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RESEARCH ARTICLE

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Experimental study of stratified lean burn characteristics on a dual injection gasoline engine

Abstract Due to increasingly stringent fuel consumption and emission regulation, improving thermal efficiency and reducing particulate matter emissions are two main issues for next generation gasoline engine. Lean burn mode could greatly reduce pumping loss and decrease the fuel consumption of gasoline engines, although the burning rate is decreased by higher diluted intake air. In this study, dual injection stratified combustion mode is used to accelerate the burning rate of lean burn by increasing the fuel concentration near the spark plug. The effects of engine control parameters such as the excess air coefficient (Lambda), direct injection (DI) ratio, spark interval with DI, and DI timing on combustion, fuel consumption, gaseous emissions, and particulate emissions of a dual injection gasoline engine are studied. It is shown that the lean burn limit can be extended to Lambda = 1.8 with a low compression ratio of 10, while the fuel consumption can be obviously improved at Lambda = 1.4. There exists a spark window for dual injection stratified lean burn mode, in which the spark timing has a weak effect on combustion. With optimization of the control parameters, the brake specific fuel consumption (BSFC) decreases 9.05% more than that of original stoichiometric combustion with DI as 2 bar brake mean effective pressure (BMEP) at a 2000 r/min engine speed. The NO_x emissions before three-way catalyst (TWC) are 71.31% lower than that of the original engine while the particle number (PN) is 81.45% lower than the original engine. The dual injection stratified lean burn has a wide range of applications which can effectively reduce fuel consumption and particulate emissions. The BSFC reduction rate is higher than 5% and the PN reduction rate is more than 50% with the speed lower than 2400 r/min and the load lower than 5 bar.

Keywords dual injection, stratified lean burn, gasoline engine, particulate matter emission, combustion analysis

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1 Introduction

Due to increasingly stringent fuel consumption and emission regulation, improving thermal efficiency and reducing particulate matter emissions are two main issues for next generation gasoline engine [1–3]. To meet these challenges and compensate for the disadvantages of traditional spark ignition engines, many studies have been conducted [4–6]. Lean burn mode could greatly reduce pumping loss and decrease the fuel consumption of gasoline engines. However, for gasoline engines, decreasing the mixture concentration inevitably leads to a reduction in the combustion burning rate. Therefore, increasing the combustion burning rate at low mixture concentrations is a key technical point for lean burn. Many techniques have been applied to overcome these challenges, such as increasing the compression ratio, increasing the spark energy, using multiple sparking, and using stronger reactivity fuel. Nagasawa et al. optimized the injection strategy with a high compression ratio, which could improve thermal efficiency and reduce emissions under high air-fuel ratio conditions [7]. Gong et al. studied the effects of different air-fuel ratios and compression ratios on combustion and emissions characteristics of gasoline engine. The experiment indicate that the optimized compression ratio and air-fuel ratio cannot only improve the thermal efficiency but also reduce particulate matter (PM) emissions [8]. The multiple spark strategy and high energy ignition system could promote ignition stability and improve flame propagation. Proper spark parameters can extend the lean burn limit, and thermal efficiency basically increases as excess air coefficient increases [9].

Improving the fuel reactivity to accelerate the burning rate under low mixture concentration conditions is another way to overcome the difficulties of lean burn. Yu et al. indicated that hydrogen could make combustion more stable, while reduce HC emissions and NO_x emissions simultaneously [10]. Hydrogen port injection achieves more complete combustion due to the accelerated burning rate, and the combustion of a lean burn is closer to the ideal Otto cycle, which results in a higher combustion thermal efficiency [11]. Gong et al. used dual port injection and DI with port injection to achieve lean burns on different engines. Dual port injection uses hydrogen and methanol, while the other engine uses hydrogen for port injection and methanol for DI. The combination of DI and port injection is more conducive to expanding the lean burn limit, and combustion is more stable than dual port injection. DI with port injection results in lower HC and CO emissions, and dual port injection results in lower in NO_x and particulate emissions [12]. Yu et al. investigated the combustion and emissions of ethanol port injection with gasoline direct injection under lean burn conditions. The results suggest that combustion fluctuation increases with increasing Lambda. An optimized lean burn has a higher thermal efficiency than stoichiometric combustion. Because of incomplete combustion, HC emissions and particulate emissions increase significantly, and NO_x emissions correspondingly decrease due to the reduction in the combustion temperature [13]. Li et al. studied the effects of different control strategies on combustion with dimethyl ether (DME) direct injection and gasoline port injection. The experimental results prove that different DI strategies have different impacts on combustion parameters [14]. Wang et al. added a set of intake port methanol injectors to a natural gas lean burn engine and studied the effects of the air-fuel ratio on combustion and emissions. The brake-specific fuel consumption (BSFC) decreases as Lambda increases [15]. Chen et al. studied the effect of spark timing at different methanol ratios on the combustion of lean burn engines. The experimental results show that an increase in the methanol ratio can improve the combustion thermal efficiency, and a specific spark advance angle that maximizes the combustion thermal efficiency exists for each methanol ratio [16]. The application of dual injection with gasoline under stoichiometric condition was also studied. It is found that the application of dual injection allows to yield much lower particulate emissions than single DI or port injection under cold start conditions at different speeds and loads [17–20].

