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Research on Dual Fuel Combustion Characteristics of PFI Natural Gas +DI Gasoline in Large Cylinder Diameter Engine

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Abstract

A 3D simulation model of a single cylinder of a natural gas engine was established. PFI natural gas DI gasoline was used to control the time matching between injection and ignition, so that gasoline could form a small active stratified area near the spark plug. The combustion characteristics under different matching conditions were studied, and finally compared with pure natural gas. It is found that gasoline can be used in large cylinder diameter natural gas engine in a small range, which can accelerate the combustion of natural gas fuel, shorten the time used by CA0-CA5 by 77.6%, shorten the time used by CA5-CA90 by 26.5%, and increase the indicated thermal efficiency of natural gas engine by 1.4%.

Keywords

Dual Fuel; Active Stratification; Direct Injection Gasoline.

1. Introduction

A Natural gas, as the cleanest fossil energy, is a representative of low-carbon clean energy and has attracted much attention in the current energy industry. However, the main component of natural gas is methane(CH4), which is a very stable saturated hydrocarbon, and requires large ignition energy when burning in the cylinder, and the flame spreads slowly. Especially in the use of thin natural gas as fuel in large cylinder diameter engines, it will face the problems of non-ignition, large cycle changes in the process of thin combustion, and high thermal load, resulting in low efficiency of the engine, and sometimes even knock, which will cause great damage to the engine. In order to solve the above problems, many scholars have done extensive research[1-5]. In the above many methods, adding flammable fuel to the natural gas engine to form stratified combustion can effectively improve the low concentration of natural gas ignition difficulties, slow combustion problems.

Hydrogen has the advantages of low minimum ignition energy, fast combustion speed and wide flammability limit. Natural gas, coal bed gas and hydrogen are all gas fuels, which are easy to be premixed and blended. Mixing hydrogen in a fixed proportion can improve the ignition stability of the engine and reduce the cycle variation of the engine. Hydrogen mixing can also accelerate the flame propagation speed, improve the combustion efficiency of the engine and reduce the generation of harmful emissions such as CO and HC[6]. Hydrogen mixing accelerates flame propagation and improves flame quenching resistance. Both the thin burn limit and EGR tolerance of the engine increase with the increase of hydrogen mixing ratio. Hydrogen is also an excellent antiknock agent, and hydrogen mixing of gas can effectively reduce the knocking tendency of the engine[7]. Biffiger and Soltic[8] compared the combustion characteristics of combined injection of methane/hydrogen PFI, methane PFI+ a little hydrogen DI, and methane PFI+ a little methane DI in reciprocating piston internal combustion engines, and found that PFI+DI stratification can improve the combustion efficiency of the engine.

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Direct injection of diesel fuel in the cylinder can also ensure the stable operation of the diesel ignition natural gas engine under all working conditions, and can achieve high thermal efficiency and low emission targets [9-10]. Fasching et al.[11] conducted a study on an in-cylinder direct injection diesel piloted natural gas engine with Euro VI emission level through experimental means. The results show that the engine can operate steadily at medium to high loads, while significantly reducing HC emissions, which decreased by 91% to 2.2 g/kw·h at 1750 RPM (r/min) and an average effective pressure of 5bar. Huang et al.[12] found through experimental research that, within a certain range of crankshaft Angle, increasing diesel injection pressure or advancing diesel injection timing can improve the thermal efficiency of diesel-ignited natural gas engines and reduce CO2 emissions.

Zhan et al.[13] studied combustion and emissions from dual-fuel engines fueled by natural gas/methanol, natural gas/ethanol, and natural gas/n-butanol. The results show that the combustion rate of natural gas can be accelerated by adding three alcohol fuels, and the thermal efficiency of natural gas can be significantly improved by adding methanol. Roberto et al.[14] studied and analyzed the natural gas-ethanol dual-fuel spark ignition engine, and found that compared with pure natural gas, dual-fuel mode improved fuel conversion efficiency and better combustion effect, but dual-fuel would increase NOx emission.

To sum up, most fuels mixed in natural gas engines are hydrogen, diesel and alcohol fuels, etc. For gasoline, gasoline is also more flammable than natural gas, but it is prone to knock when used in large cylinder diameter engines, so few studies have been done when it is used as blended fuel. In this paper, PFI natural gas DI gasoline is proposed. A small amount of gasoline is injected directly to form an active stratification area in a large-bore natural gas engine. The influence of different injection and ignition matching on combustion characteristics of the natural gas engine is studied to realize the small-range application of gasoline in a large-bore natural gas engine.

2. Model and Methods

Table 1. Main performance parameters of the engine

Project	Parameter
Engine style	In-line, water-cooled, four-stroke
Number of cylinders × diameter × stroke	6×160mm×240mm
Compression ratio	10.5: 1
Rated speed	1000r/min
Rated power	326kW

In this paper, a diesel engine transformed into a natural gas engine model is adopted, which is simulated and calculated in CONVERGE. The specific parameters of the engine and the simulation scheme are shown in Table 1. The selected 3D calculation model is shown in Table 2. The 3D simulation model is shown in Figure 1.

