Simulation study on the influence of fuel injection strategy on the soot emission of dual-injection engine

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Abstract. A 2.0T gasoline engine was modified to a dual-injection engine. Numerical simulation method is used to study the influence of fuel type and fuel injection strategy on the in-cylinder mixture formation process, combustion process and soot emission of dual-injection gasoline engine. The simulation results show that an appropriate increase in the intake port fuel injection ratio can improve the uniformity of the air-fuel mixture. Increasing the fuel injection ratio in the intake port of the dual-injection system, the proportion of fuel directly injected into the cylinder decreases and the oil film distribution in the cylinder is reduced. Among the three injection schemes, the flame front of the D85 injection scheme reaches the cylinder wall earliest, and the flame spreads faster; the soot mass fraction generation decreases with the decrease of the GDI injection ratio. In particular, the amount of soot generated under the D65GDI injection ratio is significantly reduced.

1 Introduction

GDI gasoline engine has been the most popular type of engine used in automobiles in recent years. But the soot emission problem of GDI engine conflicts with the current environmental protection problem. The China VI that have been implemented across the country at 2020 has higher requirements for particulate emissions, then the problem of particulate emissions is more serious. Relevant studies have shown that dual-injection gasoline engines combined with port injection and direct injection can solve the problem of particulate emissions, fuel economy, and power to a certain extent[1].

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Dual-injection technology includes two injection technologies: port injection and in-cylinder injection, which are usually modified from traditional GDI gasoline engine, that is, a set of PFI injection system is introduced into the intake port of a GDI gasoline engine to achieve dual injection. Studies have shown [1] that using dual injection technology and a reasonable organization of fuel injection strategies can not only obtain a better mixture, but also reduce engine particulate emissions.

A mixture of gasoline and alternative fuel in a certain proportion are used in the traditional dual fuel injection system. Among them, ethanol is an ideal alternative fuel. Because ethanol has a high octane number, it has higher knock resistance, and as an oxygenated fuel, ethanol can effectively reduce particulate matter emissions. Scholars both national and abroad have also done some related researches on dual-injection engines.

Through experimental methods, Daniel et al. studied the gas emissions and soot emissions when injecting a mixture of ethanol and gasoline in the intake port and cylinder. Their research confirmed that dual-injection gasoline engines can indeed reduce the emission of soot particles, reduce the emission of carbon oxides and HC, be compared with direct-injection gasoline engines, but the cooling effect of the in-cylinder impulse is reduced due to the direct injection spray in the cylinder[2]. In addition, Daniel and the others also carried out the injection strategy of direct injection of methanol in the cylinder and injection of gasoline in the intake port of the engine on the dual-injection engine, and studied the influence of fuel type and fuel injection method on the anti-knock of the dual-injection system[3].

Stien et al. Studied on the dual-injection system , which was also modified by adding an injection system to the intake port on a GDI engine, and successfully increased the compression ratio to 12:1[4]. Its research found that injecting E85 into the engine cylinder can effectively suppress the engine's combustion knock, which can be used to expand the engine's operating load.

Yuan et al. studied the influence of the injection timing of direct injection of ethanol in the cylinder on the effect of suppressing knock [5]. It shows that direct injection of ethanol into the cylinder after the intake valve of the engine is closed can improve the combustion resistance of the engine. However, correspondingly, the quality of the mixed gas is degraded, resulting in a decrease in the combustion efficiency of the engine and a corresponding deterioration in emissions; and before the intake valve of the engine is closed, ethanol is injected into the cylinder, and the combustion efficiency of the engine is improved. .

In addition, Francesco et al. conducted a series of studies on the particulate emission of the dual-injection system using experimental methods [6-8]. Its research showed that compared with the mixed fuel injection strategy for direct injection, the dual injection system can improve the thermal efficiency of the engine. Compared with in-cylinder direct injection, the engine test using dual-injection fuel injection strategy reduces the quality of particulates emitted by the engine. However, the number of nuclear particles produced is increasing.

Domestic scholars have done relevant research on the gaseous emissions and non-gaseous emissions of dual-injection gasoline engines of pure gasoline and gasoline and ethanol. However, few people have used numerical simulation to study the state of the in-cylinder mixture and the formation process of the main component soot of only-gasoline and gasoline-ethanol dual-injection engines. There are few studies on the influence of fuel injection ratio and fuel injection timing changes on the state of the cylinder mixture and the combustion process and the formation process of soot emissions.

