S3–25

Study of the effect of combustion chamber shapes on the mixture formation and combustion characteristic on CNG–DI engine

Yang Miao, *Lin Xuedong*[*], *Jiang Tao*
State Key Laboratory of Automotive Simulation and Control, Jilin University, Changchun 130022, China
[*]Corresponding author: Email: xdlin@jlu. edu. cn

Summary

In order to refit 2.0L type direct injection diesel engine into CNG–DI engine, a spark plug is arranged between two exhaust valves under the condition that the original four–valve structure and injector position are unchanged, and several combustion chambers are designed, the compression ratio is reduced from 17.2 of the original engine to 12. The combustion system model of CNG–DI engine was established by using the 3D CFD software FIRE, and the turbulence model and combustion model of CNG–DI engine are verified by the experimental results of CNG–DI optical engine. On this basis, in order to design a reasonable combustor for CNG–DI engine, the effects of different combustor shapes on micro–physical fields such as turbulence in cylinder are simulated and analyzed, and the effects of different combustor shapes on mixture formation, combustion process and NO formation are studied. The results show that the shape of the combustion chamber has a great influence on the turbulent characteristics and the distribution of the concentration field, especially the concentration field and the turbulence intensity near the spark plug play a decisive role in the whole combustion process. When the entrance structure of the combustion chamber was designed to be straight entrant and the bottom structure of the combustion chamber was designed to be properly raised, it can not only inhibit the production of NO, but also be favorable to lean combustion, and the rate of combustion heat release can also be effectively controlled. The amount of NO is closely related to the reaction rate, the reaction duration and the size of the reaction area when the CNG fuel is injected directly into the cylinder, while the reaction rate of NO is closely related to the oxygen concentration and temperature in the local region, it can be controlled by reasonable design of combustion chamber structure.

Keywords: DI–diesel engine; refit; CNG–DI engine; combustion chamber structure; simulation analysis;heat release rate; NO formation rule

1. Introduction

The diversifications of energy sources and rarified combustion technology in cylinder have been the trend of vehicle engine with the development of automobile low–carbon. The natural gas has been one of the widely used automotive alternative fuels comparative maturity technically due to its high H/C ratio, clean combustion, low HC, CO and CO_2 and abundant reserves [1]. The development of low–carbon has been effected directly by the loss of volumetric efficiency, and when the natural gas is injected in intake port, the output power loss of the low heat value of mixture [2]. CNG–DI lean combustion technology is the effectively way to the question [3, 4]. How to improve the combustion stability and reduce the NO production at the same time is the core problem of the CNG–DI lean combustion. The mixture formation mechanism, flame propagation characteristic of lean combustion and NO formation law is the key to deal with the core problem of the lean combustion of CNG–DI engine when the lean combustion of CNG fuel in cylinder is organized.

In recent years, the effects of combustion characteristics and flow characteristics of CNG engines on the combustion process of natural gas have been deeply studied [5–7]. However, when CNG fuel is injected directly into the cylinder, there are few reports on the influence of combustion chamber structure on mixture formation mechanism and combustion process, as well as on the formation of NO during lean combustion.

In this paper, based on the study on flame propagation characteristics of rarified combustion on CNG–DI optical experimental prototype, the simulation model of 2.0L direct injection four–valve engine combustion system is established by using software FIRE. On the basis of this, the microscopic physical field and its dynamic distribution characteristics of two kinds of gas flow in cylinder with different combustion chamber shapes are simultaneously studied. The mixture formation law of CNG–DI engine, rarified combustion characteristic and NO generation rule are also studied on simulation.

According to the preview project of the that is improved 2.0L type direct injection four–valve diesel engine (prototype) to CNG–DI engine, combined with the previous visual experimental results of flame propagation characteristics of CNG–DI optical engine, the in cylinder turbulence model and combustion model of CNG–DI engine are verified. On this basis, the simulation model of four–valve CNG–DI engine combustion system with improved prototype is established by using CFD software FIRE., On this basis, the influence of different combustor structures on the microphysical field and its dynamic distribution in the cylinder is analyzed when the CNG fuel is directly injected into the cylinder, and the formation mechanism of the mixture gas in the CNG–DI engine is also analyzed by simulation. The combustion characteristics and NO generation law are analyzed by simulation too, which provides a theoretical basis for the correct design of CNG–DI engine combustion system.

