Comparative study on effects of injection mode on combustion and emission characteristics of a combined injection n-butanol/gasoline SI engine with hydrogen direct injection

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Abstract

In this paper, on the basis of combined injection technique, three injection modes with five n-butanol/gasoline volume ratios (ν_N) were put forward, aiming to compare and evaluate the relationship between the injection mode and the combustion and emission characteristics of an n-butanol/gasoline SI engine equipped with an extra hydrogen direct injection system. The results indicated that under low load condition, compared with NG_P (n-butanol/gasoline port injection) mode, the NG_{PD} (n-butanol/gasoline combined injection) mode and NG_{PH} (n-butanol/gasoline with hydrogen combined injection) mode can both improve the combustion performance. The peak cylinder pressure of NG_{PD} and NG_{PH} mode was 5.58% and 11.63% higher than that of NG_P mode on average, respectively. Furthermore, the combined injection modes concentrated the process of heat release, shortened the combustion duration, increased the power performance and improved the combustion stability at all ν_N. As for emissions, the NG_{PD} and NG_{PH} combined modes significantly decreased the HC and CO emissions, but increased the NO_X emissions. Among all the test conditions, the NG_{PH} mode with 25% ν_N obtained the best combustion quality. Additionally, the effect rules of ν_N on combustion and emission characteristics under NG_P mode were found to be consistent with those under

NG_{PD} and NG_{PH} combined modes.

Keywords: N-butanol/gasoline blends; N-butanol volume ratio; Hydrogen direct injection; Combined injection; Combustion and emission

1. Introduction

The port fuel injection (PFI), which is regarded as the main conventional injection method of SI engines, is limited by the use of a throttle valve for load control and fuel delivery delay. To overcome these major problems associated with PFI, the gasoline direct injection (GDI) technology has been introduced [1,2]. Injecting fuel directly into the cylinder improves power performance and fuel efficiency, but requires high-pressure injector and increases NO_X and PM emissions [3-5]. Thus, under the premise that automobiles will remain as the main transport system, finding high-efficiency and low-emission injection methods and appropriate alternative fuels are of great significance to improve the current problems of atmospheric pollution and reduce the dependence on petroleum [6,7].

As a biomass-based renewable fuel, n-butanol is considered to be among the most competitive alternative fuels for engines [8]. The favorable properties of n-butanol, including higher octane rating, laminar flame speed and

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Abbreviations: SI, spark-ignition; PFI, port fuel injection; GDI, gasoline direct injection; HDI, hydrogen direct injection; ν_N, n-butanol volume ratio; MBT, minimum advance for best torque; CA, crank angle; BTDC, before top dead center; ATDC, after top dead center; TDC, top dead center; BMEP, brake mean effective pressure; COV, coefficient of variation; CO, carbon monoxide; HC, hydrocarbon; NO_X, nitrogen oxide; PM, particulate matter.

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oxygen content in comparison with gasoline are shown in Table 1. Additionally, due to the close physico-chemical similarities between n-butanol and gasoline, and the n-butanol intersolubility characteristics with gasoline in any proportion [9]; blending n-butanol into gasoline as a hybrid fuel can be easily integrated into the existing fuel distribution infrastructure and match the regular gasoline engine well without much complicated modification [10]. Significantly, n-butanol is superior to methanol and ethanol due to its higher heating value and viscosity, lower vaporization heat and auto-ignition temperature, which indicates that the n-butanol has potential to surmount the disadvantages of short-chain alcohols [11]. Thus, choosing n-butanol blending gasoline as alternative fuel instead of pure gasoline in SI engine is an effective and implementable way to relieve the dependency on fossil fuels, obtain favorable combustion characteristics and reduce exhaust emissions [12,13].

Since the study of applying n-butanol as engine fuel, there are two major hybrid approaches for n-butanol and gasoline, premixed or independent dual injection. Elfasakhany [14] concluded that using small blend rate of n-butanol (below 10 vol%) into gasoline significantly improved the CO and HC emissions with few drawbacks on burning behavior of an unmodified single-cylinder SI engine. Chen et al. [15,16] performed plenty of correlative investigations through a turbocharged GDI engine fueled with high n-butanol content of gasoline blends. The tests showed that adding n-butanol increased cylinder pressure and its rise rate, promoted ignition and combustion stability, and shortened burning duration under wide-open throttle condition. Furthermore, these trends turned to be more significant when the n-butanol content increased from 30 vol% to 50 vol%. He et al. [17] conducted experimental studies on an SI engine equipped with n-butanol/gasoline port injection and gasoline/n-butanol direct injection, and detected that their indicated mean effective pressures were almost the same when the total energy released per cycle was kept constant, but their combustion processes were dependent on fuel injection methods. Feng et al. [18] evaluated the burning behavior of a dual-injection engine utilizing gasoline port injection and n-butanol direct injection at stoichiometric ratio. The tests showed that the dual-injection strategy with n-butanol addition was an effective way to improve the fuel efficiency and engine output in comparison with PFI and GDI. But the anti-knock ability degraded at small n-butanol DI ratios. Venugopal et al. [19] found that pure n-butanol at higher throttle position while pure gasoline or B50S (50% by mass) at lower throttle position could improve the combustibility and exhausts on a SI engine with two independent injectors in the intake port. Furthermore, injecting n-butanol just before the beginning of gasoline injection could reduce HC and CO emission even more than simultaneous injection at 60% throttle. From the studies described above, it was noticed that the independent injection could change the ratio and formation mode of n-butanol and gasoline flexibly in a working cycle while allowing full play to the advantages of different fuel properties in combustion process.

Although n-butanol as a substitute fuel can bring more benefits than short-chain alcohols, some potential issues still exist in n-butanol fueled engines. N-butanol has lower saturation pressure, higher latent heat of vaporization and longer ignition delay than that of gasoline [8]. So the problems of n-butanol fueled engines are mainly focused on slow evaporation rate, dissatisfactory atomization capability and ignition problems, which can offset the n-butanol positive properties and cause negative influence on combustion performance [20,21]. Hence, the application of n-butanol hybrid SI engines needs to be further improved for overcoming these drawbacks.

In comparison with conventional hydrocarbon fuels, hydrogen is another promising clean and potentially renewable energy source for engines, which has numerous advantages of fast propagation speed, wide flammability limits, short quenching distance and low ignition energy [22,23]. However, hydrogen fueled engines would inevitably suffer the effects of uncontrolled NO_X emissions and weak power output due to its high flame temperature and low energy density [24,25]. Besides, the absence of infrastructures for hydrogen refueling and the difficulties facing implementation of hydrogen storage systems on vehicles limit the widespread use of pure hydrogen engines in the immediate future [26,27]. Therefore, adding a small proportion of hydrogen as auxiliary fuel is a favorable approach to improve the combustion properties of main fuel and then the engine performance, especially under small and medium load condition [28,29,30,31].

