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Study on Combustion and Emissions of a Combined Injection Spark Ignition Engine with Natural Gas Direct Injection Plus Ethanol Port Injection under Lean-Burn Conditions

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ABSTRACT: Conventional ethanol spark ignition (SI) engines have poor fuel atomization and mixture formation. The objective of this paper is to improve the combustion and emission performance of ethanol SI engines under lean-burn conditions through the dual-injection mode with ethanol port injection and compressed natural gas (CNG) direct injection (CDI+EPI). This paper studies the engine performance at 1500 rpm under five CNG direct injection ratios (CDIr) and five excess air ratios (λ). The results show that as the CDIr increases under lean-burn conditions, the following occurs: the minimum advance for best torque (MBT), the coefficient of variation (CoV_{IMEP}), and CO and HC emissions decrease; the crankshaft rotation or time with cumulative heat release rate ranging from 10% to 90% (CA 10−90) and NOx emissions first decrease and then increase; and torque, peak in-cylinder pressure (Pmax), and the λ limit first increase and then decrease. The larger the CDIr is, the less influence λ has on the MBT. When CDIr = 15%, the CoV_{IMEP} can be effectively reduced, the engine can still work stably in all lean-burn conditions, and the λ limit will reach the maximum value of 1.73, 19.31% higher than that of the

original engine (CDIr = 0). When λ = 1.1, CO emissions decrease the most and HC emissions decrease the least. At this time, CO and HC emissions decrease by 1.56 vol % and 30 ppm, respectively, on average for every 0.1 decrease in λ. For CA 10−90, torque, and Pmax, λ = 1.1, 15% CDI, and 85% EPI is the optimal combination under lean-burn conditions. When CDIr ≥ 15%, NOx emissions are at an ideal level. Under lean-burn conditions, direct-injection CNG can form a good stratified natural gas/ethanol mixture in the cylinder, effectively improving the engine's power and stability and reducing emissions. The λ = 1.1, 15% CDI, 85% EPI combination provides a cutting-edge and outstanding solution for a natural gas/ethanol combined injection SI engine.
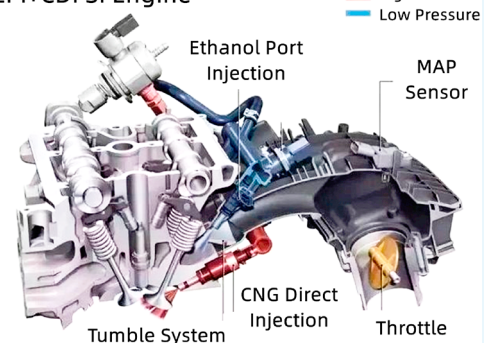
1. INTRODUCTION

With the aggravation of environmental pollution and the increasing risk of energy shortages, the development of alternative fuels provides a solution to these crises. In the process of development, many alternative fuels have entered people's vision, and there have been many research on alternative fuels, such as liquefied petroleum gas (LPG),[1,2] compressed natural gas (CNG),[3,4] methanol,[5,6] ethanol,[7−9] hydrogen,[10,11] etc. Among of these, ethanol fuel is a renewable energy, and the manufacturing process is relatively mature. It has been popularized and used in many countries and so has become one of the mainstream alternative fuels. Due to its huge storage capacity and superior combustion characteristics, CNG has become a research direction for alternative fuels. In addition, it can be realized as an auxiliary fuel through the combined injection system to improve engine performance.

1.1. Dual-Fuel Combined Injection. The dual-fuel combined injection system has been regarded as an effective way to improve most aspects of engine performance. The combined injection system divides the fuel consumed by the

engine in each cycle into two parts. One part adopts the method of port fuel injection to form the homogeneous and lean mixture, and the other part injects the fuel directly into the cylinder for local enrichment, so as to form a stratified mixture. The ideal stratified mixture can improve the power and emissions of the engine. Thomas Wallner et al.[12] studied the distribution of the mixture and the influence of the mixing process on the whole combustion process in an optical engine with a gasoline port injection and a hydrogen direct injection. The experimental results show that different in-cylinder mixture distributions can be realized by changing the direct injection strategy, which can bring the advantages of the two fuels into full play. Huang et al.[13,14] researched the characteristics of an ethanol−gasoline

combined injection engine. The results show that, compared with a pure-gasoline engine, ethanol−gasoline engines have less heat loss during the combustion process. Additionaly, they also found that later ethanol injection times increased HC emissions but reduced NO and CO emissions. Yu X et al.[15−18] deeply studied the effect of using hydrogen as an auxiliary fuel on the performance of an engine equipped with a dual-fuel combined injection system. The results show that a stratified mixture will form in the cylinder after adding hydrogen, which will improve the combustion and significantly reduce the CO and HC emissions. However, NOx emissions will increase.

1.2. Lean-Burn. The excess air ratio (λ) is one of the key parameters that characterize engine operating conditions. It represents the concentration state of the mixture in the cylinder; the greater the excess air ratio, the thinner the air−fuel mixture. At present, there has been more and more research on the lean-burn characteristic of engines. Lean-burn has great potential to imrpove engine emission, thermal efficiency, and fuel consumption. Ji et al.[19] studied engine combustion and emission performance under the lean-burn condition, and the results show that NOx emissions decrease significantly with the increase of λ due to the average decrease in temperature. Yu X et al.[20−23] did a lot of research on the dual-fuel combined injection engine under the lean-burn condition. The results show that when engine runs in the pure gasoline mode under the lean-burn condition, the emissions performance improves significantly and NOx, HC, and CO emissions decrease, however, the power of the engine decreases at this time. What's more, the coefficient of variation (CoV) increases, deteriorating the engine stability. However, the researchers also put forward effective solutions. Some auxiliary fuels such as H_2 and ethanol have been applied in the engine through the dual-fuel combined injection system, and the results show that the CoV and the power performance significantly improved after the addition of the auxiliary fuels. Meanwhile the emissions decreased further.

