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Research on Combustion and Emission Characteristics of a N-Butanol Combined Injection SI Engine

Weiwei Shang [1,2], Xiumin Yu [1], Kehao Miao [1], Zezhou Guo [1,*], Huiying Liu [2] and Xiaoxue Xing [2]

1 College of Automotive Engineering, Jilin University, Changchun 130022, China; shangww15@mails.jlu.edu.cn (W.S.); yuxm@jlu.edu.cn (X.Y.); miaokh1521@jlu.edu.cn (K.M.)
2 Electronic Information Engineering College, Changchun University, Changchun 130012, China; huiying20@mails.jlu.edu.cn (H.L.); xingxx@ccu.edu.cn (X.X.)
* Correspondence: guozz@jlu.edu.cn

Abstract: Using n-butanol as an alternative fuel can effectively alleviate the increasingly prominent problems of fossil resource depletion and environmental pollution. Combined injection technology can effectively improve engine combustion and emission characteristics while applying combined injection technology to n-butanol engines has not been studied yet. Therefore, this study adopted butanol port injection plus butanol direct injection mode. The engine test bench studied the combustion and emission performance under different direct injection ratios (NDIr) and excess air ratios (λ). Results show that with increasing NDIr, the engine torque (Ttq), peak in-cylinder pressure (Pmax), peak in-cylinder temperature (Tmax), and the maximum rate of heat release (dQmax), all rise first and then drop, reaching the maximum value at NDIr = 20%. The θ0-90 and COV_{IMEP} decrease first and then increase as NDIr increases. NDIr = 20% is considered the best injection ratio to obtain the optimal combustion performance. NDIr has little affected on CO emission, and the NDIr corresponding to the lowest HC emissions are concentrated at 40% to 60%, especially at lean burn conditions. NOx emissions increase with increasing NDIr, especially at N20DI, but not by much at NDIr of 40–80%. With the increase in NDIr, the number of nucleation mode particles, accumulation mode particles, and total particle decrease first and then increase. Therefore, the n-butanol combined injection mode with the appropriate NDIr can effectively optimize SI engines' combustion and emission performance.

Keywords: n-butanol; combined injection mode; combustion; gaseous emissions; particle number

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1. Introduction

In recent years, with the rapid development of automotive industries, the depletion of fossil sources and the aggravation of environmental pollution are becoming increasingly prominent [1,2]. Thus the research on green, alternative, and renewable fuels has attracted the attention of many scholars [3,4]. At the same time, advanced technical methods in the field of internal combustion engines are also very important in improving engine performance [5]. Therefore, the effective combination of alternative fuel characteristics and fuel injection modes will play a crucial role in the engine's performance, which is worth investigating.

The alternative fuel of internal combustion engines can be divided into two categories according to the different physical states of fuel: gas and liquid. Among the alternative fuels studied and used in the internal combustion engine at present, liquid alternative fuel mainly includes biomass fuel, alcohol fuel (ethanol, methanol, butanol), gas alternative fuel mainly includes liquefied petroleum gas (LPG), compressed natural gas (CNG), dimethyl ether hydrogen and so on [6]. Among all alternative fuels, alcohol fuel, as a renewable oxygen-containing biofuel, has been widely concerned by scholars at home and abroad because of its wide range of sources and clean combustion. Alcohols are considered one of the most promising alternative fuels for ignition engines, and their application in automobiles has been extensively studied [7]. Table 1 shows the properties of typical alcohol fuels [8]. Compared with methanol and ethanol fuel, butanol fuel has the advantages of higher caloric value and low latent heat of evaporation [9]. Therefore, butanol as a new alternative energy for engines has attracted more and more attention in recent years [10,11].

Table 1. Properties of alcohol fuels.

Fuel Properties	Methanol	Ethanol	N-butanol
Molecular formula	CH_3OH	C_2H_5OH	C_4H_9OH
Viscosity (Pa·s) at 20 °C	0.61	0.789	0.808
Research octane number	109–136	108–129	96–98
Laminar flame speed (cm/s)	52	48	48
Latent heat of evaporation (kJ/kg)	1103	840	582
Lower caloric value (MJ/kg)	19.7	26.8	33.1
Flammability limits (% vol.)	6.0–36.5	4.3–19	1.4–11.2
Stoichiometric air–fuel ratio	6.49	9.02	11.21

There are four isomers of butanol, and the different ways the carbon chain and hydroxyl group are connected the lead to the different uses of the isomers. N-butanol has been widely used as a promising alternative fuel for gasoline because of its better combustion performance and the similar physical and chemical properties to gasoline [12]. In addition, n-butanol has many advantages compared with Methanol and Ethanol, such as higher heating value and viscosity and lower volatility and vaporization heat. N-butanol can be obtained not only from coal but also by biological methods [13]. The raw materials are from various sources, including wheat, corn, and other crops, as well as agricultural waste straw [14]. According to Butyl Fuel Company's research data, 270 mL n-butanol can be produced by microbial fermentation with 1 L corn as raw material, and the cost is only 0.317 US dollars/liter. Additionally, with the continuous development of n-butanol production technology, the production cost is expected to be further reduced [15], which provides strong support for using n-butanol as an alternative fuel for engines [16].

To achieve the goal of high-efficiency and low emission of engines, new technologies of engines have been introduced in recent years, such as the combined injection technique [17]. The common injection mode of internal combustion engines includes port fuel injection (PFI) and direct injection (DI) [18]. Compared to port fuel injection, GDI enables precise fuel injection control in all operating conditions, reducing fuel consumption [19]. DI can form a localized concentration area near the spark plug and form a layered mixture in the whole cylinder to promote a better combustion effect and improve combustion thermal efficiency [20]. However, DI also has problems such as higher demand for fuel supply pressure, HC, and more emission of particulate matter [21]. PFI allows fuel and air to mix better in the inlet [22]. Therefore, the two injection methods have their own advantages and disadvantages. So, combining two injection modes, that is, the realization of compound injection in one engine through two injection systems, is of great significance to optimizing engine performance.

The research on butanol as engine fuel mainly focuses on n-butanol blends with other fuels and the pure n-butanol engine. Concerning engines of n-butanol blends with other fuels, J. Yang et al. [23] explored the combustion and emission characteristics of the hydrogen/n-butanol and hydrogen/gasoline rotary engines. They found that hydrogen addition improved the combustion and emission characteristics of gasoline and n-butanol rotary engines. Additionally, for both n-butanol and gasoline rotary engines, adding hydrogen can improve the brake thermal efficiency, shorten the development and propagation periods, reduce the coefficient of variation, and reduce the emission of HC and CO. However, NOx emission increased slightly after blending hydrogen. V. Thangavel et al. [9] studied the performance of a single-cylinder SI engine with both ethanol and gasoline injected into the inlet and performed a comparative analysis with n-butanol gasoline operation with the combined injection strategy in the intake port of a single-cylinder SI engine.

They found that at high torque conditions, the benefits of mixing ethanol with gasoline are significant. Compared to ethanol, the amount of n-butanol must be increased with increasing torque for better performance and low emissions. R. a. Ravikumar et al. [24] studied the combustion and emission characteristics of a twin spark ignition engine using n-butanol and gasoline dual fuel. They found that blending B35 resulted in lower carbon monoxide emissions, lower unburned hydrocarbon emissions, and lower nitrogen oxide emissions than pure gasoline. E. Agbro et al. [25] investigated the impact of n-butanol addition on the combustion performance and knocked properties under supercharged spark-ignition engine conditions; their results indicated that blending n-butanol can improve the anti-knock properties of gasoline and improve engine efficiency by using a higher compression ratio. Z. Guo et al. [19] tested gasoline/n-butanol blends with n-butanol volumetric ratios of 0%, 20%, 40%, 60%, 80%, and 100% in a DI SI engine. Results showed that a 20% butanol blending ratio could effectively reduce particle and gaseous emissions. M. Saraswat et al. [26] investigated the effect of different volume basis of butanol-gasoline and butanol/diesel mixture on IC engines. The results show that blending butanol can improve the power, torque, brake specific energy consumption, Hydrocarbons, Carbon-mono-oxides and NO emissions, but the NOx and CO_2 emissions are higher than those of gasoline and diesel. F. Meng et al. [27] investigated the combustion and emissions performance of a combined injection hydrogen/n-butanol dual-fuel engine with hydrogen addition fractions (0%, 2.5%, and 5%). The obtained results demonstrated the power and fuel economy performance of n-butanol engines are improved after adding hydrogen, and the HC and CO emissions drop while the NOx emissions sharply rise. T. Su et al. [28] researched the performance of a hydrogen/n-butanol dual fuel rotary engine with a n-butanol and hydrogen port-injection system. The test results indicated that blending hydrogen increased the brake thermal efficiency and in-cylinder temperature and reduced CO and CO_2 emissions effectively. Compared to n-butanol blends with other fuels, the pure n-butanol as an SI engine fuel is less researched. S. S. Merola et al. [29] studied the influence of injection timing on a wall-guided direct injection SI engine fueled with n-butanol. They found that late injection timing reduced soot but resulted in higher HC emissions and poorer performance than the optimum point. Additionally, they proved that fuel impingement on the piston crown is the main influencing factor for soot formation. N. S. a. Sandhu, X. a. Yu, and S. a. Leblanc et al. [30] analyzed the combustion characteristics of neat n-butanol under spark ignition operation using a single-cylinder SI engine. They found that n-butanol has similar physicochemical property and fuel characteristics to that of gasoline and has lower NOx, unburnt HC emission and CO_2 emissions. The above research indicates that pure butanol is feasible to replace traditional fuel as engine fuel. However, the performance of pure butanol engines needs to be further optimized.