Most previous investigations of hybrid hydrogen SI engines have been concentrated on improving the properties of common fossil fuels, such as gasoline, diesel and LNG for main working fuels [32-34]. Therefore, the potential of hydrogen improving the n-butanol combustion performance and enhancing its adaptability to combustion environments can be seen from the evidence presented by previous studies. However, due to the inability of n-butanol to be premixed with hydrogen like gasoline, combined injection technique is applied as injecting two types of fuel respectively through two independent injectors per working cycle. Studies have shown that engines demonstrated different advantages with different hydrogen injection locations [35]. The hydrogen direct injection (HDI) brings unique superiorities for avoiding low volume efficiency and two major abnormal combustion phenomena, backfiring and pre-ignition, which are caused by hydrogen port injection [36,37]. At the same time, n-butanol port injection can eliminate the undesirable cooling effect caused by injecting directly into cylinder. So an n-butanol/gasoline port injection combined with HDI method is put forward, which expects to utilize the hydrogen's favorable combustion properties to offset the shortcomings of n-butanol blending engines.

However, there are few previous studies have been investigated on the impact of hydrogen supplement on n-butanol fueled engines. Su et al. [38] found that a more stable combustion with shortened ignition delay and rapid combustion duration could be gained by 3vol% hydrogen supplement on an n-butanol rotary engine under lean burn condition. Also an additional benefit of reducing NO_X emission was achieved through lean combustion strategies on both n-butanol and hydrogen/n-butanol rotary engines. Yang et al. [39] compared the performance of a hydrogen/gasoline and hydrogen/n-butanol fueled rotary engine with dual-fuel port injection. They found that the hydrogen/n-butanol shared highly similar performance with hydrogen/gasoline and their performance could be improved by hydrogen enrichment. Sukjit et al. [40] investigated the impact of butanol and hydrogen supplement on combustion performance of a biodiesel compression ignition engine. The test indicated that the hydrogen supplement could counteract the increased HC emission caused by butanol blends and further reduce the CO and PM emissions. Raviteja et al. [41] conducted experimental studies on a gasoline/n-butanol blends fueled SI engine with hydrogen supplement at stoichiometric ratio. They concluded that 30vol% n-butanol blend was a better alternative to pure gasoline and both enriched hydrogen and oxygen content could realize more complete combustion, higher thermal efficiency and lower CO and HC emissions, but increased the NO_X emissions. However, it is noticed that all the studies mentioned above were performed using hydrogen intake port injection.

From our previous work, the combustion characteristic and emission performance showed great improvement by hydrogen direct injection on a pure gasoline or pure n-butanol SI engine [28,35,42,43,44]. Some aspects such as hybrid approaches, gaseous and particle emissions on an n-butanol blending gasoline engine have also been conducted [45,46]. However, the combustion performance of n-butanol/gasoline blended fuel combined with hydrogen supplementary fuel has not been studied yet. Therefore, in this study, to compare and analyze the relationship between hydrogen and n-butanol addition and other variables, three injection modes with five n-butanol volume ratios were considered on the basis of combined injection technique, including n-butanol/gasoline port injection (NG_P) mode, n-butanol/gasoline combined injection (NG_{PD}) mode and n-butanol/gasoline with hydrogen combined injection (NG_{PH}) mode. Comparing the NG_P mode and NG_{PD} mode means contrasting the difference between single injection and combined injection, and further comparing the NG_{PD} mode and NG_{PH} mode can obtain the effect of hydrogen direct injection.

Table 1

Specifications of alcohols [8], gasoline [8] and hydrogen [23].

	Methanol	Ethanol	n-Butanol	Gasoline	Hydrogen
Molecular formula	CH_3OH	C_2H_5OH	C_4H_9OH	C_5-C_{12}	H_2
Density (kg/L) at 20°C	0.792	0.789	0.808	0.72-0.78	0.09 (at 1 bar)
Viscosity (Pa·s) at 20°C	0.61	1.20	3.64	0.28-0.59	-
Research octane number	109-136	108-129	96-98	80-98	-
Cetane number	3-5	5-8	17-25	5-25	-
Lower heating value (MJ/kg)	19.9	26.8	33.1	42.7	119.7
Heat of vaporization (MJ/kg)	1.2	0.90	0.43	0.31-0.34	-
Saturation pressure (kPa) at 38°C	31.69	13.8	2.27	31.01	-
Stoichiometric ratio	6.49	9.02	11.21	14.7	34.5
Laminar flame speed (cm/s) at 25°C	-	-	48-53	37-43	185 (at 1 bar)
Flammability limits (%vol.)	6.0-36.5	4.3-19	1.4-11.2	0.6-8	4-76

Oxygen content (% weight)	50	34.8	21.6	0	0
Auto-ignition temperature (◦C)	470	434	385	~300	585
Minimum ignition energy (mJ)	0.215	-	-	0.24	0.02

2. Experimental setup and procedure

2.1 Experimental setup

The experiment was conducted on a combined injection SI engine with primary technical specifications as listed in Table 2 and its experimental setup as shown by the scheme in Fig. 1. The n-butanol/gasoline mixed fuel port supply system and direct supply system were used for pre-mixing and directly injecting the mixed fuel, respectively. Meanwhile an extra gas transportation line was added to the direct supply system for hydrogen direct injection. For separately controlling the engine ignition timing, n-butanol/gasoline and hydrogen injection pulse width and timing, a self-developed ECU was adopted. Fig. 2 illustrates the methods of combined injection.

In terms of the experimental equipment, a CW160 eddy current dynamometer was coupled with the engine to control and measure the torque and engine speed. The in-cylinder pressure was obtained from a GU13Z-24 pressure transducer while the crank angle signals were collected through a Kistler-2614B4 crank angle encoder. A Lambda meter LA4 was installed to monitor the excess air ratio. The mass flow rate of n-butanol/gasoline was acquired by DF-2420. A SIRIUS combustion analyzer was adopted to analyze the combustion properties. The HC, CO and NO_X emissions were measured via a DIGas 4000 light five-component exhaust gas analyzer. Detailed information about the experimental facilities is provided in Table 3. Additionally, the experiment used 99.7% purity n-butanol blending RON 97 standard gasoline with 99.99% volume fraction hydrogen.

