

Chapter 2

Study of the Effect of Combustion Chamber Shapes on the Mixture Formation and Combustion Characteristic on CNG-DI Engine

Tao Jiang, Xuedong Lin and Miao Yang

Abstract A combustion system model of CNG-DI engine which was modified from a 2.0 L diesel engine was built with software FIRE, the turbulence combustion model of CNG-DI engine was verified with the experimental result of CNG-DI optical engine, on the condition of unchanged 4 valves structure of original engine and the jet location, the spark plug was arranged between the two exhaust valves, the effect of different combustion chamber shapes on the microscopic physical fields such as turbulence characteristic in cylinder was simulated and studied, based on which the effect of combustion chamber shapes on the formation and combustion of mixture and the formation rule of NO was researched. The result shows that the combustion shape has a tremendous influence on the turbulence characteristic of cylinder and distribution characteristics of concentration fields especially besides the spark plug, which plays a decisive role on the whole process of combustion; Not only the production of NO can be inhibited and the combustion propagation speed which is in favour of lean burn can be controlled effectively when the straight mouth combustion chamber was used at the bottom of which has a proper embossment. The amount of NO is closely related to the size of the reaction area and the duration of reaction besides the reaction rate of NO when the CNG fuel is injected into cylinder directly and burned thin, which can be controlled by the rational design of the structure of combustion chamber.

Keywords CNG-DI engine · Lean burn · Injection mode · Mixture formation mechanism · Flame propagation characteristic · No_x formation rule

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The diversification of energy sources and lean combustion technology in cylinder has 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 CO₂ and abundant reserves [1]. The development of low-carbon has been effected directly by the loss of volumetric efficiency 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 the recent years, the combustion characteristic of CNG engine and the effect of flow characteristic on the combustion process of natural gas is thorough researched domestic and overseas [5–7]. But the report of the study the mixture formation mechanism and its influence factor is rare.

In this paper, the effect of different combustion chamber shape on the micro physical field of two gas flow in cylinder and its dynamic distribution characteristic was researched when the 2.0 L DI engine with 4 valves and 4 cylinders was instead with CNG-DI engine with software FIRE base on the optical research of flame propagation characteristic of lean combustion carried out in CNG-DI experimental prototype, based on which the mixture formation law, lean combustion characteristic and NO_x formation law of CNG-DI engine was studied.

2.1 Research Condition and Method

2.1.1 Research Condition of Optical Engine

The main technically parameter of single-cylinder optical prototype used for model validation is shown as Table 2.1. The arrangement diagram of optical test system for the observation of flame propagation characteristic in optical prototype is shown as Fig. 2.1. The arrangement diagram of spark plug and injector is shown as Fig. 2.2, two injector and two spark plug was arranged in the cylinder head; The air intake and exhaust system was arranged in the side of cylinder head for the

Table 2.1 Technical parameters of experimental prototype

Type	Four
Cylinder diameter/mm	135
Stroke/mm	280
Displacement/L	4.0
Compression ratio	6.13