Therefore, in the context of the world energy crisis, environmental pollution, and increasingly stringent emission regulations, it is of vital significance to adopt emerging technologies to improve the combustion performance of pure butanol engines and provide better theoretical and experimental support for the practical application of pure butanol engines [31]. Although n-butanol as an SI engine fuel has been partially investigated, most of them focus on n-butanol blended with other fuels. The performance of traditional pure butanol engines is unsatisfactory. Therefore, we proposed a new combustion idea as follows: the combined injection mode of partial butanol direct injection in the cylinder and partial butanol port injection, which has no research on SI engines fueled with n-butanol. We studied the influence of the combined injection mode on the combustion and emission characteristics of the butanol engine and explored the optimization potential of advanced technology on butanol engine performance. In this experiment, the combined injection mode adopts part of the n-butanol port injection plus part of the n-butanol direct injection. By engine test bench, the variations in combustion characteristics, we examined gaseous and particle emissions of SI engines due to changes in the direct injection ratio (DIr) and λ have been studied in this paper.

Although n-butanol as an SI engine fuel has been partially investigated, most focus on n-butanol blended with other fuels. The performance of traditional pure butanol engines is not satisfactory. Therefore, the effects of compound injection on combustion and emission characteristics of pure butanol engines were experimentally studied in this paper. The main contributions of this paper are as follows: (1) we proposed a new injection mode that is the combined injection mode of partial butanol direct injection in the cylinder and partial butanol port injection, which has no research on SI engines fueled with n-butanol. (2) The effects of direct injection ratio (DIr) and λ on combustion characteristics, gas emission, and particle emission of SI n-butanol engine under combined injection mode were investigated. (3) The potential of the combined injection technique in optimizing n-butanol engine performance was explored. Therefore, the purpose of this study is to use combined injection technology to solve the problem that the low saturated steam pressure of butanol may form wall flow when pure port butanol injection, while pure butanol direct injection may cause wall oil film due to the high viscosity of butanol. Coupling the two injection modes, butanol combined injection may improve their evaporation, mixing, and combustion, optimizing the butanol engine's performance.

2. Experimental Setup and Procedure

2.1. Engine and Instrument

In this study, the experiments were performed on a four-cylinder four-stroke water-cooled combined injection SI engine, a dual injection engine with port and direct injection. The main technical parameters of the original engine are listed in Table 2. Since the original direct injection system was designed for gasoline fuel, the fuel injected in the cylinder directly in this paper is butanol. Therefore, the high-pressure oil pump with direct injection in the original cylinder was not used in this test, but the high-pressure nitrogen cylinder pressurized method was used, and the direct injection pressure was 5 MPa. The original engine's oil pump still supplies the low-pressure port injection with an injection pressure of 0.3 MPa. The dual injection systems of the engine included partial n-butanol direct injection and partial n-butanol port injection. The experiment system structure image of the test engine is shown in Figure 1. Figure 1 shows that engine control parameters such as injection timing and injection durations for n-butanol and the throttle opening were all accurately controlled by the electronic control unit (ECU). The direct injection ratio is controlled precisely by adjusting the injection pulse width of the injector in real time.

Table 2. Main parameters of Engine.

Engine Parameter	Parameter Values
Engine Type	four cylinders; combined injection; naturally aspirated; water cooled; spark-ignition
Compression ratio	9.6
Bore/mm	82.5
Stroke/mm	92.8
Displaced volume/mL	1984
Maximum power/kW	132 (5000–6000 rpm)
Maximum torque/N·m	320 (1800–5000 rpm)

The specific information on the experimental instruments is listed in Table 3. The test engine is coupled to a CW160-type eddy current dynamometer to maintain a constant speed and measure torque. The values of each test point in the actual measurement were tested three times to take the average value. The n-butanol fuel flow was acquired by DMF-1-1AB and Ono Sokki DF-2420 flow meters. The λ was measured by an ETAS Lambda Meter 4 broadband oxygen sensor, and crank angle signals were collected with a Kistler-2614B4 crank angle encoder. CO, HC and NO_x exhaust gas were measured by an AVL DICOM 4000 five component tail gas λ analyzer.The in-cylinder pressure was obtained by an AVL GU13Z-24 pressure transducer mounted in cylinder 2. The collected cylinder pressure and crank angle signals are fed into the Dewesoft SRIUSi combustion analyzer for analyzing and calculating the combustion data. The combustion analyzer collects 200 cycle values after the test is stabilized for every test point to take the average value. The gaseous emissions were measured by an AVL-DICOM 4000 five-component tail gas analyzer. DMS500 fast particle analyzer was used to measure particle emissions.

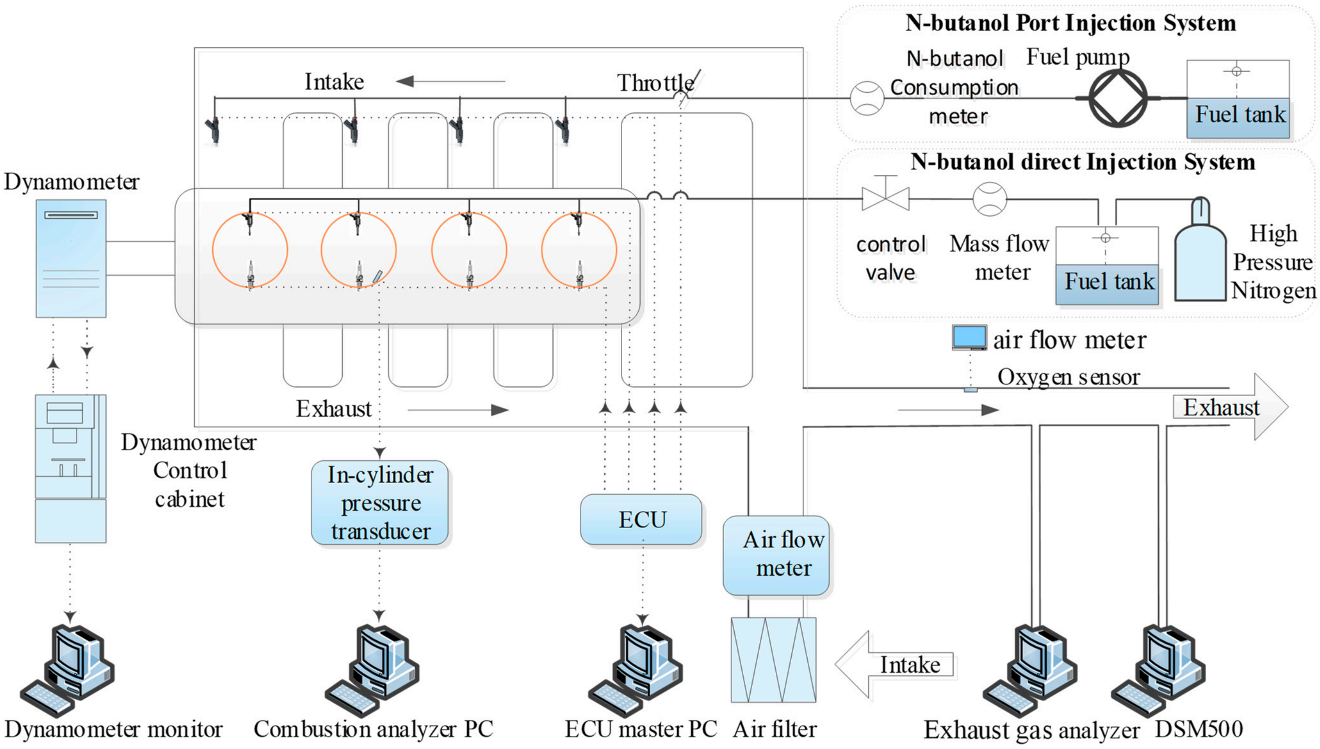


Figure 1. A schematic diagram of the test engine.

