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Effects of Ignition Timing on Combustion Characteristics of a Gasoline Direct Injection Engine with Added Compressed Natural Gas under Partial Load Conditions

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Abstract: The gasoline/natural gas dual-fuel combustion mode has been found to have unique advantages in combustion. The ignition timing has a significant impact on the combustion characteristics of gasoline engines. Thus, here we study the combustion characteristics of gasoline/natural gas dual-fuel combustion mode to determine the details of their respective advantages under cooperative combustion. A direct-injection turbocharged gasoline engine was modified, and an engine experimental platform was built for the coordinated control of gasoline direct-injection and natural gas port injection. A low-speed and low-load operating point was selected, and the in-cylinder pressure, heat release rate, pressure rise rate, combustion temperature, ignition delay, and combustion duration under the coordinated combustion of gasoline and natural gas dual fuel at the ignition moment were studied through bench tests among other typical combustion parameters. The results show that with the increase of the ignition advance angle, the maximum cylinder pressure, heat release rate, pressure rise rate, and maximum combustion temperature increase. The ignition advance angle is 28°CA-BTDC , and PES40 has the best fuel synergy effect and the best power performance improvement. The effect of the advance of the ignition advance angle on the ignition delay and the combustion duration reaches the peak at 20°CA-BTDC – 22°CA-BTDC , and the improvement of the two periods is more significant at PES60.

Keywords: gasoline/natural gas engines; CNG; cooperative combustion; ignition timing; replacement rate



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1. Introduction

Since the creation of gasoline engines, the research on gasoline engines has been in-depth and extensive. The introduction of technologies such as electronic control, in-cylinder direct injection, homogeneous compression ignition, variable compression ratio, intermittent cylinder deactivation, and variable valve has further accelerated the development of gasoline engines [1]. However, there are still many problems to be solved in order to achieve more efficient and cleaner combustion in gasoline engines, and there are still many technologies that need to be improved and studied. Compared with gasoline engines, natural gas engines have natural advantages in fuels. Natural gas has abundant reserves and cheap price. Today, when energy conservation and environmental protection are promoted globally, natural gas is considered to be a relatively ideal clean energy and a promising alternative to vehicle engine fuel, which is increasingly being valued by people [2–4]. The products of natural gas combustion are very clean, and carbon emissions are relatively low. This is because the H/C ratio of natural gas is lower than that of gasoline, and more stoichiometric combustion can reduce CO₂ emissions [5–9]. Because natural gas has a higher-octane number than gasoline and has better anti-knock properties, natural gas can be used in engines with relatively large compression ratio [10]. In addition, although the temperature in the combustion natural gas cylinder increases, which leads to

an increase in NO_x emissions, which reduces its durability and thermal efficiency, these problems can be solved by organizing lean combustion on spark ignition engines [11]. Lean combustion requires a dedicated catalyst. The cost of a dedicated catalyst for lean combustion is high, and the technology is relatively difficult. This is another problem of limiting lean combustion [12]. Increasingly internationally, two or more fuels are used for co-combustion, which has great potential to reduce emissions while ensuring power performance [13–15]. This mode of co-combustion of the two fuels has attracted more and more attention.

Natural gas is mainly used in automobiles in the form of compressed natural gas, making it a dual-purpose gasoline and compressed natural gas engine automobile. The gasoline/natural gas dual-purpose engine has two independent fuel supply systems, which can burn gasoline and natural gas separately [16]. Among all commercial fuels, gasoline and natural gas dual-fuel vehicles are relatively more on the market, while gasoline/natural gas synergistic vehicles are still rare, and there are few and few related researches compared to gasoline engines. In addition, due to the large difference in the physical and chemical properties of natural gas and gasoline, if natural gas is used in a gasoline engine, if some performance parameters of the engine are not adjusted properly, the power performance of the engine will decrease and the emission of nitrogen oxides will increase [17,18]. It is necessary to determine the influence law of engine adjustment parameters on gasoline/natural gas dual-purpose engine performance through experimental research, and reasonably determine the relevant influence parameters. Therefore, it is necessary to carry out related research on it. Some scholars have conducted certain research on gasoline/natural gas dual-fuel dual-use engines. Singh et al. [15] studied the combustion characteristics of a single-cylinder turbocharged SI engine using gasoline-methane dual fuel, port injection of methane, and gasoline direct injection. The study showed that with the increase of methane mass fraction, IMEP and knock intensity decrease. The study by Pan et al. [19] also found that increasing the percentage of methane would lead to an increase in the peak pressure in the cylinder. Behrad et al. [20] studied the knock characteristics of gasoline-natural gas hybrid dual fuels with a natural gas content of 0%, 10%, 20%, and 30% on a single-cylinder spark-ignition engine. The results show that as the proportion of natural gas in the mixed fuel increases, the knock intensity is reduced. Yontar et al. [16] used experiments and three-dimensional in-cylinder combustion computational fluid dynamics analysis methods to quantitatively analyze the characteristics of gasoline-compressed natural gas (90% gasoline/10% CNG) engines when the throttle is fully opened. The results show that the addition of CNG reduces torque, braking specific fuel consumption, volumetric efficiency, braking thermal efficiency, and other performance values, while reducing CO₂, CO, and HC emissions.

Some people have also studied the cooperative combustion of gasoline/natural gas dual-fuel engines. Pan et al. [21] conducted a research on the cooperative combustion of gasoline and natural gas on a spark ignition engine with direct injection gasoline/intake port injection of natural gas. They changed the mixing ratio, excess air ratio and ignition advance angle under low, medium and high loads. The results show that under low load, especially under lean burn conditions, as the proportion of natural gas increases, the peak cylinder pressure and heat release rate increase. Under medium load, as the excess air ratio and the proportion of natural gas increase, the combustion duration increases. Under the condition of high load and full throttle opening, the power performance can be improved by increasing the ignition advance angle and increasing the boost pressure of the supercharger. Catapano et al. [22,23] conducted a study on the particulates and emissions in the combustion products of CNG direct injection and gasoline port injection on an optical engine, and analyzed that the combined combustion of gasoline and natural gas compared to gasoline mode the number of particles produced is much smaller. Cooperative combustion can make the mixture stratified, the flame spread faster, and better improve the efficiency of the engine. Di Iorio et al. [24] also conducted related tests on the optical engine with direct injection of natural gas and intake manifold injection of gasoline, and measured

the spread of the flame front as well as particulates and emissions. The results show that in the dual-fuel synergy mode, with the increase of the proportion of natural gas, the flame front propagation speed is faster than that in the pure gasoline and pure natural gas modes, and HC and CO, NO_x and particulate emissions are significantly lower than in gasoline mode. Ramasamy et al. [25] installed a set of natural gas supply system on the gasoline engine, without any optimization of natural gas, and used the cooperative combustion mode of gasoline and natural gas to study the power and emission performance. The results show that the natural gas When the ratio is 65%, the torque and power of the engine increase by 8.6% on average. Compared with pure natural gas combustion, HC is reduced by 10%, CO is reduced by 75%, and NO_x is reduced by 50%. Hall et al. [26] conducted a related study on the cooperative combustion of gasoline and natural gas on a single-cylinder engine with port-injected gasoline and direct-injection natural gas. The results show that the best effect is when blended with 50% natural gas, HC emissions are significantly reduced under low load. Under high load, 100% natural gas is used because of its better anti-knock effect. Sarabi et al. [27] measured the continuous cycle in-cylinder pressure and exhaust emissions at different ignition timings on a single-cylinder gasoline-natural gas dual-fuel engine. They set the natural gas blending ratio to be 0%, 10%, 25%, and 40%, respectively, the results show that as the proportion of natural gas increases, the IMEP coefficient of variation is significantly reduced, the flame development angle increases (except for 25%), and the emissions of carbon dioxide and nitrogen oxides decrease (except for nitrogen oxides in 40%). The HC emission level at 75% is the smallest. They comprehensively analyzed the 25% natural gas blending ratio as the best combination of gasoline-natural gas dual fuel.