1.3. The Use of Ethanol in Engines. Alcohol fuels account for the largest component of alternative fuels and also have the most potential.[24] Most alcohol fuels can be made from renewable energy, which can alleviate the pressure of the rapid consumption of fossil fuels. At present, the research on alcohol fuels is mainly focused on methanol, ethanol, and butanol. As the most widely used alternative fuel, ethanol has the characteristics of a high-octane number, a fast burning speed, and a high oxygen content compared with gasoline. In addition, the current ethanol extraction technology is very mature, and continuous technical improvement has decreased its manufacturing cost. Therefore, the use of ethanol as an alternative fuel has a high practical value.[25,26]

Hsieh et al.[27] studied the performance of an ethanol−gasoline engine with a 0−30% ethanol blending ratio. As shown by the results, when the amount of ethanol increases, the torque increases slightly and the emissions of CO and HC decrease significantly. Bahattin et al.[28] studied engine performance under different ethanol blending and compression ratios. The test compared the data results of two compression ratios, namely 6:1 and 10:1, and five proportional fuels, namely E0, E25, E50, E75, and E100. In terms of power performance, when the throttle is fully open, the torque first increases and then decreases as the amount of ethanol increases, and the maximum value is obtained at E50. As the amount of ethanol increases, the emissions of CO and NOx decrease continuously, and HC emissions first decrease and then increase. Jia et al.[29] compared engine

performance using E10 fuel and unleaded gasoline in a motorcycle engine. The results show that the HC and CO emissions of the E10 fuel decrease, while the NOx emissions exhibit no obvious change. Maji et al.[30] studied the power performance, specific fuel consumption, and emissions characteristics of an engine using E15 fuel, E85 fuel, and pure gasoline in the United States. As shown by the results, when λ changes from 0.7 to 1.3, the torque of the engine using the ethanol−gasoline mixed fuel is generally higher and the CO and HC emissions are lower.

1.4. The Use of CNG in Engines. As an alternative fuel with rich reserves, CNG has a high research value. There has been much research on the use of CNG. There have been many experiments exploring the emission properties of the engine when gasoline and CNG dual-fuel is used in taxis. The research results show that the engine emissions have a great relationship with the speed. Additionally, the exhaust emissions of engine using CNG, such as CO and CO_2, are less than those of a gasoline engine, but the HC and NOx emissions are relatively high.[31] Singh et al.[32] did a lot of research on CNG−gasoline engines, and the results show that increasing the amount of CNG leads to a decrease in the combustion temperature, the torque, and the specific fuel consumption. In addition, there are many studies on the performance of CNG used as an auxiliary fuel in the engine. The research show that using CNG can effectively improve the combustion and the thermal efficiency and reduce the occurrence of knocking in an engine.[33−38] Chen et al.[39] compared the performances of different types of dual-fuel engines, such as CNG−methanol, CNG−ethanol, and CNG−butanol. The test engine was run at a fixed speed of 1600 r/min under light load, and the average effective pressure was fixed at 0.387 MPa. The experimental results show that alcohol fuels can cooperate well with CNG, and their synergistic effect can effectively reduce HC and NOx emissions and improve the thermal efficiency. Ran et al.[40] conducted a series of tests on an engine to study the lean combustion performance of the engine using three different alternative fuels: natural gas, ethanol, and a H_2−CO mixture. The experimental results show that the engine that used natural gas as the fuel had better combustion and emission performance under the lean-burn condition and a wider lean-burn limit. Liu et al.[41] studied the lean-burn characteristics of natural gas direct injection plus gasoline port injection (NDI+GPI) engines with increasing RCNG (direct injection ratio of CNG) at direct injection time (DIT) temperatures of 90−150 °CA before compression top dead center (BTDC). However, the optimal DIT of the NDI+GPI engine is not given.

1.5. Research Content and Characteristics. Although it can reduce the consumption of fossil fuels, using ethanol as the main fuel in a spark ignition (SI) engine will reduce the engine performance, especially in the lean-burn condition. To maintain the better performance of an ethanol SI engine and continue to reduce emissions under various working conditions, CNG direct injection was introduced in this paper to solve the above problems. Fortunately, ethanol and CNG both have good lean-burn characteristics, and CDI CNG direct injection (DCI) and EPI (ethanol port injection) can compensate for each other's negative effects on combustion and emissions. Therefore, it is appropriate to study the synergistic effect of CDI and EPI under the lean-burn conditions. Previously, we have studied the combustion and emission characteristics of CDI plus EPI (CDI+EPI) engines with the increase of the CDI ratio (CDIr) when the DIT was between 120 and 180 °CA BTDC under the same

conditions of DIP, speed, load, and λ. The optimal DIT of a CDI+EPI engine is 150 °CA BTDC.[42] However, with the increasingly stringent requirements of emission regulations, the pollutant emissions (CO, HC, and NOx) of the CDI+EPI engine can be further reduced. On the basis of previous studies, this paper continues to study the variation of combustion and emission characteristics of a CDI+EPI engine with the increase of the CDIr under the same DIP, speed, load, and DIT conditions when λ = 0.9, 1, 1.1, 1.2, 1.3, and 1.4. In previous papers, there has never been a study on both the lean-burn and the rich-burn of a CDI+EPI engine. This provides a new direction for the combustion and emission control strategy of a CNG−ethanol composite combustion engine. Therefore, the research results of this experiment are innovative and have high practical value.

2. EXPERIMENTAL SETUP AND PROCEDURE

2.1. Experimental Setup. In this study, we conducted various tests on a four-cylinder water-cooled engine to obtain experimental data. In addition, we partially modified the fuel supply system so that it could achieve compressed nature gas direct injection and allow ethanol to be supplied through the original port injection system. The detailed technical parameters of the engine are listed in Table 1. The detailed schematic diagram of the experimental device is shown in Figure 1.

Table 1. Main Technical Parameters of the Engine

item	characteristics
type	spark ignition, in-line, four cylinders, combined injection
displacement	1.984 L
bore × stroke	82.5 × 92.8 mm
compression	9.6:1
maximum torque	320 N·m (1600−4000 rpm)
maximum power	137 kW (5000 rpm)
ignition sequence	1−3−4−2

2.2. Test Instruments. Table 2 lists the measurement instruments and their resolutions. A CW160 eddy current dynamometer produced by CAMA was used in this experiment. With the corresponding control system, the dynamometer can monitor and collect the torque, speed, power, and other parameters of the engine in real time on the upper computer. The oxygen sensor installed in the engine exhaust pipe in this test was an ETAS LA4 sensor manufactured by ETAS. The corresponding values were collected and displayed in real time. For the analysis of exhaust emissions, an AVL Dicom4000 analyzer was used in this paper, as it could accurately and quickly measure the amount of HC, CO, NOx, O_2, and CO_2 in engine exhaust.

2.3. Experimental Procedure. This paper mainly explores the combustion performance and emission characteristics of a CNG−ethanol combined injection engine. The engine fuel supply system adopted the mode of ethanol port injection plus compressed natural gas direct injection. This research further explores the engine performance under different excess air ratio.

In this study, the calculation method for the excess air ratio is shown in eq 1.

$$\lambda = \frac{Q_{air}}{Q_{ethanol} \cdot AF_{ethanol} + Q_{CNG} \cdot AF_{CNG}} \tag{1}$$

where Q_{air} denotes the air mass flow, $Q_{ethanol}$ denotes the ethanol mass flow, and Q_{CNG} denotes CNG mass flow. The mass flow was measured in kilograms per hour. $AF_{ethanol}$ and AF_{CNG} denote the theoretical air−fuel ratios of ethanol and CNG, respectively.

