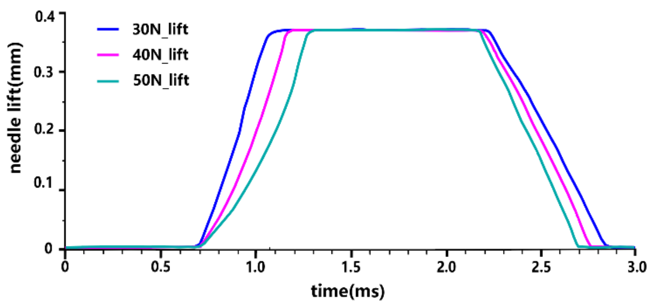

Fig.19 Change of needle valve lifting according to preload in needle spring

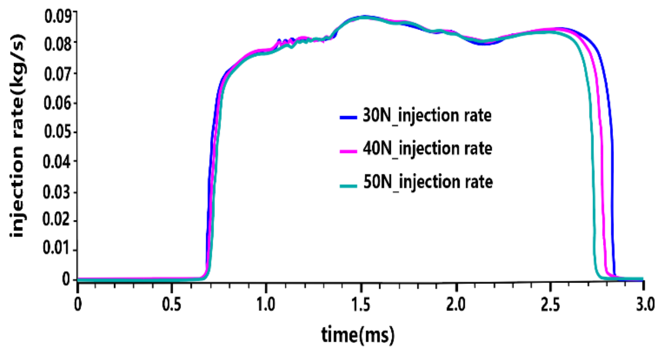

Fig.20 Change of injection rate according to preload in needle spring

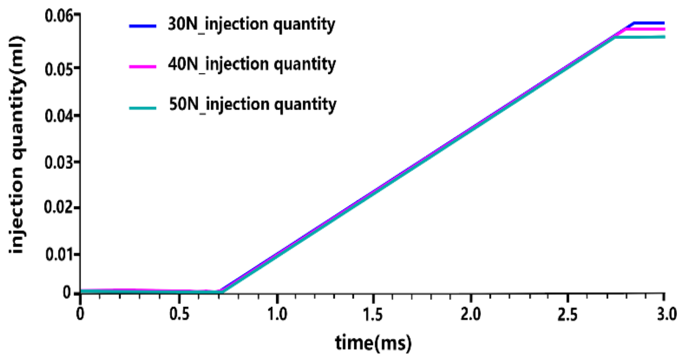

Fig.21 Change of injection quantity according to preload in needle spring

Fig.19 shows the needle valve lift curves corresponding to different preloading forces of the needle valve when injection pulse width is 1.5ms under the condition of a constant rail pressure. As can be seen from the figure, with the increase of the preloading force of the needle valve, the opening rate of the needle valve decreases, which leads to the decrease of the initial injection rate. At the same time, the closing time of the needle valve is advanced, which leads to the shortening of the injection duration and thus the fluctuation of the injection quantity per cycle. With the increase of injection pressure, the influence of the change of needle valve preload on the injection quantity per cycle decreases, and the fluctuation of injection quantity per cycle decreases.

3.3.3 Influence of armature residual clearance on the fuel injection performance

The armature residual clearance is the minimum gap between the armature and the electromagnet after the solenoid valve is fully opened.Fig.22 shows the control valve lifting curves corresponding to different armature residual clearance when injection pulse width is 0.4ms under the condition of a constant rail pressure. As can be seen from the figure, the smaller the armature residual clearance, the larger the electromagnetic force, the earlier the lifting of the control valve, thus advancing the injection time. But at the end of injection, the electromagnetic force corresponding to the small armature residual clearance is large and the decay rate is slow, which delays the setting speed of the control valve and causes the delay at the end of injection time. Therefore, with the decrease of armature residual clearance, the duration of injection increases and the amount of injection quantity per cycle increases. With the increase of injection pressure, the injection amount per unit time increases, and the fluctuation of circulating oil injection amount caused by the change of armature residual clearance also increases. In the case of small injection pulse width, the needle valve is seated and reset before reaching the maximum lift, and the change of armature residual clearance has a greater impact on the injection process, so the fluctuation of fuel flow is more obvious at this time.