Table 2. Three-dimensional computational model

Model type	Model name
Combustion model	SAGE
Turbulence model	RNG κ-ε
Spray crushing model	KH-RT
Droplet collision model	NTC
Droplet evaporation model	Frossling
Turbulent dissipation model	O'Rourke

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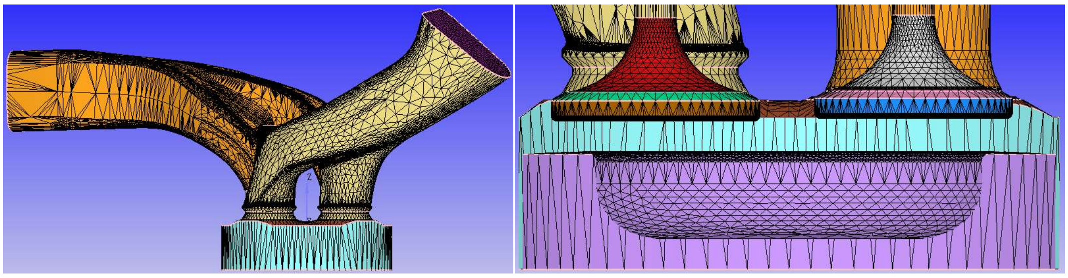

Figure 1. The 3D simulation model

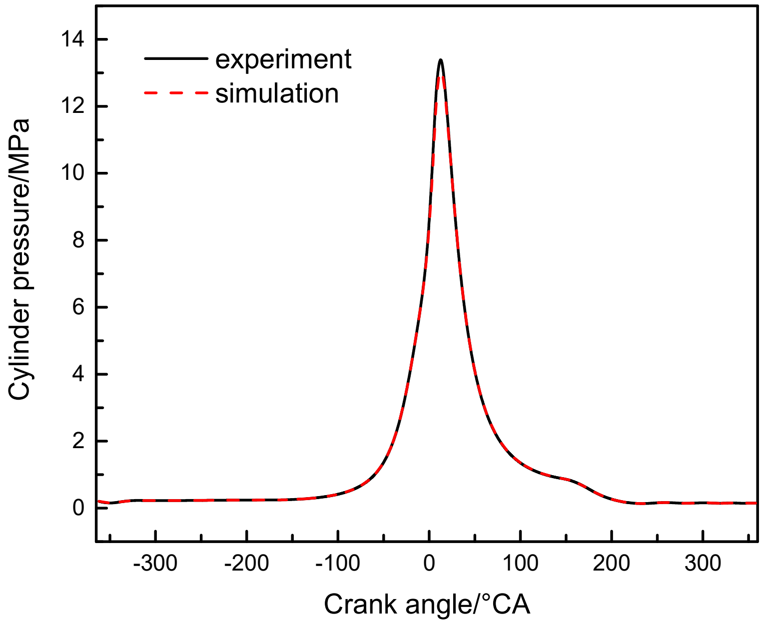

Figure 2. Comparison of cylinder pressure between experiment and simulation

Table 3. Simulation condition

Project	Parameter
Excess air coefficient	1.7
Injection pressure	35MPa
Injection pulse width	6°CA
Injection time	-20°CABTDC
Ignition time	-16°CA, -14°CA, -12°CA, -10°CA, -8°CA

In this paper, 0°CA was defined as the top dead center of compression. In order to verify the rationality of the model, the engine combustion condition was simulated under full load condition when the excess air coefficient was 1.6 and the ignition time was -30°CA. Figure. 2 shows the comparison between the simulated cylinder pressure change curve and the test value. The simulation scheme of this paper is shown in Table 3.

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3. Results and Discussion

3.1. Research on Combustion Characteristics at Different Ignition Times

3.1.1. Thermal Efficiency Analysis

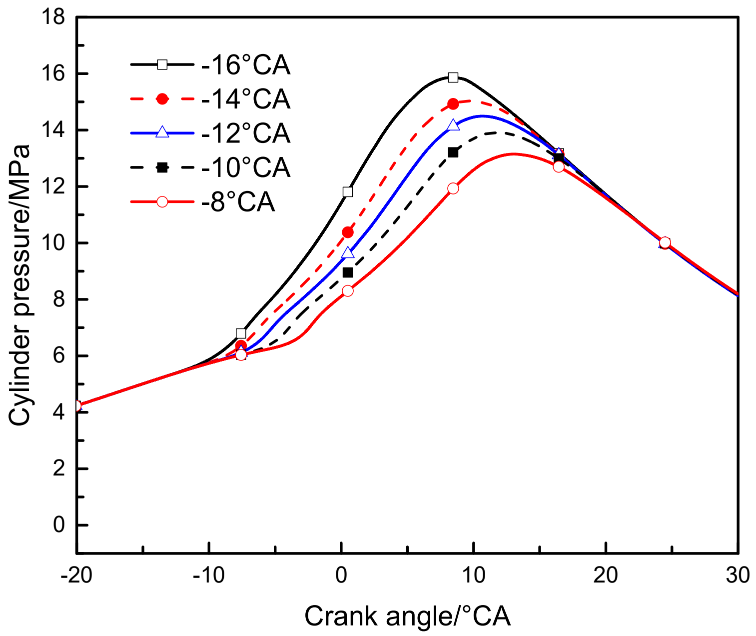

Figure 3. Cylinder pressure at different ignition times under active stratification

The injection time set in the simulation in this section is -20°CABTDC, and the injection time is not finished when the ignition time is -16°CABTDC. At other ignition times, the direct injection of gasoline in the cylinder has ended, and with the delay of the ignition time, the gasoline spray moves with the air flow to form a more uniform mixture in the cylinder and a larger active stratification area. Figure 3 shows the cylinder pressure curves corresponding to different ignition times under the active stratification of the mixture in the cylinder. The maximum cylinder pressure decreases with the delay of the ignition time, and the crankshaft Angle corresponding to the peak cylinder pressure increases with the delay of the ignition time, but the peak pressure is distributed around -10°CABTDC.