The previous analyses demonstrate that high compression ratio, high spark energy or multiple spark times, and high reactivity fuel could effectively improve the burning rate under lean burn conditions. In this study, the dual injection stratified combustion mode is applied to accelerate the burning rate of lean burn by increasing the fuel concentration near the spark plug. The effects of engine control parameters such as the excess air coefficient (Lambda), direct injection (DI) ratio, spark interval with DI, and DI timing on combustion, fuel consumption, gaseous emissions, and particulate emissions of a dual injection gasoline engine are studied. The main goal is to achieve effective stratified lean burn by adjusting the injection strategy with dual injection, in order to reduce PM emissions while improving fuel consumption.

2 Experiment setups

2.1 Engine setup

The experiment used a 2.0 L displacement supercharged gasoline engine with both direct injection and port injection. The engine parameters are summarized in Table 1. Pi Innovo OpenECU and Bosch UEGO Lambda sensor were used for Lambda closed-loop feedback control, and a Horiba MEXA-7500DEGR was used to monitor the real-time changes in Lambda while testing the gaseous emissions before three-way catalyst (TWC). A DMS500 fast particulate spectrometer was used to collect experimental data on the particle size distribution and particle number. The schematic diagram of the engine test system is displayed in Fig. 1.

Table 1 Test engine specification

Items	Specifications
Displaced volume/mL	1995
Stroke/mm	82
Bore/mm	88
Connecting rod/mm	150.5
Compression ratio	10:01
Number of cylinder	4
Number of valves	4
Intake valve open/(°CA bTDC)	340
Intake valve close/(°CA bTDC)	100
Exhaust valve open/(°CA bTDC)	244
Exhaust valve close/(°CA bTDC)	4

2.2 Experiment setup

The fuel used in the experiment was commercial RON 92 gasoline. The physical parameters of the fuel are shown in Table 2. The experimental condition is selected at 2 bar brake mean effective pressure (BMEP) at a 2000 r/min engine speed. To achieve a stratified lean burn, part of the fuel was injected into the intake manifold during the early intake stroke with port injection whose timing was selected as 350°combustion angle (°CA) before top dead center (bTDC), for which the intake valve had not been opened. The injected fuel evaporated on the intake valve and mixed with fresh air to form a lean mixture. The DI timing was

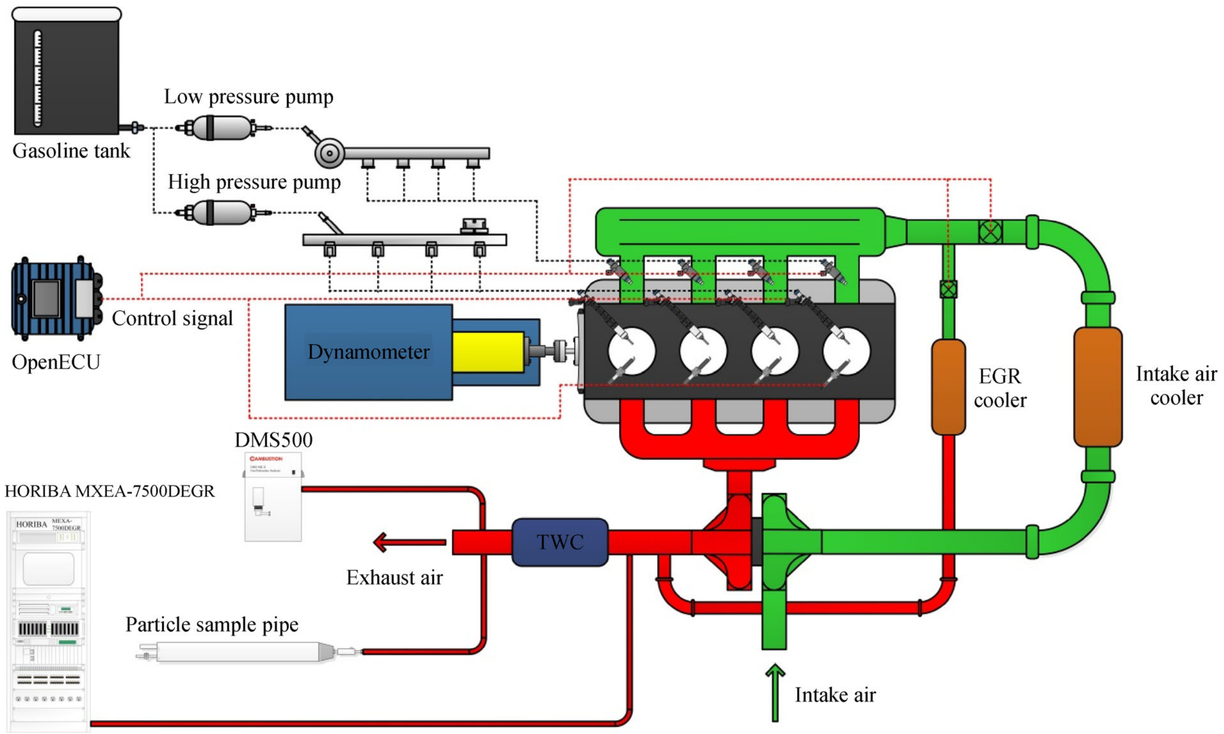


Fig. 1 Dual injection engine system and test bench.