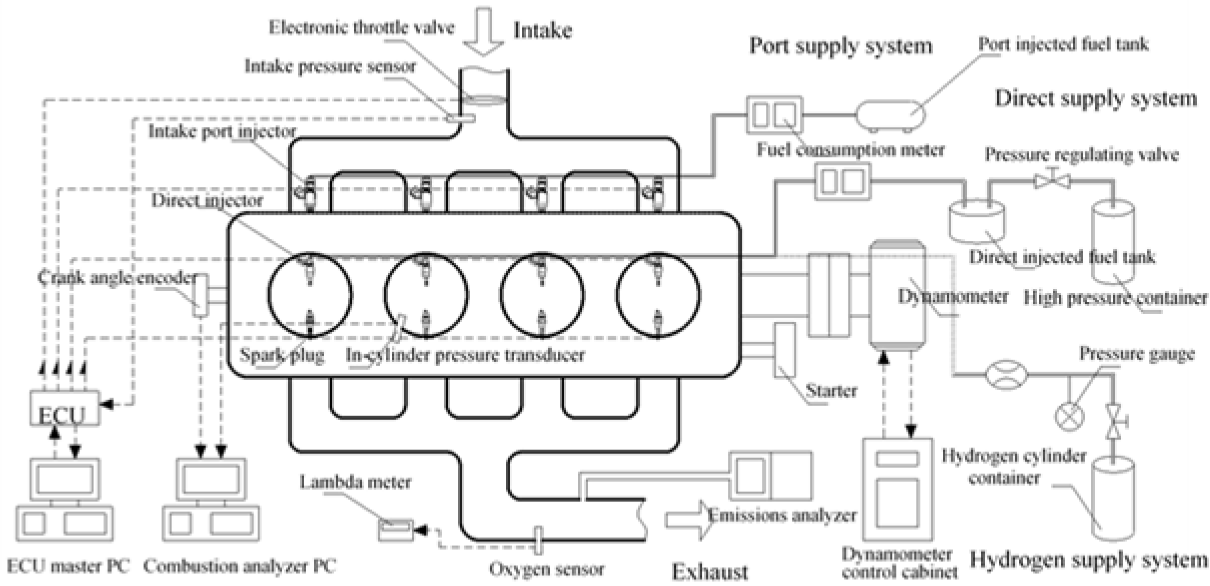


Fig. 1. Schematic diagram of the experimental setup

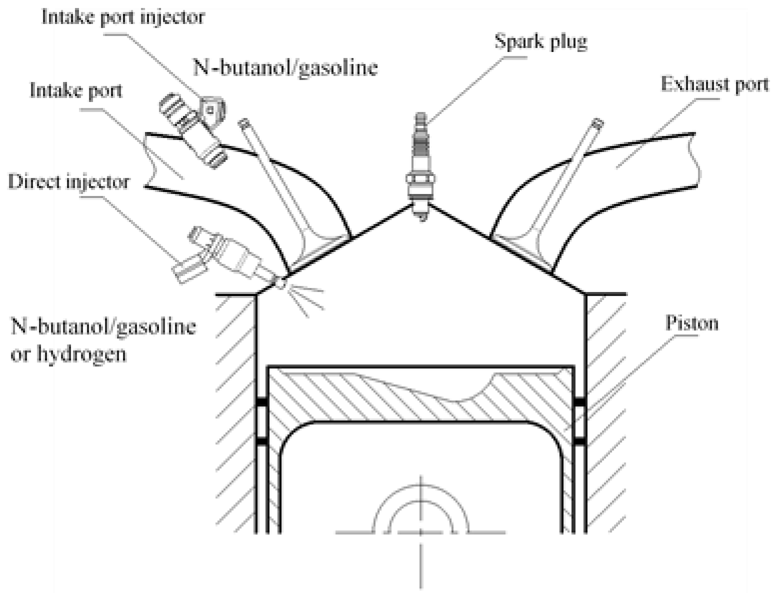


Fig. 2. Schematic diagram of the combined injection modes

Table 2

Specifications of the experimental engine.

Engine type	Inline four cylinder, spark-ignited, combined injection
Displacement	1.984 L
Bore×stroke	82.5 mm × 92.8 mm
Compression ratio	9.6 : 1
Rated power/ speed	132 kW/(4000r/min-6200r/min)
Rated torque/ speed	320 Nm/(1500r/min-4500r/min)

Table 3

Experimental apparatus specifications.

Apparatus	Parameter	Manufacturer	Type	Resolution
Dynamometer	Torque	Luoyang Nanfeng	CW 160	$\leq \pm 0.4$ %
	Engine speed			$\leq \pm 1$ r/min
Lambda meter	Excess air ratio	ETAS Engineering TOOLS	LAMBDA LA4	$\leq \pm 1.5$ %
Fuel consumption meter	Gasoline mass flow rate	ONO SOKKI	DF-2420	$\leq \pm 0.01$ g/s
In-cylinder pressure transducer	In-cylinder pressure	AVL Company	GU 13Z-24	$\leq \pm 0.3$ %
Crank angle encoder	Crank angle	Kistler Instrument Company	Kistler-2614B4	$\leq \pm 0.5$ °
Combustion analyzer	Heat release rate	DEWESoft Company	SIRIUS	$\leq \pm 0.1$ %
	Cycle-to-cycle variation			$\leq \pm 0.1$ %
Emission analyzer	CO emission	AVL Company	DIGas 4000 light	$\leq \pm 0.01$ %
	HC emission			$\leq \pm 1$ ppm
	NO_X emission			$\leq \pm 1$ ppm

2.2 Experimental procedure

In the experiments, three injection modes with five n-butanol volume ratios were selected for studying the impact of injection mode on combustion and emission performance of an n-butanol/gasoline SI engine. The ratio of n-butanol blending volume was adjusted from 0 to 100% with an interval of 25% and it was defined as follow:

$$\nu_N = \frac{V_{n-butanol}}{V_{n-butanol}+V_{gasoline}} \qquad (1)$$

In Eq. (1), ν_N denoted the n-butanol volume ratio, $V_{n-butanol}$ and $V_{gasoline}$ represented the blending volume of n-butanol and gasoline, respectively. And for verifying the difference of input heat between test conditions caused by different blending volume of n-butanol and gasoline, the total mixed fuel input heat values at five ν_N values under stoichiometric ratio condition were calculated as listed in Table 4. The total heat value only decreased by 0.838% when the ν_N was from 0 to 100%, which means the effect of total mixed fuel input heat can be negligible. Hence, to keep the total heat value constant, the fuel direct injection fraction was defined as follows:

$$\varphi_{direct} = \frac{Q_{direct}}{Q_{direct}+Q_{port}} \qquad (2)$$

In Eq. (2), φ_{direct} denoted the fraction of fuel direct injection, Q_{direct} and Q_{port} denoted the heat generated by fuel direct injection and port injection respectively. The sum of Q_{direct} and Q_{port} represented the total input heat. Based on our previous work, the φ_{direct} was set as a 10% fixed value in this experiment [47]. For NG_{PD} mode, the φ_{direct} is equal to the mass ratio of direct injection mixed fuel to total mixed fuel. And for NG_{PH} mode, the total heat value would theoretically exceed by 1.84% when the hydrogen completely replaced the direct injection mixed fuel since their different lower heating value and oxygen consumption under stoichiometric ratio condition [47]. This difference was within the allowable range of the experiment, so the total heat by three injection modes was considered to be the same. The port injection timing and direct injection timing was fixed at 300°CA BTDC and 100°CA BTDC, respectively [46,48]. The pressure of port injection and direct injection was 300 and 5,000 kPa, respectively [46,48]. The ignition timing was set at the minimum advance for best torque (MBT). The engine was operated under the speed of 1200 r/min and 30kPa intake manifold absolute pressure at stoichiometric ratio as a typical small and medium load urban condition. Each operation point was recorded for 200 cycles.

Table 4

Total heat value at different n-butanol volume ratios under stoichiometric ratio.