With the development of artificial intelligence and deep learning technology, there are many active researches applying related technologies to various fields. Test and apply the latest research results and successfully applied to engine related fields, this kind of research will be very useful [28,29]. Sotiris et al. [30] combined optical diagnosis with in-cylinder pressure analysis to study the mechanism of flame speed and stability in a dual-fuel mixture of gasoline and natural gas. The results show that the influence of Mark stein's length dominates the lean combustion process in terms of stability and speed. The influence of laminar combustion speed on the combustion process will gradually increase as the air-fuel ratio changes from a stoichiometric value to a rich value. Szabo et al. [31] used deep learning to identify the non-linearity of the knock signal of a large gas engine, and used the deep confidence network (DBN) of the knock probability sequence to discuss the framework for determining the knock non-linearity of the vibration signal. Shin et al. [32] used DNN to build a gasoline engine prediction model to predict knock, performance, combustion, and emissions. The results show that the accuracy of the knock test model is 99.0%, and the coefficient of determination (R-2) values of other test models are all higher than 0.99. The accuracy and robustness of the model prove that DNN is an effective modeling method, and the test time can be used for real-time prediction. Halit et al. [33] used DNN to predict HCCI engine cylinder pressure. They compared the results of deep learning with ANN and experimental results. The results show that the maximum accuracy of cylinder pressure predicted by ANN is 97.83% of the experimental value, and DNN can increase the accuracy to 99.84%. The difference between DNN results and experimental results is less than 1%, and the deep learning method obtains the best results.

Because the dual-fuel engine is equipped with two independent injection systems, gasoline or natural gas can operate independently, or both can be injected and burned at the same time. Compared with the use of gasoline and natural gas alone, gasoline/natural gas coordinated combustion has improved anti-knock resistance, greatly reduced pollutant emissions, significantly improved efficiency, and no power loss [27]. This advantage has become more and more obvious, ensuring power performance can also improve economy, reduce emissions, and make the combustion of internal combustion engines cleaner and more efficient [29]. Since there are few researches on the simultaneous injection of gasoline direct injection and natural gas port injection, the significance of this research is to combine

the advantages of the GDI engine with the advantages of gasoline and natural gas, and to refine the blending ratio. To understand the performance under different ignition timings and different crankshaft angles, this research first builds a test measurement and control platform, including the construction and debugging of the engine test platform, the design and development of the natural gas inlet injection system, and the development of the electronically controlled fuel injection system, which can control the parameters of the fuel supply and gas supply system. Considering that the ignition timing is an important factor that affects the engine cylinder pressure, heat release rate, pressure rise rate and cylinder temperature. The octane number of the mixed gas will change when the gasoline/natural gas is combusted together. The octane number of natural gas is higher, and the ignition advance angle can be appropriately advanced. Then, through experiments, the influence of ignition timing on the combustion process of gasoline/natural gas engines under different replacement rates is studied, and the changes in performance of gasoline and natural gas cooperative combustion engines are discussed, which provides in-depth research on the cooperative combustion of natural gas and gasoline dual-fuel engines.

2. Methods

2.1. Experimental Setup

The experiment was conducted on a four-cylinder, four-stroke and water-cooled DI engine equipped with a turbocharger. In order to realize the CNG/gasoline dual-fuel combustion mode, a port CNG injection system is introduced to the direct injection gasoline engine to provide CNG, thereby avoiding a substantial modification of the gasoline engine structure. Through the modification of the original gasoline, it mainly includes the installation and debugging of gasoline ECU, CNG ECU, and natural gas low-pressure (0.2 MPa) common rail system, so as to realize gasoline direct injection and CNG intake manifold multi-point injection, making Gasoline and CNG work together to combust. The hardware and the software are developed by the research team. Table 1 lists the main technical parameters of the engines used. Figure 1 shows the system diagram of the engine settings. The modified engine is controlled by the gasoline ECU 30 and CNG ECU 31. When operating in cooperative mode, gasoline ECU controls fuel injection timing and fuel injection pulse width, CNG ECU controls CNG injection pulse width, and ignition timing can be either gasoline ECU or CNG ECU to control. The signal of the gasoline ECU passes through two CAN lines and establishes communication with the computer through the ES581 interface. The calibration parameters can be adjusted through the control interface of the INCA calibration software to realize the ignition time, fuel injection time, fuel injection pressure, injection adjustment of gasoline pulse width and throttle opening. According to the gasoline injection pulse width and the air-fuel ratio conversion, the CNG injection pulse width can be obtained.

Table 1. Specifications of the experimental engine.

Parameter	Value
Engine type	4-cylinder, 4-stroke
Bore × stroke	74.5 × 80 mm
Displacement volume	1.395 L
Compression ratio	10.5
Max. torque/speed	250 N·m/1750 ~ 3000 r/min
Max. power/speed	110 kW/5000 ~ 6000 r/min

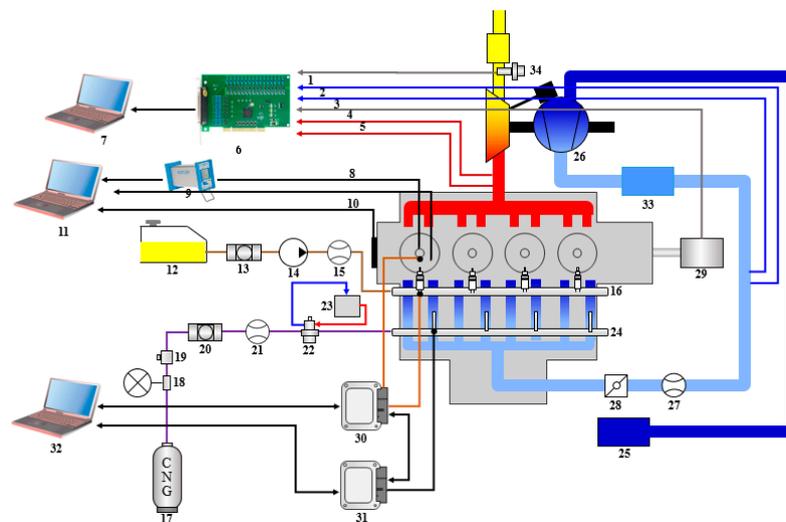


Figure 1. Schematic diagram of experimental device. 1. Intake air temperature 2. Intake pressure 3. Engine speed and torque 4. Exhaust gas temperature 5. Exhaust pressure 6. Data acquisition card 7. Computer for data acquisition 8. Cylinder pressure signal 9. Charge amplifier 10. Engine speed signal 11. Computer for combustion analyzer 12. Gasoline tank 13. Fuel Filter 14. Fuel pump 15. Fuel flow meter 16. Fuel common rail 17. CNG tank 18. Pressure gauges 19. Manual globe valve 20. CNG filters 21. CNG flowmeter 22. Pressure reducing valve 23. Heat exchanger 24. CNG common rail 25. Air filter 26. Exhaust turbocharger 27. Air flow meter 28. Throttle 29. Electric dynamometer 30. Gasoline ECU 31. CNG ECU 32. Calibration ECU 33. Intake air cooler 34. Wide-area oxygen sensor.

In addition to the engine and ECU control system used in the test, the experimental device is also equipped with modules such as a fuel supply system, a dynamometer, a data acquisition system, and a combustion analysis system. The fuel supply system is composed of gasoline fuel supply system and CNG fuel supply system, which are controlled by their respective ECUs. The former is composed of a gasoline tank 12, a fuel filter 13, a fuel pump 14, and a fuel flow meter 15, which are connected to the fuel common rail 16 through a pipeline, while the latter is mainly compressed in a CNG cylinder 17, and is compressed by a pressure gauge 18 and a manual shut-off valve 19. The CNG filter 20, the CNG flow meter 21 and the pressure reducing valve 22 are connected to the CNG common rail. It should be noted that because the pressure of CNG is relatively large, about 20 MPa, when CNG is reduced to about 0.2 MPa required for work, the temperature of the pressure reducing valve will drop rapidly. Therefore, heat exchanger 23 is connected to the pressure reducing valve to avoid freezing and failure by introducing the hot water pipe into the engine cooling circulating water. The dynamometer is used to control the engine speed and load. The function of the data acquisition system is to collect signals such as speed, torque, intake and exhaust pressure and temperature, and realize real-time monitoring and recording of the engine through A/D conversion. The function of the combustion analysis system is to monitor the engine speed and cylinder pressure signals in real time, and to obtain combustion data through computer processing.