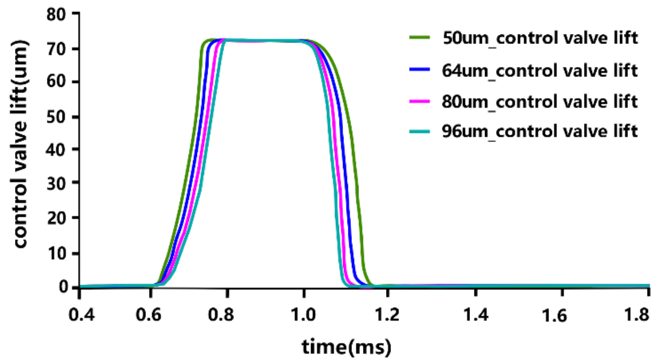

Fig.22 Change of control valve lifting according to armature residual clearance

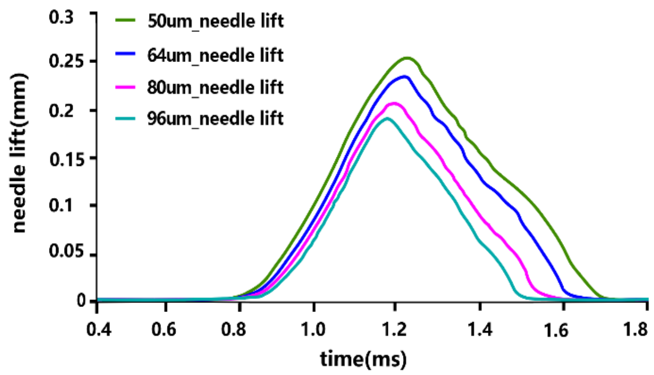

Fig.23 Change of needle valve lifting according to armature residual clearance

Fig. 24 shows the cycle injection quantity fluctuation in all working plane when the armature residual clearance is 64um, 80um and 96um respectively. The maximum cycle injection fluctuation is 4mm^3, which is obtained at the working point of 110MPa rail pressure and 0.4ms injection pulse width. Under the condition of small injection pulse width less than 0.8ms, the cycle injection quantity fluctuation is particularly significant.

The fluctuation changes sharply with the increase of pulse width, and also shows an increasing trend with the increase of rail pressure. When the injection pulse width is greater than 0.8ms, the cycle injection quantity fluctuation presents periodic fluctuation characteristics with the increase of injection pulse width, and increases slightly with the increase of rail pressure, but the influence of injection pulse width and rail pressure is very small.

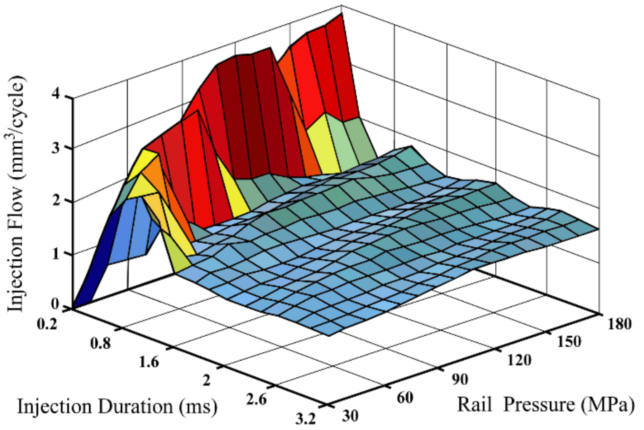

Fig.24 Cycle injection quantity fluctuation caused by variation of armature residual clearance

3.3.4 Influence of Needle Valve Lift on the engine performance

Fig. 25 shows the needle valve lift curves according to maximum lifting of needle when injection pulse width is 1ms under the condition of a constant rail pressure. As can be seen from the figure, with the increase of the needle valve lift, the opening rate of the needle valve remains unchanged, but the reaching time of the maximum lift is delayed. At the end of injection, the reset and setting time of the needle valve is extended, the duration of injection increases, and the cycle injection amount increases. At the same time, with the increase of the needle valve lift, the throttle effect between the needle valve and the nozzle hole decreases, the injection rate increases, and the injection quantity per cycle increases. Therefore, the change of needle valve lift causes the fluctuation of injection quantity per cycle, which increases with the increase of injection pressure and injection pulse width.