2. Research Condition and Method

2.1 Research condition of optical engine

The main technical parameters of the single–cylinder optical prototype for model verification are shown in Table1. The arrangement of the optical test system for observing flame propagation characteristics is shown in Fig. 1. The arrangement of spark plugs and injectors is shown in Fig. 2. Two injectors and two spark plugs are arranged on the cylinder head, and the intake and exhaust systems are arranged on one side of the cylinder head. In order to facilitate the adjustment of intake swirl, the cylinder clearance volume is large. So, the compression ratio is low to 6.13 and the engine test speed is

200r/min restricted by the prototype structure, but this does not affect the basic research of flame propagation mechanism. The main components of natural gas are shown in Table 2.

Table1. Technical parameters of experimental prototype

Type	four	Type	four
Cylinder diameter/mm	135	Displacement/L	4.0
Stroke/mm	280	Compression Ratio	6.13

Table 2. The main composition of natural gas fuel

	CH_4	C_2H_6	C_3H_8	C_4H_{10}	N_2
Vol%	85.45	4.51	3.39	3.71	2.94

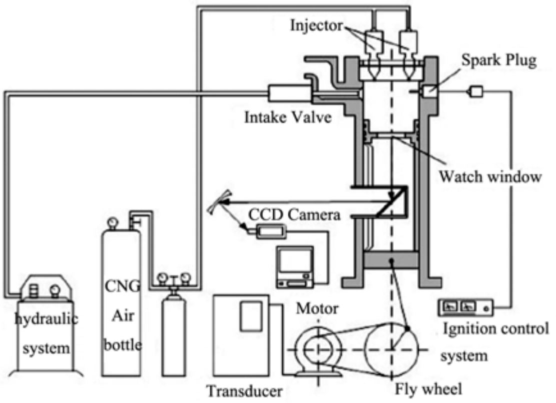

Figure 1. The diagram of experiment facilities

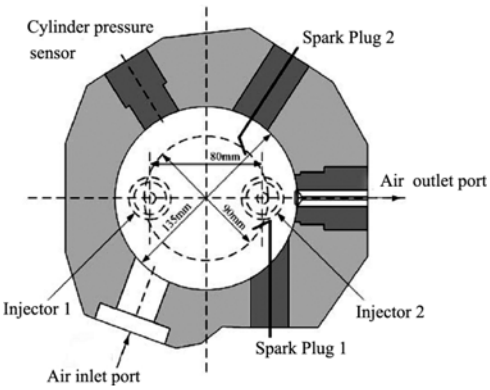

Figure 2. The diagram of the location of spark plug and injector

The intake swirl intensity is adjusted by adjusting the bias degree of the inlet air flow direction relative to the cylinder center. In the case of low engine speed, the initial airflow intensity can be improved as much as possible by adjusting the intake swirl ratio (SW=6.0). The swirl injector is used in the experiment, and the injection pressure is 5MPa. The process of flame propagation is recorded continuously by CCD high–speed camera at 563 image/sec. The measured results are that the average cylinder pressure is measured for 10 times under different working conditions.

2.2 The simulation condition and validation

The flame propagation characteristics and indicator diagram of CNG–DI engine were measured by CNG optical engine, and the turbulence and combustion models in the cylinder of CNG–DI engine were verified. Then according to the structural characteristics of the optical engine and without changing the compression ratio, the cylindrical combustion system model is generated directly by using FIRE software, which makes the combustion chamber shape consistent with the optical engine. According to the variation of cylinder volume during engine operation, the duration from intake stroke BDC to power stroke BDC is divided into four layers grid, as shown in Fig. 3.