Gasoline engine intake port and in-cylinder direct injection combined injection technology is one of the effective measures that can take into account the engine's thermal efficiency and particulate emissions. In this paper, a 2.0T gasoline engine was modified to a dual-injection engine. Numerical simulation method is used to study the influence of fuel type and fuel injection strategy on the in-cylinder mixture formation process, combustion process and soot emission of dual-injection gasoline engine.

2 3-D Set-up of engine

2.1 Geometric model

Figure 1 shows the structural model of one cylinder of a supercharged direct-injection inline four-cylinder engine, which is obtained after simplification of the model. The simplified model mainly includes inlet and exhaust valves, intake and exhaust ports, combustion chamber ceiling, piston top, cylinder wall, injector nozzle part and spark plug, etc. In addition, it is noting that due to the complicated working process in the engine cylinder, which involves the reciprocating movement of the valve switch and the piston up and down, it is necessary to set up a dynamic grid structure to analyze the fluid movement during the calculation. The calculation area is not fixed and the grid is more complicated. Use the grid generator Fire-Fame-Hexa that comes with the AVL-Fire software to set the dynamic grid division file. When setting the conditions for meshing, a boundary layer is set at the wall, so there is a body-fitted grid near the wall to more accurately reflect the changes in the boundary state of the flow area, which can more effectively improve the accuracy of numerical simulation calculations.

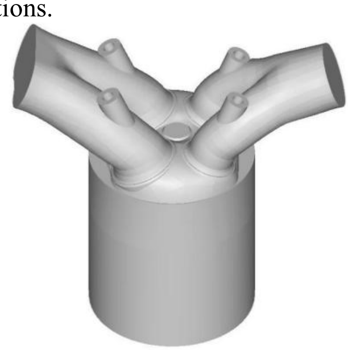


Fig. 1. 3-D Set-up of engine.

2.2 Engine grid

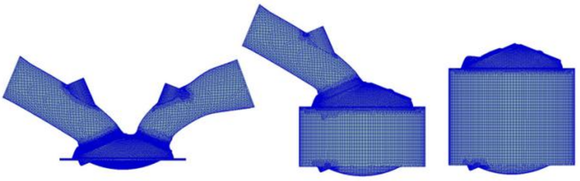
The calculation in the cylinder includes the intake stroke, compression stroke and partial expansion power stroke of the direct injection gasoline engine at 2000rpm and BMEP = 0.6MPa. The compression top dead center is selected as 720° CA, and the intake valve is opened at 322° CA, set as the starting time of the simulation calculation; the exhaust valve opening time 864° CA, set as the calculation ends. Set the basic size of the grid to 2mm. In addition, in order to improve the accuracy of the numerical simulation calculation results, and fully reflect the influence of the internal geometric structure of the engine cylinder on the fluid flow in the cylinder, the small parts and key parts are refined when the dynamic mesh is divided: in the intake and exhaust valves small or complex areas such as valve seats, spark plugs and fuel injectors and their surrounding areas have been refined to varying degrees. The minimum grid size is set to 0.125mm.

From the test process, after the intake valve is closed, the intake port does not participate in the working process in the cylinder. Therefore, after the intake valve is closed, another set of grids that remove the intake port structure is used to continue the calculation of the fluid in the cylinder. When the exhaust valve is closed, perform the same operation. The specific division method is shown in Figure 2 engine dynamic grid, the boundary condition setting is shown in Table 1, and the initial condition setting is shown in Table 2.

(a) From the opening time of the intake valve to the closing time of the exhaust valve (322° CA～362° CA), this set of grids includes intake and exhaust valves and intake and exhaust passages. The grid is the most complex, with the top stop the number of grids at a point is 1.5 million.

(b) From the closing time of the exhaust valve to the closing time of the intake valve (362° CA～549° CA), this set of grids has no exhaust channels, and the minimum number of grids is about 1.2 million.

(c) From the moment the intake valve closes to the exhaust valve opens (549° CA～864° CA), this set of grids has no intake and exhaust ports, and the minimum number of grids is about 800,000.



(a) Valve overlap period (b) Intake phase (c) Compression stage

Fig. 2. Engine dynamic grid.

Table 1. Boundary conditions of wall temperature.

Wall	T(K)
Piston	500
Cylinder head	500
Cylinder liner	450
Intake valve	400
Exhaust valve	485
Inlet port	385
Exhaust port	475

Table 2. Initial condition setting.