Gasoline is directly injected into the cylinder through an electronically controlled fuel injector with a maximum injection pressure of 20 MPa. The start of injection timing, CNG injection pulse width and ignition timing are precisely controlled by CNG ECU. The equivalence ratio of the air-gasoline mixture is determined by a universal wide range exhaust gas oxygen sensor, with a resolution of 0.001, an uncertainty within $\pm 0.8\%$, and a response time of less than 0.15 s. Connect the eddy current dynamometer (CW260-1) to the engine. The maximum torque of the dynamometer is 1395 N·m, the maximum allowable speed is 7500 r/min, and the rated absorption power is 260 kW. The output power of the engine can be calculated according to the speed and torque of the eddy current dynamometer. The eddy current dynamometer is monitored and regulated by the control

software. The pressure in the cylinder is measured with a Kistler 6115B water-cooled piezoelectric pressure sensor, which has a natural frequency of approximately 70 kHz and is flush mounted in the cylinder head. The Kistler 5018 charge amplifier is used to receive and process pressure signals, which are then collected by the National PC-6-6 data acquisition card. A photoelectric encoder connected to the crankshaft at 0.1°CA (crankshaft angle) intervals is used to trigger all pressure samples. In-cylinder pressure was measured with a Kistler pressure transducer type 6115B. Samples were taken at 0.5CAD intervals for 300 consecutive cycles. The heat release rate, pressure rise rate, combustion temperature, ignition delay and combustion duration are calculated by using a combustion analyzer (AVL IndiPar AT2645E). CNG flow rate and Gasoline flow rate is respectively measured by gas flow meter (AVL1000), fuel consumption meter (FP-2140H). The accuracy and measurement range of the experimental instrument are written in Table 2. The intake air temperature, water temperature, intake air pressure and exhaust air temperature are collected by ICP DAS 7015, 7017, and 7018 modules respectively. These temperature and pressure signals are connected to the computer via RS485 through the acquisition module, so that the data is displayed on the computer screen. The engine exhaust temperature in the test does not exceed 890 °C, the water temperature is set to 80 °C, and the change range does not exceed ± 1 °C. The coil heater keeps the coolant and oil temperature within 0.5 °C of the target value. The temperature and humidity of the air are kept the same through the air conditioning system.

Table 2. The experimental apparatus accuracy and measuring range.

Parameter	Uncertainty	Instrument Manufacturer	Type
Engine speed	$\leq \pm 5$ r/min	CAMA	CW260-1
Torque	$\leq \pm 0.2\%$ Nm	CAMA	CW260-1
Gasoline mass flow rate	$\leq \pm 0.33$ g/min	ONO SOKK	FP-2140H
CNG volumetric flow rate	$\leq \pm 0.35\%$ L/min	ToCeil	CMF025
Air volumetric flow rate	$\leq \pm 0.1$ L/min	AVL	AVL1000
Crank angle position	$\leq \pm 0.1$ °CA	Kistler	2614B
Cylinder pressure	$\leq \pm 0.1\%$ bar	Kistler	6115B

A reference gasoline with a RON of 95, which was supplied by China Petrochemical Corporation was used in the experiment. The gasoline meets Chinese National VI emission standards. The main fuel characteristics of gasoline was presented in Table 3. A commercially available natural gas with a purity of >98%, which is primarily composed of methane, was supplied by China Resources Gas Group Limited. Its LHV, RON, auto-ignition temperature, and flash point was respectively 50 MJ/kg, 120, 480 °C, and -184 °C.

Table 3. The main fuel characteristics of China-VIA gasoline.

Fuel Characteristics	Gasoline
Research Octane Number (RON)	95
Density @ 20 °C (kg/m ³)	720 ~ 775
Lower heating value, LHV(MJ/kg)	43.1
Lead content (g/L)	≤ 0.005
Manganese content (g/L)	≤ 0.002
Iron content (g/L)	≤ 0.01
Sulphur content (mg/kg)	≤ 10
Benzene content (vol%)	≤ 0.8
Aromatic content (vol%)	≤ 35
Olefin content (vol%)	≤ 18
Oxygen content (wt%)	≤ 2.7
Methanol content (wt%)	≤ 0.3
AFRStoich.	14.7
Vapor pressure (kPa)	40 ~ 85

2.2. Experimental Setup

When gasoline and CNG are injected at the same time, in order to make a better comparison, the CNG energy substitution rate PES is defined. This parameter represents the percentage of the total energy calculated by the low calorific value of CNG, as shown in Equation (1):

$$PES (\%) = \frac{m_{CNG} \cdot H_{u_{CNG}}}{m_{CNG} \cdot H_{u_{CNG}} + m_{gasoline} \cdot H_{u_{gasoline}}} \times 100\% \quad (1)$$

In the formula, m_{CNG} and $m_{gasoline}$ represent the mass flow rate of CNG and gasoline, respectively. $H_{u_{CNG}}$ and $H_{u_{gasoline}}$ represent the low heating value of CNG and gasoline, respectively, and the values are 50 MJ/kg and 44 MJ/kg, respectively.

PES refers to the percentage of CNG energy in a certain operating point in the total amount of CNG and gasoline, and also indicates the proportion of energy that CNG replaces gasoline. In theory, the higher the CNG substitution, the better the fuel economy. However, the increase in the use of CNG may lead to worsening of combustion conditions and deterioration of power and emissions. At the same time, too high PES will make the direct injection gasoline nozzle in the cylinder lack sufficient gasoline spray cooling, and the CNG coming from the intake duct will enter the cylinder to burn at a higher temperature, which is likely to cause damage to the gasoline nozzle [34–36]. Therefore, PES cannot be too high. Based on the comprehensive consideration of engine power, economy, combustion and emission characteristics, and engine reliability in gasoline/CNG dual-fuel combustion mode, PES was finally selected as a compromise of 40–70%.

In order to study the effect of ignition timing and CNG addition on the combustion characteristics of gasoline direct injection engine under partial load conditions, the experiment selects the speed of 2000 r/min and the load of 20% as the working conditions of the study, the excess air ratio is 1, and the Equivalent combustion mode. After the engine is fully warmed up, gasoline is directly injected into the cylinder for all tests. When the lubricant and coolant temperatures are 80 ± 1 °C and 90 ± 1 °C, natural gas is injected into the intake manifold, and the engine runs in a gas-gasoline dual fuel mode. The intake air temperature is maintained at 22 °C. The CNG injection pulse width, gasoline injection quantity, and air flow are respectively fixed at 2 ms, 15.3 mg/stroke, and 225 kg/h. When the PES is 40% and other parameters remain unchanged, the ignition timing is changed every 2°CA on the basis of the ignition timing of the gasoline engine. When the ignition timing is advanced, attention should be paid to whether knocking occurs. If it occurs, the ignition should be delayed immediately. When delaying ignition, pay attention to changes in torque and exhaust temperature. If the torque drops too fast or the exhaust temperature is too high, stop the strategy of delaying ignition. If the above situation does not occur, the ignition advance angle is delayed until the torque drops or the cycle fluctuation becomes larger. In this test, the adjustment angle interval of the ignition advance angle is finally set to 12°CA–30°CA. Starting from 12°CA, on the basis of increasing the ignition advance angle every 2°CA-BTDC, the cylinder pressure, heat release rate, pressure rise rate, combustion temperature, ignition delay and combustion duration under dual-fuel cooperative combustion are calculated. Then, change the value of PES and repeat the above process until the test under the four types of PES is completed. In order to obtain the correct value of an operating condition, the experiment was repeated 100 times to eliminate the fault tolerance value, keep reasonable experimental data, and find the average value under the operating condition. Finally, in order to verify the influence of the ignition advance angle on the dual-fuel combustion performance, the adjustment rule of the ignition advance angle was studied through a comparative analysis of experimental data.