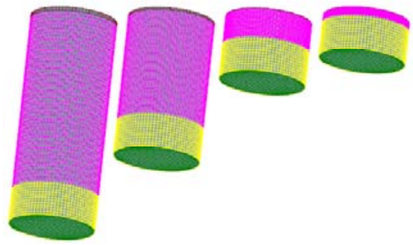

Figure 3. Simulation mesh

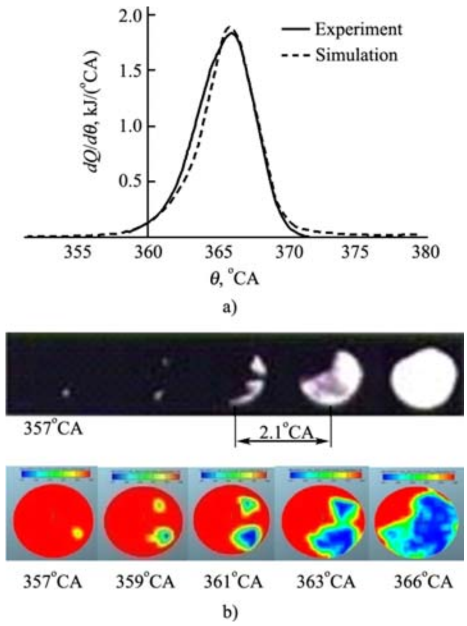

Figure 4. The comparison of experiment and simulation result
a) The comparison of heat release rate, b) the comparison of flame propagation

In the injection and combustion stages, the simulated time step is 0.5°CA, and the expansion stage is 1°CA. according to the characteristics of CNG gas fuel, the turbulent model adopts the k–ε–f model, and the combustion model adopts the ECFM model. The turbulent diffusion model is O O'Rourke model.

The air flow state of the entrance of the cylinder was supposed uniform and the measured value was adopted as the simulation boundary condition. the average intake pressure was 0.9 bar, the temperature was 350K; The top of piston was supposed to be the moving boundary and the temperature of which was 593K, the cylinder wall and the bottom if cylinder head was supposed to be the fixed boundary, the temperature of cylinder wall was 403K and the temperature of the bottom of cylinder head was 593K. The comparison of the measured value of heat release rate and flame propagation speed with the simulation result was when the equivalence ratio was $\phi=0.93$, two point ignition and ignition timing was $(\theta_{i1}, \theta_{i2})=(-4,$

–3)°CA, two injector and injection timing was θ_{inj}=–120°CA was shown as Fig. 4. The result shows that the simulation result and experimental result was in good agreement, which indicates that the built simulation model and algorithm is in accord with actual mixture formation and flame propagation of CNG fuel.

2.3 The technically condition and research program of prototype

When the CNG–DI engine is improved and designed with 2.0L DI diesel engine, the basic structure of the diesel engine prototype remains unchanged, the compression ratio is reduced from 17.2 to 12, and the shape of the combustion chamber is improved. According to the characteristics of the original engine 4 valve structure, as shown in Fig. 5, the position of the injector and spark plug is rearranged.