Program	Set
Inlet port pressure(Pa)	146965
Inlet port temperature(K)	323
In-cylinder pressure(Pa)	134379
In-cylinder temerature(K)	1065
Exhaust pressure(Pa)	134207
Exhaust temperature(K)	1035

2.3 Calculation model

The in-cylinder calculation involved in this paper includes the spray process and the combustion reaction process. It is necessary to select the spray model and the combustion model in the engine cylinder reasonably in the calculation. The spray calculation is more complicated and involves many models, including: turbulence model and turbulence diffusion model in spray model; particle interaction (including droplet collision, fusion, splashing, etc.); spray droplet evaporation model and spray droplet Broken sub-models and droplet collision models, etc. The combustion process also involves combustion models and emission models.

Three orthogonal cross-sections of X, Y and Z are selected to analyze the working process in the cylinder: Among them, the X1, X2, and X3 cross-sections in the figure are perpendicular to the x-axis, and X1 passes through the center line of the cylinder; X2 and X3 respectively pass through The axis of the intake and exhaust valves; the Y section in the figure is perpendicular to the y axis and passes through the center line of the cylinder; the Z section is perpendicular to the z axis and is close to the position of the spark plug, as shown in Figure 3. The X1 section and Z section are used to analyze the mixing situation and distribution of the mixture in the cylinder, and the X2 section is used to analyze the air flow state in the cylinder and the distribution of turbulent kinetic energy in the cylinder.

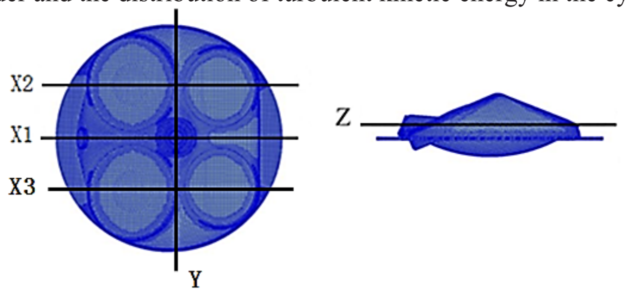


Fig. 3. Section position.

3 3-D Simulation results and analysis

The distribution and state of the mixed gas in the engine cylinder play a very important role in the quality of the combustion in the engine cylinder. Furthermore, it also affects the noise level of the engine combustion process, the generation and emission of harmful gases, and even the power performance of the engine. And economy, etc. A reasonable organization of fuel injection strategies and the use of direct injection and gas channel combined injection methods can effectively reduce engine particulate matter generation and emissions while ensuring engine power and economy.

This section uses the AVL-FIRE 3-D simulation software to study the effect of the combined injection GDI/GPI (Gasoline Direct Injection / Gasoline Port Injection) injection ratio on the mixture and soot emissions in the cylinder. Table 3 shows the fuel injection scheme to study the influence of different fuel injection ratios on the working process in the cylinder.

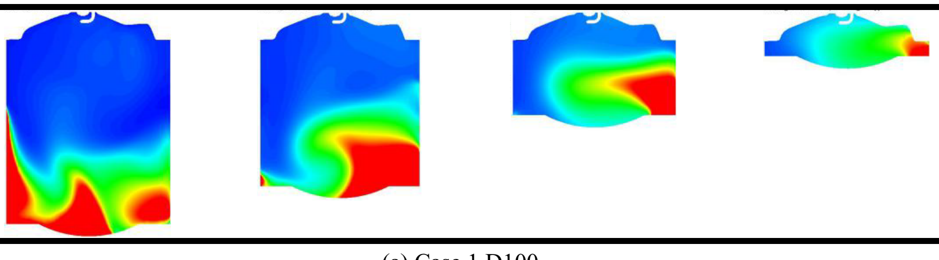
Table 3. Fuel injection scheme.

Case	GDI:GPI	DI Injection time (° CA BTDC)	PIInjection time (° CA BTDC)	FUEL
Case 1 D100	100:0	300	-	Gasoline
Case 2 D85	85:15	300	390	Gasoline
Case 3 D65	65:35	300	390	Gasoline