Figure. 4 shows the curves of instantaneous heat release rates corresponding to different ignition times under the active stratification of the mixture in the cylinder. As can be seen from the figure, the instantaneous heat release rates corresponding to different ignition times all rise and then decline with the passage of time to form a small peak with a small peak instantaneous heat release rate, and then the instantaneous heat release rate rises with the passage of time to form a large peak with a large peak instantaneous heat release rate. The overall performance is a small instantaneous heat release rate curve with a large double instantaneous heat release rate peak. The reason for this phenomenon is that the first peak of the instantaneous heat release rate is caused by the active stratified region formed by gasoline around the spark plug. Compared with methane, gasoline is easier to ignite and burns faster. In the early stage of combustion in the cylinder, the relatively dense mixture with a large amount of gasoline vapor distribution near the spark plug starts to burn as the main fuel. The instantaneous heat release rate rises faster. However, only a small amount of gasoline was injected into the cylinder, forming an active stratification in a small area. Over time, when most of the gasoline fuel was

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consumed, the main fuel methane in the cylinder began to burn, but because of the low concentration of methane in the cylinder, the amount of methane involved in combustion was small at the beginning, and the combustion speed was slow, resulting in a decrease in the instantaneous heat release rate and the formation of the first instantaneous heat release rate peak. The appearance of the second peak instantaneous heat release rate is because after most of the gasoline in the cylinder is consumed, methane as the main fuel participates in the combustion in a small amount from the beginning, with the expansion of the flame front, more methane begins to burn, resulting in a rapid increase in instantaneous heat release rate. When the amount of methane in the cylinder decreases after a period of time, the instantaneous heat release rate decreases, forming the second instantaneous heat release rate peak.

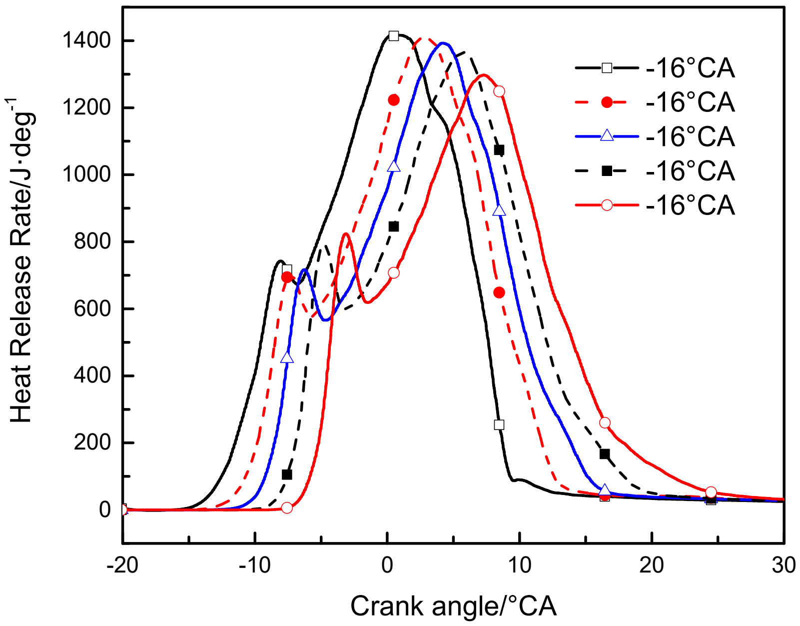

Figure 4. Instantaneous heat release rate at different ignition times under active stratification

It can also be seen from Figure 4 that the peak value of the large peak decreases with the delay of the ignition time, while the peak value of the instantaneous heat release rate of the small peak decreases first and then increases with the delay of the ignition time. The change of the peak value of the large peak with the ignition time is a phenomenon caused by normal combustion, while the peak value of the -16°CABTDC ignition time in the small peak is higher than that of the -14°CABTDC ignition time. This is because when the ignition time is -16°CABTDC, the gasoline starts to burn before being sprayed into the cylinder. Finally, a large amount of gasoline fuel burns rapidly in a small range, which also causes the combustion of surrounding methane earlier. When most gasoline is burned, the amount of methane involved in combustion also increases, so the small peak value is high. When the ignition time is -14°CABTDC, the gasoline injection has been completed. Compared with the ignition of -16°CABTDC, the gasoline distribution is relatively wide, the gasoline consumption is less in the same time, and the moment when methane officially starts to burn is also relatively late. Therefore, the instantaneous heat release rate peak value is relatively small, forming a phenomenon that the peak value of the instantaneous heat release rate falls first.