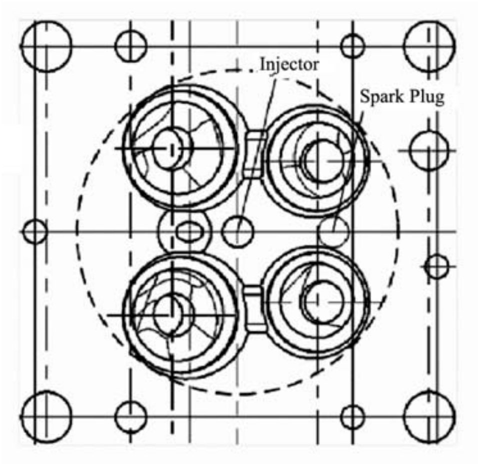

Figure 5. The diagram of injector and spark plug

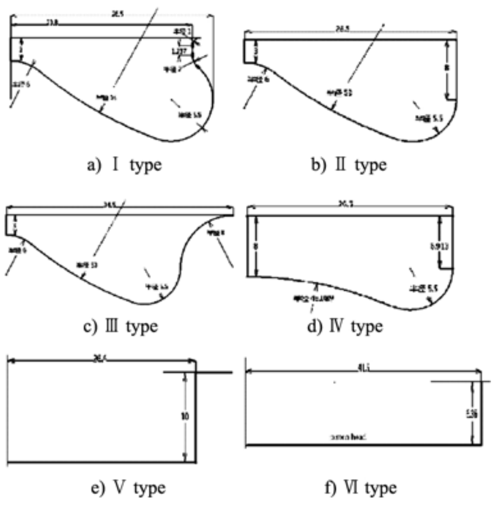

Figure 6. The different combustion chamber shape

The 6 types different kinds of combustion chamber shape was designed aimed at the scheme of injector in center of the cylinder and spark plug in exhaust valve. Among them, the structure of I type combustion chamber keeps the same as the original diesel engine type; The throat ratio of II type combustion chamber was changed to 1 compared to I type combustion chamber; The throat combustion chamber was changed to displaced one in III type combustion chamber. The effect of throat shape on the mixture formation mechanism was studied on these three combustion chamber; The shape of the bottom of combustion chamber was changed on the fundamental of Straight Port combustion chamber in IV~V combustion chamber: The shallow convex shape in bottom for IV type combustion chamber, flat bottom in V type combustion chamber and flat in piston top for VI type combustion chamber, based which the effect of combustion chamber shape in bottom on mixture formation mechanism was studied.

The establishment of mesh model structure, simulation calculation method and simulation model of the simulation prototype was basically consistent with the optical engine. The boundary conditions of the simulation calculation are determined by the measured results in the experimental study of 2.0L diesel engine. The main technical parameter and simulation condition of simulation prototype was shown in Table 3. The maximum torque engine speed (2 200r/min) working condition was selected as simulation calculation condition, the cycle injection quantity was supposed to be 34.77mg, the injection timing was supposed to be 158°CA BTDC, and the ignition timing was 6°CA BTDC.

3. Simulation Results and Analysys

3.1 The effect of combustion chamber on turbulence characteristic in cylinder

When the CNG fuel is injected directly into the cylinder, and the injecting timing was 158°CABTDC and the injection ending timing was 10°CABTDC, The influence of different combustion chamber shapes on the turbulent characteristics in the cylinder was shown as Fig. 7. It follows that, The TKE in cylinder increased quickly at the start period of injection increased quickly due to the input jet kinetic energy of the certain amount of injected fuel gas during the early stage of compression stroke, which decreased quickly with the development of compression stroke after peak value. The variation law of TKE would be different due to the different combustion chamber shape with the same jet kinetic energy.

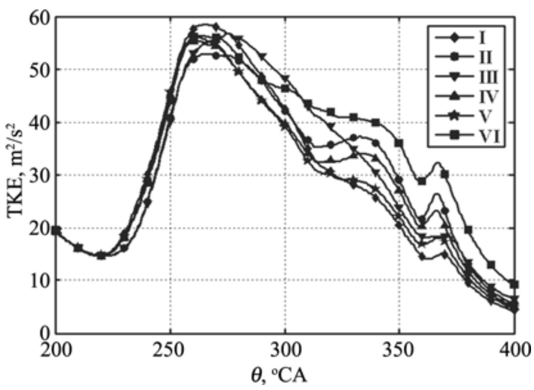

Figure 7. The effect of combustion chamber on TKE

Table 3. The technical parameter and simulation condition

Type	2.0L 4 cylinder CNG–DI engine
Cylinder bore x stroke	83×92mm
Compression ratio	17.2
Rated power	100kW/(3 800r/min)
The maximum torque	310N·m/(2 200r/min)
The simulation working condition	
Rotate speed	2 200r/min
Jet mass	34.8mg
Injection timing	158°CA BTDC
Injection duration	148°CA
Ignition timing	6°CABTDC
The simulation condition	
Pressure in the entrance	1.08bar
Induction swirl ratio	2
The initial TKE	33.443m^2/s^2
Temperature in piston top	553K
Cylinder wall	403K
Cylinder head	403K