3.1 The influence of DI/PI injection ratio on the process of mixture formation

In the following, D100, D85 and D65 are used to represent the three fuel injection ratios. Figure 4 shows a distribution diagram of the equivalence ratio of the in-cylinder mixture under three injection schemes with different proportions. The fuel injection scheme is different, the state of the fuel entering the cylinder is different. Because the sprayed fuel enters the cylinder in liquid form under the direct injection mode, the droplets break and evaporate in the cylinder, while the fuel evaporates in the intake port under the port injection mode, than after mixing with intake air, it enters the cylinder with the intake air. In the combined injection mode of port and direct injection, the fuel can be injected into the cylinder through a variety of combined ways. Comparing Figure 4 (a), (b), (c), it can be seen that the mixed gas concentration in the cylinder with D65 injection ratio is relatively uniform. At 170° CA BTDC, when the in-cylinder injection is just completed, the fuel in the D100 injection scheme is concentrated under the cylinder, and the upper part of the cylinder has less fuel. Compared with the D100 injection scheme, the D85 injection scheme is at 170° CA For BTDC, there is a small amount of fuel distribution in the upper part of the cylinder, and the mixed gas concentration is between 0.2-0.4. D65 fuel injection scheme at 170° CA BTDC crankshaft angle mixed gas distribution is not much difference from D85 fuel injection scheme. At the moment of ignition (27° CA BTDC crankshaft angle), the three fuel injection ratios form a richer mixture on the intake side and a leaner exhaust side. The latter two fuel injection schemes have a mixture equivalent ratio of 0.8-1.2 in most areas of the cylinder. It can be seen that an appropriate increase in the intake port fuel injection ratio can appropriately improve the uniformity of the air-fuel mixture.

Figure 5 shows the distribution of the flow field in the cylinder and the distribution of the oil film in the cylinder at the time the intake valve is closed. Figure 6 shows the variation of fuel wall collision amount and oil film quality with crankshaft angle for three fuel injection schemes. In the GDI injection mode, the oil film in the cylinder is mainly distributed on the piston crown and the cylinder wall on the exhaust side. There is a small amount of oil film on the intake valve due to the airflow in the cylinder. Among them, most of the oil film is distributed on the top of the piston. After the intake valve is closed, there is still fuel that collides with the top of the piston under the action of airflow and forms a fuel film on it. There is a very small amount of fuel film on the intake valve. After increasing the fuel injection ratio of the intake port of the dual-injection system, as the ratio of fuel directly injected into the cylinder is reduced, the oil film distribution in the cylinder is significantly reduced. At the same time, there are oil film areas on the intake port and the back of the valve, and the thickness of the oil film is related to the air passage. The proportion of fuel injection is positively correlated. The air-fuel mixture in the cylinder is pushed to the left of the cylinder with the airflow in the cylinder, that is, the exhaust valve side, so that the air-fuel mixture concentration in the left side of the cylinder is higher than the air-fuel mixture concentration in other parts of the cylinder. As the piston moves up, the oil and gas in the cylinder is gradually mixed under the action of the airflow. At 27° CA BTDC crankshaft angle (ignition time), a mixture of denser on the intake valve side and relatively leaner on the exhaust valve side is formed in the cylinder. .



(a) Case 1 D100

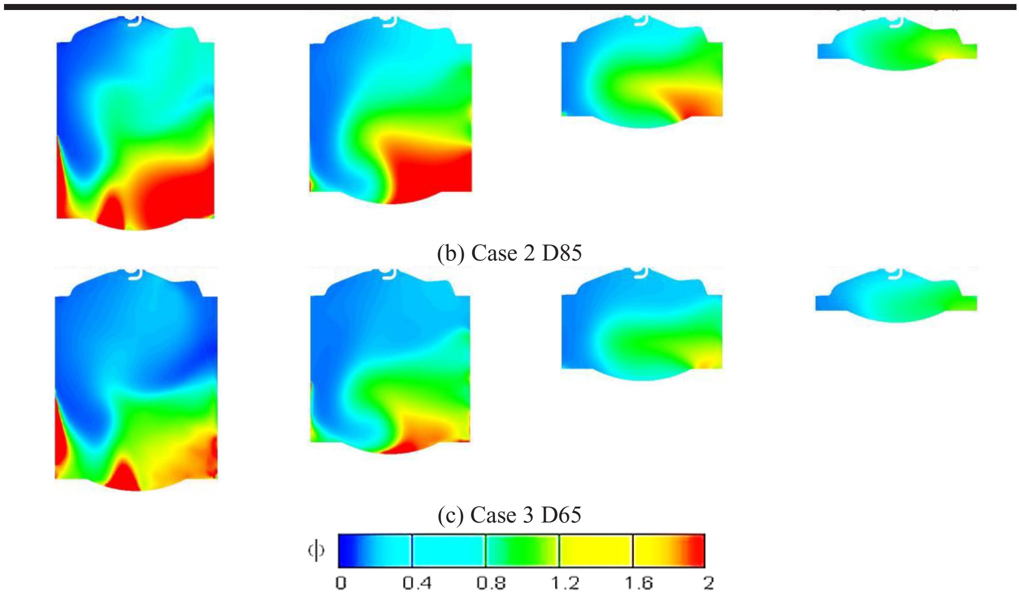


Fig. 4. Equivalent ratio of mixed gas in cylinder with different fuel injection ratio.