N-butanol volume ratio (%)	0	25	50	75	100
Volumetric heating value (kJ/L)	31.538	31.478	31.408	31.346	31.273
Heating value reduction in comparison with pure gasoline (%)	0	-0.190	-0.412	-0.609	-0.838

3. Results and discussion

3.1 Effect of injection mode on combustion characteristics

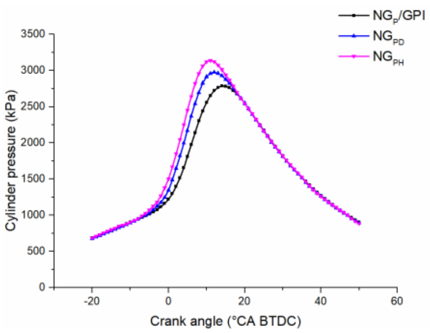
3.1.1 Cylinder pressure

Fig. 3 shows the variation of cylinder pressure with crank angle for three modes at various n-butanol volume ratios.

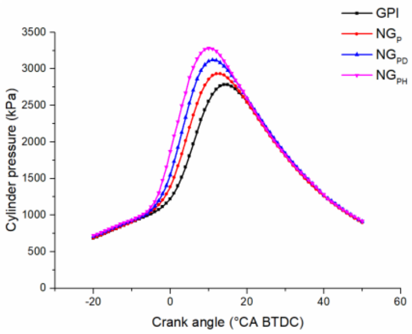
When the ν_N=0, the NG_P mode can be deemed as pure gasoline port injection (GPI) mode. So its cylinder pressure curve can regard as the reference line to compare the pressure curves of other ν_N under different modes. Under each ν_N, Fig. 3 shows that the peak cylinder pressure of NG_{PD} and NG_{PH} combined modes is all higher than that of NG_P single mode and the corresponding crank angle is slightly advanced even at the MBT ignition timing. The reason may lie in the concentration distribution of mixture inside cylinder [49]. Through combined injection, a stable homogeneous lean mixture can fill the cylinder during the intake stroke and subsequently a local rich concentration area around the spark plug can be concentrated during the compression stroke, forming a stratified mixture that realizes the reliable ignition and complete fuel combustion. So the cylinder pressure increases and even the pure n-butanol can maintain better combustion performance than GPI through combined injection modes. In further comparison of the two combined injection modes, the peak cylinder pressure continuously increases with the addition of hydrogen under each ν_N. This is mainly because the favorable combustion characteristics of hydrogen [23]. Its wide flammability limits, fast diffusion rate and low minimum ignition energy can make up for the innate problems of poor evaporation and atomization and ignition delay caused by n-butanol, allowing the cylinder pressure to reach a higher peak more rapidly.

In addition, the values of the peak cylinder pressure and its corresponding phase at different ν_N under three injection modes are shown in Fig. 4. It can be seen form Fig. 4, under each injection mode, the peak cylinder pressure initially rises and subsequently decreases with the increasing ν_N. This result may be explained by the physico-chemical properties of n-butanol [50]. Firstly, the oxygen atom contained in n-butanol can lead to a more complete combustion. Secondly, the high laminar flame speed of n-butanol can increase the degree of constant-volume combustion. These advantageous factors bring the cylinder pressure to the maximum at ν_N=25% and still promote a slight beneficial effect when the ν_N=50%. However, as the ν_N continues to increase, the drawbacks of slow evaporation rate and dissatisfactory atomization capability of n-butanol gradually reveal out, leading to a drop in cylinder pressure. Fig. 4 also shows that the corresponding phases of peak cylinder pressure are all concentrated within the scope of 11 to 14°CA ATDC since the ignition timing is set to MBT to obtain the optimal combustion phase.

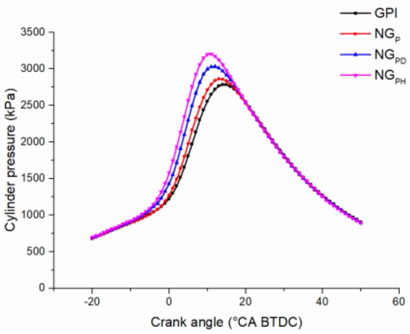
When the ν_N is constant, the main difference in three modes is the concrete values of the peak cylinder pressure. Compared to NG_P mode, the peak cylinder pressure increases 6.80%, 6.43%, 5.98%, 4.55% and 4.13% at ν_N=0, ν_N=25%, ν_N=50%, ν_N=75% and ν_N=100% respectively under NG_{PD} mode, which represents a sustained downward trend with the increasing ν_N. This tendency occurs because the fuel injected directly into the cylinder becomes harder to vaporize around the spark plug with the increase of ν_N, leading to the less significant effect of NG_D. Meanwhile the peak cylinder pressure of NG_{PH} mode is 11.63% higher on average than that of NG_P mode. Hence, the peak cylinder pressure is increased by combined injection and further adding hydrogen can make a greater improvement under each ν_N.



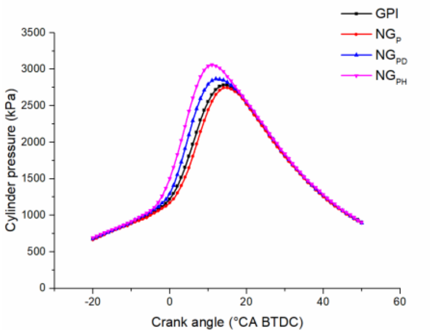
a. ν_N=0



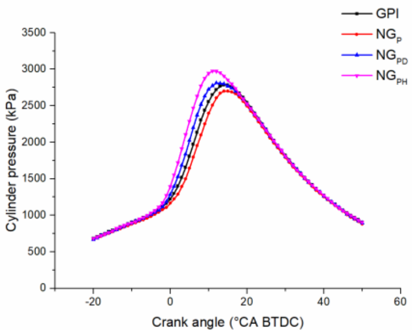
b. ν_N=25%



c. ν_N=50%



d. ν_N=75%



e. ν_N=100%

Fig. 3. Cylinder pressure versus crank angle at different n-butanol volume ratios under different modes

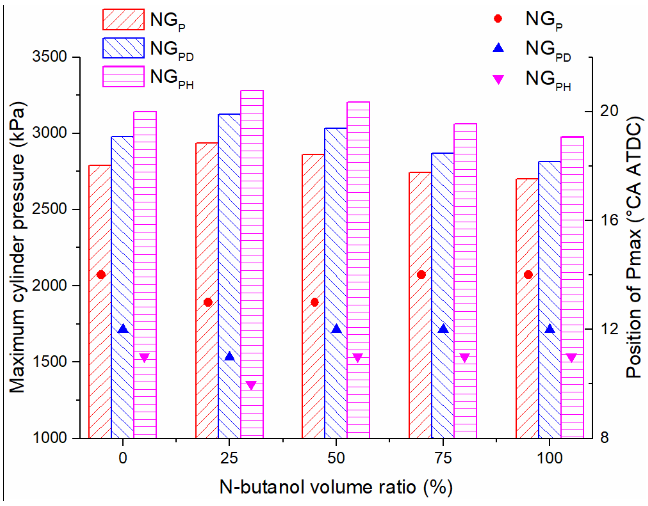


Fig. 4. Peak cylinder pressure and its phase at different n-butanol volume ratios under different modes

3.1.2 Heat release rate

Fig.5 illustrates that the regularity of peak heat release rate versus ν_N under each injection mode, which reveals a similar tendency to the peak cylinder pressure. And when ν_N is constant, for example, at ν_N=50%, the peak heat release rate increases and its phase moves forward to the TDC simultaneously in a sequence of NG_P, NG_{PD} and NG_{PH} injection mode as shown in Fig. 6. Through combined injection, a local mixture-enriched area near the spark plug can be formed because of the geometric structure of piston top and moderate tumble motions, which contributes to obtaining a steady flame kernel. Through the reliable ignition and rapid outward propagation of the flame front benefitted from the steady flame kernel, the surrounding mixture can be quickly ignited, thereby increasing the heat release rate and facilitating the concentration of heat release process. Furthermore, hydrogen supplement can increase the mixture combustion rate, especially in the flame propagation inception phase [51],which indicates the slope of heat release rate curve can become steeper as more mixture heat releases in a shorter period. Therefore, an accelerated and concentrated heat release process can be achieved by NG_{PD} and NG_{PH} injection mode.