The influence of different combustion chamber shapes on the turbulent characteristics in cylinder when the CNG fuel is injected directly into the cylinder, and the injection timing was 158°CABTDC and the injection ending timing was 10°CABTDC, was shown as Fig. 7. The increasing rate of TKE in I type combustion chamber was the fast during the early stage of injection and the peak value of which was the highest; The increasing rate and peak value of TKE in II type combustion chamber decreased obviously. In the middle stage of the compression stroke(270~320°CA), the average TKE attenuation degree is different due to the different flow states in different combustion chamber; In the expansion stage (360~400°CA), the turbulent intensity of TKE reaches the second peak due to combustion, and the peak value is different due to different shape of the combustion chamber, and the peak value of the VI type combustion chamber is the highest and the peak value of the I type combustion chamber is smallest.

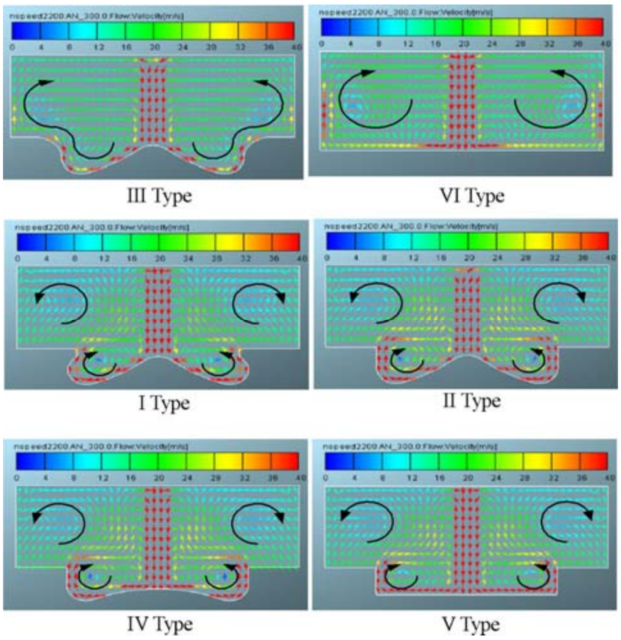

Figure 8. The air flow distribution characteristic at 300°CA

The distribution of cylinder air flow state in different combustion chamber at 300°CA was shown in Fig. 8. With the compression stroke, the turbulence intensity in the cylinder attenuates to varying degrees, and the tumble flow forms symmetrical distribution relative to the center of the combustion chamber, The tumble flow direction of III combustion chamber and VI combustion chamber is toward the center of the cylinder, and the turbulent intensity decay rate is slow, while the tumble flow direction of the type of combustion chamber (I\II\IV\V) is backward to the center of the combustion chamber, and the turbulence intensity decreases rapidly.

The cylinder clearance volume could be ignored near the end of compression stroke (320~360°CA), the air flow was focused on the combustion chamber; the top of piston of VI type combustion chamber was flat, the turbulence intensity was higher due to the directly compression of tumble in cylinder at the end of the compression stroke; the peak of turbulence intensity is the highest at the beginning of the compression stroke. Because the type I combustor has a throat at the ingress and a bulge at the bottom, but it weakens rapidly during the compression process and the minimum at the end of the compression process. The turbulent intensity changes of type I combustor and V type combustor with flat bottom are basically the same at the later stage of compression stroke; the attenuation degree of turbulence intensity at the end of compression stroke is improved when the bottom shape of the straight opening combustion chambers (II\IV\V) changes from flat bottom to convex shape.

3.2 The effect of combustion chamber shape on concentration field and combustion characteristic

The combustion process include ignition process and flame propagation process according to the ignition mode of CNG–DI engine, which is close related to the turbulence characteristic in cylinder and distribution characteristic of concentration field. The effect of concentration field and turbulence intensity near the spark plug is of special importance to lean combustion process.