Table 3. The main test equipment of the experiment.

Parameter	Manufacturer	Range	Precision	Production Type
Speed	Luoyang Nanfeng Electromechanic Equipment Manufacturing Co., Ltd. (Luoyang, China)	0–6000 rpm	≤±1 rpm	CW160
Crank angle	Kistler Instrument China Ltd. (China)	0–720°	≤±0.5°	Kistler-2614B4
Excess air ratio	ETAS Engineering TOOLS (Germany)	0.700–32.767	≤±1.5%	LAMBDA LA4
Torque	Luoyang Nanfeng Electromechanic Equipment Manufacturing Co., Ltd. (Luoyang, China)	0–600 N m	≤±0.28 N·m	CW160
Cylinder pressure	DEWETRON GmbH. (Austria)	0–20 MPa	≤± 0.3%	AVL-GU13Z-24
n-butanol mass flow rate	Ono Sokki DF-2420 (Japan)	0.2~82 kg/h	≤± g/s	DF-2420
Carbon monoxide (CO)	AVL List GmbH (Austria)	0–10% vol	≤±0.01% vol	AVL DICOM 4000
Hydrocarbon (HC)	AVL List GmbH (Austria)	0–20,000 ppm vol	≤±1 ppm	AVL DICOM 4000
Nitrogen oxides (NOx)	AVL List GmbH (Austria)	0–5000 ppm vol	≤±1 ppm	AVL DICOM 4000
Particle number concentration	British combustion (England)	0–1011 dN/dlogDp/cm^3	≤±1.4 × 10^4 dN/dlogDp/cm^3	DMS500

2.2. Injection Modes and Definitions

The fuel used in the experiment was 99.7% purity n-butanol. For ease of expression and understanding, the naming of different injection modes of n-butanol tested is reported in Table 4.

Table 4. The naming of different injection modes.

The Naming of Different Injection Modes	NPI	N20DI	N40DI	N60DI	N80DI	NDI
N-butanol direct injection ratio (NDIr)	0%	20%	40%	60%	80%	100%

2.3. Experimental Method

The experiment was mainly focused on the combustion and emission characteristics of an n-butanol SI engine adopting different injection modes under lean burn conditions. The experiments were conducted at n-butanol direct injection ratios 0%, 20%, 40%, 60%, 80%, 100%, and five excess air ratios 0.9, 1, 1.1, 1.2, 1.3. The engine was run at a constant speed of 1500 rpm, and the throttle opening was kept at 10%, a typical urban condition. The engine coolant temperature was kept at (85 ± 5) °C. The injection timings of the cylinder direct injector and port fuel injector were 180° and 300 °CA BTDC. The direct injection pressure was set at 5 MPa, and the injection pressures of the port fuel injector were 0.3 MPa. The injection duration is adjusted constantly according to different NDIr. To highlight the influence of NDIr, the ignition advance angles were all fixed at 15 °CA unless otherwise specified.

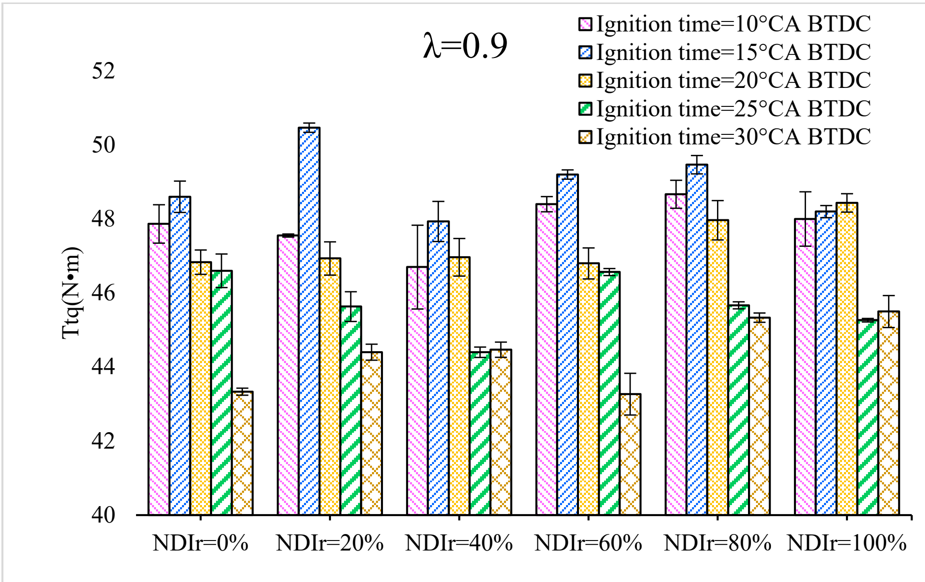
3. Results and Discussion

Fuel injection modes, NDIr, and λ have an important influence on the combustion and emission characteristics of the engine. This paper discusses the influence of direct injection mode on butanol engine performance from three aspects: combustion characteristics, gas emission characteristics, and particle emission characteristics.

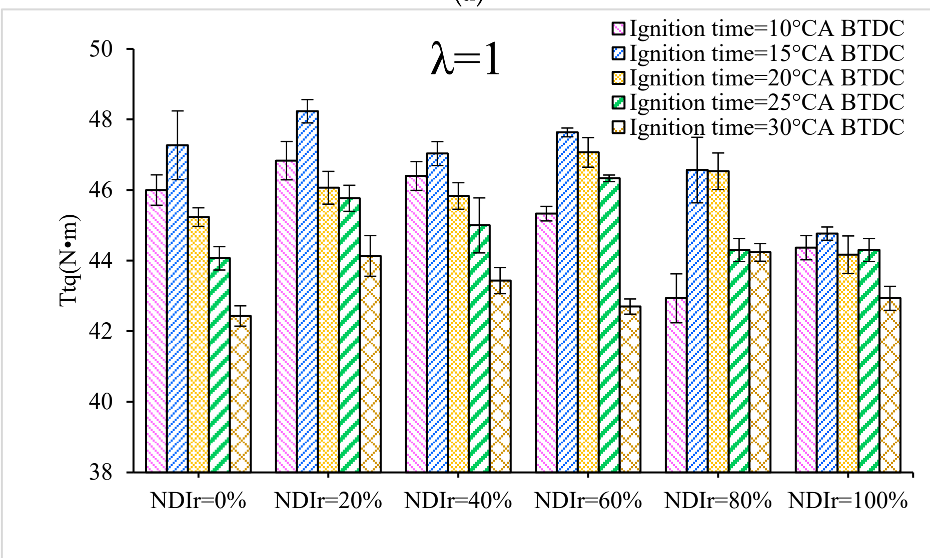
3.1. Combustion Characteristics

Figure 2 shows Ttq with ignition time at different NDIr and λ. The error bar indicates standard deviations for each experimental data. The value of λ is set to 0.9, 1, and 1.2, which, respectively, represent the three conditions of rich burn, stoichiometry, and lean born. It can be seen that Ttq increases first and then decreases with the advance of ignition time at different NDIr and λ. This can be explained by the following reasons. With the increase in the ignition advance angle, more fuel burns before the TDC leads to a rise in the compression negative work. When the ignition time is too late, post-combustion occurs, resulting in a decrease in torque. We also can see that the torque of N20DI is nearly the highest among different NDIr. The specific reasons will be analyzed in detail in the next section. In addition, Figure 2a–c show that under different λ, the maximum torque corresponding to the ignition time is different. When the values of λ are 0.9 and 1, the MBT is 15 °CA BTDC. While MBT is 20 °CA BTDC at λ = 1.2. This is mainly caused by the slow combustion speed under lean burn conditions. The slow combustion speed makes the whole combustion duration long, and the ignition time needs to be advanced to obtain the appropriate combustion phase.