The control methods for the different fuel injection ratios in this study are as follows. Under a certain researched λ, the engine first operated in the ethanol fuel mode alone, and the corresponding fuel consumption was measured as a benchmark. Then, different ethanol/CNG ratios were achieved by reducing the amount of ethanol and expanding the content of compressed natural gas until λ returned to the specific value.

Because ethanol's calorific value is different from that of CNG, the influence of the above control method on the calorific value of the mixed fuel is not very clear. Therefore, the following analysis is based on whether the total energy of mixed fuel changes. The total energy formula of mixed fuel was obtained from eq 2.

$$Q_{total} = \frac{H_u(ethanol) \cdot Q_{air} \cdot EPIr}{AF_{ethanol}} + \frac{H_u(CNG) \cdot Q_{air} \cdot CDIr}{AF_{CNG}} \tag{2}$$

where $H_u(ethanol)$ denotes low heat value of ethanol and $H_u(CNG)$ denotes low heat value of CNG; both were measured in meajoules per kilogram. EPIr denotes the proportion of ethanol, and CDIr denotes the proportion of CNG.

To more intuitively analyze the change of the calorific value of the mixture after the proportion of CNG was increased, the energy change rate was calculated using eq 3.

$$a = \frac{\dfrac{H_u(ethanol) \cdot Q_{air}}{AF_{ethanol}} - \dfrac{H_u(ethanol) \cdot Q_{air} \cdot EPIr}{AF_{ethanol}} + \dfrac{H_u(CNG) \cdot Q_{air} \cdot CDIr}{AF_{CNG}}}{\dfrac{H_u(ethanol) \cdot Q_{air}}{AF_{ethanol}}} \tag{3}$$

Through the calculation of eq 3, the value of a under different proportions of CNG can be obtained. It was found that the total energy change rate can be maintained within 1.05%; detailed data are shown in Table 3.

The compressed nature gas direct injection ratio was calculated using eq 4.

$$CDIr = \frac{M_{CNG} \times H_u(CNG)}{M_{CNG} \times H_u(CNG) + M_{ethanol} \times H_u(ethanol)} \tag{4}$$

where M_{CNG} and $M_{ethanol}$ respectively denote the mass (kg) of compressed natural gas and ethanol, respectively.

The CoV is a crucial parameter for evaluating the stability of an engine.[43,44] The relevant calculation formulas are shown as follows:

$$CoV_x = \frac{\sigma_x}{\bar{x}} \times 100\% \tag{5}$$

where

$$\bar{x} = \frac{\sum_{i=1}^{N} x_i}{N} \tag{6}$$

$$\sigma_x = \sqrt{\frac{\sum_{i=1}^{N} (x_i - \bar{x})^2}{N}} \tag{7}$$

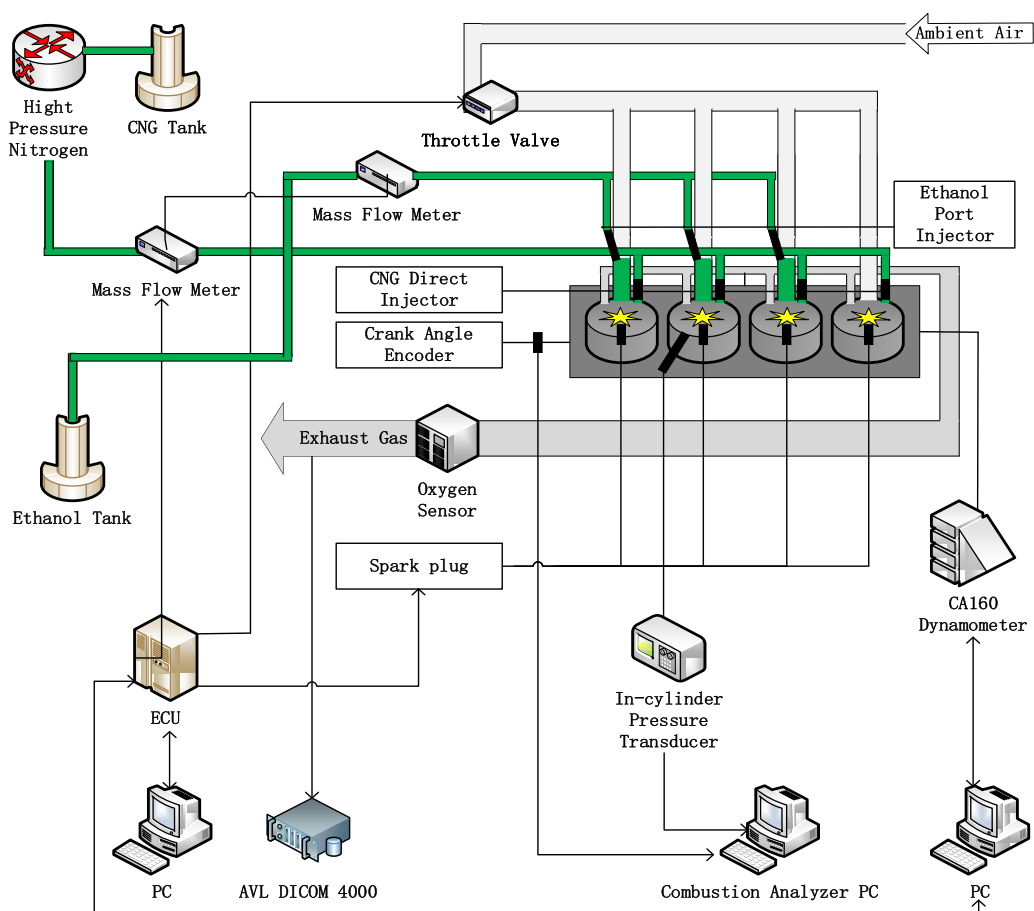where x denotes IMEP and N denotes the number of the sampling test cycles.

Figure 1. Schematic diagram of the experimental setup.