Table 2 Physical parameters of fuel

Parameters	Value
Temperature recovered to 80%/°C	144.6
Temperature recovered to 90%/°C	165.3
Temperature recovered to 95%/°C	178.3
Final boiling point (FBP)/°C	191.7
Residue/vol.%	1.1
Loss/vol.%	1.3
Evaporated at 70°C/vol.%	34.3
Evaporated at 100°C/vol.%	55. 8
Evaporated at 150°C/vol.%	83.8
Evaporated at 180°C/vol.%	96.7

selected to occur at the end of the compression stroke, which was close to top dead center (TDC), and the strong in-cylinder airflow movement could promote the rapid atomization of the DI fuel. The stratified mixture with different fuel concentrations was formed in the combustion chamber. To ensure stable ignition, the spark timing was made different from that of general gasoline engines. The DI fuel diffused to the surroundings as the piston moved upward, and the local high fuel concentration zone near the spark plug gradually expanded where the fuel concentration gradually decreased. The spark timing was selected within a certain range after the DI timing to ensure that the local high fuel concentration zone is near the spark plug. To study the effects of different control parameters on stratified lean combustion, the excess air coefficient (Lambda), DI timing, and spark timing were divided into three groups of experiments with different DI ratios (RDI). When the influence of one parameter was studied, the other two parameters remained unchanged. The experimental test conditions are listed in Table 3 in which aINJ means after injection which is another way to express the spark timing relative to DI timing.

2.3 Mixture formation process of dual injection stratified lean burn

Figure 2 exhibits the schematic diagram of dual injection stratified lean burn. As shown in Fig. 2, Phase I corresponds to the intake stroke (port injection end at 350°CA bTDC, intake valve open at 340°CA bTDC), while Phase II corresponds to the compression stroke. Gasoline was directly injected into the intake manifold as the intake valve was nearly closed. The high temperature of the intake valve could accelerate the evaporation process to form a homogeneous mixture. After the intake valve opened, the homogeneous mixture formed in the intake manifold and more fresh air were drawn into the cylinder, because of the larger throttle opening. Subsequently, the lean mixture was formed continuously when the piston moved upward. As shown in Phase II, the DI fuel was directly injected into the cylinder in the late stage of the compression stroke. The fuel could quickly diffuse to form

Table 3 Test conditions

Speed /(r·min^{-1})	BMEP /bar	PFI timing /(°CA bTDC)	DI ratio /%	Lambda	DI timing /(°CA bTDC)	Spark timing /(°CA aINJ)	Spark timing /(°CA bTDC)
2000	2	350	50 75	1.2	50	8	42
				1.4			
				1.6			
				1.8			
			50	1.4	50	4	46
						6	44
						8	42
						10	40
						12	38
			25 50 75	1.4	30	8	22
					40		32
					50		42
					60		52

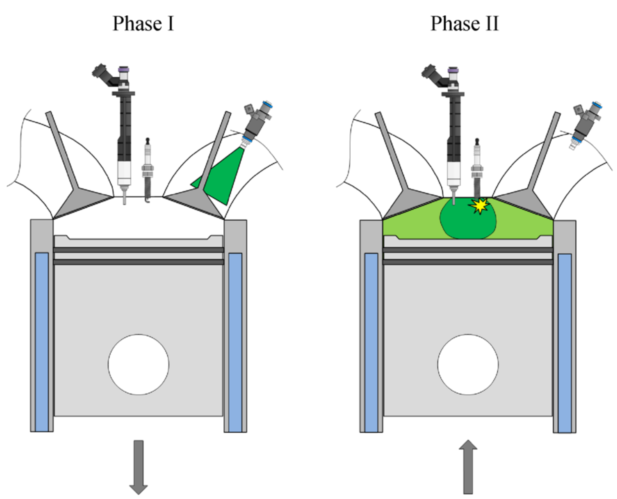


Fig. 2 Schematic diagram of dual injection stratified lean burn.

a local fuel rich zone. Moreover, in order to ensure a proper air-fuel ratio near the spark plug, the spark timing had to be selected within a limited range after the direct injection.

3 Results and discussion

3.1 Effect of Lambda on combustion and emissions for dual injection stratified lean burn

The effect of Lambda on combustion pressure and combustion heat release rate at different DI ratios is presented in Fig. 3 while that on combustion phase and combustion duration at different DI ratios is given in Fig. 4.

It can be observed from Fig. 3 that the in-cylinder combustion pressure increases with increasing Lambda at different DI ratios caused by the increase of intake air, but the trends of Lambda are not consistent at different DI ratios. The in-cylinder combustion pressure is basically linear with Lambda at the high DI ratio, but it becomes relatively closer at the low DI ratio. The results of combustion duration show that the low DI ratio leads to a greater combustion duration variation with Lambda than the high DI ratio. At a low DI ratio, the majority of fuel evaporates in the intake manifold to form the lean mixture, while the rest is injected to form the local rich mixture through the DI system. After ignition, the combustion flame rapidly diffuses from the local rich mixture to the surrounding lean mixture as the piston moves down. For the high DI ratio, the surrounding lean mixture fuel concentration decreases as the local rich mixture fuel concentration increases. Lambda has little effect on the combustion process in the local rich mixture, which leads to less combustion duration variation with Lambda. For the low DI ratio, there is more fuel in the surrounding lean mixture than in the local rich mixture, and Lambda greatly affects the combustion flame propagation speed in the surrounding lean mixture, which leads to a greater combustion duration variation. Because of the decrease in fuel concentration in the local rich mixture, the proportion of the rapid combustion period (CA50–CA10) in the entire combustion process at a low DI ratio is larger than that at a high DI ratio, while the proportion of the after combustion period (CA90–CA50) in the entire combustion process at a high DI ratio is much larger than that at a low DI ratio. For homogeneous lean burn, the combustion flame propagation speed decreases with increasing Lambda, which means that the interval between the spark and ignition becomes longer as the fuel mixture concentration decreases [21–24]. For dual injection