Figure 3 displays the Ttq with NDIr at different values of λ. It can be seen that Ttq rises first and then drops with the increase in NDIr, reaching the maximum value at NDIr = 20%. The Ttq values of N20DI increase by about 3.3%, 0.4%, 9.1%, 4.1%, and 2.6% for λ values of 0.9, 1.0, 1.1, and 1.2, respectively. This can be attributed to the following reasons. On the one hand, when n-butanol is directly injected into the cylinder, it will cause a localized over-dense layered mixture near the spark plug, promote combustion in the cylinder and increase Ttq [32]. On the other hand, the n-butanol injected directly into the cylinder evaporates and absorbs heat, causing a drop in the temperature of the cylinder. Additionally, with the increase of NDIr, the local rich-fuel region increases, resulting in the formation of an inhomogeneous mixture and, finally, mass-spread combustion [21]. When NDIr is lower (20%), all these beneficial factors predominate and encourage the Ttq to rise. Therefore, Ttq rises first and then drops as NDIr increases. Moreover, compared with the NDI, the NPI injection mode allows more time to obtain a better air-fuel mix, resulting in a more complete combustion of the in-cylinder mixture and a higher Ttq. In addition, Ttq decreases gradually with the increase in λ, which could be attributed to the rise in λ making the mixture of n-butanol and air lean so that cycle fuel feeding decreases, resulting in the decrease in Ttq.

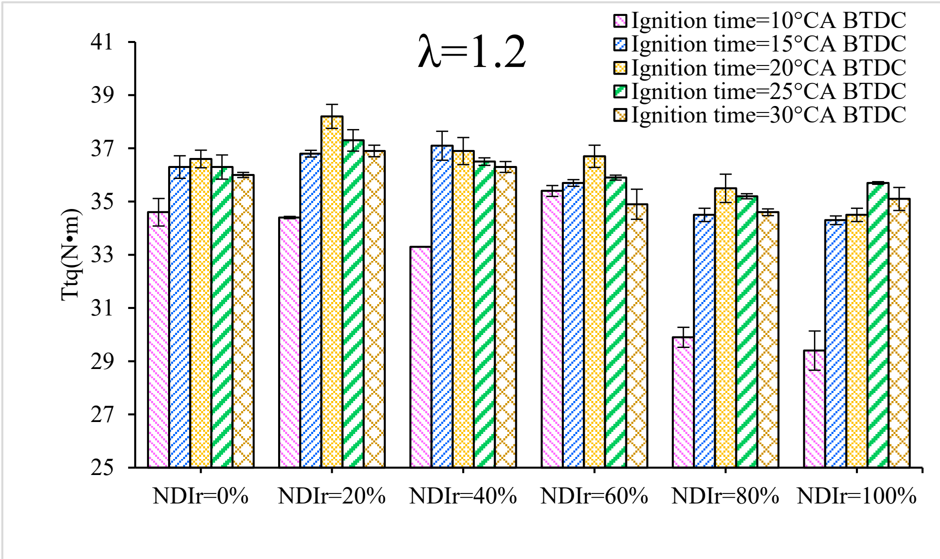


(**a**)



(**b**)

Figure 2. *Cont.*



(**c**)

Figure 2. Ttq with ignition time at different NDIr under λ = 0.9, 1, 1.2. (**a**) λ = 0.9; (**b**) λ = 1; (**c**) λ = 1.2.

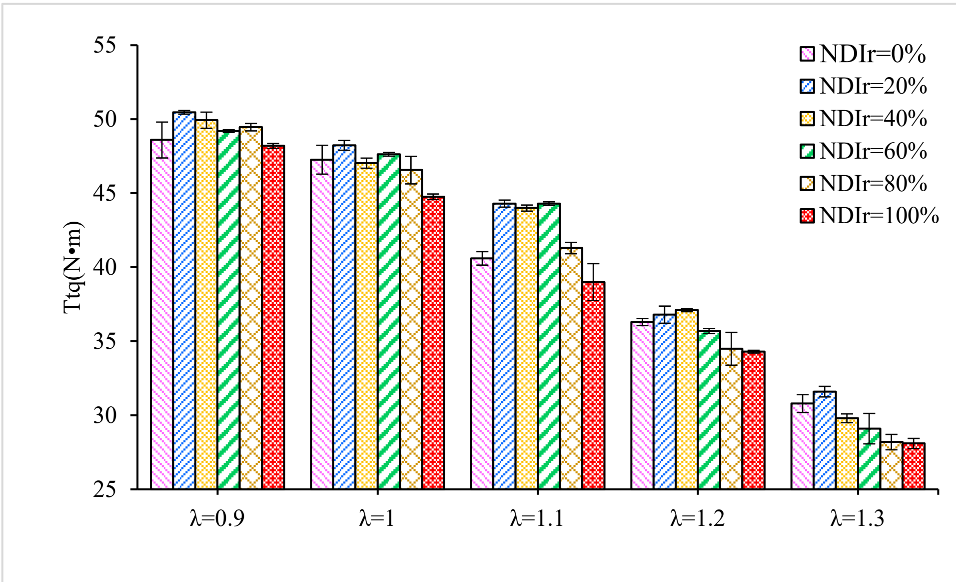


Figure 3. Ttq with NDIr at different values of λ.

Figure 4 displays the Pmax with NDIr at different values of λ. From Figure 4, we can see that Pmax shows similar trends as Ttq. Pmax goes up slightly and then down with the rising NDIr values, reaching the maximum value at NDIr = 20% and minimum value at NDIr = 100%. A small amount of butanol directly injected into the cylinder can result in a localized concentrated stratified mixture near the spark plug, which improves the combustion speed and promotes the combustion more completely. Thus Pmax is increased. As NDIr increases further, most of the fuel is injected directly into the cylinder resulting in a shorter time for the fuel to mix with the air. Thus the heterogeneous mixture weakens the advantage of the stratified mixture in the ignition, which results in incomplete combustion and decreases the Pmax. Therefore, Pmax increases first and then gradually decreases. In addition, Pmax decreases gradually with the increase in λ, which could be attributed to the rise in λ making the mixture of n-butanol and air lean so that cycle fuel feeding decreases, resulting in the decrease in Pmax.

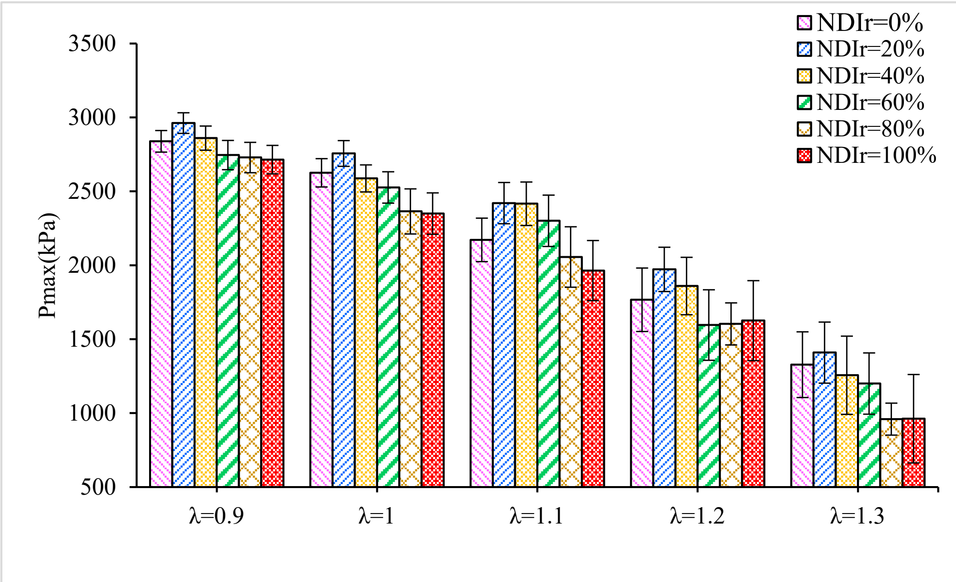


Figure 4. Pmax with NDIr at different values of λ.

From Figure 5, it can see that Tmax with NDIr at different λ. It can be seen that the Tmax shows a slight increase and then a gradual decrease with the increasing DIr, reaching the maximum value at NDIr = 20%. The results may be explained in two ways. Firstly, a dual injection could effectively combine the benefit of NDI and NPI, a small amount of n-butanol is injected directly into the cylinder, causing a localized concentration of the layered mixture near the spark plug, which promotes the combustion in the cylinder and leads to an increase in Tmax [13]. On the other hand, it can be attributed to is the properties of butanol itself. Because of the higher latent heat of vaporization, the n-butanol injected directly into the cylinder evaporates and absorbs heat, resulting in a decrease in Tmax [30]. The latter predominates when NDIr exceeds 20%, so the Tmax increases first and then decreases as the NDIr increases.