3. Results and Discussion

3.1. In-Cylinder Pressure

Adjusting the ignition timing can affect the cylinder pressure, thereby affecting the power of the engine. The advance or retard of the ignition advance angle can control the length of the ignition delay of combustion. By adjusting the ignition advance angle, the best ignition advance angle can be found to increase the pressure in the cylinder to improve the power performance of engine. The pressure peak in the cylinder can characterize the magnitude of the mechanical load of the combustion, and the crank angle corresponding to the pressure peak can be used to analyze the influence of the pressure peak on the mechanical load. Figure 2 shows the cylinder pressure curve at different ignition timings.

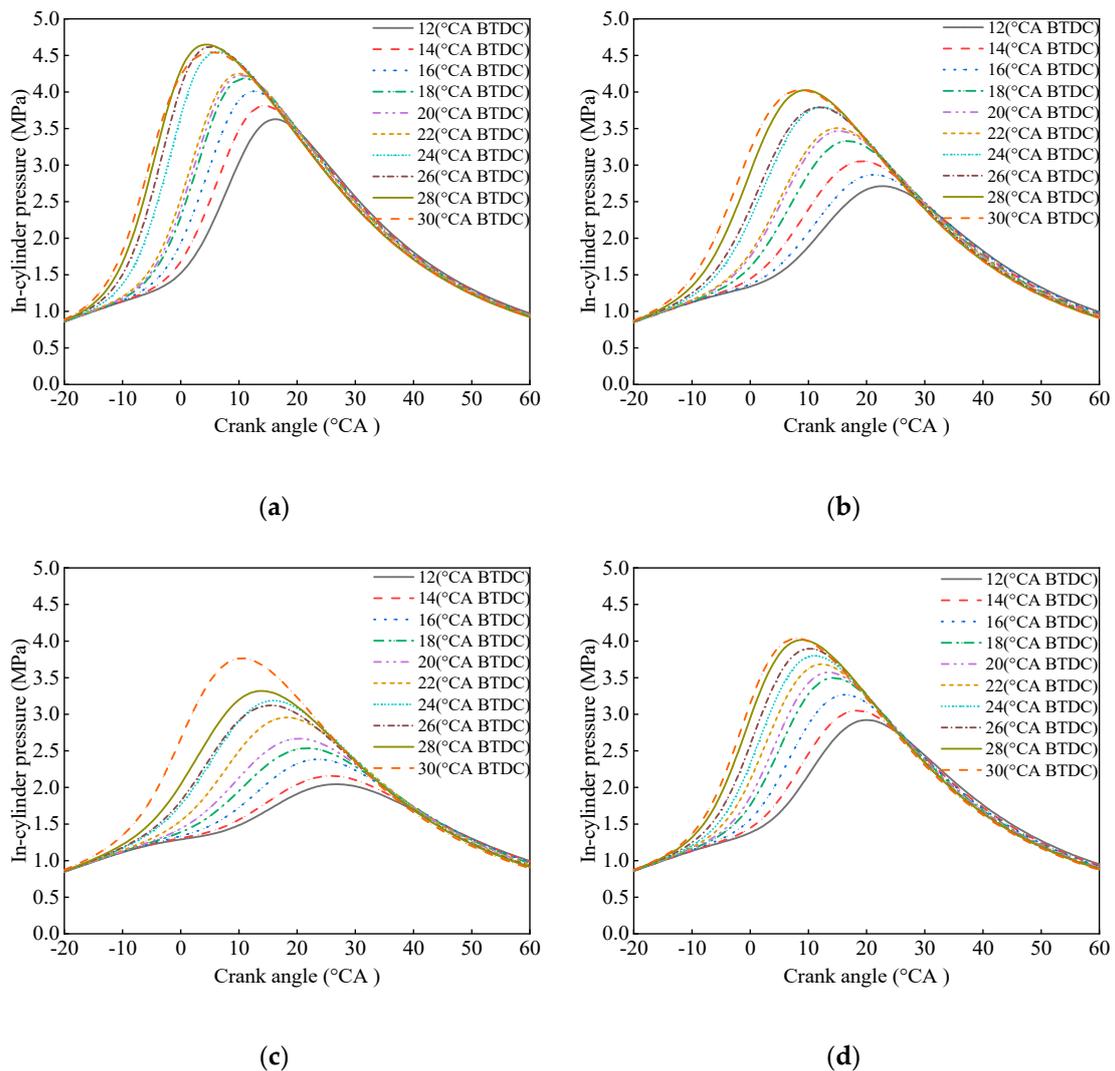


Figure 2. (a) In-cylinder pressure at PES40; (b) In-cylinder pressure at PES50; (c) In-cylinder pressure at PES60; (d) In-cylinder pressure at PES70.

At PES40, as the ignition advance angle advances, the peak pressure in the cylinder gradually increases, and the peak phase of the pressure in the cylinder also gradually advances, getting closer and closer to the top dead center. This is because as the ignition timing advances, the combustion phase advances, which strengthens the mixing of gasoline and air, increases the combustion efficiency, and increases the pressure in the cylinder [15]. At the ignition advance angle of 28° CA-BTDC, the in-cylinder pressure peak and peak phase reach the extreme values, 4.65 MPa and 4° CA-BTDC, respectively. After continuing

to increase the ignition advance angle, the pressure peak in the cylinder begins to decrease, and the crankshaft angle corresponding to the pressure peak no longer decreases, but starts to increase. The ignition advance angle is usually used to control the position of the pressure peak in the cylinder. It can also be seen from Figure 2a that when the ignition timing is 12°CA-BTDC , the crankshaft angle corresponding to the peak pressure in the cylinder is far from the top dead center, and the peak pressure in the cylinder is the lowest. At this time, the mixed gas ignites later, the expansion ratio under the power stroke decreases, the combustion volume increases, and the heat transfer surface area increases, which increases the heat transfer loss, thereby reducing the cycle thermal efficiency. When the ignition timing is 28°CA-BTDC , the crank angle corresponding to the peak pressure in the cylinder is closer to the top dead center. However, the pressure peak phase appears too early, indicating that the mixture is ignited prematurely, resulting in an increase in the negative compression work. At the same time, the pressure peak in the cylinder increases, which leads to an increase in mechanical load, which in turn reduces power and economy.

In PES50, PES60, and PES70, the change trend of the in-cylinder pressure peak and peak phase with the ignition advance angle is basically the same as that under PES40. However, the difference is that the in-cylinder pressure peaks all appear at 30°CA-BTDC . This is because the increase in substitution rate increases the anti-knock resistance of the mixture, which is 2°CA earlier than the ignition timing of PES40. With the increase of PES, the pressure in the cylinder shows a decreasing trend, and the maximum pressure peak gradually decreases. This is mainly because the increase in the proportion of CNG in the fuel leads to a decrease in the calorific value of the mixed gas, a decrease in the combustion temperature, and a decrease in the mechanical load of the combustion, which leads to a decrease in the peak pressure in the cylinder [16]. The degree of incomplete combustion of the mixed gas is strengthened. However, with the increase of CNG content, natural gas combustion takes the leading role in PES70. Since the quality of CNG and air mixture is higher than that of gasoline and air, the efficiency of mixed combustion is improved, the mixing is more complete, and the combustion effect is better. Well, the pressure in the cylinder rises. It can be seen that both the ignition timing and the substitution rate can have a greater impact on the cylinder pressure. Properly advance the ignition timing to 28°CA-BTDC , and the pressure can be increased under PES40 to improve dynamics.