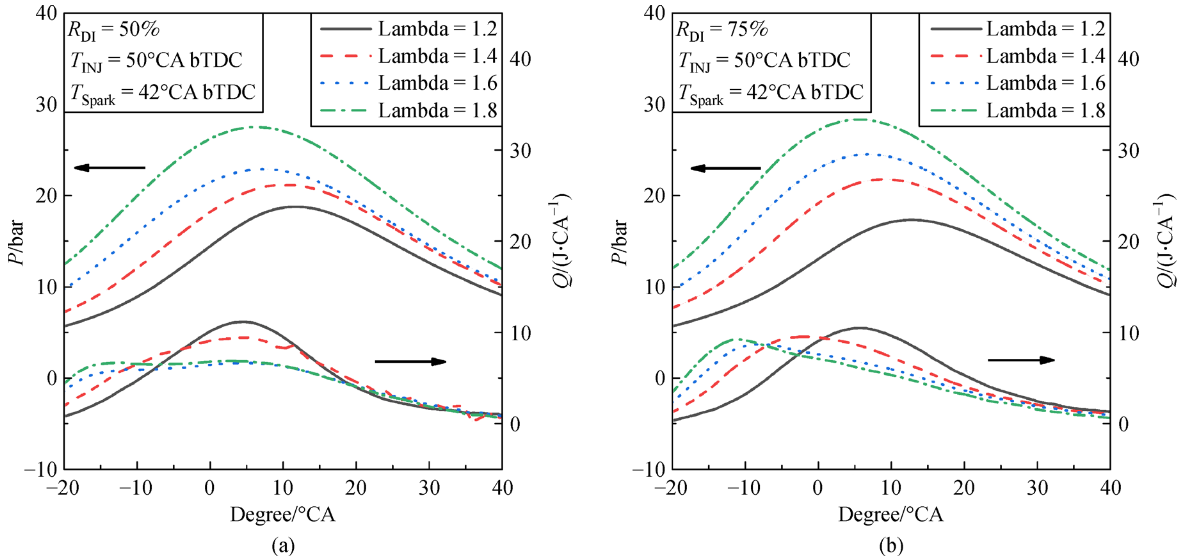


Fig. 3 Effect of Lambda on combustion pressure and combustion heat release rate at different DI ratios. (a) 50%; (b) 75%.

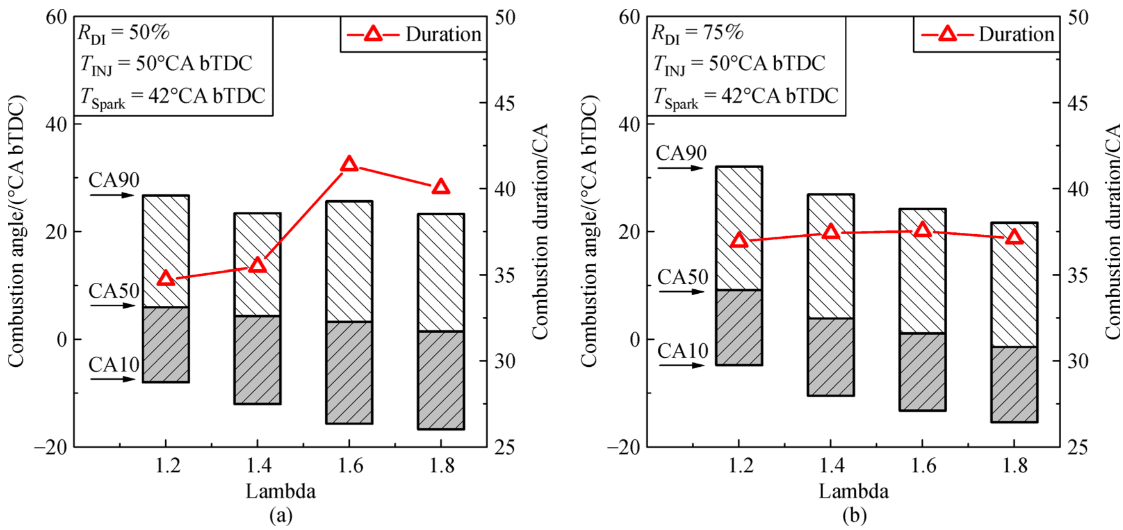


Fig. 4 Effect of Lambda on combustion phase and combustion duration at different DI ratios. (a) 50%; (b) 75%.

stratified lean burn at both high and low DI ratios, the CA10 shifts forward with increasing Lambda, while the interval between the DI timing and spark timing remains constant. The ignition delay period (CA10–T_{Spark}) becomes shorter with increasing Lambda, which is in contrast to the results for the homogeneous lean burn. In the stratified mixture, the flame first spreads in the local rich mixture with a high fuel concentration, and an increase in Lambda enhances the in-cylinder airflow movement intensity. The enhanced in-cylinder airflow movement accelerates the combustion flame propagation speed in the local rich mixture. Therefore, the ignition delay period of the dual injection stratified lean burn decreases with increasing Lambda. In the process of gasoline combustion, the combustion flame propagation speed is dominated by many elementary reactions of small radicals at high temperatures. A high fuel concentration is beneficial to the generation of radicals such as H and OH and accelerates the combustion flame propagation speed [25–28]. In homogeneous lean burn, the fuel has been uniformly mixed before sparking, which means that the mixture fuel concentration is fixed. The concentration of radicals generated after ignition is proportional to the mixture fuel concentration. Therefore, the combustion flame propagation speed decreases with increasing intake air volume. As a result, the chemical kinetics are the main combustion driving factor for homogeneous lean burn. For the dual injection stratified lean burn, because regions with different fuel concentrations are present, the enhanced in-cylinder airflow movement accelerates the diffusion velocity of the radicals. An increase in Lambda shortens the ignition delay period, which means the driving factor of aerodynamics is strengthened during the combustion process for dual injection stratified lean burn.