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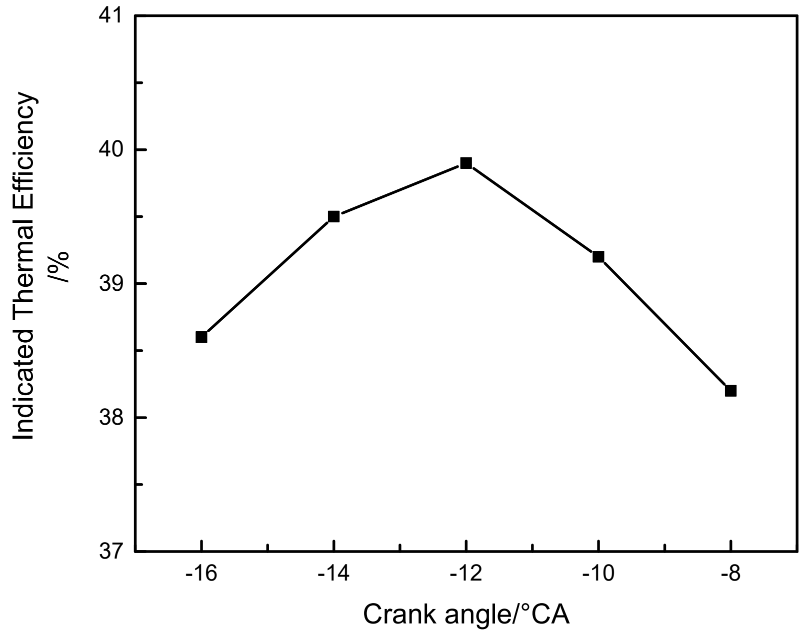

Figure 5. The efficiency is indicated at different ignition times under the active layering pattern

When the ignition time is delayed from -14°CABTDC to -8°CABTDC, the peak of the instantaneous heat release rate of the small peak increases with the delay of the ignition time. This is because after the gasoline injection ends, the mixture distribution in the cylinder becomes more uniform with the passage of time, and the active stratification area of the uniform mixture suitable for rapid combustion expands. As the ignition time is delayed, more gasoline/natural gas mixture is involved in the combustion at the initial stage of combustion, so that the peak instantaneous heat release rate of the small peak increases with the delay of the ignition time.

Figure 5 shows the indicated thermal efficiency under different matching conditions of fuel injection and ignition under active stratification in the cylinder, in which the highest indicated thermal efficiency is 39.9%, and the corresponding ignition time is -12°CABTDC, with an interval of 8°CA between the time of fuel injection.

3.1.2. Analysis of Combustion Duration and Flame Propagation

Figure 6 shows the cumulative heat release corresponding to different ignition times under the active stratification state of the mixture in the cylinder, and the overall law is normal. The figure in Table 4 is calculated according to the corresponding data in Figure 6, where CA5 is defined as the starting point of combustion, indicating the crankshaft Angle corresponding to the cumulative heat discharge of 5% of the total heat discharge; CA90 is defined as the combustion end point, indicating that the cumulative heat release accounts for 90% of the total heat release. It can be seen from the table that the time from ignition to CA5 gradually decreases with the delay of ignition Angle, while the combustion duration from CA5 to CA90 lengthens. The main reason for the shortening of the time from ignition to combustion starting point CA5 is that the mixture in the cylinder forms a more uniform and larger active stratified region near the spark plug with the development of time, which makes the flame propagate faster and the instantaneous heat release rate rise faster in the early stage of combustion. As a result, the time from ignition in the cylinder to the starting point of combustion is shortened with the delay of

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ignition time. The gradual extension of the time used in the combustion duration CA5-CA90 is because with the delay of the ignition time, the temperature and pressure in the cylinder are relatively low, the maximum peak value of the instantaneous heat release rate decreases, and the overall instantaneous heat release rate of the combustion duration decreases. Therefore, the combustion duration CA5-CA90 is extended with the delay of the ignition time.

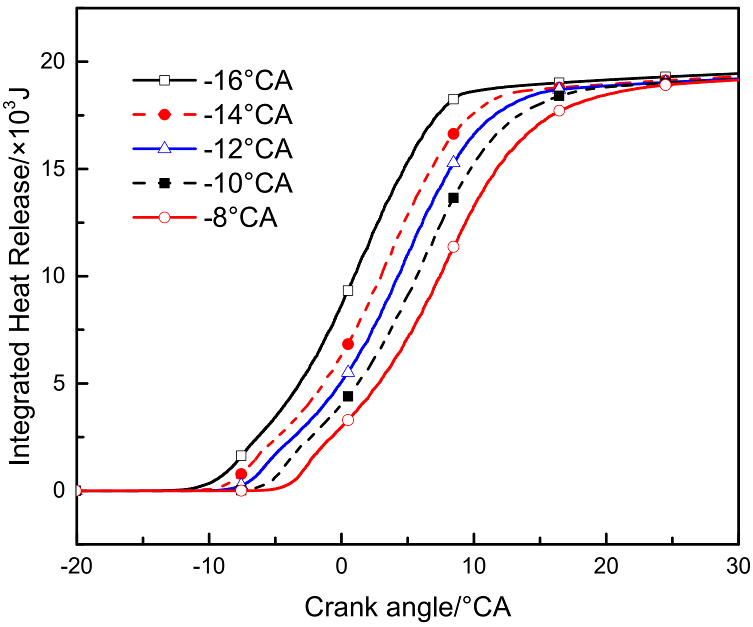
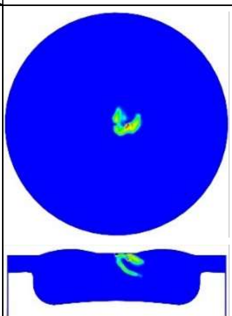
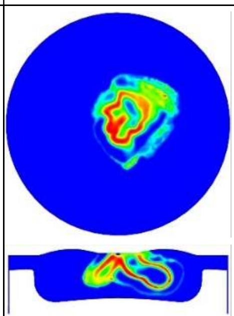
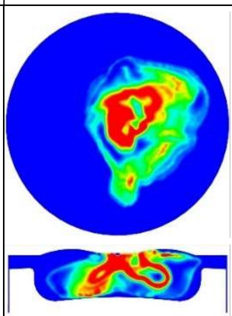
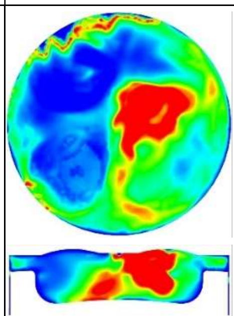