Table 2. Test Equipment of the Experiment

parameters	type	precision	range
torque	CW160	$\leq \pm 0.28$ N·m	0−600 N·m
speed	CW160	$\leq \pm 1$ r/min	0−6000 r/min
cylinder pressure	AVL-GU13Z-24	$\leq \pm 0.5\%$	0−20 MPa
crank angle	Kistler-2614B	$\leq \pm 0.5°$	0−720°
excess air rate (λ)	LAMBDA LA4	$\leq \pm 0.1$	0.700−32.767
carbon monoxide (CO)	AVL-DICOM 4000	$\leq \pm 0.01\%$	0−10% vol
hydrocarbon (HC)	AVL-DICOM 4000	$\leq \pm 1$ ppm	0−20000 ppm
nitrogen oxides (NOx)	AVL-DICOM 4000	$\leq \pm 1$ ppm	0−5000 ppm
mass flow meter	DF-2420	$\leq \pm 0.01$ g/s	0.21−82 kg/h

2.4. Operating Conditions. During the test, the relevant test parameters set to ensure the stable operation of the engine include the temperature of cooling water, which is stable at 80 ± 1.5 °C, and the temperature of engine lubricating oil, which is stable at 90 ± 5 °C. To ensure that the mixture could burn well and the engine could obtain a large torque, the injection timing of the ethanol port injection was set at 300 °CA BTDC and the injection pressure was fixed at 0.5 MPa. The CNG direct injection pressure was 5 MPa. The ignition timing iwas s fixed at the maximum brake torque timing of the test point. Detailed operating conditions are shown in Table 4. Over the entire test

Table 4. Summary of Operating Conditions

engine speed (r/min)	1500
intake manifold pressure (kPa)	48
λ	0.9, 1.0, 1.1, 1.2, 1.3
ignition timing (°CA BTDC)	maximum brake torque timing (MBT)
CNG injection timing (°CA BTDC)	150
ethanol injection timing (°CA BTDC)	300
CNG injection pressure (MPa)	5
ethanol injection pressure (MPa)	0.5
CNG direct injection ratio (%)	0, 5, 10, 15, 20
dual-injection mode	CDI+EPI

process, the combustion analyzer will collect 200 cycles of data at each working condition point for an average analysis. This multicycle acquisition method can greatly reduce the test error caused by unstable engine work and increase the test repeatability and reliability. At the same time, the engine should

Table 3. Total Energy and the Energy Change Rate

CDIr	0%	5%	10%	15%	20%
Q_{total} (KJ)	$460.44Q_{air}$	$458.87Q_{air}$	$458.14Q_{air}$	$456.99Q_{air}$	$455.84Q_{air}$
a (%)	0	0.25	0.5	0.55	1.05

work stably for 2 min during all the tests, and multiple groups of records and average values should be taken after the emission values of the tail gas analyzer are stabilized so as to ensure the accuracy and stability of the test data.

3. RESULTS AND DISCUSSION

3.1. Combustion Characteristics. *3.1.1. Minimum Advance for Best Torque (MBT).* The minimum advance for best torque (MBT) is the minimum ignition advance angle when the engine has maximum power and minimum fuel consumption. It is mainly related to engine speed, load, fuel type, cylinder temperature, cylinder pressure, λ, and other influencing factors. Figure 2 shows the variation of the CDIr on the MBT at different
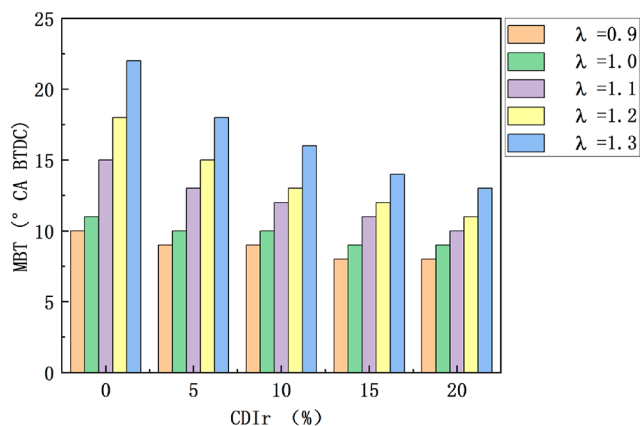

Figure 2. MBT versus the CDIr at different values of λ.

values of λ. As shown in the Figure. 2, when the CDIr is constant, the MBT increases with the increase of λ. This is mainly because increasing λ will increase the fresh air in the cylinder and decrease the mixture concentration, leading to the nonobvious stratification of the combustible mixture and a decrease in the combustion speed of the fuel. At this time, increasing the ignition advance angle can give more time to the natural gas/ ethanol mixture combustion after the spark plug ignition and make the Pmax in the engine cylinder close to 10−12 °CA ATDC of the compression stroke, increasing the MBT. Although the MBT increases with the increase of λ, this increasing trend slows down with the increase of the CDIr. This is mainly because the direct injection of more CNG effectively reduces the negative effects of lean-burn on in-cylinder mixture formation and combustion. At the same time, CNG's good lean-burn stability enables the mixture to burn stably at a relatively small ignition advance angle. The larger the CDIr is, the less influence λ has on the MBT.

In addition, the MBT decreases with the increasing CDIr when λ is constant. Addtionally, with the increase of λ, the MBT decreases more obviously. This is mainly because of the longer the time required to form the stratified mixture at the appropriate concentration in the cylinder when more CNG is injected directly and the corresponding decrease in the MBT. When $\lambda \leq 1$, the natural gas/ethanol mixture is thicker, and it is easier to ignite the spark plug . The increase of the CDIr has little influence on the formation of the stratified mixture, so the decrease of the MBT is small. However, when $\lambda > 1$, the natural gas/ethanol mixture becomes thinner and thinner, and the gradual increase of λ prolongs the mixing time of the mixture more obviously, so the MBT decreases more obviously. At the same time, under the condition of lean-burn, directly injecting

more CNG can more easily form a good stratified mixture for diffusion combustion with the ethanol sprayed by the inlet in the cylinder and effectively accelerate the combustion speed of the mixture, thus increasing the reduction of the MBT. The larger λ is, the more influence the CDIr has on the MBT.

3.1.2. Crank Angle (CA 10−90). Crank angle (CA) 10−90 refers to crankshaft rotation or time with a cumulative heat release rate ranging from 10% to 90%. In the CA 10−90 stage, most of the fuel in the engine cylinder will burn quickly, reflecting the combustion of the engine to a large extent. Figure 3
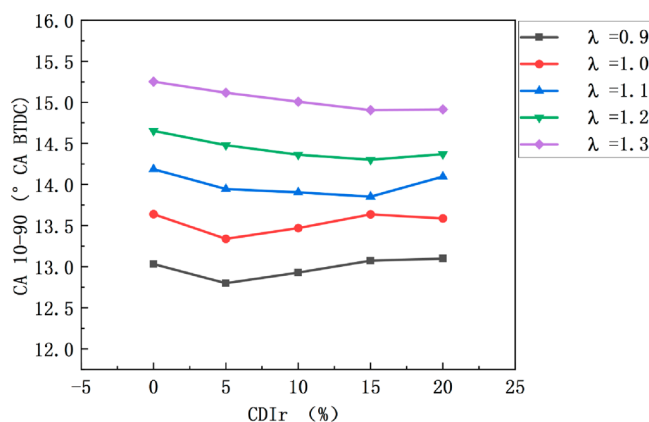

Figure 3. CA 10−90 versus the CDIr at different values of λ.

shows the effect of the variation of the CDIr on CA 10−90 at different values of λ. As shown in the Figure 3, when the CDIr is constant, CA 10−90 extends with the increase of λ. This is due to the fact that as the gas/ethanol mixture becomes thinner and thinner, the flame can no longer maintain good penetration and its propagation speed decreases significantly, which in turn lengthens CA 10−90.