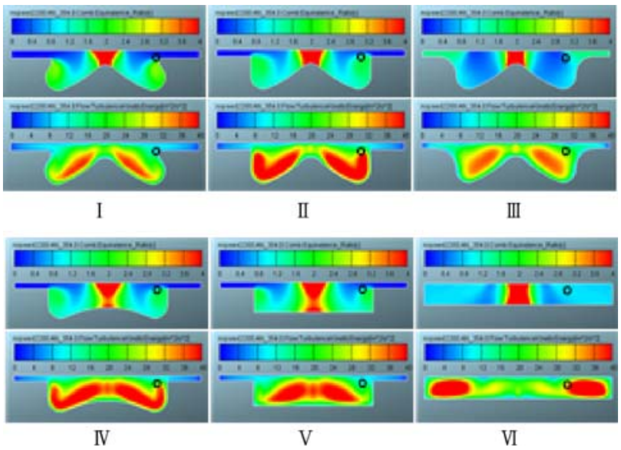

Figure 9. The distribution characteristic of concentration field (up) and TKED (down) at 354°CA

The effect of different combustion chamber on mixture concentration field and distribution characteristic of turbulence when the injection timing was 6°CA BTDC was shown as Fig 9, the black circle showed the installation site of spark plug. From which we can know that the combustion chamber has an obvious influence on mixture concentration field and TKE distribution characteristic, especially near the spark plug.

Table 4. The equivalence ratio and TKE at spark plug at 354°CA

Type	I	II	III	IV	V	VI
Equivalence ratio	0.443	1.369	0.651	1.373	1.012	0.926
TKE, m^2/s^2	10.12	21.33	21.4	20.98	11.73	35.76

The mixture concentration and turbulence intensity near the spark plug at ignition timing was shown as Table 4. From which we can know the mixture concentration near the spark plug of I type and III type combustion chamber was lean which was not easy to be ignited. The mixture concentration in V and VI type combustion chamber was most easy to be ignited, the turbulence intensity in V type combustion chamber was lower while the VI type combustion chamber was the highest. So the flame propagation speed was the fastest after the formation of flame nucleus in VI combustion chamber, whose heat release rate was the fastest and peak value was the highest as shown in Fig. 10; The turbulence intensity was lower relatively although the mixture concentration in V type combustion chamber was fit for ignition and the fast formation of flame nucleus, so the flame propagation speed is slowly and the peak value and heat release rate was decreased; In the III combustion chamber, although the TKE around the spark plug is strong, because of the dilute gas concentration and the same as the type I combustor, it is not suitable for ignition, so the formation of the flame core is slower, the flame propagation speed is low, and the heat release rate is slow. The equivalent ratio and TKE near the spark plugaround are about 1.37 and $21m^2/s^2$ respectively when type Ⅱ and type Ⅳ combustor are ignited. Although the combustion rate is slower than that of VI type combustion chamber, the combustion rate can still be controlled effectively.

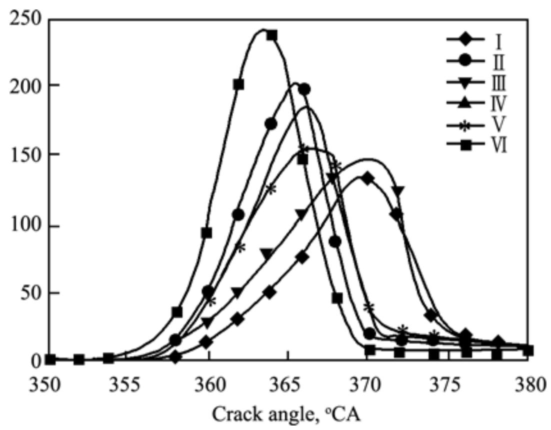

Figure 10. The effect of combustion chamber shape on heat release rate

The results show that the design of combustor structure not only can effectively control the mixture concentration and turbulence intensity near spark plug, but also can improve the characteristics of ignition and the distribution of mixture concentration and turbulence intensity in cylinder. This is beneficial to the improvement of flame propagation characteristics in late combustion.