Figure 6. Cumulative heat release at different ignition times under active stratification

Table 4. Combustion duration at different ignition times under active stratification

	CA5 (°CABTDC)	CA0-CA5(°CA)	CA90 (°CAATDC)	CA5-CA90(°CA)
-16°CA	-8	8	8.3	16.3
-14°CA	-6.7	7.3	11.0	17.7
-12°CA	-5.5	6.5	12.8	18.3
-10°CA	-4	6	14.8	18.8
-8°CA	-2.3	5.7	17.4	19.7

	5°CA	10°CA	15°CA	25°CA
-16°CABTDC				

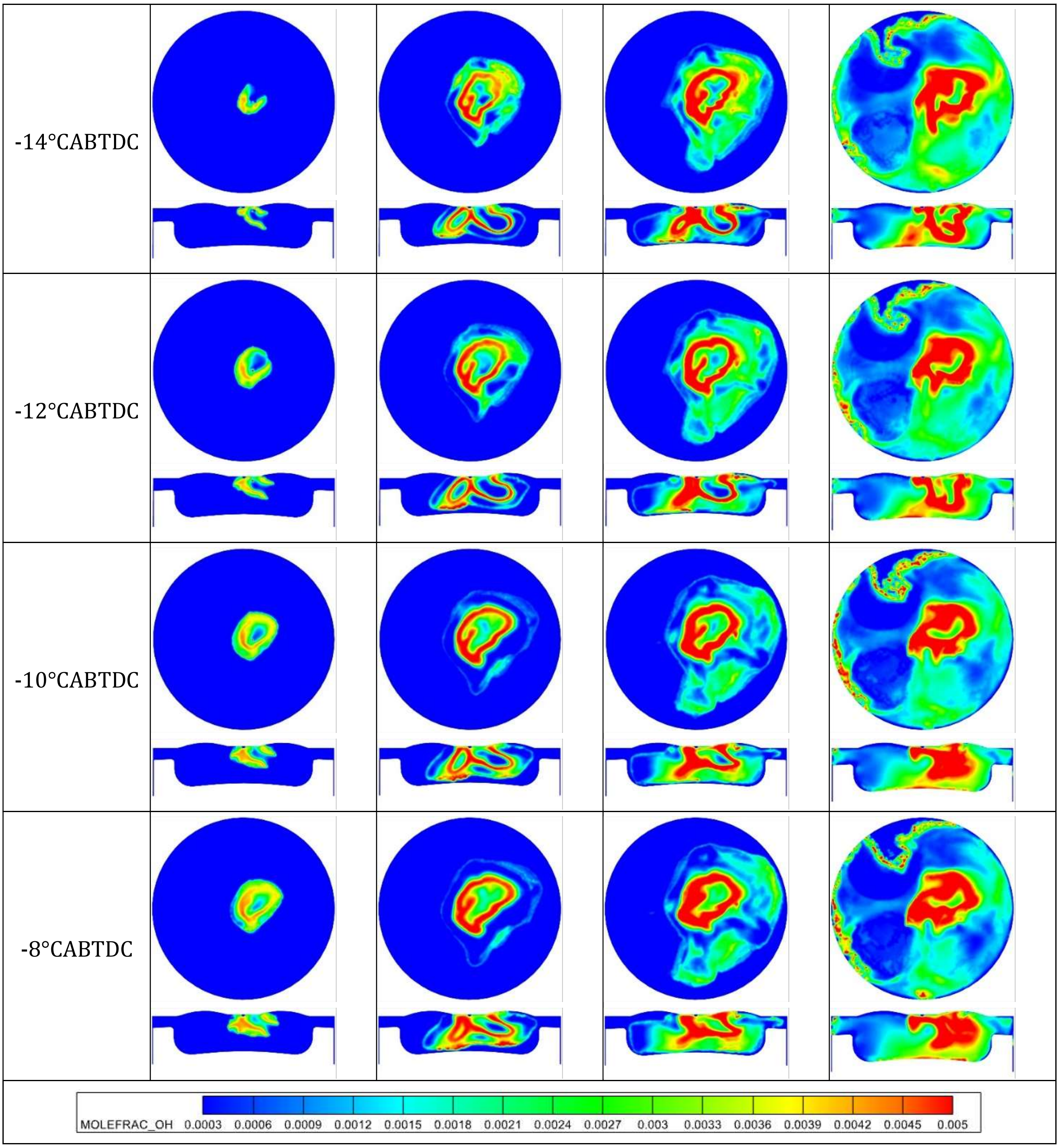


Figure 7. Cloud image of flame propagation in the cylinder at different ignition times under the active stratification pattern

Figure 7 shows the flame propagation cloud image at different ignition times under the active stratification state in the cylinder. The ordinate represents the ignition time, and the abscissa represents the time interval between the crankshaft Angle where the cross section cloud image is located and the ignition time. As can be seen from the figure, at the same ignition time, the flame propagation is mainly carried out along the penetration path of gasoline spray at the beginning, and spreads from the spray path to the surrounding areas at the same time. Compared with different ignition times, when the time interval is 5°CA, it is obvious that with the delay of ignition time, the flame propagation speed is faster in the initial combustion stage, which is due to the influence of gasoline spray in the cylinder airflow movement over time to form a more uniform and more widely distributed active stratification region of the mixture.