In the study of fuel consumption, this paper uses the BSFC reduction rate (ΔBSFC), whose definition is shown in Eq. (1).

$$\Delta\mathrm{BSFC} = (\mathrm{BSFC}_{\mathrm{Origin}} - \mathrm{BSFC}_{\mathrm{Exp}})/\mathrm{BSFC}_{\mathrm{Origin}}, \quad (1)$$

where $\mathrm{BSFC}_{\mathrm{Origin}}$ is the fuel consumption of the original gasoline direct injection (GDI) engine using the original control parameters under stoichiometric combustion, $\mathrm{BSFC}_{\mathrm{Exp}}$ is the fuel consumption measured under the experimental test condition. Figure 5 shows the change of BSFC and ΔBSFC with Lambda. The BSFC decreases with the increase of Lambda, and the high DI ratio decreases ΔBSFC. For lean burn, combustion and pumping loss are the two major factors which can affect fuel consumption. As Lambda increases, the throttle opening becomes larger and the pumping loss is reduced, which is more beneficial to reduce fuel consumption. The increase in Lambda will gradually aggravate the combustion. When the fuel consumption lost by the combustion deterioration exceeds the fuel consumption saved by the pumping loss reducing, ΔBSFC will deteriorate or even become negative. The lowest BSFC is achieved at a DI ratio of 50% with Lamda = 1.4 which is reduced by 9.05% compared to the original engine. When Lambda≥1.6–Lambda≥1.6, although the throttle opening is larger, ΔBSFC is greatly reduced compared to other operating conditions. When Lambda = 1.8, BSFC has greatly exceeded the original fuel consumption. It can be seen from the combustion that the combustion angle moves advanced which leads to the fact that CA50 is too close to TDC, which results in the fact that the burning mixture work process coincides with part of the piston upward process. At the same time, the combustion flame propagation speed in the surrounding lean mixture decreases and the ignition delay period increases, both of which lead to a deteriorated combustion. Therefore, fuel consumption does not decrease with the further pumping loss reduction but increases with the combustion deterioration.

For lean burn, incomplete combustion generates a large amount of total hydrocarbon (THC), and Lambda has exceeded the operating range in which the TWC can efficiently remove NO_x. Thus, these two gaseous emissions are two important indicators in the research. The main sources of THC are incomplete combustion, cylinder wall quenching, the slit effect formed by pistons and cylinder walls, and the adsorption of fuel films and deposits [29,30]. For the dual injection stratified lean burn, a large number of small radicals are generated by the fuel in the local rich mixture at the beginning of combustion. Some of these radicals participate in the oxidation reaction at the flame front, and the others diffuse to the unreacted area to form unburned THC and nucleation particles. NO_x is mainly derived from the reaction between the activated nitrogen molecules and the excessively rich oxygen molecules at a high temperature. Therefore, high temperature and oxygen enrichment are two indispensable conditions for NO_x generation. As shown in Fig. 6, the THC emissions gradually increase with increasing Lambda, while the NO_x emissions gradually decrease. When Lambda is small, the THC emissions are relatively close at different DI ratios, while the NO_x emissions change greatly with the DI ratio. When Lambda is large, the NO_x emissions are relatively close at different DI ratios, while the THC emissions change greatly with the DI ratio. The combustion analysis reveals that an increase in Lambda leads to a decrease in the surrounding lean mixture fuel concentration, and thus, the combustion flame propagation speed decreases, and the

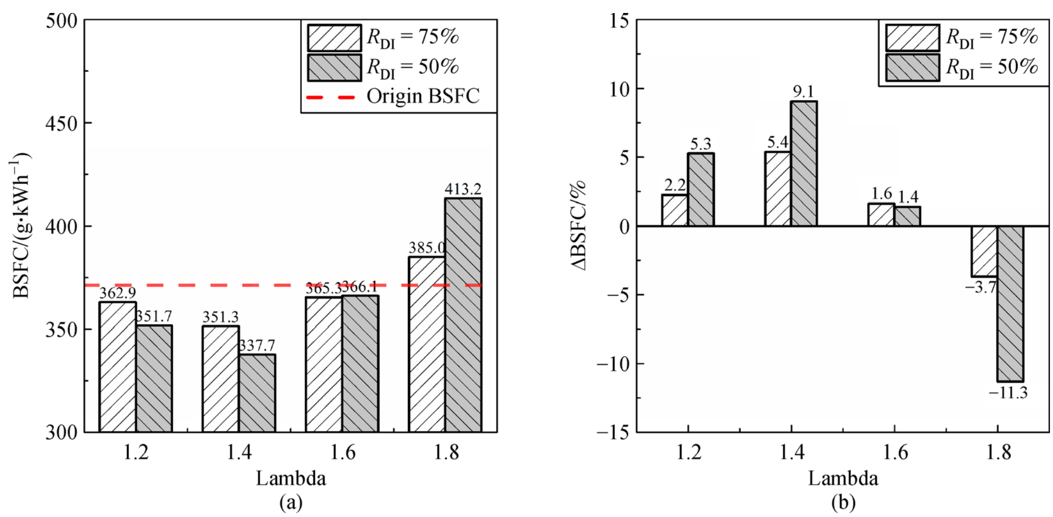


Fig. 5 Effect of Lambda on BSFC and ΔBSFC at different DI ratios.
(a) BSFC; (b) ΔBSFC.