3.2. Heat Release Rate

The heat release rate curve can characterize the control of the heat release law, and can be expressed by the crankshaft angle corresponding to the center of the heat release rate curve. Figure 3 shows the variation curve of heat release rate with ignition timing under different replacement rates. The face center value is the value of the crankshaft angle obtained by multiplying the heat release rate and the corresponding crankshaft angle and dividing by the integral of the heat release rate. The upper and lower limits of the integral are the combustion start point and the combustion end point respectively. The face center value is the integrated result of the heat release rate and the corresponding crankshaft rotation angle. The face center value indicates the distance from the top dead center of the combustion. A large face center value indicates that the combustion heat efficiency is lower if it is far from the top dead center. On the contrary, a small face center value indicates that the combustion heat efficiency is higher near the top dead center. After calculation, under different replacement rates, as the ignition advance angle advances, the overall face center value first increases and then decreases. This is because as the ignition timing advances, the combustion can be improved. The larger the natural gas combustion area, the more fully the combustion, resulting in an increase in the heat release rate [20]. However, the excessive advancement of the ignition timing deteriorates the combustion process and reduces the heat release rate. The crank angle corresponding to the peak heat release rate moves forward, and the peak value first rises and then decreases. The main combustion is dominated by natural gas. Advance ignition means that the crankshaft angle corresponding to combustion moves forward, and the combustion of natural gas

moves forward. Therefore, the crankshaft angle corresponding to the peak heat release rate moves forward. Under PES40, when the ignition time is 24°CA-BTDC, the absolute value of the face center value is smaller and closer to the top dead center. Compared with other ignition moments, the combustion heat efficiency is higher, which can also be used at other substitution rates get the same law. As the PES increases, the heat release rate decreases. This is because the increase in the proportion of CNG in the fuel leads to a decrease in the calorific value of the mixed gas [29]. This means that the more CNG is burned, the heat energy released decreases significantly, and the peak value of the heat release rate curve decreases. After PES60, the natural gas in the cylinder burns faster at this time, the combustion rate of the mixture is faster, the fuel utilization rate is good, and the heat release rate is improved.

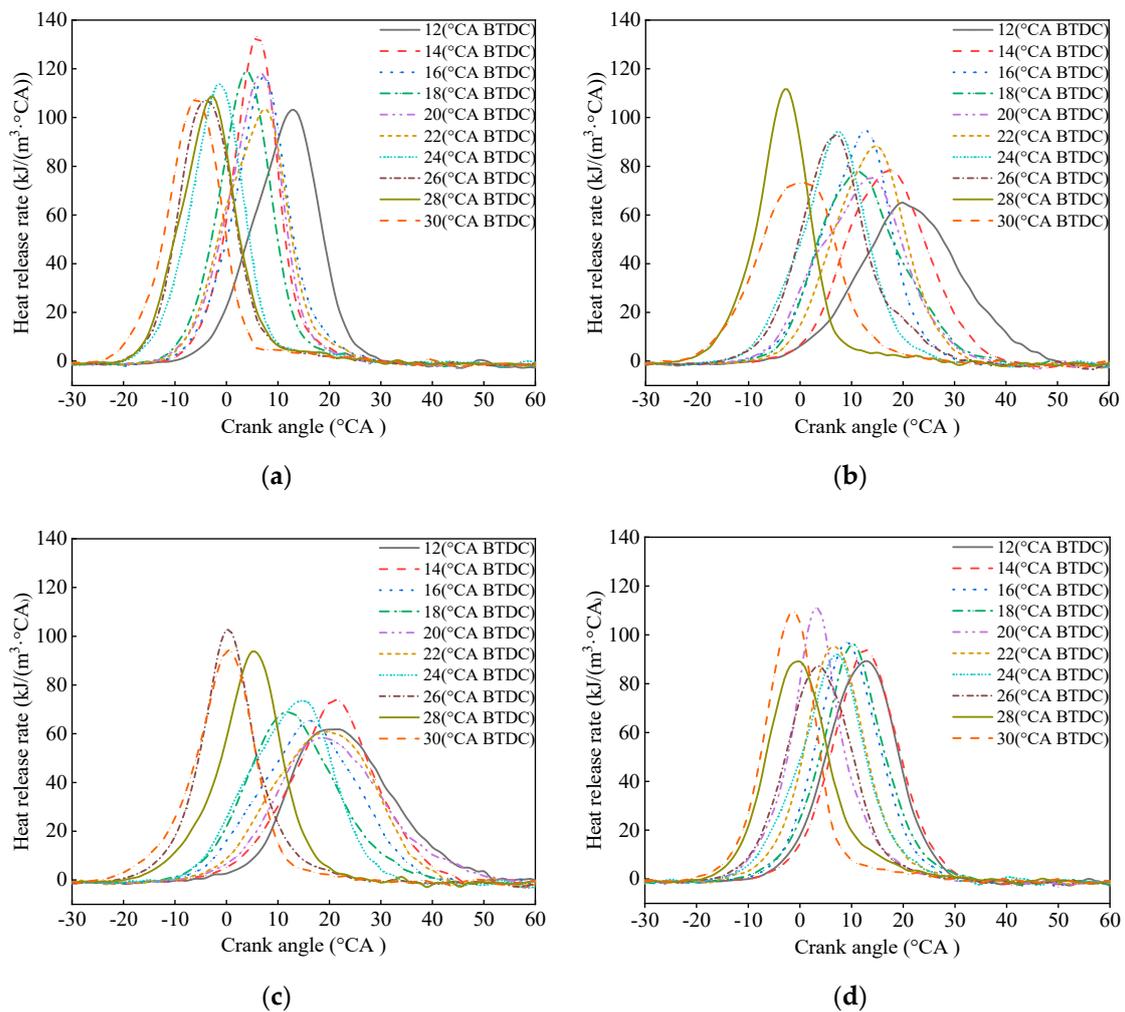


Figure 3. (a) Heat release rate at PES40; (b) Heat release rate at PES50; (c) Heat release rate at PES60; (d) Heat release rate at PES70.

3.3. Rate of Pressure Rise

The rate of pressure rise can be used as an important indicator for evaluating the speed of flame propagation in the cylinder. The faster the flame propagation speed, the greater the degree of change in the pressure increase in the cylinder, and the increase in pressure rise rate. The faster the combustion, the longer the combustion duration. The more heat is released by the combustion, the better the power and economy. However, an excessively high pressure rise rate will make the engine work rough, and the vibration and noise caused by combustion will increase. Figure 4 shows the pressure rise rate curves

at different ignition timings. The change of the ignition advance angle under PES70 has little effect on the overall fluctuation of the pressure rise rate. This is because the mixing efficiency of gasoline and CNG is improved, so that it burns sufficiently, shortens the afterburning period of the mixed fuel combustion process, and makes the change of the maximum pressure rise rate smooth [16]. In general, as the ignition advance angle advances to the optimal ignition advance angle, the pressure rise rate increases. This is because the combustion phase is advanced to strengthen the mixing of gasoline and air, and the high-octane performance of CNG improves the combustion speed of CNG to a certain extent, and the pressure rise rate increases [15]. As the ignition advance angle increases, the crankshaft angle corresponding to the maximum pressure rise rate also moves before the top dead center. For example, when the ignition timing of PES40 is 30°CA-BTDC, the crank angle corresponding to the maximum pressure rise rate is 7°CA-BTDC, and the corresponding maximum cylinder pressure is also advanced accordingly. Taking a comprehensive look at the ignition timing at different replacement rates, a higher pressure rise rate can be maintained at 28°CA-BTDC.