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From the flame propagation cloud images of the three ignition times of -16°CABTDC, -14°CABTDC and -12°CABTDC, it can also be seen that in the early stage of combustion, the flame is mainly wrapped with a stronger mixture. It can be seen from the flame propagation cloud image of different ignition advance angles at time intervals of 5°CA, 10°CA and 15°CA that the flame propagation speed is accelerated with the delay of ignition advance Angle in a certain flame propagation time. It can be seen from the flame propagation cloud image of different ignition advance angles at the time interval of 25°CA that the final combustion duration is extended with the delay of ignition advance Angle. The above flame propagation phenomena are consistent with the numerical changes shown by CA0-CA5 and CA5-CA90 in Table 4.

3.2. Influence of GDI on Combustion Characteristics of Natural Gas Engine

3.2.1. Thermal Efficiency Analysis

In the previous section, the combustion characteristics of the gasoline mixture in the active stratified form after direct injection in the cylinder were studied. This section mainly compares the ignition conditions of -12°CABTDC under active stratification conditions with the combustion characteristics of pure natural gas according to the research results in the previous section, and explores the influence of active stratification on the combustion characteristics of low-concentration natural gas engines. The operating conditions of the simulation for pure natural gas in this section remain the same as before. After calculation, when the excess air coefficient is 1.7, the optimal ignition advance Angle of pure natural gas is -34°CABTDC. The comparison of various pure natural gas combustion curves and cloud images in this section are all fired at -34°CABTDC.

According to the comparison of cylinder pressure changes and instantaneous heat release rate between active combustion and pure natural gas combustion in Figure 8 and Figure 9, it can be seen that the rise rate of instantaneous heat release rate is increased by active stratification, and the corresponding peak value of instantaneous heat release rate is also greatly increased. According to the calculation results in the first two sections, the indicated thermal efficiency of in-cylinder combustion in the active stratified form is 39.9%, and the indicated thermal efficiency of pure natural gas combustion in this section is 38.5% when the ignition advance Angle is -34°CABTDC. The indicated thermal efficiency of active stratified combustion was increased by 1.4%.

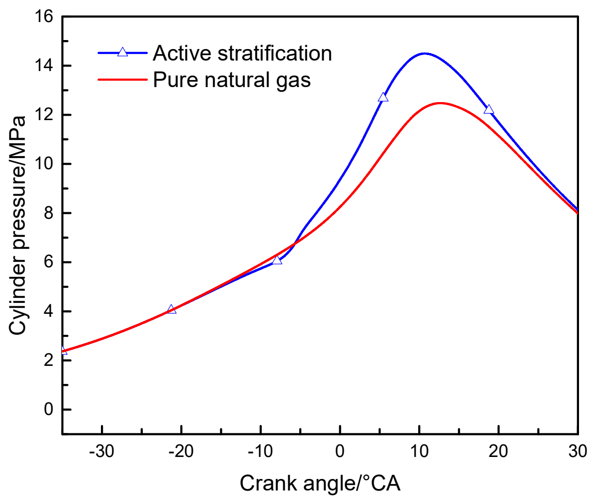

Figure 8. Comparison of cylinder pressure between active stratified combustion and pure natural gas combustion

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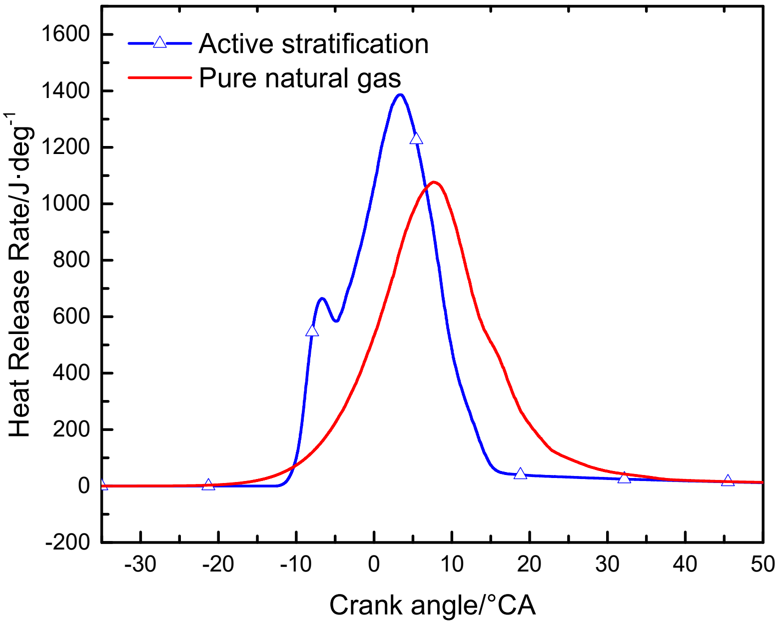

Figure 9. Comparison of instantaneous heat release rate between active stratified combustion and pure natural gas combustion