In addition, when λ is constant, CA 10−90 shortens first and then lengthens with the increase of the CDIr. This is because the direct injection of appropriate amount of CNG will form a mixture stratification state. With an appropriate concentration in the cylinder, the fuel combustion speed is accelerated and the combustion environment in the cylinder is optimized, so CA 10−90 decreases. However, with the continuous increase of the CDIr, the excessive direct injection of CNG will cause the mixture in local areas to be too thick and the good stratified combustion mode of the mixture will be destroyed, so CA 10− 90 will rebound. At the same time, because ethanol is an oxygen-containing fuel, and its LPS is larger than that of CNG, the direct injection of CNG will greatly reduce the oxygen content and flame propagation speed of the mixture in the cylinder, which will lead to an increase in CA 10−90. In particular, under the condition of $\lambda \leq 1$, the valleys of CA 10−90 all appear when CDIr = 5% at 12.80 °CA and 13.34 °CA, respectively, which are 1.80% and 2.21% lower than the pure EPI mode. Under the condition of $\lambda > 1$, the valley values of CA 10−90 all appear when CDIr = 15% at 13.85, 14.30, and 14.91 °CA, respectively, which are 2.36%, 2.39%, and 2.26% higher than the original machine. It can be seen that the optimum CDIr for CA 10−90 lengthens with the increase of λ. This is because 5%CDI can form a good stratified combustion environment of the natural gas/ethanol mixture in the cylinder when the mixture is relatively thick. At the same time, 5% CDI can decrease the loss of the oxygen content caused by the reduction of the ethanol proportion as much as possible. As λ continues to increase, the oxygen content

of the mixture is no longer an important factor hindering combustion, but the concentration of the mixture continues to decrease. To maintain a good stratified combustion environment, the direct injection of more CNG is needed. Therefore, the optimum CDIr of CA 10−90 goes from 5% to 15%. In summary, for CA 10−90, λ=1.1, 15% CDI, and 85% EPI is the optimal combination under lean-burn conditions.

3.1.3. Torque and Peak In-Cylinder Pressure (Pmax). Figures 4 and 5 show effect of the variation of the CDIr on
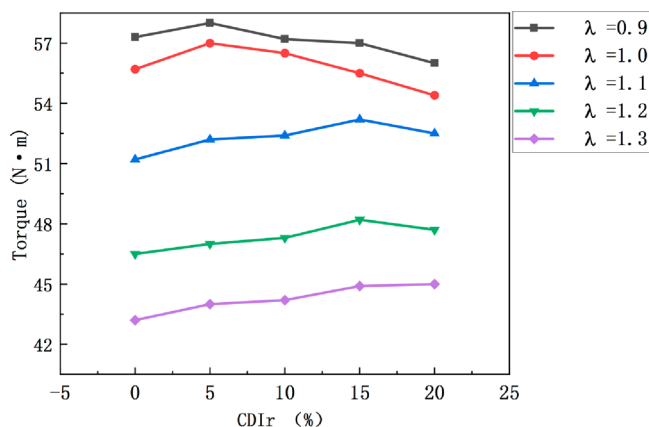

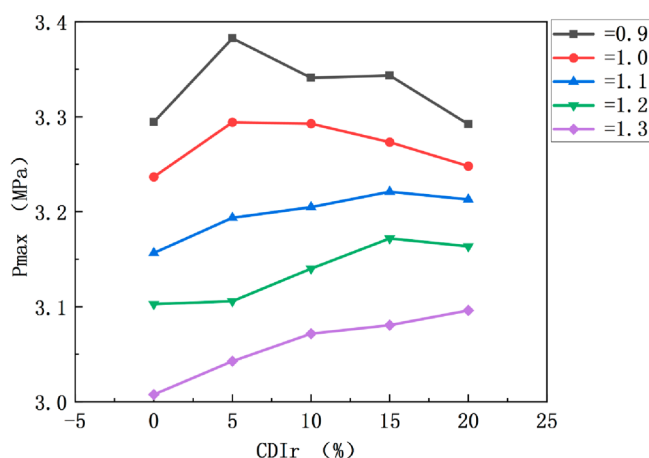**Figure 4.** Torque versus the CDIr at different values of λ.

Figure 5. Pmax versus the CDIr at different values of λ.

the torque and Pmax, respectivelyat different values of λ. As shown in the Figures 4 and 5, when the CDIr is constant, both torque and Pmax reduce with the increase of λ. This is mainly because an increase in λ reduces both the concentration of the fuel mixture in the cylinder and the heat released by fuel combustion. The in-cylinder combustion gradually deteriorates, resulting in a decrease in engine power that reduces both the torque and Pmax. At this moment, the increase of λ makes the natural gas enrichment area that forms near the spark plug become thinner and thinner, the fuel-stratified combustion state formed by CDI+EPI in the cylinder begins to deteriorate, and the torque and Pmax both decrease. It is worth noting that the torque remains at a high level when $\lambda \leq 1$. It can be seen that the thicker mixture is more conducive to the stratified combustion of natural gas and ethanol, thus improving the power performance of the engine.

In addition, the torque and Pmax both increase first and then decrease with the increase of the CDIr when λ is constant.

Combined with the above analysis of CA 10−90, a direct injection of the appropriate amount of CNG will lead to the stratified combustion of the CNG-rich area near the spark plug and the homogeneous ethanol surrounding it. Compared with the homogeneous combustion of pure ethanol, the stratified combustion mode can further improve the combustion state of the engine, which increases both the temperature and pressure during the combustion in the cylinder and then increases the torque and Pmax of the engine. However, as the CDIr continues to increase, the torque and Pmax start to decrease. The main reason is that the LPS of CNG is less than that of ethanol, and the continued injection of CNG will lead to a decrease in the flame propagation speed throughout the cylinder. This negative influence on the flame propagation even exceeds the promotion effect of fuel stratification on combustion and finally leads to a decrease in the torque and Pmax.