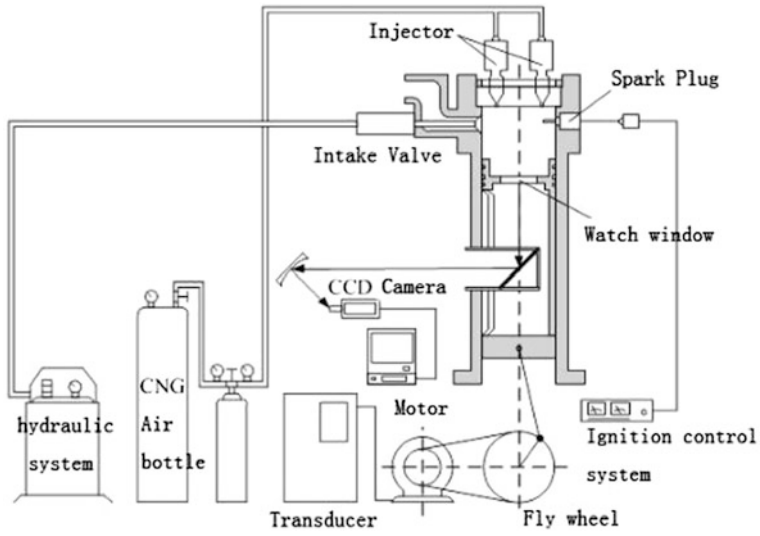


Fig. 2.1 The diagram of experiment facilities

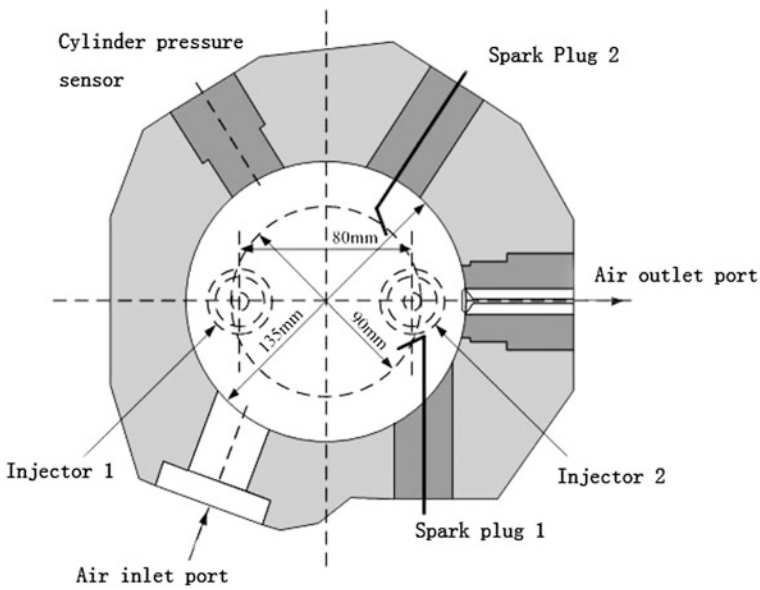


Fig. 2.2 The diagram of the location of spark plug and injector

Table 2.2 The main composition of natural gas fuel

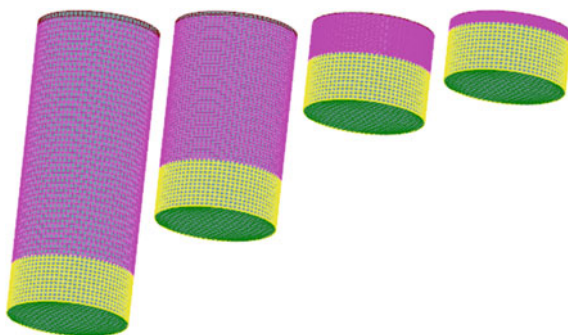
	CH ₄	C ₂ H ₆	C ₃ H ₈	C ₄ H ₁₀	N ₂
vol%	85.45	4.51	3.39	3.71	2.94

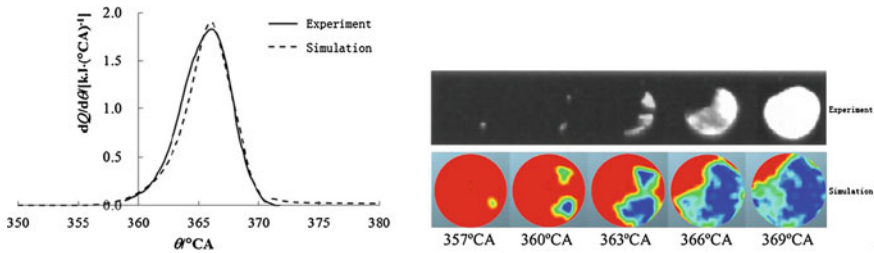
convenience of adjustment of intake eddy, which caused the large of clearance volume, so the combustion ratio was 6.13; The engine rotate speed was 200 r/min which was limited by the characteristic of prototype structural features, but the fundamental research of flame propagation mechanism would not be influenced. The main composition of natural gas was shown as Table 2.2.

The intake vortex intensity was adjusted by the offset of intake axis relative to the center line of cylinder. The initial air flow intensity was improved as high as possible by the adjustment of intake swirl ratio to $sw = 6.0$ according to the low engine rotate speed. The swirl injector was adopted in the experiment, the injection pressure was 5 MPa. The flame propagation process was continuous recorded in 563 picture/s with CCD high-speed camera. The made up indicated diagram was the average result of cylinder pressure measured 10 times in every working condition.