3.2.2. Analysis of Combustion Duration and Flame Propagation

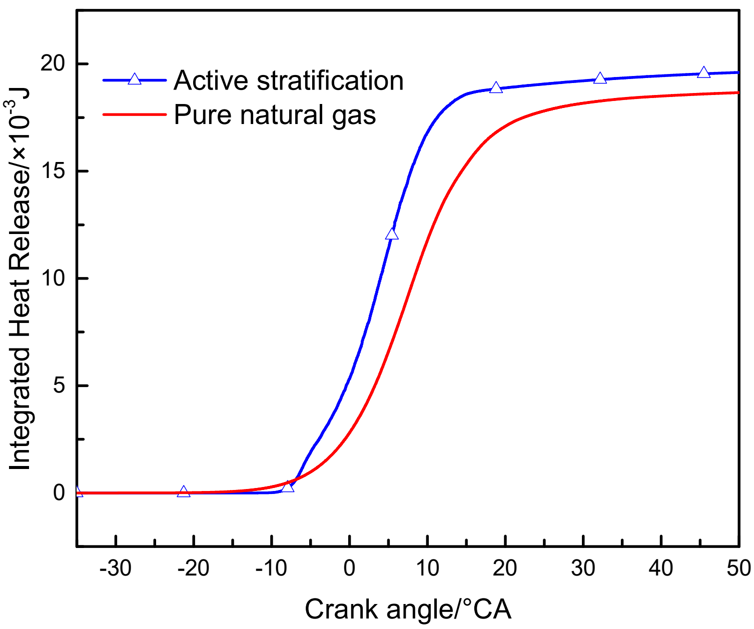

Figure 10. Comparison of cumulative heat release from combustion of different mixture forms and pure natural gas

Since 20mg gasoline was injected directly into natural gas with an excess air coefficient of 1.7 in this study, the final value of accumulated heat release in active stratification was higher in the comparison diagram of cumulative heat release in Figure 10. Combined with Table 5 and

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Figure 10, it can be seen that after direct injection of gasoline, the time used by CA0-CA5 under the active stratification condition is 22.5°CA shorter than that of pure natural gas, and the shortened time accounts for 77.6% of pure natural gas. The time used by CA5-CA90 is reduced by 6.6°CA and accounts for 26.5% of pure natural gas. Figure. 11 shows the in-cylinder flame propagation cloud image of active stratified combustion and pure natural gas combustion, where the horizontal coordinate represents the time interval between the cross section time and the ignition time. Compared with the time interval of 5°CA and 10°CA, the flame propagation range of active stratified combustion is larger than that of pure natural gas in the early stage of combustion. Compared to the time interval of 20°CA and 30°CA, in the case of active stratification, the cylinder flame has spread to the edge of the cylinder, while the pure natural gas flame only occupies a small area of the piston. In summary, the activity stratification promotes the flame propagation of the natural gas engine and makes the low-concentration natural gas engine burn faster.

Table 5. Combustion duration of different mixture combustion and pure natural gas combustion

	CA5 (°CABTDC)	CA0-CA5 (°CA)	CA90 (°CAATDC)	CA5-CA90 (°CA)
Active stratification (-12°CABTDC)	-5.5	6.5	12.8	18.3
Pure natural gas (-34°CABTDC)	-5	29	19.9	24.9