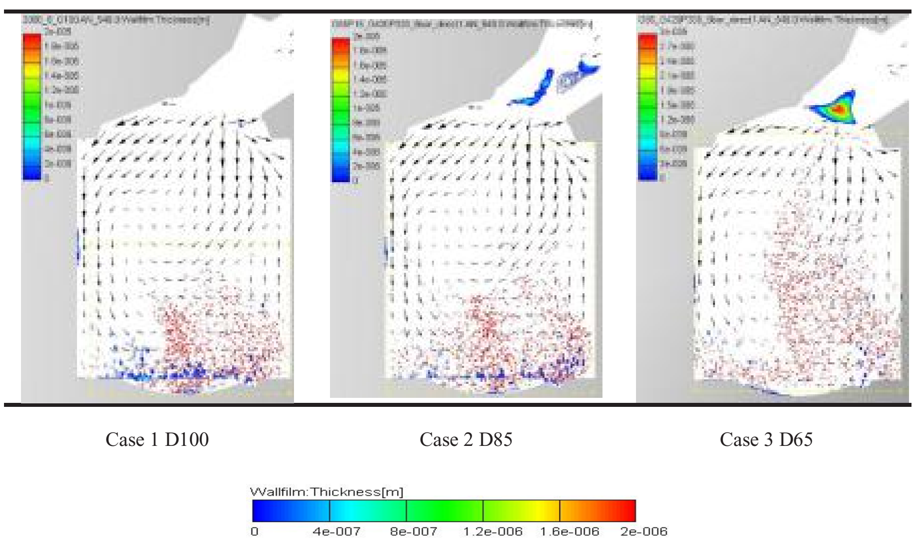
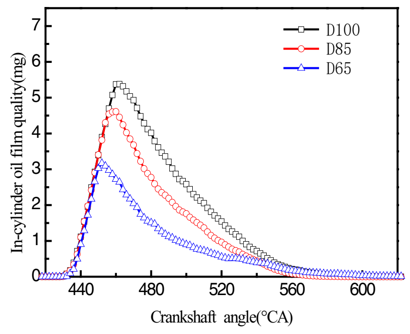
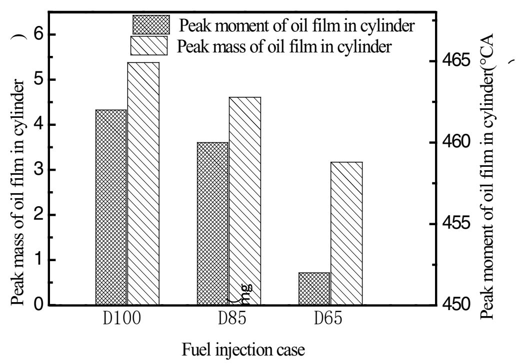


Fig. 5. The oil film in the cylinder at the intake valve close timing(549° CA).



(a) In-cylinder oil film quality



(b) Peak mass and peak moment of the oil film in the cylinder

Fig. 6. In-cylinder oil film quality.

3.2 The influence of DI/PI injection ratio on the combustion process

Figure 7 shows the combustion pressure curves in the cylinders under the three fuel injection ratios. It can be seen that the cylinder pressures in each working condition are not much different. Figure 8 shows the distribution of turbulent kinetic energy at the ignition moment under three fuel injection ratios. When the turbulent kinetic energy in the cylinder is relatively high and its distribution is reasonable, it can shorten the flame development period and improve combustion period of combustion in the cylinder to a certain extent. In turn, can appropriately and effectively improve the engine's working power. According to Figure 8, when only the fuel injection ratio is changed, the turbulent kinetic energy in the cylinder changes very little, so under the research conditions, the fuel injection ratio has a small effect on the turbulent kinetic energy. Figure 9 shows the flame development process with different fuel injection ratios, with the exhaust side on the left and the intake side on the right. It can be seen from Figure 9(a) that at compression top dead center 10° CA, the flame surface density at the spark plugs of the D85 and D65 injection schemes is around 1000, and the temperature at the spark plugs of the D100 injection scheme is lower than 900K. This is because the development of fire cores is more sensitive to lean mixtures. Compared with thicker mixtures, lean mixtures are more likely to hinder the development process of fire cores[9]. As shown in Figure 4, jetting at D85 and D65 In the scheme, the mixed gas concentration near the spark plug is higher than the mixed gas concentration in the D100 injection scheme. Among the three injection schemes, the flame front of the D85 injection scheme reaches the cylinder wall earliest, and the flame propagation speed is

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faster. The flame propagation speed of the D100 and D65 injection schemes is similar, and the D65 injection scheme reaches the cylinder wall earlier than the D100 injection scheme.