2.1.2 The Simulation Condition and Validation

The flame propagation characteristic and indicator diagram was measured with CNG optical engine to validate the cylinder turbulence and combustion model of CNG-DI engine, then a cylindrical combustion system model was generated directly without the change of compression ratio with software FIRE based on the structure characteristic of optical engine to bring combustion chamber shape into correspondence with prototype. The duration from intake stroke BDC to power stroke BDC was divided into four layers mesh based on the variation law of cylinder volume during the working process of engine was shown as Fig. 2.3.

Fig. 2.3 Simulation mesh



a)The comparison of heat release rate b)The comparison of flame propagation

Fig. 2.4 The comparison of experiment and simulation result

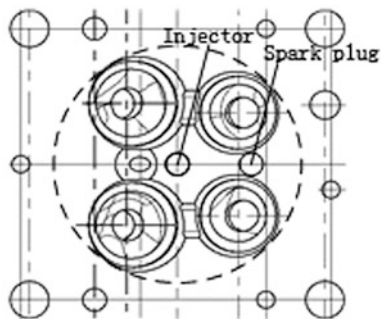
The 0.5°CA was adopted during the injection and combustion period and 1°CA was adopted during the expansion period for simulation time step; The $k-\zeta-f$ model was adopted as turbulence model, the ECFM model was adopted as combustion model and O'Rourke model was adopted as turbulence diffusion model based on the characteristic of CNG gas fuel.

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 350 K; The top of piston was supposed to be the moving boundary and the temperature of which was 593 K, the cylinder wall and the bottom if cylinder head was supposed to be the fixed boundary, the temperature of cylinder wall was 403 K and the temperature of the bottom of cylinder head was 593 K. 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 $(\theta_{i1}, \theta_{i2}) = (-4, -3)^{\circ}\text{CA}$, two injector and injection timing was $\theta_{inj} = -120^{\circ}\text{CA}$ was shown as Fig. 2.4. 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.

2.1.3 The Technically Condition and Research Program of Prototype

The basic structure of prototype remains unchanged when the 2.0 L DI diesel engine was instead with CNG-Di engine, only the compression ratio was changed from 17.2 to 12 and the improved design was proceed in combustion chamber shape. The arrangement of injector and spark plug was shown as Fig. 2.5 based on the characteristic of 4 air valve structure of original engine. The 6 different kinds of combustion chamber shape was designed aimed at the scheme of injector in center and spark plug in exhaust valve. Among them, the structure of I type combustion

Fig. 2.5 The diagram of injector and spark plug



chamber keep 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 dispark 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 structure model, simulation method and simulation model of simulation prototype was in accordance with the optical prototype, while the simulation boundary condition was confirmed by the measured result during the experimental study of 2.0 L diesel engine. The main technical parameter and simulation condition of simulation prototype was shown in Table 2.3. The maximum torque engine speed working condition was selected as simulation working condition, the cycle injection quantity was supposed to be 34.77 mg, the injection timing was supposed to be 158°CA BTDC and the ignition timing was 6°CA BTDC (Fig. 2.6).

2.2 Simulation Result and Analysis

2.2.1 *The Effect of Combustion Chamber on Turbulence Characteristic in Cylinder*

The effect of different combustion chamber on turbulence characteristic in cylinder when the injection timing was 158°CA BTDC and injection ending timing was 10°CA BTDC of CNG fuel was shown as Fig. 2.7; 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

Table 2.3 The technical parameter and simulation condition

<i>The technical parameter</i>	
Type	2.0L 4 cylinder CNG-DI engine
Cylinder bore × stroke	83 × 92/mm
Compression ratio	17.2
Rated power	100 kW/(3800 r/min)
The maximum torque	310 Nm/(2200 r/min)
<i>The simulation working condition</i>	
Rotate speed	2200 r/min
Jet mass	34.8 mg
Injection timing	158°CA BTDC
Injection duration	148°CA
Ignition timing	6°CA BTDC
<i>The simulation condition</i>	
Pressure in the entrance	1.08 bar
Induction swirl ratio	2
The initial TKE	33.443 m ² /s ²
Temperature in piston top	553 K
Cylinder wall	403 K
Cylinder head	403 K