It is worth noting that, similar to the variation of CA 10−90, the optimal CDIr the torque and Pmax also increases with λ. This is mainly because in the lean-burning condition more CNG needs to be injected directly into the in-cylinder mixture to form a CNG-rich zone near the spark plug and achieve the stratified combustion of the mixture. At this moment, the continuous stratified combustion can efficaciously reduce the negative impact of lean combustion on the engine power performance. Therefore, as the mixture gets thinner, the optimal CDIr increases from 5% to 20%. Furthermore, under the condition of $\lambda > 1$, the engine torque and Pmax after the direct injection of CNG are always higher than the torque of the original engine. This indicates that although the torque of CDI+EPI decreases under the condition of lean burn, CNG has good lean-burning characteristics, and the disadvantage of a slower LPS will be reduced when the mixture is relatively thin. On the contrary, the direct injection of more CNG can make up for the heat release loss caused by lean-burning. Overall, the power performance of the engine is optimal at λ=1.1, 15% CDI, and 85% EPI in all lean-burn conditions.

3.1.4. Coefficient of Variation (CoV). CoV_{IMEP} is the main parameter used to evaluate the combustion stability and driveability of vehicles. It is generally believed that CoV_{IMEP} should not exceed 5%. Overall, the injection mode of CDI+EPI keeps the engine running within a stable range. However, CoV_{IMEP} still changes in lean-burning conditions. Therefore, this paper analyzes the influence of λ and the CDIr on CoV_{IMEP}. As shown in Figure 6, when the CDIr is constant, CoV_{IMEP} increases with λ. This is because the concentration of the
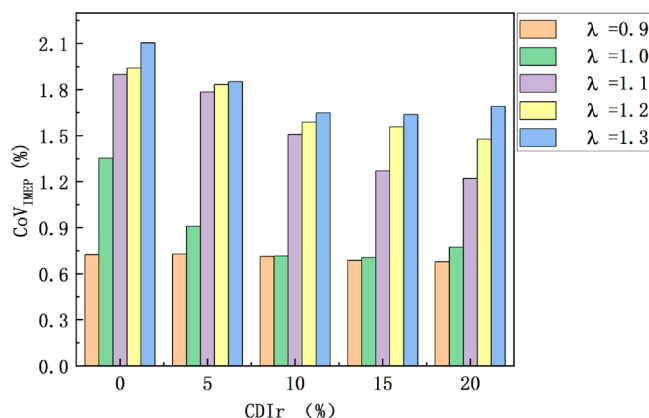

Figure 6. CoV_{IMEP} versus the CDIr at different values of λ.

combustible mixture in the cylinder will decrease with the increase of λ. The CNG originally enriched near the electrode gap of the spark plug becomes uneven during ignition, which affects the formation of the early flame center and later flame development. At the same time, there will be misfire in the cylinder, which makes the combustion situation in the cylinder worse and then increases CoV_{IMEP}. It is worth noting that because the mixture is richer the higher combustion temperature and pressure in the cylinder make the flame propagation speed faster, which in turn reduces CoV_{IMEP}. In general, when $\lambda=0.9$, CoV_{IMEP} is not sensitive to the size of the CDIr and always remains at about 0.7%.

In addition, CoV_{IMEP} decreases with the increase of the CDIr when λ is constant. This is mainly because the direct injection of more CNG can efficaciously improve the stratified combustion state of the mixture and make the engine more stable. At the same time, the direct injection pressure is as high as 9 MPa, and the high-pressure injection of more CNG strengthens the in-cylinder vortex intensity to a certain extent, thus speeding up the flame development and propagation speed and the fuel combustion speed and reducing CoV_{IMEP}. However, with the increase of λ, the decreasing range of CoV_{IMEP} gradually decreases. Even in the case of $\lambda=1.3$, CoV_{IMEP} shows a slight increase when CDIr > 15%. According to previous studies, the change rule of CoV_{IMEP} is related to the change of CA 10−90. It is known from a previous study on CA 10−90 that when $\lambda=1.3$, excessive CNG will lead to excessive mixture concentrations in local areas as the CDIr increases. This will destroy the good stratified combustion mode of the mixture and greatly reduce the oxygen content of the mixture, making the engine unstable. At the same time, the negative side of lean-burning begins to trump the good lean-burning characteristics of CNG, slowing the flame propagation and resulting in an increase in CoV_{IMEP}. In general, when CDIr = 15%, CoV_{IMEP} can be effectively reduced and the engine can still work stably in the lean-burn conditions.

3.1.5. Excessive Air Coefficient Limit (λ Limit). The λ limit is of guiding significance to the optimization of operating points. In this paper, CoV_{IMEP} > 5% was defined as the criterion to determine the attainment of the λ limit. To explore the λ limit of a CDI+GPI engine, the experimental λ range was extended to 1.8. It was found that the engine jitter was very strong and CoV_{IMEP} is far more than 5% when λ exceeded 1.8. As can be seen from Figure 7, the λ limit increases first and then decreases

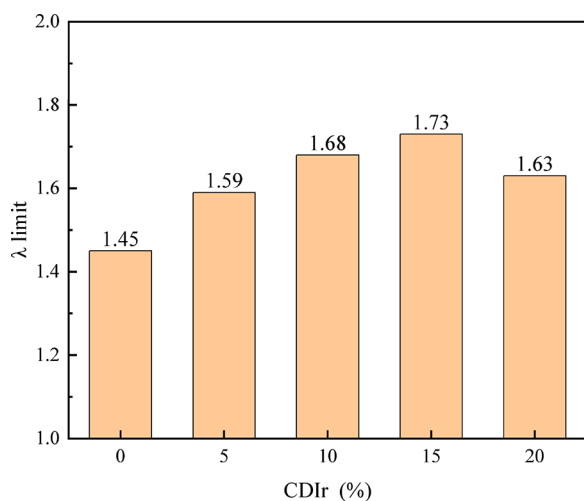**Figure 7.** λ limit versus the CDIr.

with the increase of the CDIr. Combined with the previous study on CA 10−90 and Pmax, it can be seen that the simultaneous increase of λ and the CDIr does not lead to a decrease of the oxygen content in the cylinder mixture but does enable the continuous distribution of enriched CNG around the spark plug. This maintains the stratified combustion environment of the CNG−ethanol mixture in the cylinder, and the engine stability is always relatively good. Therefore, when CDIr = 15%, the engine can tolerate a λ increase to the maximum value of 1.73 under the condition of stable operation, and the λ limit increases by 19.31% compared with that of the original engine (CDIr = 0). In the process of the CDIr increasing from 0 to 15%, the λ limit is positively correlated with the CDIr, which fully reflects the good lean-burn characteristics of CNG. However, with the continuous increase of the CDIr, although increasing λ can compensate for the decrease of the oxygen content in the mixture caused by the decrease in the ethanol injection volume, increasing λ can not fully compensate for the shortcomings of the low LFS of CNG. A too-thin mixture also has a negative effect on the flame propagation speed, which can lead to combustion deterioration in the cylinder. Therefore, it is necessary to improve the propagation speed of the mixed flame by reducing λ and restoring the stability of the engine. It is worth noting that the λ limit is negatively correlated with the CDIr in the process of the CDIr increasing from 15% to 20%. The λ limit decreases slightly but still increases by 12.41% compared with that of the original machine (CDIr = 0).