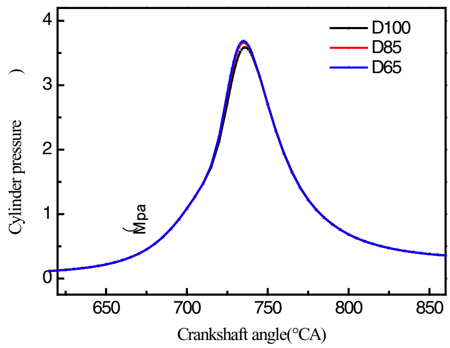


Fig. 7. Pressure in cylinder.

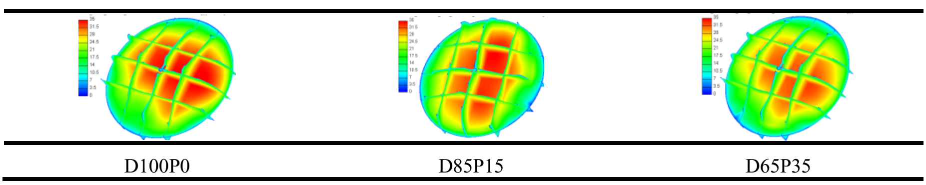


Fig. 8. Turbulent kinetic energy distribution in the cylinder at the moment of ignition.

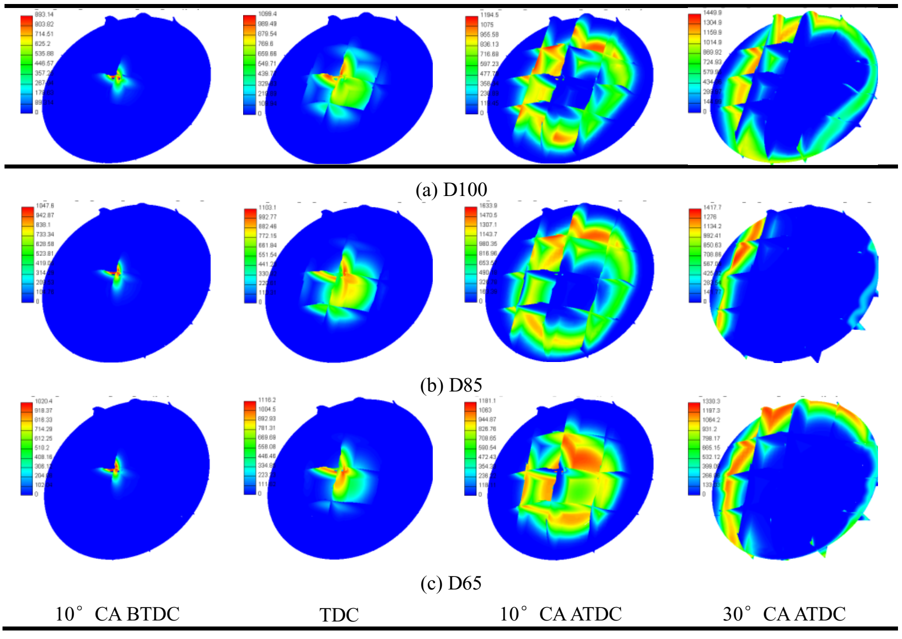


Fig. 9. Flame propagation process with different fuel injection ratios.

Figure 10 shows the variation process of the temperature field in the cylinder with the crankshaft angle after ignition (The left side is the exhaust side and the right side is the intake side). The flame propagates from the center of the cylinder (spark plug) to the periphery of the cylinder wall, after top dead center 20° CA, the flame surface spreads to the cylinder wall. It can be seen from the figure that since the mixture concentration in the exhaust side area is lower than that in the intake side area, the flame propagates from the center to the cylinder wall after ignition, and the flame front reaches the cylinder wall on the intake side first, and then reaches the cylinder wall on the exhaust side.