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. 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. The degree of attenuation of average TKE was different due to the different air flow state in different combustion chamber during the middle stage of compression stroke (270–320°CA). The distribution of cylinder air flow state in different combustion chamber at 300°CA was shown in Fig. 2.8. The turbulence intensity decreased slowly due to the entirely tumble of air flow in the top of piston and air flow in the III and VI type combustion chamber, while the air flow motion state was reverse in other type of combustion chamber(MIVV), so the turbulence intensity decreased quickly. 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 value of turbulence I type combustion chamber with lug boss at the bottom of throat at the early stage of compression stroke was the highest but was weakest at end of the compression process, while the turbulence intensity of I type combustion remained the same level with V type combustion chamber with Straight mouth flat at the end of compression process; The degree of attenuation of turbulence intensity could be improved at the end of compression

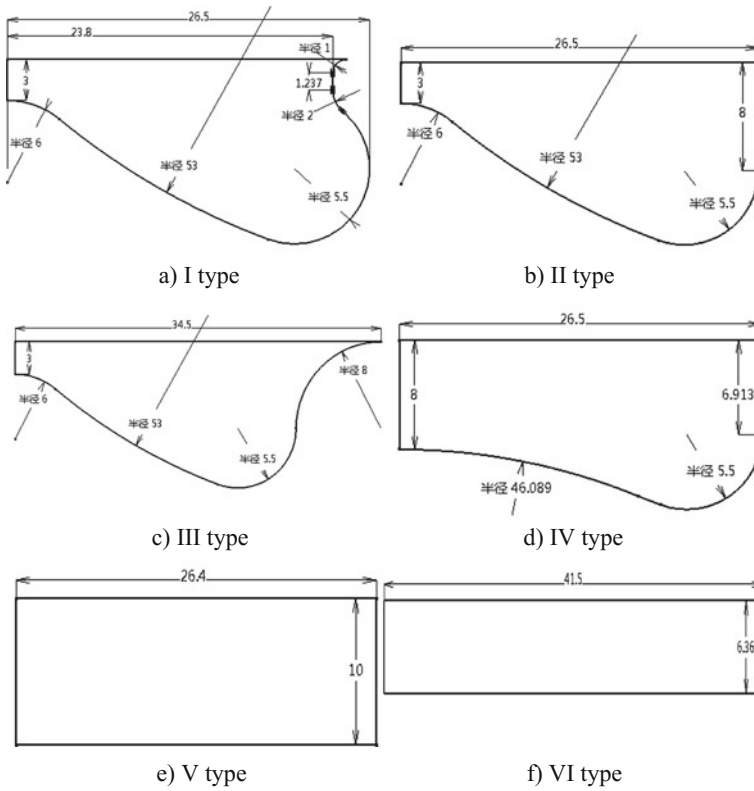
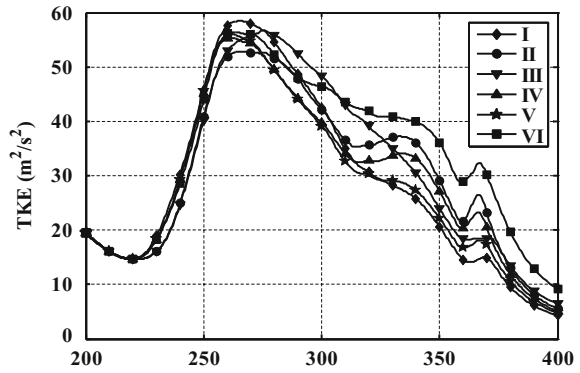