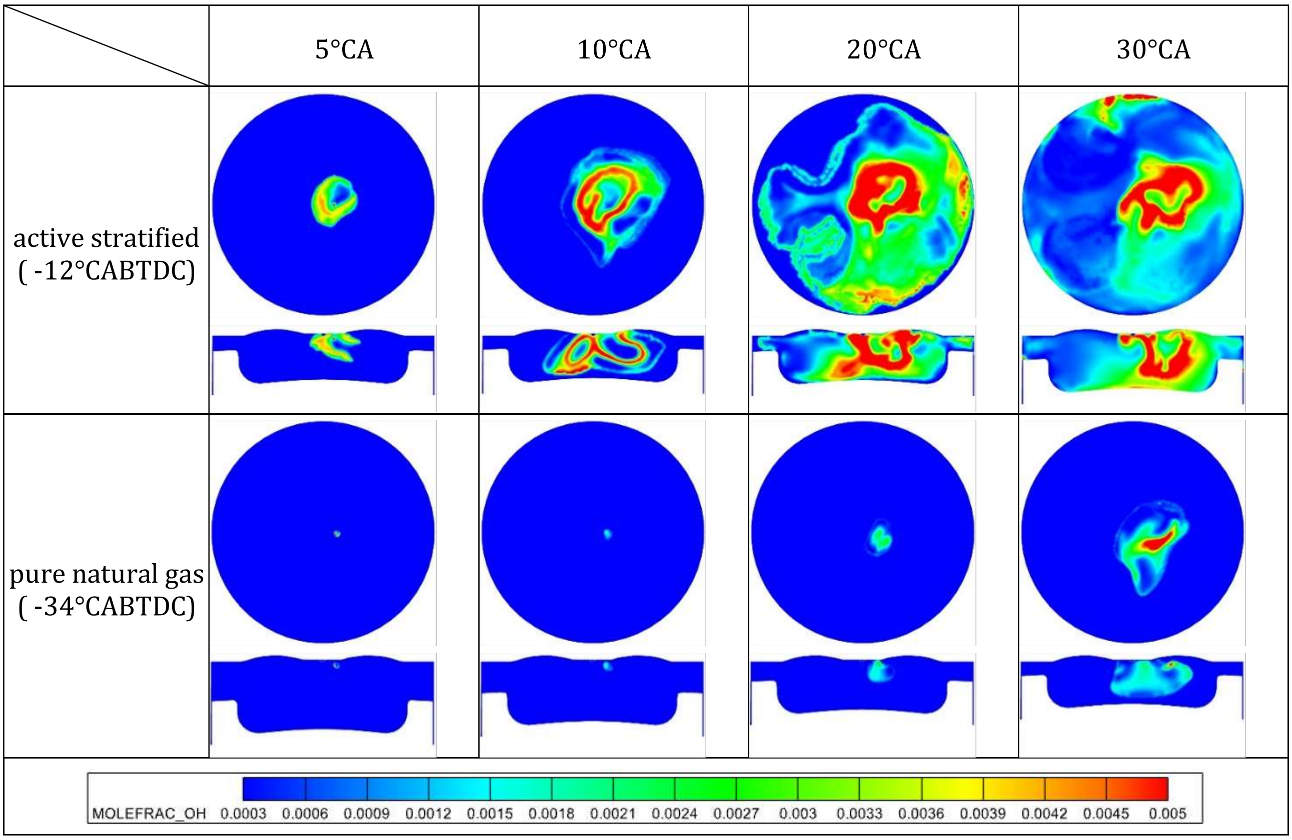

Figure 11. Comparison of flame propagation between combustion of different mixture forms and pure natural gas

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4. Conclusion

This paper mainly studies the in-cylinder combustion characteristics under the active stratification mode after direct injection of gasoline, and then compares it with pure natural gas combustion to get the effect of gasoline direct injection ignition on the combustion characteristics of low concentration natural gas engine. Specific conclusions are as follows:

(1) The small-range application of gasoline in large-bore natural gas engines is realized. In the study, it was found that under the condition of activity stratification, the instantaneous heat release rate in the cylinder showed a double peak phenomenon, and the small peak was formed by the main combustion of gasoline/natural gas mixture in the early stage of combustion. Under the active stratification condition, the initial combustion time in the cylinder gradually decreases with the extension of the injection ignition interval to the combustion starting point CA5, but the time from CA5-CA90 gradually increases.

(2) Comparing the in-cylinder combustion characteristics of active stratification and pure natural gas, it is found that active stratification can improve the indicated thermal efficiency of natural gas engine by 1.4%. In low concentration natural gas engine, direct injection of gasoline can accelerate the flame propagation in cylinder, increase the combustion rate, and shorten the time from CA0-CA5 by 77.6% and from CA5-CA90 by 26.5% under active stratification condition.

5. References

[1] Duan XB, Zhang SH, Liu YQ, Li YT, Liu JP, Lai M-C, Deng BL. Numerical investigation the effects of the twin-spark plugs coupled with EGR on the combustion process and emissions characteristics in a lean burn natural gas SI engine[J]. Energy, 2020,206:118181.

[2] Li YY, Yang FY, Xu LX, Liu JP, Wang JF, Duan XB. Influences of the control parameters and spark plug configurations on the performance of a natural gas spark-ignition engine[J]. Fuel, 2022,324:124728.

[3] Attard WP, Toulson E, Huisjen A, Chen X, Zhu G, Schock H. Spark ignition and pre-chamber turbulent jet ignition combustion visualization[J]. Sae paper, 2012,2012-01-0823.

[4] Mazacioglu A, Gross MC, Kern J, Sick V. Infrared borescopic characterization of corona and conventional ignition for lean/dilute combustion in heavy-duty natural-gas engines[J]. Proceedings of the Combustion Institute, 2019,37(4):4993-5001.

[5] A MG, B RVR. In-cylinder flow and combustion modeling of a CNG-fuelled stratified charge engine[J]. Applied Thermal Engineering, 2019;149:425-38.

[6] Reyes M, Tinaut FV, Melgar A, Pérez A. Characterization of the combustion process and cycle-to-cycle variations in a spark ignition engine fuelled with natural gas/hydrogen mixtures[J]. International Journal of Hydrogen Energy, 2016,41(3):

[7] 2064-2074.

[8] Nitnaware PT, Suryawanshi JG. Effects of Exhaust Gas Recirculation on Performance and Emission Characteristic of SI Engine using Hydrogen and CNG Blends[J]. Journal of the Institution of Engineers, 2018(12):1-6.

[9] Biffiger H, Soltic P. Effects of split port/direct injection of methane and hydrogen in a spark ignition engine[J]. International Journal of Hydrogen Energy, 2015,40(4): 1994-2003.

[10] Taggart-Cowan GP, Rogak SN, Munshi SR, et al. The influence of fuel composition on a heavy-duty, natural-gas direct-injection engine [J]. Fuel 2010, 89(3): 752-759.

[11] Wei LJ, Geng P. A review on natural gas/diesel dual fuel combustion, emissions

[12] and performance [J]. Fuel Processing Technology 2016; 142: 264-278.

[13] Fasching P, Sprenger F, Eichlseder H. Experimental Optimization of a Small Bore

[14] Natural Gas-Diesel Dual Fuel Engine with Direct Fuel Injection [J]. SAE International Journal of Engines 2016; 9(2): 1072-1086.

ISSN: 2710-0723

[15] Huang H, Zhao J, Guo X, et al. Study on reducing carbon dioxide and harmful emissions of diesel-ignited natural gas engine[J]. Chemosphere, 2022. 306: 135586.
[16] Chen ZM, Wang L, Zeng K. A Comparative study on the combustion and emissions of dual-fuel engine fueled with natural gas/methanol, . natural gas/ethano1， and natural gas/n-butanol[J]. Energy Conversion and Management, 2019,192,11-19.
[17] Roberto Berlini Rodrigues da Costaa, Combustion， performance and emission analysis of a natural gas-hydrous ethanol dual-fuel spark ignition engine with internal exhaust gas recirculation[J]. Energy Conversion and Management, 2019,195,1187-1198.