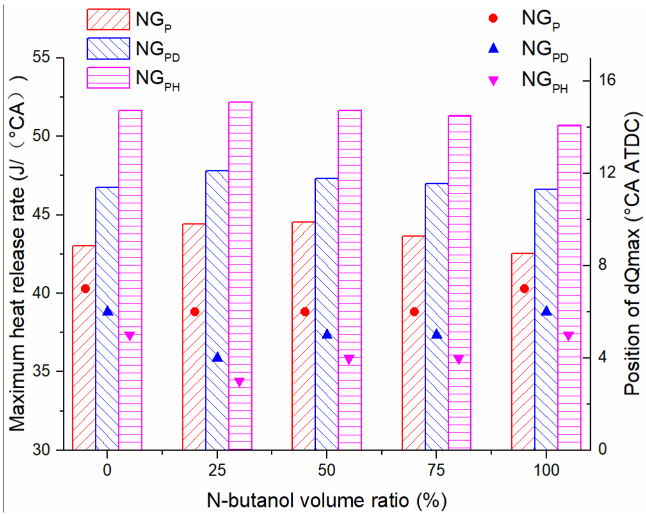


Fig. 5. Peak heat release rate and its phase at different n-butanol volume ratios under different modes

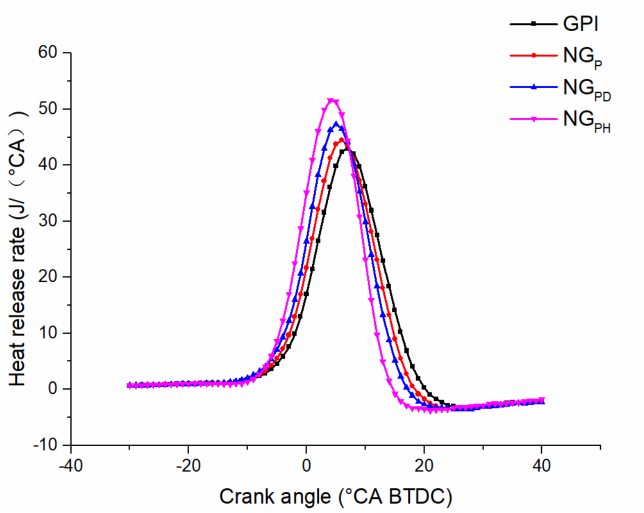


Fig. 6. Heat release rate versus crank angle at 50% n-butanol volume ratio under different modes

3.1.3 Flame development duration

The flame development duration (CA0-10) is defined as the crank angles from the beginning of spark ignition to the mass of burned fuel reaching 10%, and the rapid combustion duration (CA10-90) is defined as the crank angles from 10% to 90% of the mass of burned fuel in this paper. The variation of flame development duration and rapid combustion duration with ν_N under three modes are shown in Fig. 7 and 8, respectively.

Fig. 7 shows that with the increasing ν_N, the flame development duration first shortens then gradually lengthens under NG_P mode, whereas the effect of ν_N on flame development duration is subdued under NG_{PD} and NG_{PH} mode.

The flame development duration obtains the minimum at ν_N=25% can be explained by the dehydrogenation reaction of n-butanol during the early combustion phase, which provides large amounts of OH free radicals to facilitate the chemical reaction rate of gasoline in the initial combustion stage [52]. Meanwhile, the gasoline with small ν_N has shorter flame development duration also because the n-butanol has a faster laminar flame speed [53]. However, the high latent heat of vaporization and low saturated vapor pressure are two other important thermodynamic properties of n-butanol that affect early combustion [54]. The strong cooling effect of in-cylinder charge and the long time of complete evaporation can offset the positive properties of n-butanol and cause negative influence on shortening flame development duration when the ν_N continues to increase.

For NG_{PD} and NG_{PH} mode, the same as analyzing the heat release process, the steady flame kernel formed by combined injection and the favorable combustion characteristics of hydrogen can contribute to the reliable ignition and early flame propagation, weakening the effect of ν_N on flame development duration and shortening the initial combustion phase. The flame development duration of NG_{PD} and NG_{PH} mode is shortened by 3.28 °CA and 6.61°CA on average respectively as compared to NG_P mode.

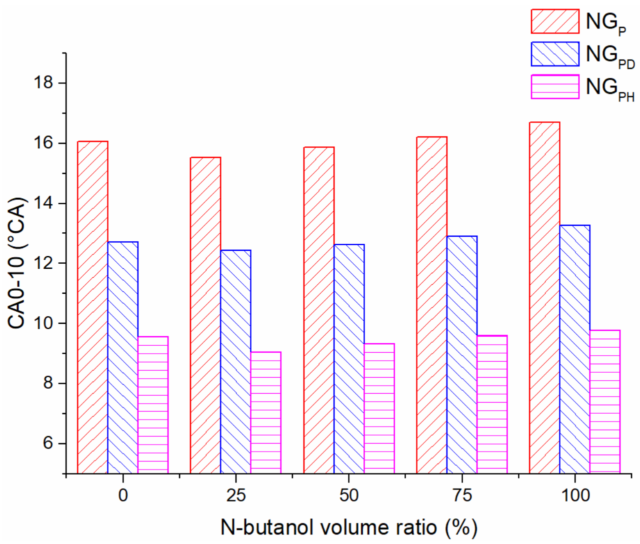


Fig. 7. Flame development duration versus n-butanol volume ratios under different modes

3.1.4 Rapid combustion duration

Fig. 8 shows that the rapid combustion duration continuously shortens with the increasing ν_N under each injection mode. Due to the molecular chain of n-butanol is shorter than the average molecular chain of gasoline, once the n-butanol is ignited, coupled with its faster laminar flame speed, the rapid combustion duration will be shortened [55]. Also the rapid combustion duration of NG_{PD} and NG_{PH} mode is shortened by 0.64 °CA and 1.74°CA on average respectively than that of NG_P mode. Thus, it can be concluded that the combined injection and hydrogen addition have a definite effect on total combustion duration (CA0-90).