3.3 The effect of combustion chamber shape on NO formation law

The effect of combustion chamber shape on NO formation law was shown as Fig. 11, According to the formation rule of high temperature oxygen–enriched no in Zeldovich, the VI combustion chamber has the fastest combustion speed and the highest exothermic rate. The high temperature oxygen–enriched zone is formed rapidly in the cylinder, and the mass fraction of no increases rapidly, and the mass fraction of no begins to decrease slowly after reaching the peak value at the time of 364CA. At the end of combustion, the mass fraction of no remained at the level of 4.8×10^{-3}.

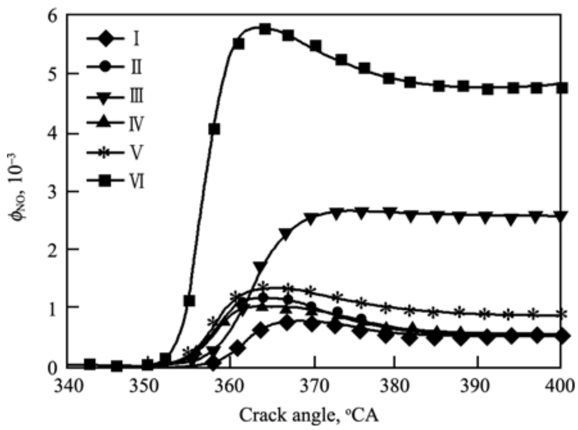

Figure 11. The effect of combustion chamber shape on NO fraction

The heat release rate of type Ⅲ combustor is smooth at the beginning of combustion, but the heat release rate is higher at the later stage of combustion, so the NO formation rate was higher, NO emission level is about half of that of type VI combustor, although the other combustors have different heat release rate curves, However the variation of no mass fraction was not different, and the total change trend was kept at a lower level.

Fig. 12 shows the variation characteristics of the mixture concentration field, temperature field and NO production rate in the combustion process of the three typical combustion chambers of III, IV and VI. From this the effects of different combustion chambers on the formation mechanism of NO were analyzed.

From which we can be seen that the amount of NO produced is not only related to the rate of NO formation, but also to the size of the rapid formation region of NO and the reaction time in the cylinder, in which the rate of no formation depends on the local temperature and the concentration of the mixture.

The VI combustion chamber produces more NO during combustion, which is not only due to the fast heat release rate and high NO formation rate, but also has a wide high temperature and oxygen–rich region. Therefore, by designing the throat and bottom shape of the III and IV type combustion chamber, the distribution characteristics of the mixture concentration field in the combustion chamber can be improved effectively. As a result, the size and duration of no production region are reduced, so the amount of no production is effectively controlled.