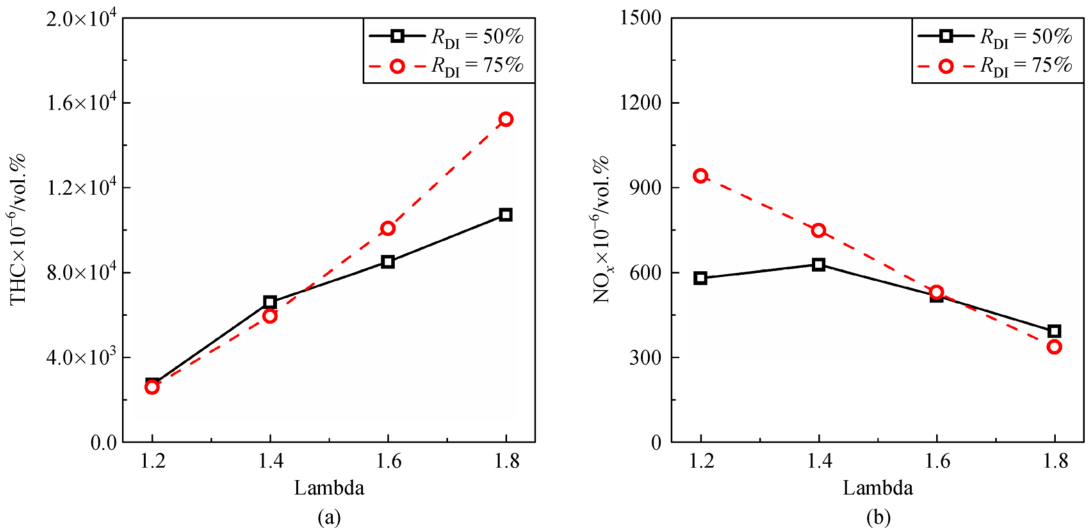


Fig. 6 Effect of Lambda on gaseous emissions at different DI ratios. (a) 50%; (b) 75%.

proportion of unburned fuel increases. The heat capacity of the mixture increases with increasing Lambda, and the heat release rate decreases with combustion deterioration. As a result, the total combustion heat release decreases, which means that the in-cylinder combustion temperature decreases. Therefore, the THC emission increases because of the increase in incomplete combustion, and NO_x decreases because of the decrease in the in-cylinder combustion temperature. At a high DI ratio, relatively more small radicals decompose in the local rich mixture. These radicals increase the combustion flame propagation speed and diffuse to the unburned area, resulting in a higher THC emission. When Lambda is small, the fuel concentration in the local rich mixture is higher at a high DI ratio, which results in a relatively high combustion temperature with oxygen enrichment. Therefore, at a Lambda of 1.2 and 1.4, the NO_x emissions at high DI ratios are both higher than those at low DI ratios. When Lambda increases, the combustion temperature decreases rapidly, and the NO_x generation process is suppressed. Therefore, the NO_x emissions at both high and low DI ratios remain at relatively low and similar levels.

In general, particles in the range of 5–30 nm are classified as being in the nucleation mode, and particles in the range of 30–1000 nm are classified as being in the accumulation mode [31,32]. The small radicals generated during the combustion process are referred to as intermediate combustion products. The nucleation mode mainly consists of the nucleation of intermediate products, and the accumulation mode mainly consists of the accumulation of ash and sulfate adsorbed by the intermediate products. For the dual injection stratified lean burn, as the fuel concentration stratification provides a large number of intermediate combustion products, the combustion duration and temperature directly affect the process of the intermediate combustion products transforming into accumulation particles. As shown in Fig. 7, the particle size distribution at a high DI ratio is unimodal, while the DI ratio is reduced, and the distribution becomes bimodal. The larger the DI ratio is, the higher the peak of the accumulation particles. The particle size peak position decreases with the DI ratio. Reducing the DI ratio can significantly reduce the particle size, and the geometric mean diameter (GMD) corresponding to a low DI ratio is much smaller than that of a high DI ratio as shown in Fig. 8. The combustion phase results show that the first half of the combustion at a low DI ratio constitutes a greater proportion of the entire combustion duration and that the flame front spreads more stably, resulting in most of the intermediate combustion products being directly converted into nucleation particles, which inhibits the particle generation progress. Therefore, the nucleation mode ratio at a low DI ratio is much higher than that at a high DI ratio, while the particle number (PN) is much lower than that at a high DI ratio as shown in Fig. 9. At a high DI ratio, because the combustion process proceeds faster in the local rich mixture, the first half of the combustion constitutes a lower proportion of the entire combustion duration. The higher fuel concentration in the local rich mixture can provide more intermediate combustion products. Some of these products begin to transform into accumulation particles in the latter half of the combustion with a higher temperature and longer duration. Therefore, the accumulation mode accounts for the majority of intermediate combustion products at a high DI ratio. At a low DI ratio, the change in Lambda has minimal effect on the particulate emissions, while at a high DI ratio, the increase in Lambda greatly reduces particulate emissions. This occurs because an increase in Lambda reduces the in-cylinder combustion temperature, which inhibits the conversion of the