3.2. Gaseous Emissions. *3.2.1. CO Emissions.* Figure 8 shows the effect of the variation of the CDIr on CO emissions at
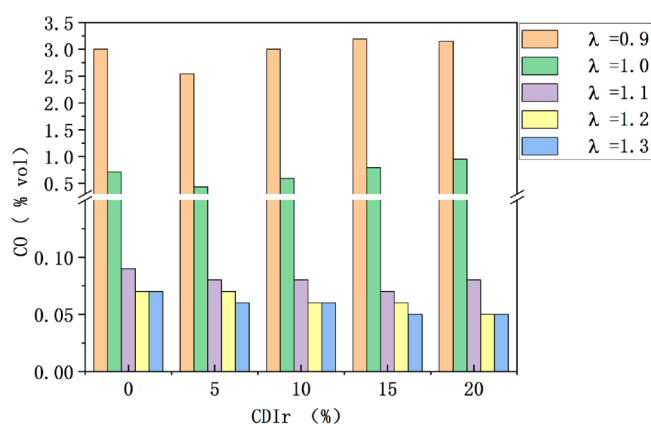

Figure 8. CO emissions versus the CDIr at different values of λ.

different values of λ. As shown in Figure 8, when the CDIr is constant, CO emissions decrease with the increase of λ. This is mainly because the increase of λ adds the partial pressure of oxygen to the mixture, decreases the proportion of carbon fuel, and hinders the generation of CO. At the same time, CO will be further oxidized to CO_2. It is noteworthy that the larger λ is, the smaller the reduction of CO emissions is. This is because most of the CO produced by fuel combustion has been oxidized to CO_2 by the increased amount of oxygen at $0.9 < \lambda < 1.1$, although a larger λ increases the oxygen content of the mixture in the cylinder. At the same time, the negative effect on combustion of increasing λ also hinders the further oxidation of some CO. Under all lean-burn conditions, when $\lambda=1.1$ and CDIr = 15%, the reduction of CO emission is the largest, and the average reduction of CO emission is 1.56 vol % for every 0.1 amplification of λ.

In addition, with the increase of the CDIr, CO emissions first decrease and then increase when $\lambda \leq 1$; however CO emissions continue to reduce when $\lambda > 1$. Combined with the above analysis of CA 10−90, when the mixture is thick, the direct injection of 5% CNG will cause the stratified combustion of the CNG-rich area near the spark plug and the homogeneous ethanol around it, which will increase the cylinder temperature during combustion and then reduce CO emissions. With the continuous increase of the CDIr, ethanol, as a high oxygen content fuel, is greatly reduced and CO cannot continue to be oxidized to CO_2, so CO emissions increase, even beyond the level of the original machine. However, this phenomenon does not occur under lean-burn conditions. Because of the increase of λ, there is enough oxygen in the mixture for the further oxidation of CO. The cylinder can continue to maintain the high-temperature and oxygen-rich stratified combustion state, so the CO emissions continue to decrease.

3.2.2. HC Emissions. Figure 9 shows the effect of the variation of the CDIr on HC emissions at different values λ. As shown in
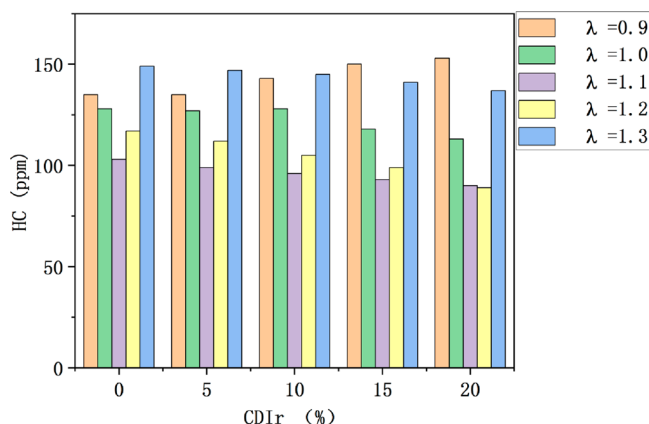

Figure 9. HC emissions versus the CDIr at different values of λ.

the Figure 9, when the CDIr is constant, HC emissions first decrease and then increase with the increase of λ. When the combustible mixture is thicker, the fuel cannot be completely burned, and HC emissions are produced in large quantities. As λ increases to 1.1, the good lean-burn characteristics of CNG come into play. Meanwhile, the stratified mixture of natural gas/ethanol can be fully burned as far as possible to further oxidize unburned HC, thus reducing HC emissions. However, with the continuous increase of λ, more and more missing fire cycles (large volume quenching and misfire) appear in the cylinder and part of the fuel is discharged without combustion, leading to a sharp rise in HC emissions.

In addition, with the increase of the CDIr, HC emissions increase when $\lambda = 0.9$; however, HC emissions all decrease when $\lambda \geq 1$. Especially when $\lambda = 1.1$ and 1.2, HC emissions decrease the most. When the mixture is thick, the direct injection of more CNG not only fails to completely burn the mixture but also decreases the oxygen content of the mixture due to the reduction of ethanol, which leads to the failure of further HC oxidation and an increase in HC emissions. However, the oxygen supply in the tank is abundant under all lean-burn conditions. The higher latent heat of vaporization of ethanol will lead to a lower wall temperature, thus aggravating wall-quenching effect and the narrow gap effect. Therefore, the reduction of ethanol can effectively reduce HC emissions caused by these two effects. At this moment, when $\lambda = 1.1$ and 1.2, ethanol and CNG form a

better stratified combustion mixture, and the unburned HC is further oxidized. On the whole, HC emissions are the lowest under the condition of $\lambda = 1.1$, and the reduction amplitude increases with the CDIr. When CDIr $\geq 15\%$, the average reduction of HC emissions is 30 ppm for every 0.1 amplification of λ. For HC emission, $\lambda = 1.1$ and CDIr $\geq 15\%$ is the optimal combination.