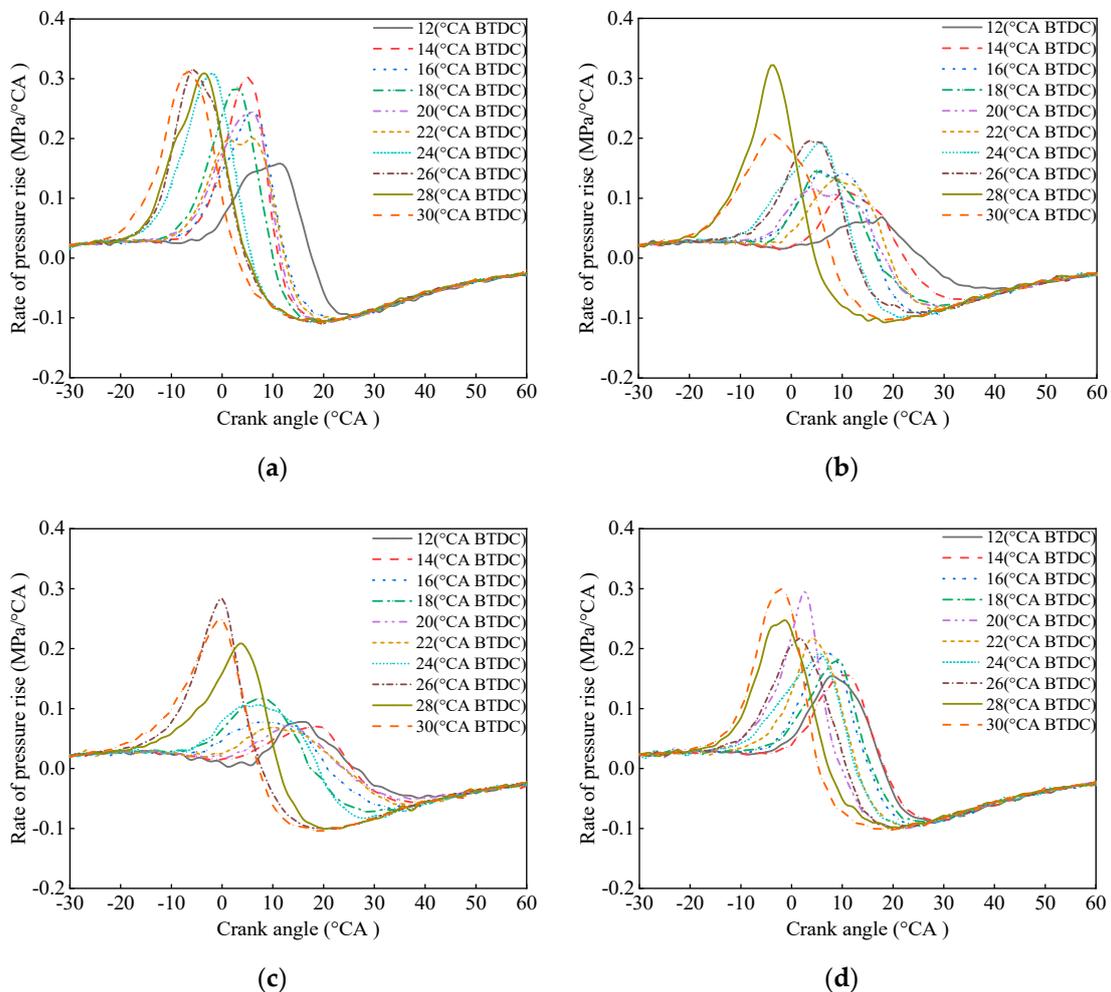


Figure 4. (a) Rate of pressure rise at PES40; (b) Rate of pressure rise at PES50; (c) Rate of pressure rise at PES60; (d) Rate of pressure rise at PES70.

It can also be seen from Figure 4 that with the increase of PES, the maximum pressure rise rate begins to gradually decrease. This is because gasoline has a higher combustion speed than CNG during the combustion process. With the increase of CNG content, the combustion flame propagation rate of the mixed fuel decreases, which affects the afterburning period of the mixed gas combustion, and the maximum pressure rises [20].

The high rate gradually decreases. Another reason is that the maximum pressure rise rate is mostly determined by the amount of combustible mixture formed during the ignition delay. The short chemical delay of CNG in the mixed fuel reduces the atomization quality of the mixed fuel, thereby reducing the ignition delay of the engine. The amount of mixed gas during the stroke causes the maximum pressure rise rate to decrease. However, with the increase of CNG content, when PES70, the CNG content dominates. Since the quality of CNG and air mixture is higher than that of gasoline and air, the efficiency of mixed combustion is improved, and the mixing becomes more complete, making high the combustion effect is better with CNG content, and the pressure rise rate increases. It can be seen that adjusting the ignition advance angle and PES can affect the pressure increase rate, and reasonably control the ignition advance angle, thereby controlling the crankshaft angle corresponding to the maximum pressure increase rate, so as to achieve reasonable control of the combustion speed.

3.4. Combustion Temperature

The maximum pressure in the cylinder can reflect the size of the mechanical load during the combustion process, and the maximum combustion temperature in the cylinder can reflect the size of the thermal load during the combustion process. The higher the temperature, the greater the thermal load, and vice versa. The combined result of the highest pressure in the cylinder and the highest combustion temperature can affect the thermal efficiency of the engine, and the crankshaft angle corresponding to the two can evaluate the timeliness of the combustion process organization. Figure 5 shows in-cylinder temperature at different ignition timings.

It can be seen that as the ignition delay advance angle increases, the phase corresponding to the highest temperature in the cylinder advances accordingly. This is because as the ignition timing advances, the combustion phase advances, which strengthens the mixing of gasoline and air, increases the combustion efficiency, and increases the temperature and pressure in the cylinder, so the maximum combustion temperature increases [29]. With the increase of PES, the combustion temperature shows a decreasing trend, and the maximum combustion temperature decreases. This is mainly because the increase in the ratio of CNG mainly changes the heating value of the mixed fuel, thereby changing the overall heating value of the fuel and lowering the combustion temperature [29]. However, with the increase of PES under PES50-PES70, the change of the maximum combustion temperature in the cylinder is not very significant. This is because the low-speed working condition of the study reduces the mixing effect of gasoline and CNG, thereby reducing the combustion efficiency. Therefore, the change in the calorific value of the mixed fuel has no obvious effect on the decrease of the combustion temperature under low-speed operating conditions. The maximum combustion temperature in the cylinder of PES40 did not change significantly, maintained at about 2300 K. The maximum temperature in the cylinder of PES50-PES70 increased with the advancement of the ignition timing, and the maximum value was basically around 2200 K. Therefore, the highest in the cylinder under the PES40 the combustion temperature is higher than other replacement rates.

3.5. Ignition Delay

The ignition timing will affect the ignition delay under the gasoline/CNG cooperative combustion. In principle, the more advanced the ignition advance angle, the shorter the flame retardation period. However, the higher of the cylinder pressure and the bigger of the ignition advance angle, may lead to the longer of the ignition delay, which affect the gasoline engine work. In the current work, the ignition timing is selected as the baseline point, namely CA0. CA10 represents the crank angle that releases 10% of the overall heat energy, and also represents the turning point of the initial laminar flame core formation [37]. Therefore, the crank angle interval CA0-10 is defined as the ignition delay. Figure 6 shows the ignition delay at different ignition timings. From the overall point of view in the figure, at the same PES, with the advance of the ignition timing, the ignition delay shows a trend

of first increasing and then decreasing. Under PES40, PES50 and PES60, when the ignition advance angle is increased from 12°CA-BTDC, the ignition delay first increases and then decreases. When the ignition advance angle is 22°CA-BTDC, the ignition delay is the longest, reached 18.1°CA, 24.35°CA, and 26.1°CA respectively. After the ignition advance angle is greater than 22°CA-BTDC, the ignition delay begins to gradually decrease. This is because the main factors that affect the ignition delay are pressure and temperature. With the advancement of the ignition timing, the temperature and pressure in the cylinder are lower when the mixture enters the cylinder, which prolongs the physical and chemical preparation phase before the combustion of the mixture, resulting in prolonged ignition delay [21]. However, as the ignition timing advances again, the ignition advance means that the crankshaft angle corresponding to the combustion advances, which is beneficial to the efficiency of the mixed combustion of the in-cylinder mixed gas driven by natural gas, the combustion effect is better, and the ignition delay is reduced. Under PES70, with the continuous advancement of the ignition advance angle, the ignition delay presents a gradual increase trend, and the ignition delay tends to stabilize after the ignition advance angle is greater than 22°CA-BTDC. This is because, as the ignition advance angle increases, the time interval between the injection and the ignition will be shortened and the ignition delay will change.