Fig. 2.6 The different combustion chamber shape

Fig. 2.7 The effect of combustion chamber on TKE



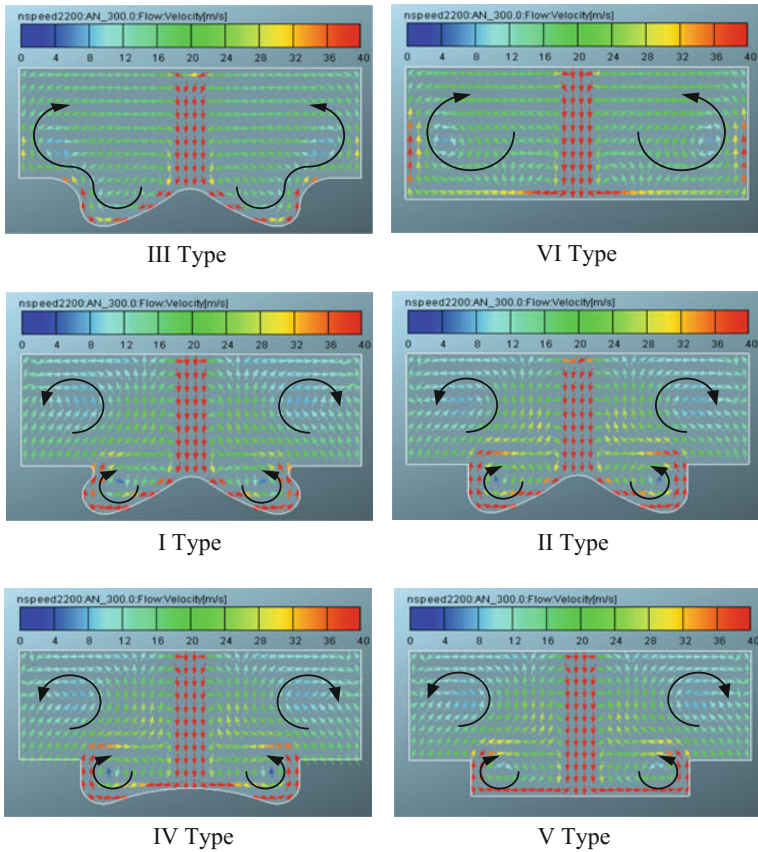


Fig. 2.8 The air flow distribution characteristic at 300°C

stroke when the straight mouth combustion chamber changed from a little heave one in bottom to an open one. The turbulence TKE in cylinder reach the second peak due to the flame propagation after ignition point at 6°CA BTDC.

2.2.2 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

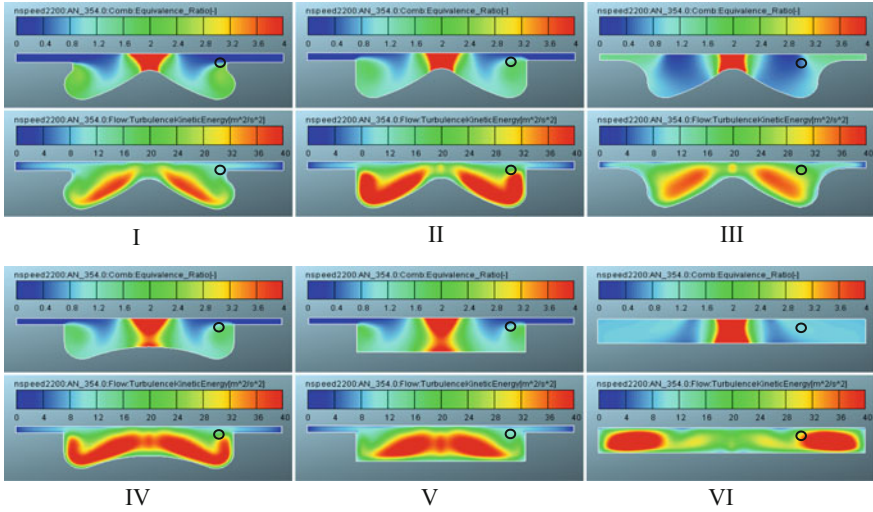


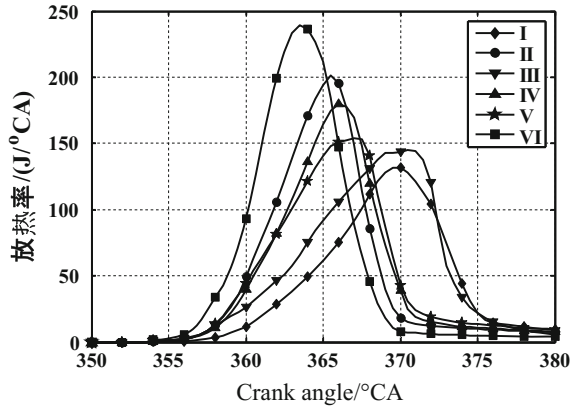
Fig. 2.9 The distribution characteristic of concentration field (up) and TKED (down) at 354°C