3.2.3. NOx Emissions. NOx emissions are mainly composed of thermal NO generated in the high-temperature region after flame. According to the extended Zeldovitch mechanism,[45,46] the main factors that affect thermal NO generation are the oxygen concentration in combustion products and the combustion temperature of the mixed gas. Figure 10 shows
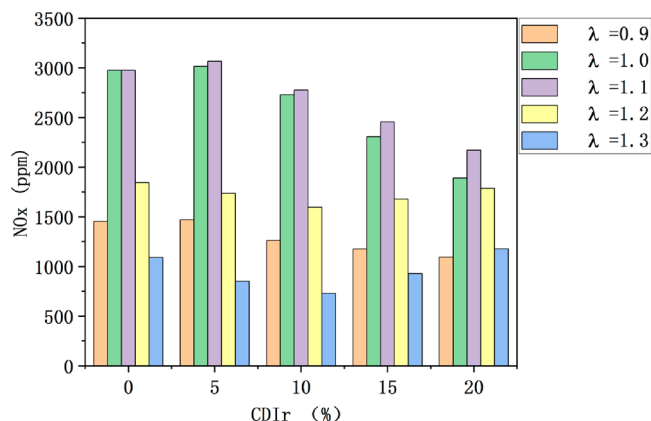

Figure 10. NOx emission versus CDIr at different values of λ.

the effect of the variation of the CDIr on NOx emissions at different values of λ. As shown in Figure 10, when the CDIr is constant, NOx emissions increase first and then decrease with the increase of λ. When the combustible mixture is thick, although the temperature of the burned gas in the cylinder is high, the oxygen concentration of the burned gas is very low, which inhibits the generation of NOx. When λ is increased to 1 or 1.1, the oxygen concentration increases rapidly. The stratified combustion of the CNG and ethanol mixture effectively obstructs the reduction of the cylinder temperature. At this time, the effect of the increasing oxygen partial pressure overcomes the effect of the decreasing temperature, leading to a significant increase in NOx emissions. However, with the further increase of λ, the stratified combustion state is destroyed, and the trend of temperature decline cannot be retrieved. At this time, the effect of temperature reduction overcomes the effect of the oxygen partial pressure increase, leading to the reduction of NOx emissions.

In addition, when $\lambda \leq 1.1$, NOx emissions increase first and then decrease with the increase of the CDIr. However, when $\lambda > 1.1$, NOx emissions decrease first and then increase with the increase of the CDIr. The reduction of NOx emissions is mainly due to the continuous reduction of ethanol as an oxygen-containing fuel due to the increase of the CDIr, which increases the reduction effect of the oxygen partial pressure of the mixture. The increase in NOx emissions is mainly due to the stratification of the mixture formed by CNG and ethanol, which optimizes the combustion in the cylinder and increases the temperature of the burned gas. The two factors always conspire against each other. When $\lambda \leq 1$, 5% CDI and 95% EPI can form the best mixed stratified combustion. At this time, the effect of the temperature increase overcomes the effect of the oxygen partial pressure

reduction, so NOx emissions increase. However, at $\lambda > 1.1$, the rarer mixture requires more CNG to maintain the good stratified combustion of the mixture, thus allowing the temperature increase effect to remain dominant. Therefore, the direct injection of a large amount of CNG at this time will lead to an increase in NOx emissions. Overall, under all λ conditions, when CDIr $\geq$ 15%, NOx emissions are at an ideal level.

4. CONCLUSIONS

In this test, by changing CDIr and a, the combustion and emission characteristics of a four-cylinder SI natural gas/ethanol engine under lean burn conditions were studied. After careful analysis, the following conclusions can be drawn from the experimental results:

(1) The MBT increases with the increase of λ. The larger λ is, the more influence the CDIr has on the MBT. However, the MBT decreases with the increase of the CDIr. The larger the CDIr is, the less influence λ has on the MBT.

(2) CA 10−90 increases with the increase of λ, but decreases first and then increases with the increase of the CDIr. Under all lean-burn conditions, the valley values of CA 10−90 all appear when CDIr = 15%. For CA 10−90, λ=1.1, 15% CDI, and 85% EPI is the optimal combination under lean-burn conditions.

(3) Both the torque and Pmax decrease with the increase of λ, but first increase and then decrease with the increase of the CDIr. In addition, with the increase of λ, the optimal CDIr increases from 5% to 20%. Overall, the power performance of the engine is optimal at λ=1.1, 15% CDI, and 85% EPI in all lean-burn conditions.

(4) CoV$_{IMEP}$ increases with the increase of λ, but decreases with the increase of the CDIr. When λ=0.9, CoV$_{IMEP}$ is not sensitive to the size of the CDIr and always remains at about 0.7%. In general, when CDIr = 15%, the CoV$_{IMEP}$ can be effectively reduced and the engine can still work stably in the lean-burn conditions.

(5) The λ limit first increases and then decreases with the increase of the CDIr. When CDIr = 15%, the λ limit reaches the maximum value of 1.73, 19.31% higher than that of the original engine (CDIr = 0), which fully reflects the good lean-burn characteristics of CNG.

(6) CO emissions decrease with the increase of λ. With the increase of the CDIr, CO emissions first decrease and then increase when $\lambda \leq 1$; however, CO emissions continue to decrease when λ>1. When λ=1.1 and CDIr = 15%, the reduction of CO emissions is the largest, and the average reduction of CO emissions is 1.56 vol % for every 0.1 amplification of λ.

(7) HC emissions first decrease and then increase with the increase of λ. With the increase of the CDIr, HC emission increase when λ=0.9; However, HC emissions all reduce when $\lambda \geq 1$. HC emissions are the lowest under the condition of λ=1.1. When CDIr $\geq$ 15%, the average reduction of HC emissions is 30 ppm for every 0.1 amplification of λ. For HC emissions, λ=1.1 plus CDIr $\geq$ 15% is the optimal combination.

(8) NOx emissions first increase and then decrease with the increase of λ. With the increase of the CDIr, NOx emissions first increase and then decrease when $\lambda \leq 1.1$; however, NOx emissions first decrease and then increase when λ> 1.1. Under all λ conditions, NOx emissions are at an ideal level when CDIr $\geq$ 15%.

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Notes

The authors declare no competing financial interest.

■ NOMENCLATURE

SI spark ignition
CDI CNG direct injection
CDIr CNG direct injection ratio
CNG condensed natural gas
Pmax peak in-cylinder pressure
EPI gasoline port injection
GDI gasoline direct injection
IMEP indicated mean effective pressure
CA crank angle
BTDC before compression top dead center
λ excess air ratio
CoV coefficient of variation
HC hydrocarbon

MBT minimum advance for best torque
CO carbon monoxide
LFS laminar flame speed
NOx nitrogen oxides
DIT direct injection time
DIP direct injection pressure

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