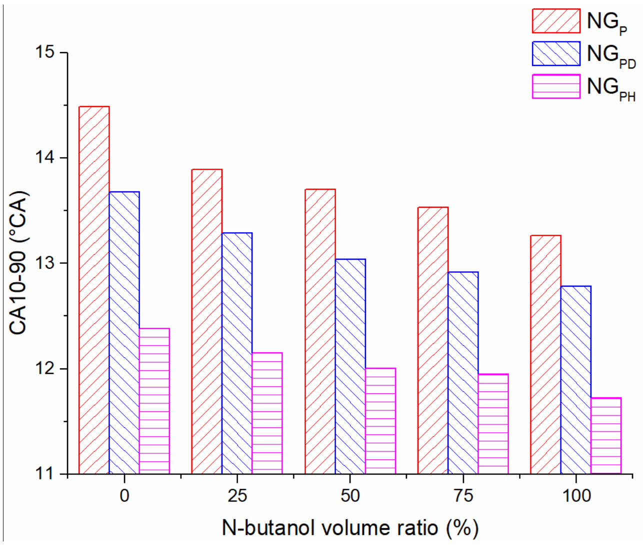


Fig. 8. Rapid combustion duration versus n-butanol volume ratios under different modes

3.1.5 Peak cylinder temperature

As shown in Fig. 9, the peak cylinder temperature initially rises a little before continuously declining with the increase of ν_N under all injection modes. Since n-butanol has a lower stoichiometric air-fuel ratio and lower heating value then gasoline, the mass of mixed fuel injected per cycle needs to be increased as the ν_N is incremental under the same excess air coefficient condition. Plus the n-butanol's high latent heat of vaporization makes a more prominent heat absorption of the blends with large ν_N in comparison with pure gasoline [56]. Thus the in-cylinder temperature decreases under large ν_N condition, while the mixture formation quality turns worse and the incomplete combustion becomes obvious, also reflecting in the lower heat release and lower cylinder pressure. The peak cylinder temperature at ν_N=25% is a little higher than that at ν_N=0% can be explained by the positive properties of n-butanol, which is the same as the process of analyzing the combustion performance. In addition, the in-cylinder temperature is accumulated by the heat generated from combustion, implying that the concentrated heat release process and less heat transfer loss will result in a higher peak cylinder temperature. Therefore, the peak cylinder temperature of NG_{PD} and NG_{PH} modes is higher than that of NG_P mode due to the combination of high peak release rate and short combustion duration, which can also be demonstrated by Fig. 5 and Fig. 8.

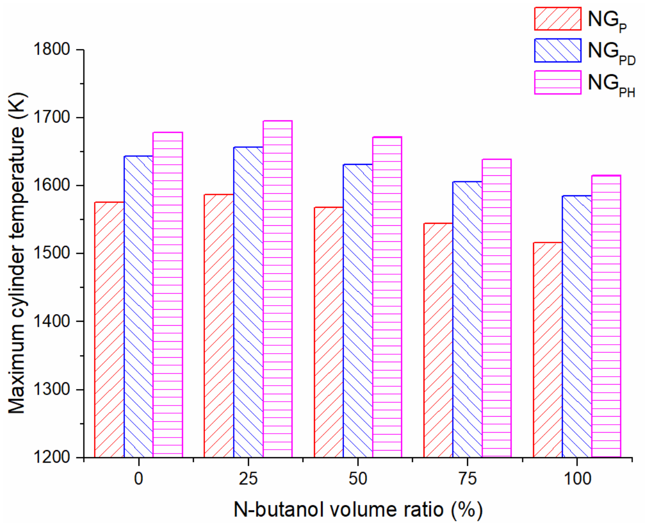


Fig. 9. Peak cylinder temperature versus n-butanol volume ratios under different modes

3.1.6 Brake mean effective pressure

The brake mean effective pressure (BMEP) represents the effective work output of the engine under per piston swept volume, which is a key parameter for measuring the engine power performance [25]. In addition, as the total heat generated remains the same for n-butanol/gasoline mixed fuel or with hydrogen, the brake thermal efficiency is directly proportional to the BMEP under three modes. Fig. 10 shows that the trend in which the BMEP first rises and then falls with the increasing ν_N is the same for all of the modes. The physico-chemical properties of n-butanol can explain this result. Firstly, the n-butanol can make the combustion more complete as an oxygenated fuel. Secondly, the high latent heat of vaporization of n-butanol can reduce the mixture temperature inside cylinder, thus reducing the heat transfer loss through the cylinder wall as well. Thirdly, the n-butanol has a faster laminar flame speed than gasoline, which can increase the degree of constant-volume combustion. These beneficial characteristics of n-butanol can still lead to a higher BMEP compared to pure gasoline when the ν_N is 50%. As the ν_N continues to increase, the disadvantages of poor evaporation and atomization performance of n-butanol gradually emerge, causing a decline in the BMEP.

Fig. 10 also shows that the BMEP of NG_{PD} mode is all higher than that of NG_P mode under each ν_N and the BMEP can continue to increase with adding hydrogen. The reason is that the mixture concentration distribution inside cylinder formed by combined injection and the favorable combustion characteristics of hydrogen can shorten the total combustion duration and lessen the incomplete combustion, allowing the combustion much closer to ideal constant-volume combustion and more fuel energy can be used for engine working [43]. On the other hand, as shown in fig.11, the lower exhaust temperature of combined injection modes can also reflect that more heat is used to produce work as the same energy input [47]. Besides, the direct injection does not occupy the intake charge and can increase the volumetric efficiency. It is also noted that the BMEP values of NG_{PD} and NG_{PH} mode at ν_N=100% are both higher than that of NG_P mode at ν_N=0%, so using combined injection and hydrogen supplement can evidently improve the engine power performance.

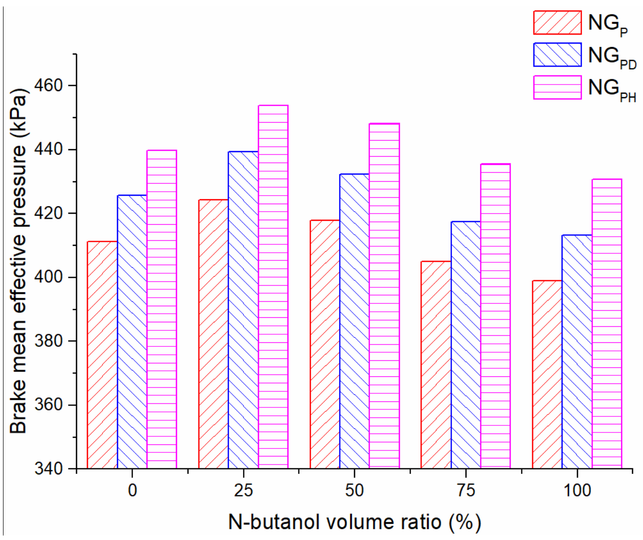


Fig. 10. Brake mean effective pressure versus n-butanol volume ratios under different modes