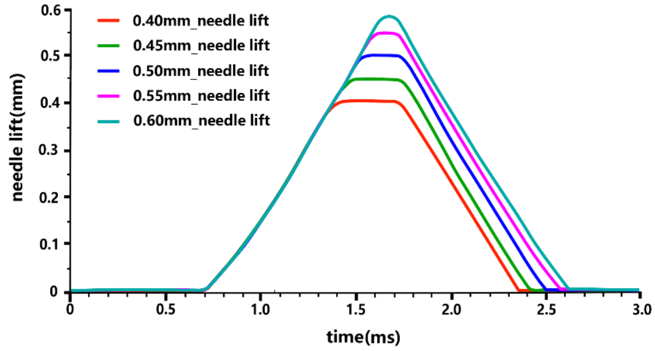

Fig.25 Needle lift according to maximum lifting of needle valve

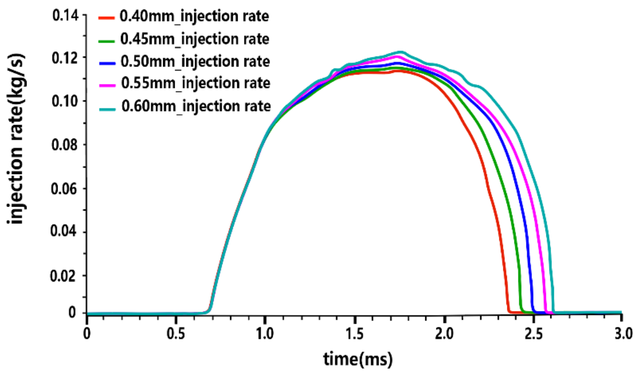

Fig.26 Change of injection rate according to needle valve lift

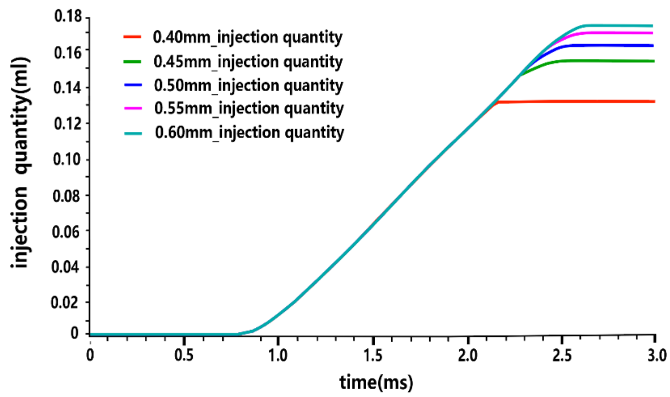

Fig.27 Change of injection quantity according to needle valve lift

Fig. 28 shows the cycle injection quantity fluctuation in all working plane when the needle valve lift is 64um, 80um and 96um respectively. The maximum cycle injection fluctuation is 7.05mm³, which is obtained at the working point of 180MPa rail pressure and 3.2ms injection pulse width. It can be seen that in the area of low rail pressure and small injection pulse width, the cycle injection quantity fluctuation caused by the change of needle valve lift is 0, while at the working point of high rail pressure and large injection pulse width, the cycle injection quantity fluctuation caused by the change of needle valve lift increases approximately linearly with the increase of rail pressure and injection pulse width.

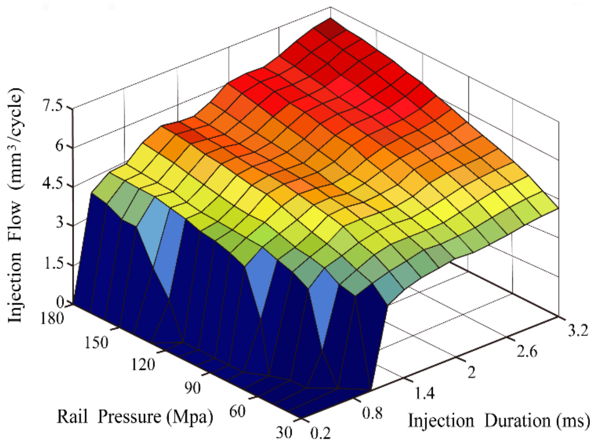

Fig.28 Fuel injection quantity fluctuation caused by variation of needle lift

At the operating point of low pressure and small injection pulse width where the needle valve cannot reach the maximum lift, the change of needle valve lift will not affect the injection quantity per cycle, so the cycle injection quantity fluctuation is 0. With the increase of rail pressure and injection pulse width, the cycle injection quantity fluctuates linearly.

IV. CONCLUSION

The fluctuation of engine speed caused by the non-uniformity of the injection amount of each injectors is the basic cause for lowering power of engine, reducing the life of the engine and increasing fuel consumption and emission. The co-simulation method of high-pressure fuel system and engine performance system and control system in common rail diesel engine can be simulated not only engine performance in the steady state condition but also fluctuation of engine speed in the transitional process. The verified co-model through experiments can accurately simulate the fast transient response of fuel injection rate on engine performance. Therefore, influence of the injector parameters on the injection quantity per cycle was analyzed by the co-simulation method. Through quantitative analysis and simulation some conclusions are obtained.

(a) The injector control valve lift has important influence on engine performance. In idle speed condition, the increasing of control valve lift will result in the increasing of fluctuation amplitude of engine speed. In the maximum torque condition, the increasing of control valve lift will result in the decreasing of engine speed and power output.

(b) The variability of needle valve preload has also great influence on engine performance variability. In the idle speed condition, with the decreasing of preload in needle spring, the fluctuation amplitude of engine speed and rail pressure increases. In the maximum torque condition, with the decreasing of preload in needle spring the engine speed increases.

(c) The armature residual clearance has important influence on injection performance. With the decrease of the armature residual clearance the duration of injection increases and the amount of cycle injection quantity increases. Under the condition of small injection pulse width, the cycle injection quantity fluctuation is particularly significant.

(d) The injector needle valve lift has a certain influence on injection performance. With the increase of the needle valve lift, the duration of injection increases, and the cycle injection amount increases. Therefore, the change of needle valve lift causes the fluctuation of injection quantity per cycle. The cycle injection quantity fluctuation caused by the change of needle valve lift increases approximately linearly with the increase of rail pressure and injection pulse width.

Thus, because the change of these parameters will make the injection quantity per cycle of the fuel injection system fluctuate obviously, so it is necessary to improve the quality control standards of these parameters in the process of production, processing and operating, so as to minimize the fluctuation of injection quantity per cycle caused by the parameter inconsistency caused by production, processing and operating.

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