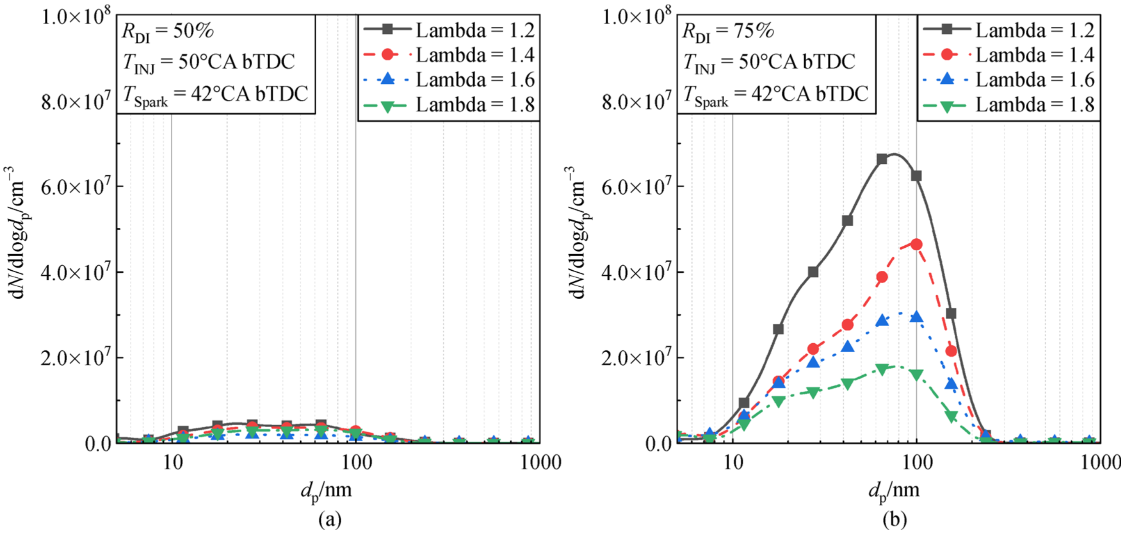


Fig. 7 Effect of Lambda on particle size distribution at different DI ratios.
(a) 50%; (b)75%.

intermediate combustion products to the accumulation mode. Thus, an increase in Lambda reduces the PN and increases the proportion of the nucleation mode in the total particulates. The PM at a low DI ratio is significantly smaller than that that at a high DI ratio, and Lambda has little effect on the PM emission in this condition.

3.2 Effect of DI timing on combustion and emissions for dual injection stratified lean burn

In the dual injection stratified lean burn, to ensure reliable ignition, the spark timing is selected to be near the end of DI. Therefore, the phase interval between the spark timing and the DI timing is used as the spark interval to study the influence of the spark timing. During the experiment, when the spark interval is lower than 4°CA aINJ, the fuel concentration near the spark plug is too high, which makes ignition more difficult and leads to an unstable combustion. When the spark interval is greater than 12°CA aINJ, the stratified mixture becomes more homogeneous, which makes the combustion flame propagation speed slow. The spark interval between 4°CA aINJ and 12°CA aINJ is called the spark window. Figure 10 shows the effect of the spark interval on combustion in dual injection stratified lean burn.

The research on spark timing shows that the spark interval within the spark window has relatively weak effects on combustion and emissions. Therefore, although the spark timing follows the DI timing, the changes in combustion and emissions are mainly caused by the DI

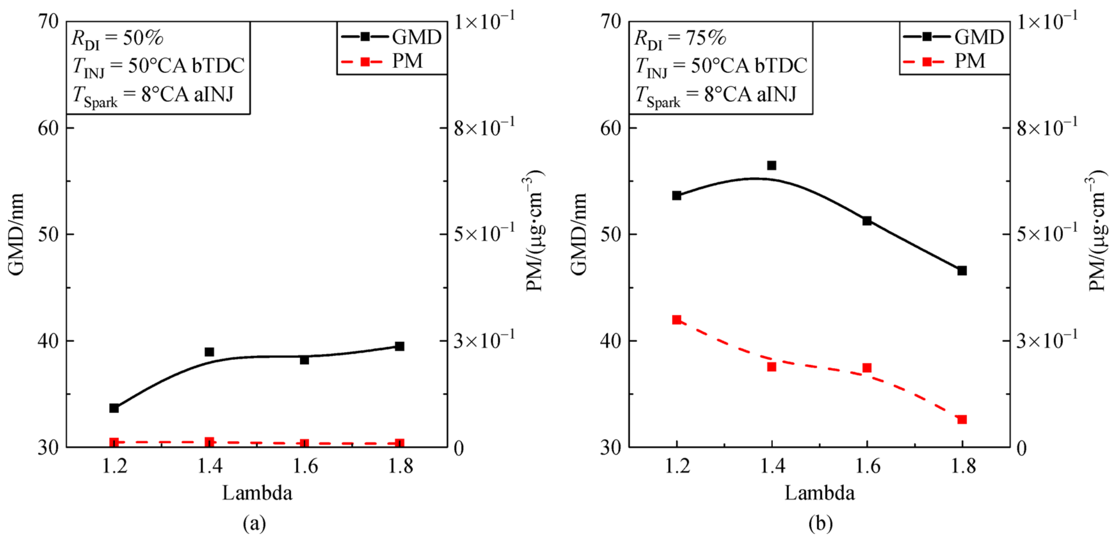


Fig. 8 Effect of Lambda on GMD and PM at different DI ratios.
(a) 50%; (b)75%.