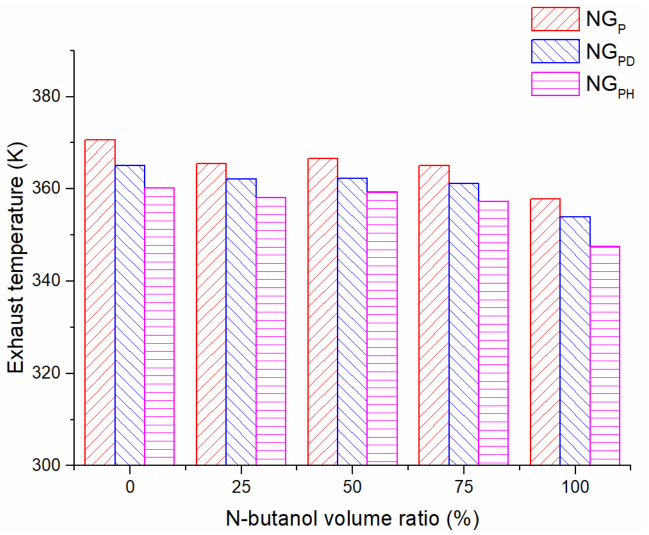


Fig. 11. Exhaust temperature versus n-butanol volume ratios under different modes

3.1.7 Coefficient of variation

To compare the combustion stability of three modes, Fig. 12 shows that the effect of cyclic variation on coefficient of variation (COV) versus ν_N at stoichiometric ratio. As shown, the COV initially falls and subsequently rises as the ν_N is incremental under each injection mode. When the ν_N=25%, because of the small amount of n-butanol blending into gasoline, the increased oxygen content and laminar flame speed of mixed fuel can improve the combustion performance. Then, as ν_N keeps increasing, the increased latent heat of vaporization and the reduced saturated vapor pressure of the mixed fuel result in a worse atomization quality, which is not favorable to forming the mixture inside cylinder and makes the incomplete combustion more obvious. Thus, there is a minimum COV at ν_N=25% and a maximum COV at ν_N=100% under each injection mode.

Fig.12 also shows that the COV of two combined injection modes is all lower than that of port injection mode and the COV can be further controlled below 1.0% by hydrogen addition. The same as analyzing the heat release process, through combined injection, the local rich concentration area around the spark plug can heighten the reliability of ignition and accelerate the speed of flame propagation, which is shown to weaken the interference from combustion environment and reduce the COV. For NG_{PH} injection mode, the hydrogen addition can reduce the ignition energy, increase the combustion rate and further make up for the issues of poor combustion performance of n-butanol. Hence, a stable flame developing process during the inception phase of ignition can be realized by means of hydrogen, which is effective in improving the combustion stability [51].

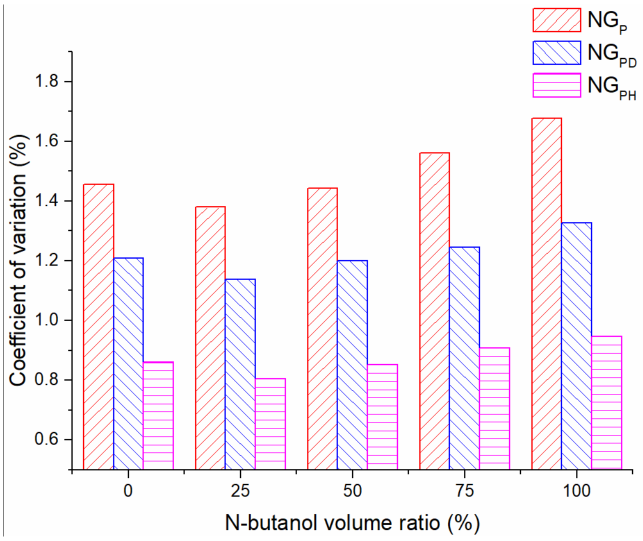


Fig. 12. Coefficient of variation versus n-butanol volume ratios under different modes

3.2 Effect of injection mode on emission characteristics

3.2.1 HC emission

Fig. 13 shows the HC emissions, which initially fall and then rise with the increasing ν_N for all injection modes. This occurs because the increase of oxygen content introduced by n-butanol can produce large amounts of OH free radicals [52]. Its strong oxidizing property promotes the process of C-H chain reaction and helps the mixed fuel burn more completely, and can even continuously oxidize the unburned hydrocarbons in the late combustion stage. As the ν_N continues to increase, the low combustion temperature inside cylinder, the local fuel concentrated area and the flame quenching at combustion chamber wall become the primary sources of HC emissions. Thus, under each injection mode, the HC emissions obtain the minimum at ν_N=25% and reach the maximum at ν_N=100%. It can also be said that the ν_N lower than 50% will all contribute to reducing the HC emissions compared to pure gasoline.

Under each ν_N, Fig. 13 also shows that the HC emissions of NG_{PD} and NG_{PH} combined modes are both lower than that of NG_P mode. The evident reduction of HC emissions through combined injection can be attributed to the steady flame kernel formed in initial stage, which is conducive to igniting reliably and setting fire to the ambient mixed fuel rapidly, enhancing the combustion quality and decreasing the HC emissions. For NG_{PH} injection mode, on the one hand, the intermediates of hydrogen combustion, O and OH free radicals, can promote the oxidation of HC [57]. On the other hand, the favorable combustion characteristics of hydrogen such as low ignition energy and short quenching distance can lead to more HC emissions reduction [58]. The HC emissions of NG_{PD} and NG_{PH} modes decreased by 9.18% and 21.71% on average respectively than that of NG_P mode. Therefore, an effective method for reducing HC emissions is through combined injection with hydrogen addition.

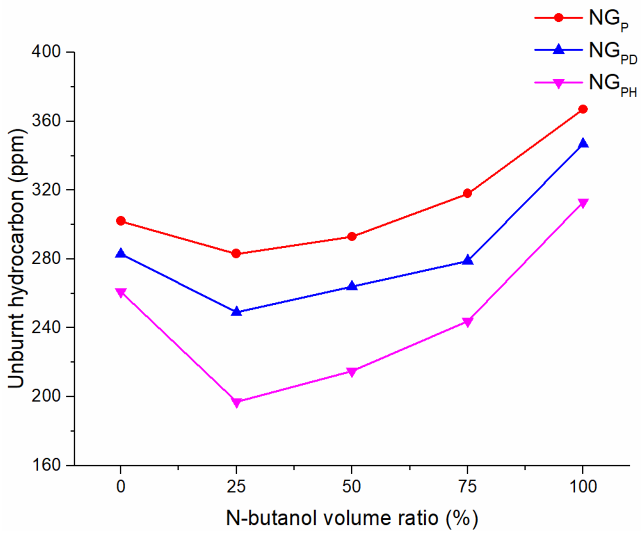


Fig. 13. HC emissions versus n-butanol volume ratios under different modes

3.2.2 CO emission

Fig. 14 shows that the CO emissions continuously reduce with the increasing ν_N under each injection mode. CO is an intermediate product formed by fuel incomplete combustion and its generated quantity is strongly associated with the oxygen concentration. Therefore, the oxygen lacking regions inside cylinder can be reduced due to the high oxygen content brought by n-butanol, which causes the decrease of CO emissions. Also the fast flame speed of n-butanol is conducive to reducing the CO emissions [59]. Fig. 14 also shows that the CO emissions of NG_{PD} and

NG_{PH} combined modes are both lower than that of NG_P mode under each v_N. The same as analyzing the combustion process, the favorable mixture concentration distribution inside cylinder formed by combined injection and the addition of hydrogen can curb the incomplete combustion. The CO emissions of NG_{PD} and NG_{PH} mode decreased by 19.22% and 32.63% on average respectively than that of NG_P mode. Additionally, hydrogen has no carbon element, but this is not the main reason for such a sharp CO emissions decrement. Hence, the NG_{PD} and NG_{PH} combined modes have an obvious effect on the reduction of CO emissions.