of special importance to lean combustion process. 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. 2.9, 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 distribution characteristic of TKE, especially near the spark plug. The mixture concentration and turbulence intensity near the spark plug at ignition timing was shown as Table 2.4. From which we can know the mixture concentration near the I type and III type combustion chamber near the spark plug was thinner which was not easy to be ignited. The mixture concentration in V and VI type 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. 2.10; 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; The formation of flame nucleus was slow, flame propagation speed was low and heat release rate was

Table 2.4 The equivalence ratio and TKE at spark plug at 354°C

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

Fig. 2.10 The effect of combustion chamber shape on heat release rate

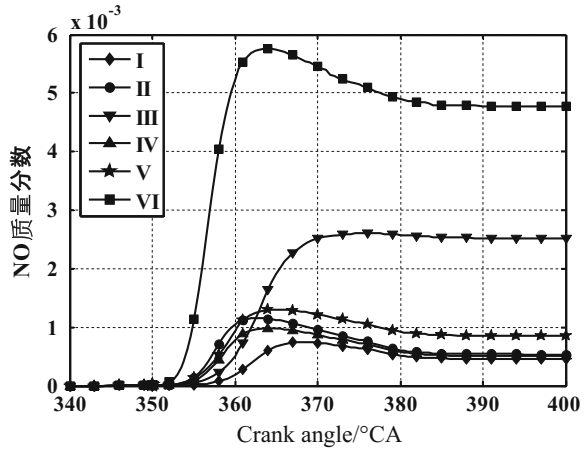


tardiness because the mixture concentration was as lean as the one in I type combustion chamber and not fit for the ignition although the TKE around the spark plug was more powerful in III type combustion chamber. The equivalence ratio and TKE near the spark plug around the ignition timing in II type and IV type combustion chamber was 1.37 and $21 \text{ m}^2/\text{s}^2$ approximately, whose combustion rate could be controlled effectively in spite of its slow combustion rate relative to VI type combustion chamber. The research result shows that not only the mixture concentration and turbulence intensity near the spark plug could be controlled effectively which is convenient for the improving of ignition characteristic but the one in cylinder could be improved which is convenient for the later flame propagation characteristic by the rational designation of throat of combustion chamber and shape of lug boss in bottom.

2.2.3 The Effect of Combustion Chamber Shape on NO Formation Law

The combustion rate of VI type combustion chamber was fastest and its heat release rate was the highest based on the high temperature and oxygen enrichment NO formation law of Zeldovich, so the mass fraction of NO formation increased quickly due to the high temperature and oxygen enrichment zone formed in cylinder, which decreased slowly after the reaching the peak at 364°CA , the NO mass fraction keep the level of 4.8×10^{-3} at the end of the combustion; The heat release rate in III type combustion at the later period of the combustion was higher while at the early stage of the combustion was flat, and the NO production was higher due the lesser mixture concentration distribution gradient in cylinder, but the appearance moment of the peak value of NO production was late relatively, whose NO emission level was about the half of the VI type combustion chamber; The dynamic variation characteristic of NO mass fraction had little difference in spite of the different heat

Fig. 2.11 The effect of combustion chamber shape on NO fraction



release rate curves of other combustion chambers, and the total variation trend keep lower level. The variation characteristic of mixture concentration field, temperature field and NO formation rate during the combustion process in the representative combustion chamber such as VI, III and IV type combustion chamber was shown as Fig. 2.12 to investigate the effect of different combustion chamber on the NO formation mechanism. From which we can know that NO production was related to not only the NO production rate but the size of NO fast production zone and duration of reaction in cylinder. The NO production rate depends on the local temperature and mixture concentration. The NO production was rich due to not only fast heat release rate and high NO formation rate but the wide high temperature and oxygen enrichment zone in VI type combustion chamber during the combustion process; The distribution characteristic of mixture concentration field in combustion chamber could be effectively improved by the designation of combustion chamber throat and bottom shape in III type and IV type combustion chamber, which can effectively control the NO production by reducing the size of zone and duration of NO formation in different degrees. The combustion rate in IV type combustion chamber was fast especially which caused the high temperature and larger mixture concentration gradient in high temperature zone, so the size of the formed NO reaction zone was smaller in spite of the high NO formation rate in local area, which made the final NO emission level lower; The high temperature and oxygen enrichment zone formed in cylinder was wide in spite of low heat release rate and local combustion temperature in III type combustion chamber at the early stage, caused the higher final NO production due to the larger formed reaction zone area in spite of low NO formation rate (Fig. 2.11).