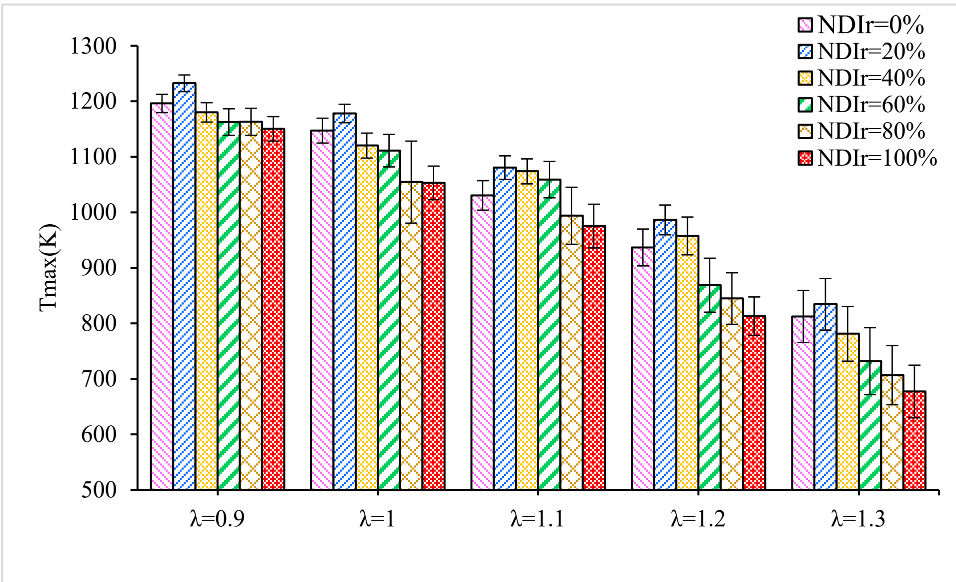


Figure 5. Tmax with NDIr at different values of λ.

Figures 6 and 7 show the maximum heat release rate (dQmax) and the dQmax (AdQmax) position with NDIr under different λ. It is shown that the dQmax has the same tendency as the Tmax. Additionally, the AdQmax decreases initially and then increases. It is clear that dual injection with 20% and 40% NDIr results in a higher dQmax. Meanwhile, the peak phase moves forward correspondingly, which enables combustion to release more energy around the TDC. This is mainly because the locally stratified mixture formed near the spark plug due to the small amount of n-butanol injected directly into the cylinder can ensure the ignition's reliability, creating a more stable flame core and promoting the early propagation of flame, thus promoting the exothermic process of the mixture.

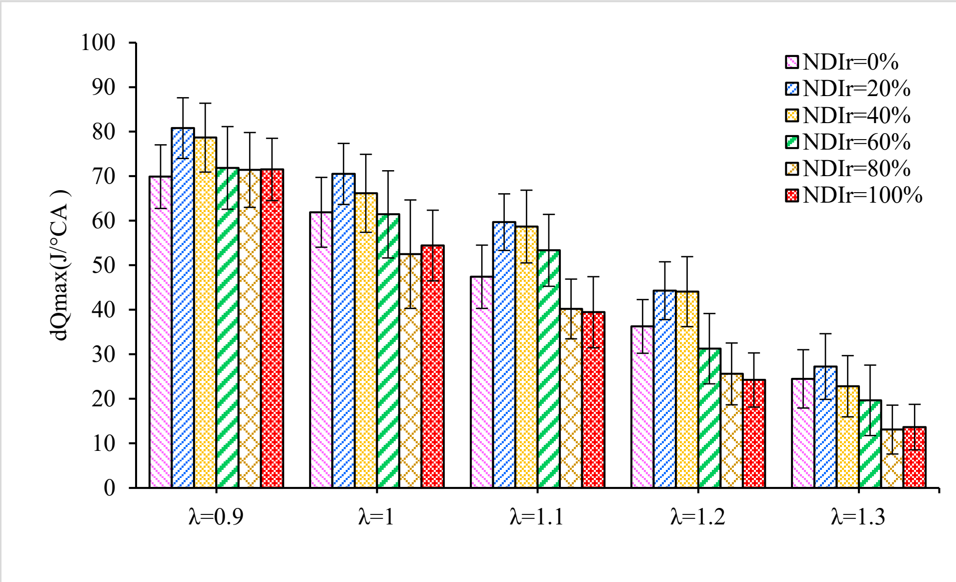


Figure 6. dQmax with NDIr at different values of λ.

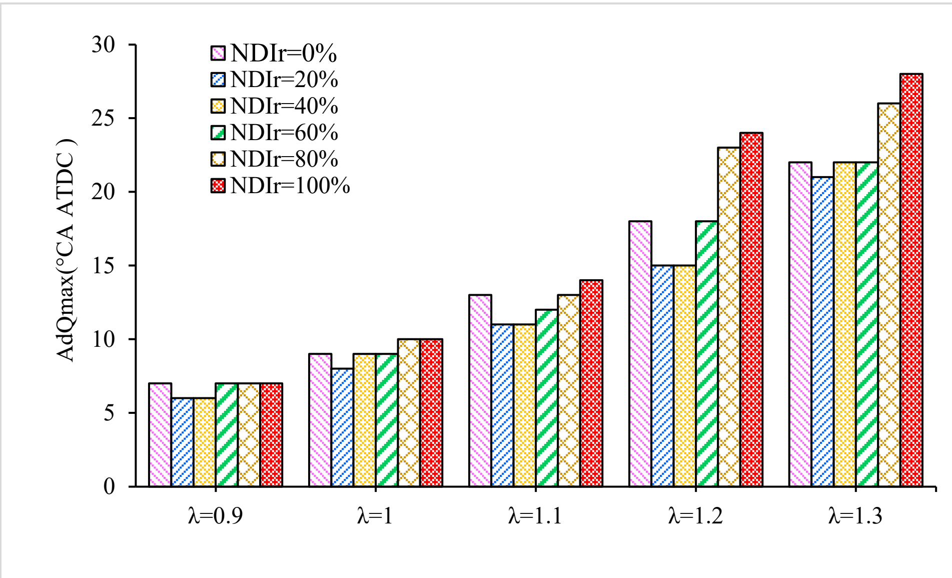


Figure 7. AdQmax with NDIr at different values of λ.

In this paper, the θ0-90 is defined as the crank angles for which 0–90% of the fuel mass has been burned. Figure 8 shows that the θ0-90 decreases first and then increases as NDIr increases. Figure 8 shows that compared to the single injection mode, the dual injection mode with NDIr values of 20% and 40% has obvious advantages in terms of θ0-90, especially under the larger λ. Compared to the results obtained with NPI and NDI, at λ = 0.9, 1, 1.1, 1.2, 1.3, the θ0-90 of N20DI decreases by 2 °CA, 2 °CA, 3 °CA, 4 °CA, 5 °CA, and, 2 °CA, 4 °CA, 7 °CA, 15 °CA, and 14 °CA, respectively. Combined with the previous analysis, dual injection mode with a relatively small NDIr can form a good layered mixture, which is beneficial to the ignition and the formation of a stable flame nucleus in the initial stage. Therefore, the smaller NDIr greatly increases the combustion speed. In addition, when the NDIr is constant, θ0-90 extends with the increase in λ. The increase in λ means the mixture gets leaner and the fuel burning rate slower, thus extending θ0-90.