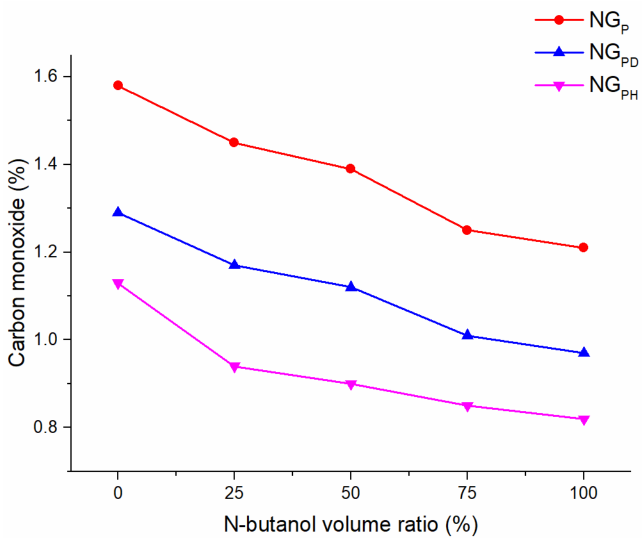


Fig. 14. CO emissions versus n-butanol volume ratios under different modes

3.2.3 NO_X emission

As shown in Fig. 15, the NO_X emissions initially rise and then fall with the increasing v_N for all injection modes. There is no nitrogen contained in the n-butanol blending gasoline fuel, so the generation of NO_X emissions are primarily form the nitrogen in the air that involved in the combustion process inside cylinder. The formation mechanism of NO_X is complicated and it is mainly affected by two factors, the temperature of combustion condition and the duration of staying at high temperature condition. As such, NO_X emissions rise the maximum at v_N=25% when the combustion quality of the mixed fuel performs the best and the peak heat release rate reaches the highest among all n-butanol volume ratios. Then as the v_N continues to increase, the physico-chemical properties of n-butanol leads to the falling temperature inside cylinder and the shortened rapid combustion duration, which are unfavorable for the formation of NO_X, causing the decrease of NO_X emissions. Although the oxygen content brought by n-butanol is still rising with the increase of ν_N, the NO_X emissions continuously decrease as a consequence of the low temperature.

On the basis of Fig. 9 above, the in-cylinder combustion temperature can be increased by combined injection as well as the addition of hydrogen, which is one of the main factors in generating NO_X emissions. Hence, also as shown in Fig. 15, the NO_X emissions of NG_{PD} and NG_{PH} combined modes are both higher than that of NG_P mode under each ν_N. The NO_X emissions of NG_{PD} and NG_{PH} mode increased by 6.79% and 16.83% on average respectively than that of NGP mode. Hence, the addition of n-butanol above ν_N=50% can reduce the NO_X emissions, however, the NG_{PD} and NG_{PH} combined modes will aggravate the NO_X emissions.

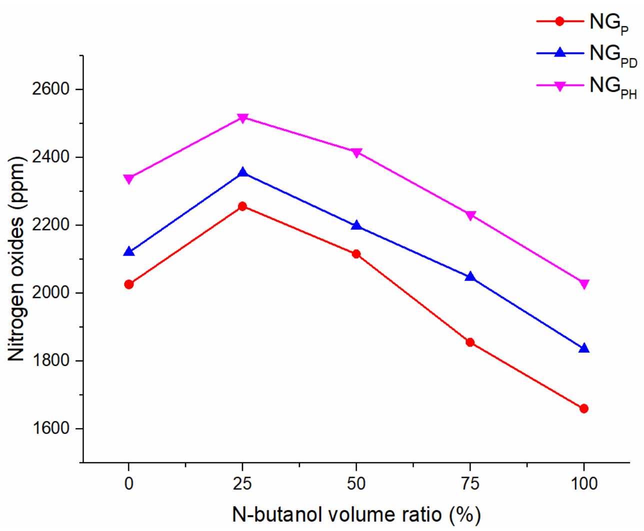


Fig. 15. NO_X emissions versus n-butanol volume ratios under different modes

4. Conclusions

In this paper, the influence of the combination of combined injection and hydrogen supplement on combustion and emission characteristics of an n-butanol/gasoline blended SI engine has been compared and evaluated under the speed of 1200 r/min and 30kPa intake manifold absolute pressure. The performance of cylinder pressure, heat release rate, flame development duration, rapid combustion duration, cylinder temperature, BMEP, COV and HC, CO and

NO_X emissions with five n-butanol volume ratios under three injection modes are carried out. The main conclusions are summarized as follow:

1. Compared to NG_P mode, the NG_{PD} mode can achieve more reliable ignition and complete fuel combustion. As for NG_{PH} mode, a local hydrogen-enriched area around the spark plug can be generated through HDI, which contributes to further improving the combustion performance.
2. The peak cylinder pressure of NG_{PD} and NG_{PH} mode is 5.58% and 11.63% higher than that of NG_P mode on average, respectively. Also the combined injection modes concentrate the process of heat release, shorten the flame development duration and rapid combustion duration, increase the peak cylinder temperature and BMEP.
3. The effect rules of ν_N on combustion and emission performances for NG_P mode are consistent with those of the NG_{PD} and NG_{PH} combined modes. When the ν_N=25%, the small amount of n-butanol can increase the oxygen content and laminar flame speed of mixed fuel and improve the combustion performance. At ν_N=50%, these advantageous physico-chemical properties can still promote a slight increase. But as the ν_N continues to increase, the disadvantage of poor evaporation and atomization performance of n-butanol gradually reveal out, resulting in a worse combustion quality.
4. The combined injection modes can significantly improve the combustion stability, while the COV can be further controlled below 1.0% by hydrogen addition. For another, the COV obtains the minimum at ν_N=25% under all injection modes.
5. In terms of emissions, the HC and CO emissions of NG_{PD} and NG_{PH} combined modes are both significantly less than those of NG_P mode, and the NG_{PH} mode has the lowest level among three injection modes. However, the NG_{PD} and NG_{PH} mode obviously increase the NO_X emissions. For another, under each injection mode with the increasing ν_N, the HC emissions initially fall and then rise, obtaining the minimum at ν_N=25%. The CO emissions continuously reduce. The NO_X emissions initially rise and then fall,

reaching the maximum at ν_N=25%.

6. Among all the test conditions, the NG_{PH} mode with 25% ν_N obtains the best combustion quality. Its peak cylinder pressure, peak heat release rate, peak cylinder temperature and BMEP reach the maximum values, and the COV and flame development duration obtain the minimum values. Also the HC and CO emissions level are reduced but the NO_X emissions are at most.

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