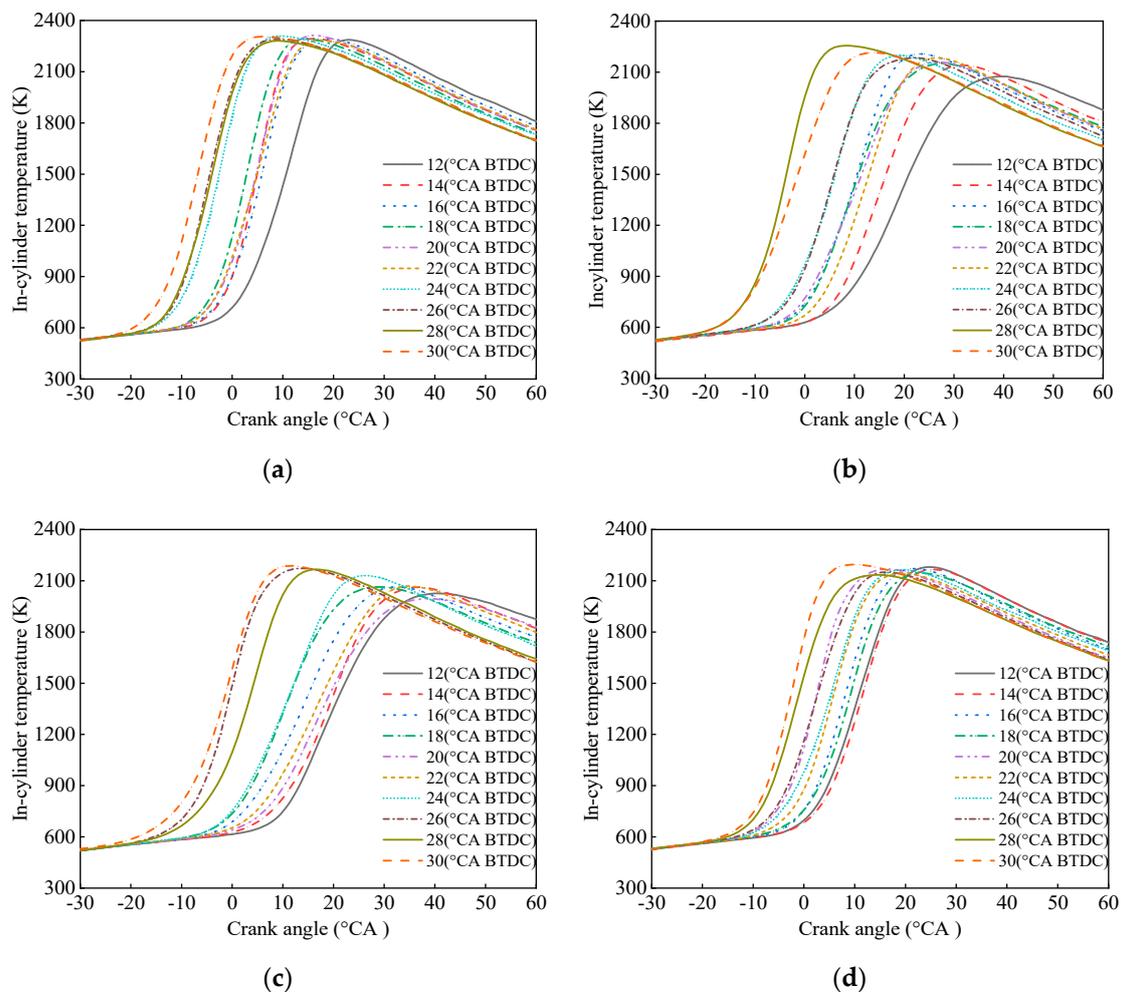


Figure 5. (a) In-cylinder temperature at PES40; (b) In-cylinder temperature at PES50; (c) In-cylinder temperature at PES60; (d) In-cylinder temperature at PES70.

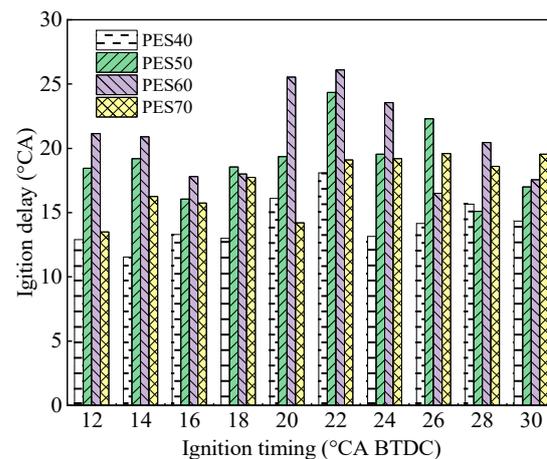


Figure 6. Ignition delay at different ignition timings.

It can also be seen from the figure that at the same ignition timing, with the increase of PES, the ignition delay is first shortened and then extended. Under small PES, proper amount of natural gas participates in the combustion to accelerate the combustion rate, and the throttle opening is increased, which increases the intake charge and intake pressure in the cylinder, thereby improving the thermodynamic state of the mixture in the cylinder and the ignition delay is reduced. When the PES further increases, the amount of natural gas injection increases, natural gas occupies a certain air volume, and the amount of air decreases. In addition, the gasoline concentration decreases and ignition is slower. In addition, natural gas has a high specific heat capacity, and natural gas has a greater activation energy and a slower flame speed than gasoline, which significantly reduces the temperature and pressure in the cylinder and prolongs the ignition delay [38]. In PES70, natural gas takes the leading role. Because CNG has better mixing characteristics and high combustion equivalent volume brought about by early ignition, the maximum combustion pressure in the cylinder increases and the ignition delay decreases [39].

3.6. Combustion Duration

In the current study, the different combustion stages are determined by the heat release rate [19]. The baseline point is CA0, and the crank angles corresponding to the overall heat release rate of 10%, 50%, and 90% are defined as CA10, CA50, and CA90. According to related physical criteria, CA10, CA50, and CA90 can respectively indicate the formation of the initial laminar flame core, the fastest development of the turbulent flame and the turning point of the end of the main combustion. According to this definition, the crank angle intervals CA10-50, CA50-90, and CA10-90 can correspond to early flame development, fast turbulent flame propagation, and combustion duration, respectively [40]. Figures 7–9 further show the effects of different injection strategies on the initial flame development duration (CA0-10), fast turbulent flame propagation duration (CA50-90), and combustion duration (CA10-90). At a certain PES, with the advance of the ignition timing, CA10-50 is basically unchanged, and CA50-90 and CA10-90 change slightly. This shows that the addition of certain natural gas and the change of ignition timing do not affect the initial development of the flame, but affect the rapid turbulent flame propagation in the later stage. On the whole, the changing trends of the three durations are basically the same. Take the duration of fast turbulent flame propagation as an example. With the increase of PES, CA50-90 showed a trend of increasing first and then decreasing. The center of gravity of combustion showed a trend of moving back and then back with the increase of PES. With PES40, the main combustion fuel is gasoline. The flow of natural gas intensifies the uniform mixing of gasoline and air. At the same time, the cylinder pressure and temperature are suitable for accelerating the combustion rate of diesel, which advances the entire combustion process and moves the center of gravity forward. With PES60, the

main combustion fuel is natural gas, and the amount of natural gas in the cylinder is relatively large. Due to the presence of natural gas, the temperature and pressure are reduced, which is not conducive to gasoline combustion, and the combustion center of gravity shifts backward. At low ignition advance angle, with the change of PES, the rapid turbulent flame propagation duration fluctuates little. This is mainly affected by the combined effect of factors such as intake air volume, temperature and pressure in the cylinder, and heat dissipation loss [41]. At a high ignition advance angle, a good cylinder environment promotes combustion, and the flame core and temperature make the diffusion combustion of gasoline and natural gas proceed quickly, resulting in a short duration of rapid turbulent flame propagation.