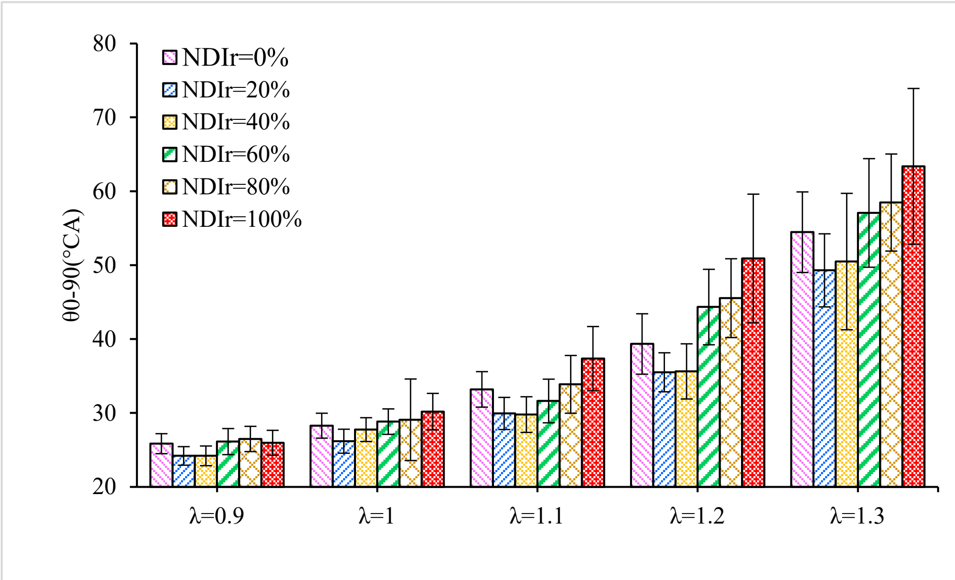


Figure 8. θ0-90 with NDIr at different values of λ.

Figure 9 shows the mean indicated pressure (IMEP) COV with NDIr under different λ. As seen from Figure 9, with the increase in NDIr, COV decreases first and then increases with NDIr, reaching a maximum at N20DI. It is interesting that COV changes significantly with DIr as λ increases. As can be seen in Figure 9, when λ is 1.2, compared with NPI and NDI, the COV of N20DI decreased by 30% and 60%, respectively. When λ is 1.3, compared with NPI and NDI, the COV of N20DI decreased by 30% and 90%, respectively. When λ is less than 1.2, the NDIr has little effect on COV under dual injection mode. However, comparing to NDI, the dual injection mode significantly improves engine stability. The reason for the above phenomenon is that the early stage of flame development is the main factor that causes the engine cyclic variation [33]. A small amount of butanol is directly injected into the cylinder, and the mixture is enriched near the spark plug to form stratified gas, which makes the ignition process more stable. In addition, it improves the flame propagation speed, reduces the influence of uncertain factors such as local misfires, and makes the engine run more smoothly. When the NDIr is relatively large, it can be seen from the previous analysis that the heterogeneity of the mixture and the fuel characteristics of butanol itself lead to the deterioration of combustion, leading to the increase in COV. The results show that the dual injection mode with NDIr of 20–40% can effectively reduce COV compared to NPI and NDI, especially under lean burn mode.

3.2. Gaseous Emissions Characteristics

Figure 10 shows the HC emissions with NDIr at λ values of 0.9, 1, 1.1, 1.2 1.3. It is shown that for a given λ. HC emissions decrease first and then increase with the increase in NDIr. This is because a small amount of n-butanol injected directly into the cylinder will cause a localized layered mixture near the spark plug, which makes it easy to form flame nuclei and makes the fuel burn faster and more completely. As the NDIr increases further, more fuel is injected into the cylinder, exacerbating the effect of preventing the formation of a homogeneous mixture. Additionally, the high latent heat of vaporization and the poor atomization of n-butanol lead to an uneven mixture. Therefore, the combustion process is incomplete, and HC emissions increase. We can also see from Figure 10 that HC emission first decreases and then increases with the increase in λ, reaching a maximum at λ = 0.9 and a minimum at λ = 1.2. This is because when λ is 0.9, more fuel is injected into the cylinder, significantly increasing localized n-butanol enrichment areas and incomplete combustion. On the other hand, the appropriate increase in λ increases the oxygen concentration in the cylinder, which is conducive to full fuel combustion. However, when λ = 1.2, the mixture is too thin, leading to misfire and incomplete combustion, and the HC emission increases accordingly. The above results show that the dual injection mode can effectively reduce HC emissions compared to NPI and NDI when the NDIr is below 60%.

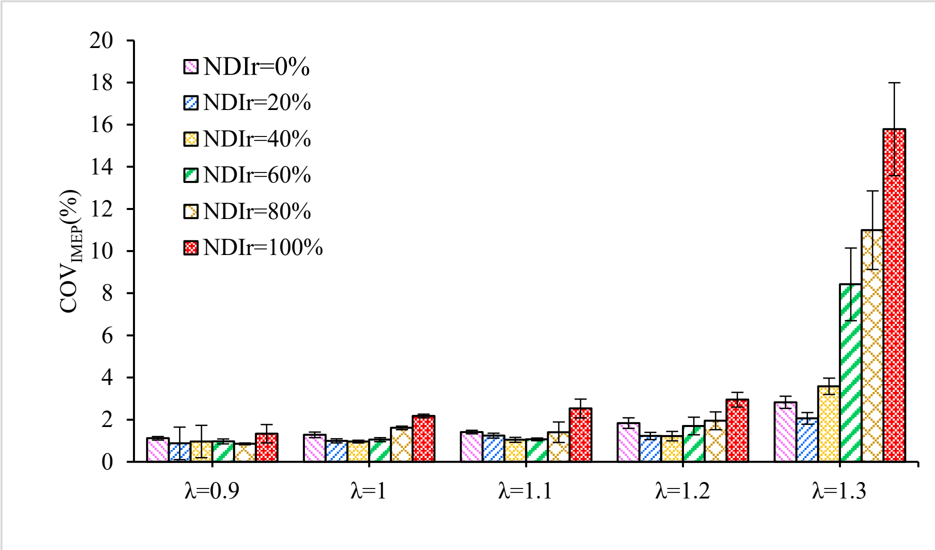


Figure 9. COV_{IMEP} with NDIr at different values of λ.

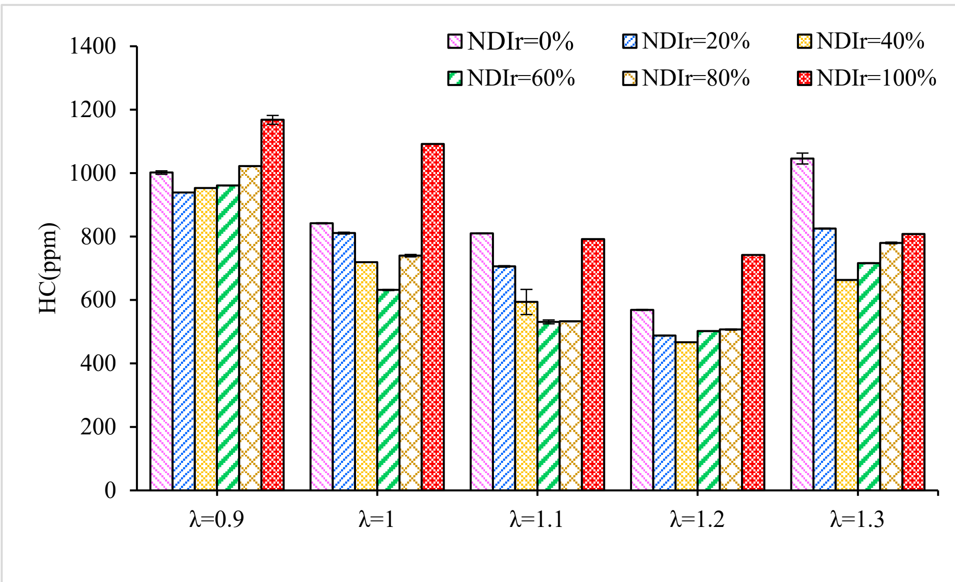


Figure 10. HC with NDIr at different values of λ.

Figure 11 shows the CO emissions with NDIr at λ values of 0.9, 1, 1.1, 1.2 1.3. Figure 11 shows that when λ is more than 0.9, CO emissions remain low and are little affected by NDIr. This may be explained by the fact that the n-butanol fuel itself contains oxygen and burns under the condition of sufficient oxygen, and the oxidation efficiency of CO is very high, so CO emissions are not affected by NDIr much. When λ is 0.9, the CO emissions are much higher than those obtained at the other four conditions. This is because when λ = 0.9, the mixture in the cylinder is rich, and the oxygen content is insufficient, leading to incomplete fuel combustion and higher CO emissions. The results indicated that compared with the in-cylinder direct injection, the dual injection mode of n-butanol can significantly reduce CO emission for the rich mixture.