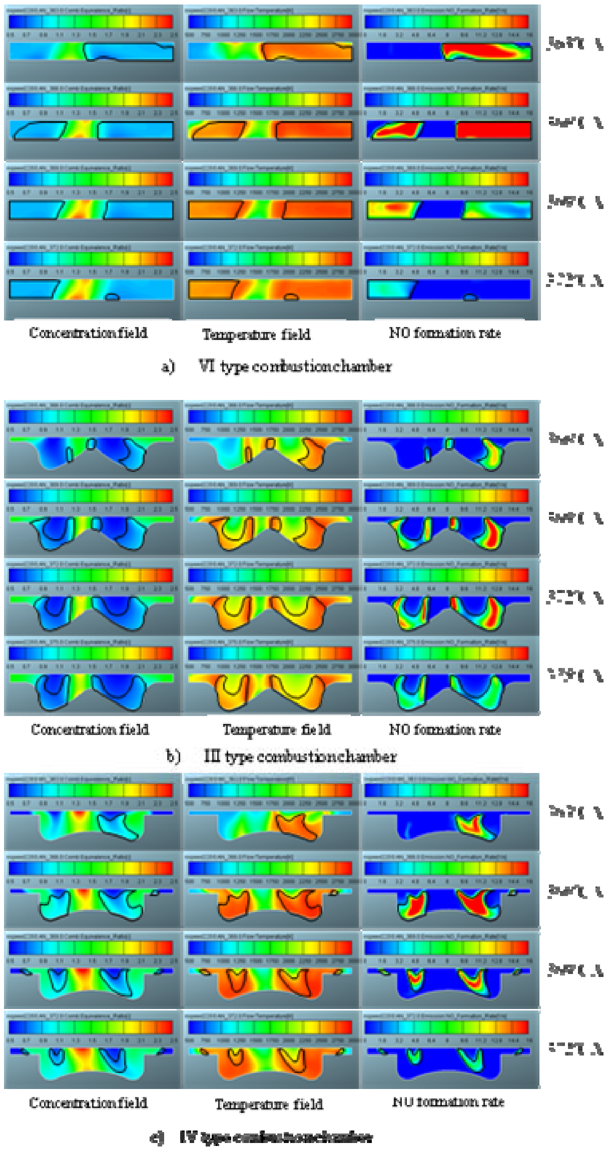

Figure 12. The effect of combustion chamber on formation condition

The combustion rate of type IV combustion chamber was relatively fast, especially in the high temperature region, the mixture concentration gradient is larger, so the local NO production rate is higher, but because of the smaller reaction area to form NO, the final NO emission level is lower.

Although the initial heat release rate and local combustion temperature of type Ⅲcombustion chamber are lower, the high temperature oxygen–enriched zone formed in the cylinder is wider. Therefore, although the formation rate of NO in the local region is lower, the reaction area formed is larger, resulting in the higher NO production.

4. Conclusion

a) The shape of the combustion chamber of CNG–DI engine has a great influence on the gas flow characteristics and the distribution characteristics of the concentration field, especially near the spark, which plays a decisive role in the ignition and flame propagation process in the later stage.

b) For the CNG–di engine, it is not appropriate to design the entrance shape of the combustion chamber into shrinkage and an opening type. When the flat piston was designed, the output of NO is the highest. The NO production can be effectively reduced when the combustion chamber structure is straight entrant and the bottom is raised(II and IV type). It can effectively control the velocity of flame propagation and is suitable for rarified combustion.

c) The formation rate of NO depends on the double conditions of temperature and oxygen concentration, and the final amount produced of NO depends on the formation rate of NO and the size and duration of the reaction region of NO formation. Reasonable design of combustion chamber structure is an important prerequisite for CNG–DI engine to improve NO emission characteristics.

References

1. Yanuandri Putrasari a, Achmad Praptijantoa, Arifin Nur, et al., 2015 Evaluation of performance and emission of SI engine fuelled with CNG at low and high load condition, Science Direct *Energy Procedia* 68, 147–156.

2. Musthafah Mohd. Tahira, M. S. Alia, M. A. Salim et al, 2015, Performance analysis of a spark ignition engine using compressed natural gas (CNG) as fuel, *Energy Procedia* 68, 355–362.

3. Mirko Baratta, 2012, Nicola Rapetto and Ezio Spessa, Numerical and Experimental Analysis of Mixture Formation and Performance in a Direct Injection CNG Engine, *SAE Paper* 01–0401, 2012.

4. 木戸口善行. 2010, 噴射制御による直接噴射式天然ガス内燃機関の希薄燃焼に関する研究[J]. 自動車技術会論文集, 41(4): 859–864.

5. Imran S., Emberson D. R., Diez A., et al. Natural gas fueled compression ignition engine performance and emissions maps with diesel and RME pilot fuels [J], *Applied Energy*, 2014, 124: 354–365.

6. 焦运景, 司鹏鹍, 杨志勇等, 2013, 气道形状对天然气发动机缸内气体流动与燃烧过程影响的研究[J].内燃机工程, 34(3): 26–31.

7. Chongmin Wu, Kangyao Deng, Zhen Wang, 2015, The effect of combustion chamber shape on cylinder flow and lean combustion process in a large bore spark–ignition CNG engine, *Journal of the Energy Institute*, xxx 1–8.