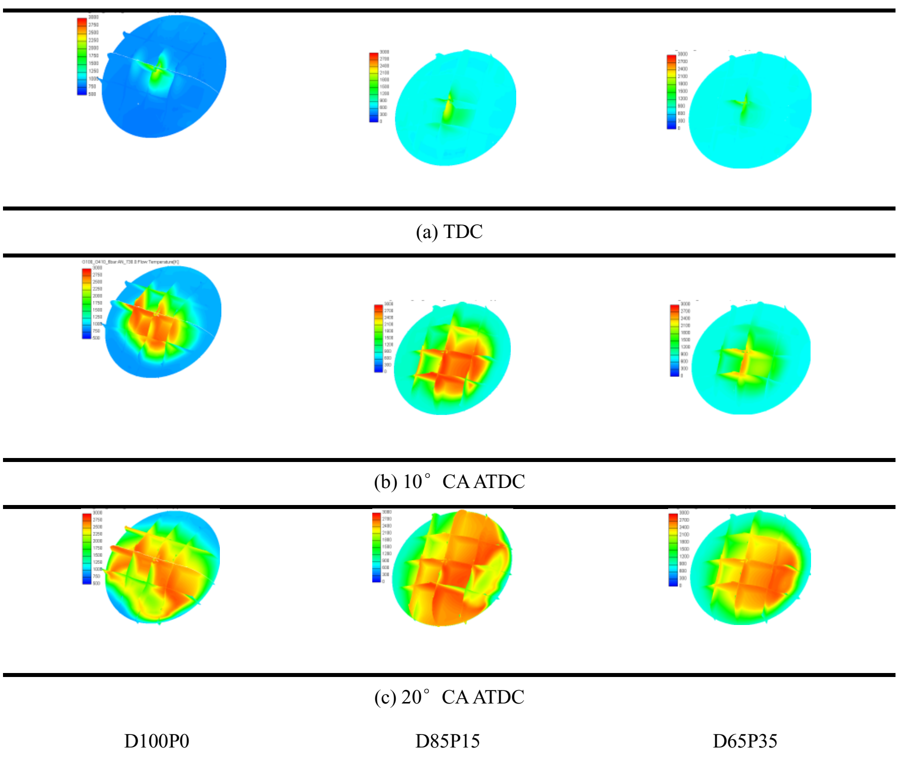


Fig. 10. Comparison of the temperature field in the cylinder with different fuel injection ratios.

3.3 The influence of DI/PI injection ratio on soot emission

Figure 11 shows the variation of the soot mass fraction in the cylinder with the crankshaft angle at different in-cylinder/airway injection ratios. Figure 12 shows the residual amount of soot in the cylinder after the exhaust valve is opened. It can be seen from the figure that under three different fuel injection ratios, the soot mass fraction in the cylinder presents a single peak distribution, and the soot peak appears near 30° CA after top dead center; the soot mass fraction peak moments of the three fuel injection schemes follow the GDI injection The reduction in oil ratio is slightly delayed.

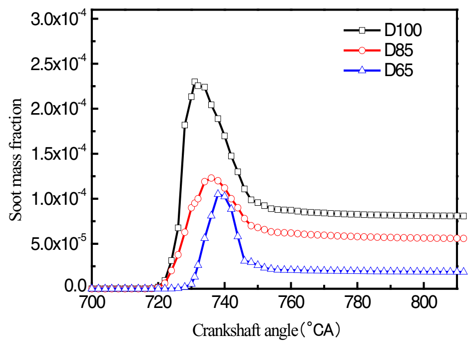


Fig. 11. Soot generation process in the cylinder with different DI/PI injection ratios

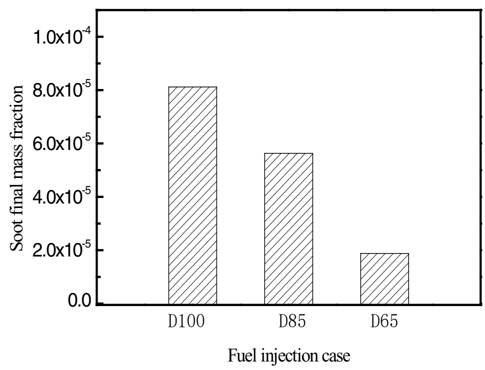
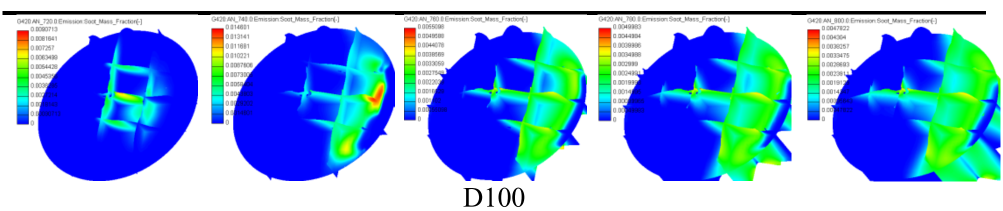


Fig. 12. The final mass fraction of soot in the cylinder with different fuel injection ratios.