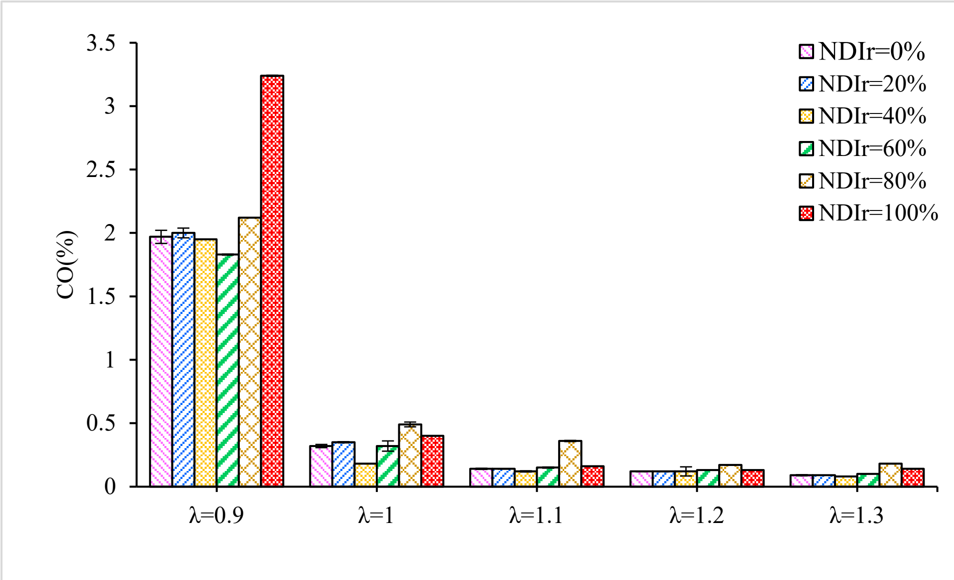


Figure 11. CO with NDIr at different values of λ.

Figure 12 shows the NOx emissions with NDIr at λ values of 0.9, 1, 1.1, 1.2 1.3. It can be seen that NOx emissions first increase slightly and then decreases s with the increasing NDIr. Especially when λ = 0.9, 1.0, and 1.1, the increase in NDIr has a significant effect on reducing NOx. NOx emission is very little at λ = 1.3. The above results may be explained by the following reasons. The formation of NOx emissions is relevant to cylinder temperature, oxygen concentration, and long residence time at high temperatures. A small amount of n-butanol direct injection promotes the combustion in the cylinder and increases the in-cylinder temperature, leading to an increase in NOx emissions at NDIr = 20%. As the NDIr increase further, NOx emissions tend to decrease. This is because a large amount of n-butanol is injected directly into the cylinder, causing the uneven mixing of the mixture because of the mixing time limitation, then leading to incomplete combustion. Secondly, a higher amount of fuel injected directly into the cylinder causes an increase in the specific heat capacity, adding to the effect of evaporation and atomization on the decrease in temperature. All of these jointly inhibit the generation of NOx emissions. So the NOx emissions first increase and then decrease. In addition, NOx emissions tend to increase initially and then decrease with λ, reaching a maximum at λ = 1. This is because when λ = 1, the oxygen content is sufficient, the combustion is sufficient, and the temperature in the cylinder is high, so the NOx emission appears the maximum. It is worth noting that the NOx emissions reduced significantly under the other four λ conditions compared with λ = 1. This indicates that the lack of oxygen and lean burn conditions can effectively reduce the NOx emission of n-butanol engines.

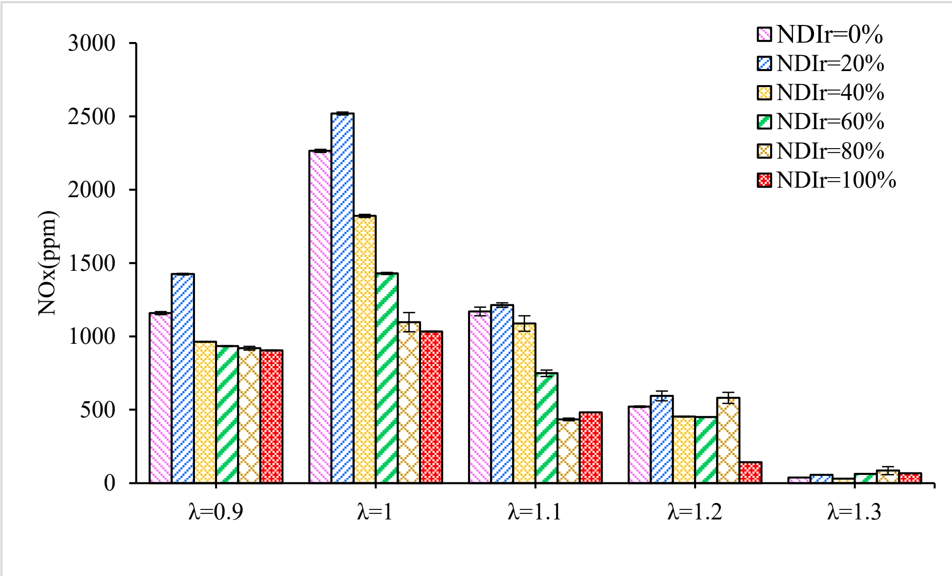


Figure 12. NOx with NDIr at different values of λ.

3.3. Particle Emissions Number

Particle emissions are important performance evaluation indexes of direct injection engines [34]. In the following section, NPN represents the number of nucleation mode particles. APN represents the number of accumulation mode particles [35]. TPN represents the total particle number [36]. NPN and APN can be used to show the number of different modes and their proportions in total particle emission, and TPN can be used to reveal the level of total particle number.

Figure 13 displays the APN with NDIr at λ values of 0.9, 1, 1.1, 1.2 1.3. When the λ is more than 0.9, the APN remains at a very low level and is little affected by NDIr. There are two possible reasons for this result. Firstly, with the increase in NDIr, the phenomenon of n-butanol spray-wall impingement increases, and the diffusion and combustion of the oil film on the wall surface intensifies, which may increase particle emission generation. Secondly, the formation of accumulation mode particles is closely related to the polycyclic aromatic hydrocarbons (PAHs). The hydroxyl radical (–OH) content produced during the combustion process of n-butanol can promote the oxidation of the PAHs [37]. Additionally, in the condition of sufficient oxygen, the formation conditions of high temperature and hypoxia of particles are inhibited. Thereby the APN remains at a very low level. At λ = 0.9, the APN increases significantly when NDIr is more than 80%. This is due to the combination of high-temperature hypoxia and direct injection mode, which leads to the increase in APN. Although the APN emission is relatively small, it is evident that when A = 0.9, compared with the in-cylinder direct injection, the dual injection mode can significantly reduce the APN.

It can be seen from Figure 14 that the NPN with NDIr at λ values of 0.9, 1, 1.1, 1.2, and 1.3. Figure 14 shows that the NPN drops first and then rises as NDIr decreases. Especially when the NDIr value is larger, NPN changes significantly. When the NDIr value ranges from 100% to 80%, NPN decreases by 73.3%, 70.9%, 65.9%, 73.5%, and 87.4% under different λ values. This can be explained in two parts. Firstly, with the decrease in NDIr, the phenomenon of n-butanol spray-wall impingement is significantly reduced, the diffusion and combustion of the oil film on the wall are improved, and the generation of particulate matter is also reduced. Secondly, according to Figure 10, HC emissions for dual injection mode with NDIr of 20% to 80% are lower than HC emissions with NDI, which should contribute to a reduction in NPN. In addition, as the λ increase, NPN decreases significantly, especially at the larger NDIr. At the condition of NDI, the increases at different λ cause a decrease in NPN by about 67.3%, 80.9%, 86.0%, and 75.9%. This is because when λ = 0.9, the fuel in the cylinder is over-rich, the mixture distribution is uneven, and the combustion is incomplete, leading to the increase in NPN. With the increase in λ, the oxygen content in the cylinder increases, and the formation conditions of hypoxia of particles are inhibited; thus, the concentration of NPN decreases. It can be concluded that, compared with the NDI, the dual injection mode can effectively reduce the NPN.