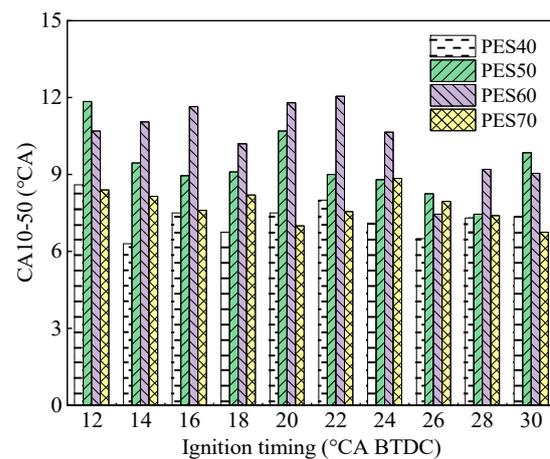


Figure 7. Duration of initial flame development (CA10-50) at different ignition timings.

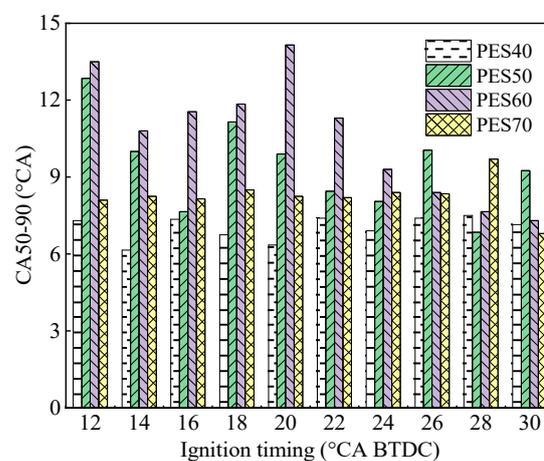


Figure 8. Duration of fast turbulent flame propagation (CA50-90) at different ignition timings.

It can be seen from Figure 9 that for the combustion duration, under the substitution of PES50 and PES60, within the range of the ignition advance angle considered, the combustion duration is relatively large due to the adjustment of the ignition advance angle, which is about 18.8°CA and 21.0°CA respectively. However, when replaced by PES40 and PES70, the impact is relatively small, and the difference is not much, maintained at about 14.3°CA and 16.0°CA , respectively. When the ignition advance angle is increased by 20°CA-BTDC , because the ignition advance angle is too large, it is possible that the engine has produced knocking and the combustion duration begins to decrease. Under PES40, the combustion duration showed a trend of first decreasing, then increasing and then decreasing. This is because with the increase of the ignition advance angle, the probability of the flame front surface contacting the cylinder wall is reduced when the

mixture is combusted, which makes the combustion complete and the combustion duration shortens. With the increase of the ignition advance angle, the time interval between the injection and the ignition is shortened. If the ignition without complete mixing, the combustion will be insufficient due to the uneven mixing and the combustion duration will increase [15]. Excessive ignition advance angle causes the mixture to ignite too early. The mixture starts to burn before the piston reaches the top dead center, and the combustion duration decreases. However, overall, the ignition advance angle has a relatively small effect on the combustion duration.

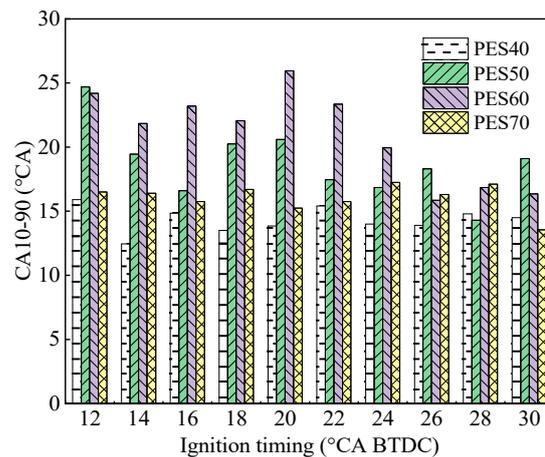


Figure 9. Combustion duration (CA10-90) at different ignition timings.

Like the ignition delay scenario, the PES has a significant impact on the combustion duration, affecting the combustion speed, and then the entire combustion process. The law of change is that under the condition that the ignition time remains the same, the combustion duration of PES40 to PES60 increases as the PES increases. The ignition delay of PES60 is the longest, and the combustion duration is relatively lagging and longer. This is because PES increases, the amount of natural gas injection increases, the amount of gasoline decreases, the temperature and pressure in the cylinder decrease, the ignition delay increases, and the combustion duration increases [37]. The combustion duration of PES60 to PES70 is significantly reduced. At this time, natural gas takes the leading role, indicating that blending gasoline with pure natural gas fuel can shorten the combustion duration. This is because the combustion and heat transfer rate of gasoline is higher than that of natural gas, and it is used as a liquid fuel. Gasoline blending will relatively reduce the actual air-fuel ratio, so that the mixture combustion heat release earlier. Under PES40, the proportion of natural gas is less than that of gasoline, and the burning speed of pure natural gas is slower than that of pure gasoline. Pure gasoline is easier to ignite than natural gas. When the ratio of gasoline is high, the gasoline/natural gas mixture is easily ignited. At the beginning of fire, when the spark plug ignites, the flame center is formed, and after a period of time, the flame core is formed [42]. Gasoline in-cylinder direct injection and natural gas inlet port injection form a stratified mixture, which accelerates the flame propagation speed. Because the temperature in the PES40 cylinder is higher than other substitution rates, the ignition delay is shorter, so the combustion duration of PES40 is shorter, and the combustion is faster at the same advance angle. In addition, the thermodynamic state of the mixture in the cylinder, the stratification of the mixture, and the competition between the air dilution effect with the introduction of natural gas can increase the combustion rate and shorten the combustion duration [21].

4. Conclusions

This paper takes an in-cylinder direct injection turbocharged gasoline engine modified from a gasoline engine with direct injection and natural gas port injection as the research object. Under the partial load conditions, the typical combustion parameters such as in-

cylinder pressure, heat release rate, pressure rise rate, combustion temperature, ignition delay, and combustion duration of the cooperative combustion of gasoline and natural gas at different ignition timings are carried out. The main conclusions drawn from the analysis and research are as follows:

- (1) Due to the high-octane number and good anti-knock characteristics of CNG, the in-cylinder pressure, heat release rate, pressure rise rate and combustion temperature of gasoline/natural gas dual-fuel engines can be increased with the advancement of the ignition timing. Moreover, under all PES, the ignition advance angle of 28°CA-BTDC has the best overall effect and the best power performance improvement.
- (2) The change of PES makes the cylinder pressure; heat release rate and pressure rise rate of gasoline/natural gas engines decrease first and then increase with the increase of PES. PES70 is smaller than PES40, and PES40 reaches the maximum value. Therefore, the fuel synergy effect under PES40 is the best, and the comparative advantage is more obvious.
- (3) At the same ignition timing, with the increase of PES, the ignition delay shows a trend of first increasing and then decreasing; while under the same PES, as the ignition timing advances, the ignition delay shows a trend of first decreasing, then increasing and then decreasing. The ignition delay and combustion duration peak at 20°CA-BTDC-22°CA-BTDC, and both periods are longer at PES60.

In order to allow dual-fuel engines to achieve the goal of reducing emissions when using different fuels, and obtaining better fuel economy, the research in this article will develop an adaptive ignition strategy. Through adaptive adjustment of the ignition advance angle, the gasoline/natural gas dual-fuel engine can work under the best ignition advance angle, and the power performance of the engine can be improved to a certain extent.

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Nomenclature

ANN	artificial neural network
BTDC	before top dead center
CA	crank angle
CAN	controller area network
CNG	compressed natural gas
CO	carbon monoxide
CO ₂	carbon dioxide
DBN	deep belief network
DI	direct injection
DNN	deep neural network
ECU	electronic control unit
GDI	gasoline direct injection
HC	hydrocarbons
HCCI	homogeneous charge compression ignition
H/C	hydrogen-to-carbon ratio

IMEP	indicated mean effective pressure
LHV	lower heating value
NO _x	nitrogen oxides
PES	percentage of energy substitution
RON	research octane number

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