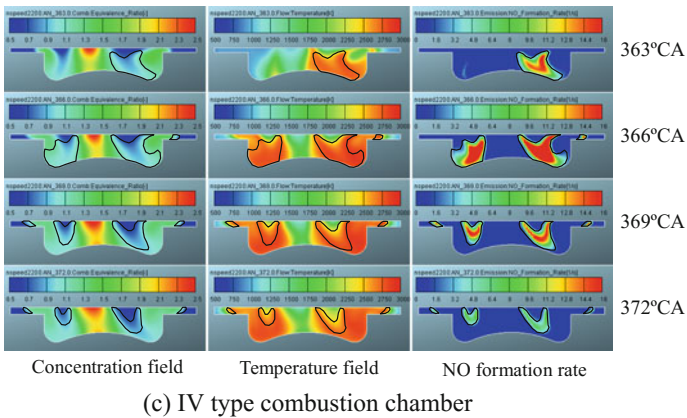
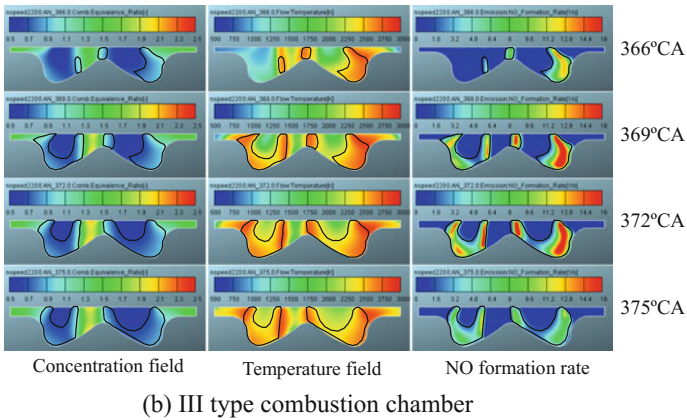
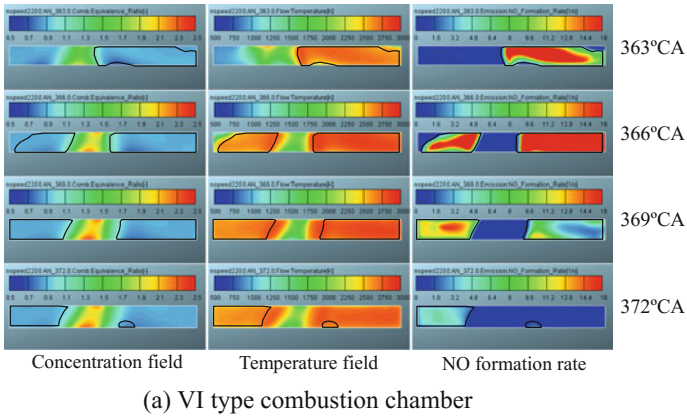


Fig. 2.12 The effect of combustion chamber on formation condition

2.3 Conclusion

- (1) The distribution characteristic of air flow characteristic and concentration field was effected by the combustion chamber of CNG-DI engine, especially near the spark which plays a decisive role on the ignition and flame propagation process in the later period.
- (2) The throat and open mouth combustion chamber is not fit for CNG-DI engine; The NO production is the highest when the flat piston is adopted; Not only the NO production can be reduced effectively but the flame propagation speed can be effectively controlled when the straight port and raised combustion chamber was adopted, which is fit for the organization of lean combustion.
- (3) The NO formation rate depends on the dual condition of temperature and oxygen concentration when the CNG-DI engine is adopted, while the NO final production depends on the NO formation rate, size of reaction zone of NO formation and reaction duration, which can be improved by the rational designation of combustion chamber structure.

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