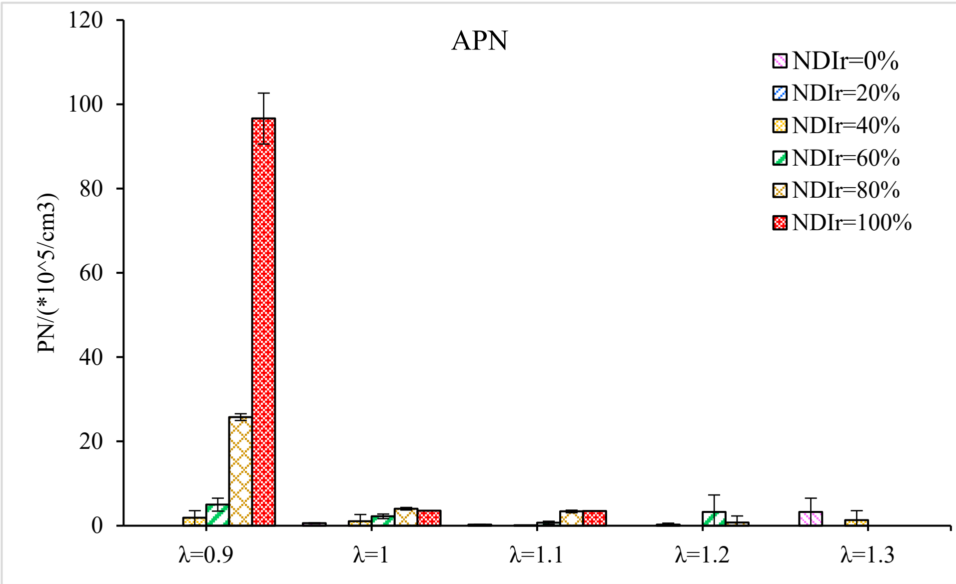


Figure 13. APN with NDIr at different values of λ.

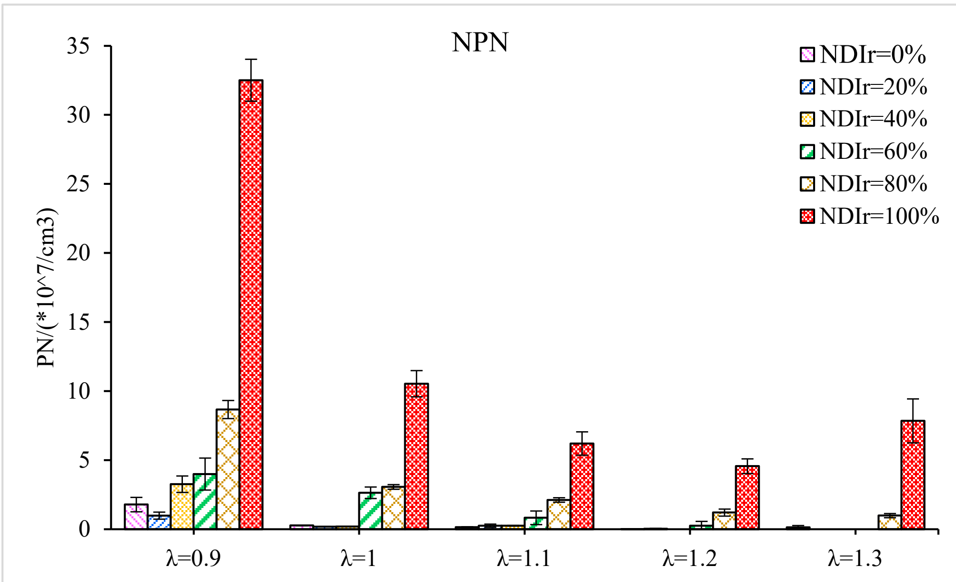


Figure 14. NPN with NDIr at different values of λ.

Figure 15 is a sum of Figures 13 and 14. The sum of NPN and APN for NDIr and different λ can be seen. The TPN shows similar trends as NPT, with an increase for increasing NDIr, and decreases initially and then increases slightly with the increasing λ. This is because APN is very low, almost negligible compared with NPN; the behavior of TNP can be explained in terms of the behavior of NPN.

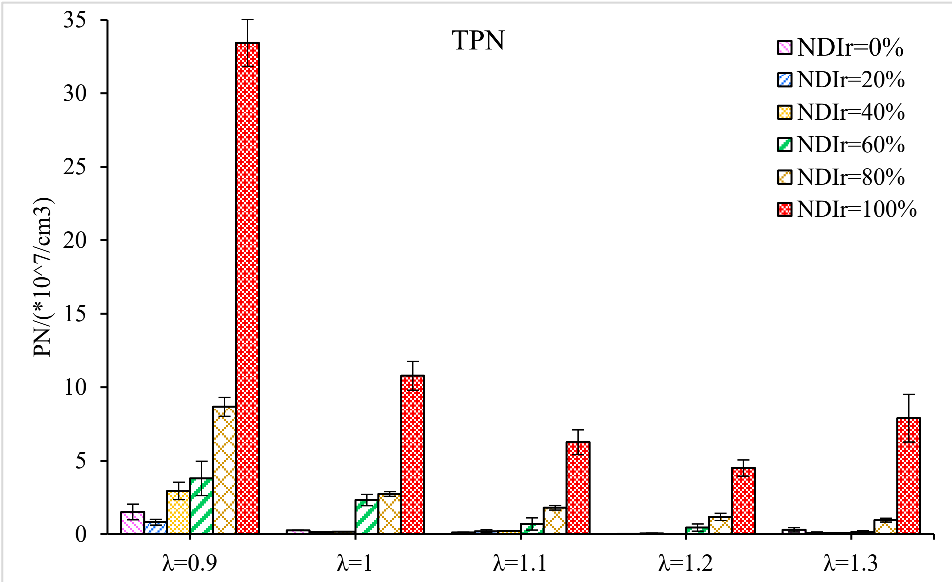


Figure 15. TPN with NDIr at different values of λ.

4. Conclusions

In this paper, the combustion and emission characteristics of the proposed combustion method were evaluated from several angles through experiments. The advanced technology of the engine and the advantages of the physicochemical properties of n-butanol were fully combined to explore a reasonable injection mode. The research in this paper will significantly contribute to butanol replacing gasoline as an effective and feasible method to reduce fossil energy consumption, obtain good combustion performance, and reduce emissions. The specific experimental conclusions are as follows:

1. Compared with the single injection mode, The dual injection of butanol engines with a smaller NDIr can form a local fuel-rich region near the spark, which is conducive to ignition and creates a more stable flame core, thus improving the comprehensive performance of butanol engines.
2. From the combustion performance index, Ttq, Pmax, Tmax, and dQmax all rise first and then drop, reaching the maximum value at NDIr = 20% and minimum value at NDIr = 100%. The AdQmax, θ0-90, and COV decrease first and then increase as NDIr increases. Thus, N20DI is considered to have an optimal combustion performance, followed by N40DI.
3. The dual injection mode has little affected on CO emission except for λ = 0.9, but can significantly reduce HC emissions than those relative to NPI and NDI. When λ is more than 0.9, the NDIr of 40–60% reduces HC emission most significantly, while the NDIr of 20% at λ = 0.9. NOx emissions increase in the dual injection mode, especially at N20DI, but not by much at NDIr of 40–80%.
4. NPN, APN, and TPN increase continuously as NDIr increases, the dual injection mode. The dual injection mode can effectively reduce particulate emissions relative to the in-cylinder direct injection mode, especially when λ = 0.9.
5. To summarize, the dual injection mode with NDIr of 20% to 40% significantly reduces HC and particulate emissions and maintains good combustion characteristics. In addition, the dual injection mode with a proper excess air ratio can effectively inhibit the increase in NOx emission caused by dual injection. Therefore, the dual injection mode with the proper NDIr can effectively optimize butanol engines' combustion and emission performance.
6. The combination of combined injection technology and butanol can reflect the excellent characteristics of butanol fuel to a greater extent, but the change of combustion state under different working conditions needs the real-time adjustment of combined

injection strategy to meet the effect of dynamic optimization under full operating conditions. Therefore, it is necessary to explore the combustion and emission characteristics of the combined injection butanol engine under more engine speed and load conditions and establish the global optimization control strategy in the next research stage.

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Nomenclature

SI	spark ignition	Pmax	peak in-cylinder pressure
NDIr	n-butanol direct injection ratio	Ttq	the engine torque
PFI	the port fuel injection	dQmax	the maximum rate of heat release
NDI	n-butanol direct injection	AdQmax	position of dQmax
NPI	port n-butanol injection	Tmax	peak in-cylinder temperature
MBT	maximum brake torque	θ0-90	total combustion duration
TDC	top dead center	COV	coefficient of variance
BTDC	before top dead center	TPN	total particle number
λ	excess air ratio	APN	accumulation mode particle number
IMEP	indicated mean effective pressure	TPN	nucleation mode particle number
HC	Hydrocarbon	CO	carbon monoxide
NOx	nitrogen monoxide or nitric oxide		

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