Figure 13 shows the distribution area of in-cylinder soot in the cylinder under different in-cylinder/airway injection ratios. Soot mass score generation decreases with the decrease of GDI injection ratio. Especially the amount of soot generated at 65% GDI injection ratio is significantly reduced. The final soot mass fraction emission of D65P35 is better than that of D100 injection.

It can be seen from the figure 13 final soot mass score that there is more soot generation on the intake side and less soot distribution on the exhaust side. This is caused by the uneven distribution of the mixture in the cylinder. For the area on the intake side, the concentration of the mixed gas is higher than other areas, so relatively more soot is generated on the intake side.



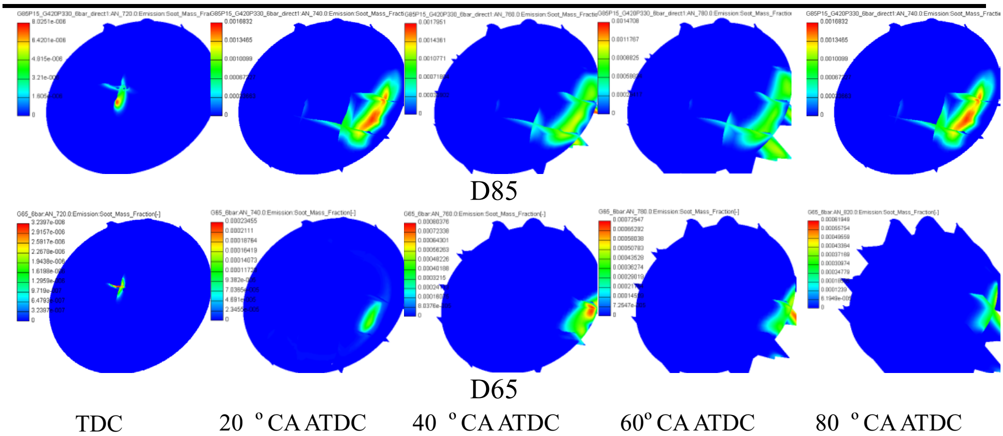


Fig. 13. The distribution of soot mass fraction in the cylinder with different DI/PI injection ratio.

Figure 14 shows the quality of the inlet port oil film. It can be seen that as the proportion of inlet port fuel injection increases, the final quality of the inlet port oil film increases. Taking into account the fuel economy of the gasoline engine and the particulate emissions generated by the gasoline engine, 85% The in-cylinder fuel injection ratio is more reasonable, and 85% of the in-cylinder direct injection fuel ratio is selected for the following research.

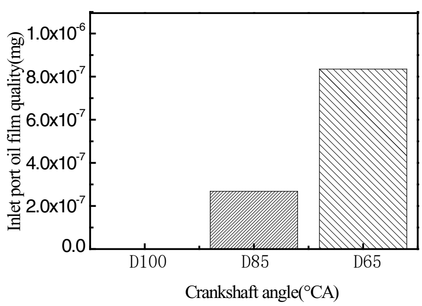


Fig. 14. Inlet port oil film quality.

4 Conclusion

Based on a 2.0T gasoline engine modified intake port and in-cylinder combination engine, numerical simulation methods are used to study the influence of fuel type and fuel injection strategy on the formation process, combustion process and soot emission of dual-injection gasoline engine in-cylinder mixture. Conclusion as below:

Appropriately increasing the intake port fuel injection ratio can appropriately improve the uniformity of the air-fuel mixture. After increasing the fuel injection ratio of the intake port of the dual-injection system, as the ratio of fuel directly injected into the cylinder is reduced, the oil film distribution in the cylinder is significantly reduced. At the same time, there is an oil film area on the intake port and the back of the valve, and the thickness of the oil film is positively correlated with the injection ratio of the air port

2) Among the three injection cases, the flame front of the D85 injection case reaches the cylinder wall earliest, and the flame propagation speed is faster. The flame propagation

speed of the D100 and D65 injection cases is similar, and the D65 injection case reaches the cylinder wall earlier than the D100 injection scheme.

3) Soot quality score generation decreases with the decrease of GDI injection ratio. Especially the amount of soot generated at 65% GDI injection ratio is significantly reduced. The final soot mass fraction emissions of D65 are nearly an order of magnitude lower than the final soot mass fraction emissions of D100 injection scheme

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