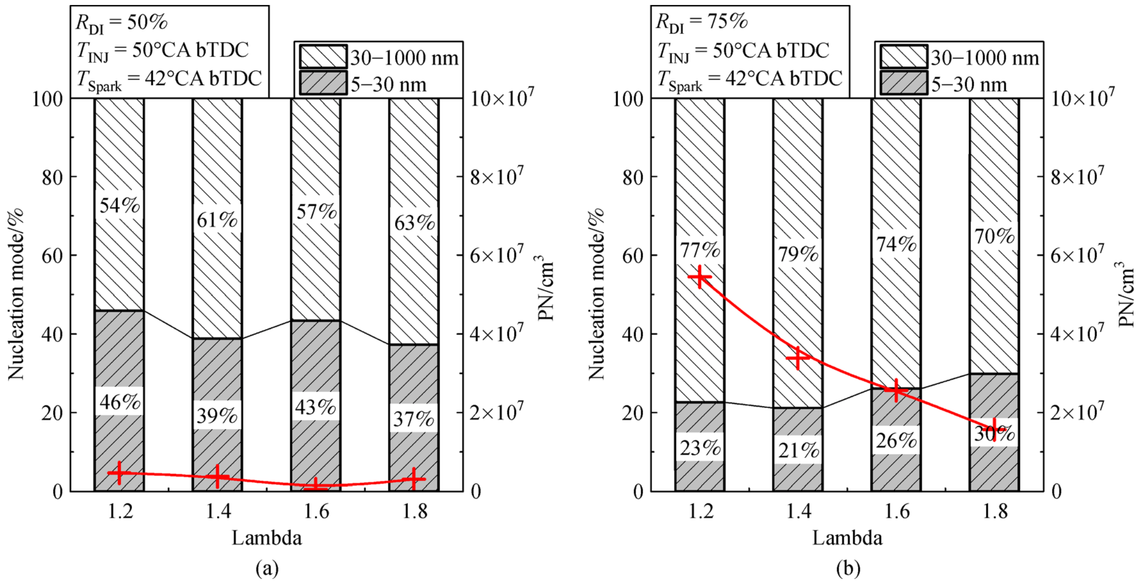


Fig. 9 Effect of Lambda on mode ratio and PN at different DI ratios.
(a) 50%; (b) 75%.

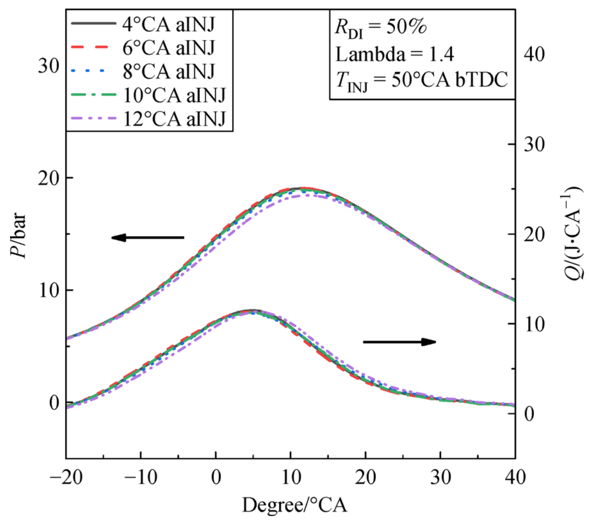


Fig. 10 Effect of spark timing on combustion pressure and combustion heat release rate in stratified lean burn.

timing. Figures 11 and 12 show that as the DI timing advances, the in-cylinder combustion pressure and in-cylinder combustion heat release rate increase as the combustion phase advances, but the combustion duration greatly decreases. The DI timing directly changes the start time of the combustion, and the combustion phase advances. The in-cylinder peak heat combustion release rate advances, which leads to an increase in the in-cylinder combustion pressure. The DI ratio changes the fuel concentration in the local rich mixture, which affects the combustion flame propagation speed in different concentration mixtures. When the fuel concentration is low, the corresponding combustion flame propagation speed is correspondingly low.

Figure 13 shows the effects of different injection timings on the BSFC and ΔBSFC. When the DI timing is close to the TDC, the combustion phase is greatly delayed with the DI timing. The burning mixture is deteriorated, leading to the increase of fuel consumption. When the DI timing is far away from the TDC, the combustion phase advances, which causes the mixture to do work while the piston is moving upward. Therefore, the maximum ΔBSFC at 60°CA bTDC is lower than that at 50°CA bTDC. The lowest BSFC point was achieved at 2 bar BMEP and 2000 r/min, with a 50% DI ratio at a Lambda of 1.4, 50°CA bTDC DI timing, and a 8–10°CA aINJ spark interval.

From Fig. 14, when the DI timing is farther from the TDC, the combustion is more stable, and the combustion phase is further forward, leading to the increase of NO_x emission because of the higher in-cylinder combustion temperature. For the dual injection stratified lean burn, the main source of THC is the incomplete combustion and evaporation of the fuel film on the top of the piston. When RDI = 50%, the incomplete combustion produces relatively less THC, and the advanced DI timing can provide more time for fuel film evaporation. Therefore, the THC emission decreases with the advancement of the DI timing. When RDI = 25%, the THC mainly comes from the incomplete combustion of the surrounding lean mixture. Because the DI ratio is low, the DI timing has a weaker impact. When RDI = 75%, the THC emissions mainly come from the evaporation of the fuel film on the top of the piston, which is thicker. The in-cylinder airflow movement affects the fuel film evaporation, especially when the piston is close to the TDC. Thus, for a high DI ratio, a

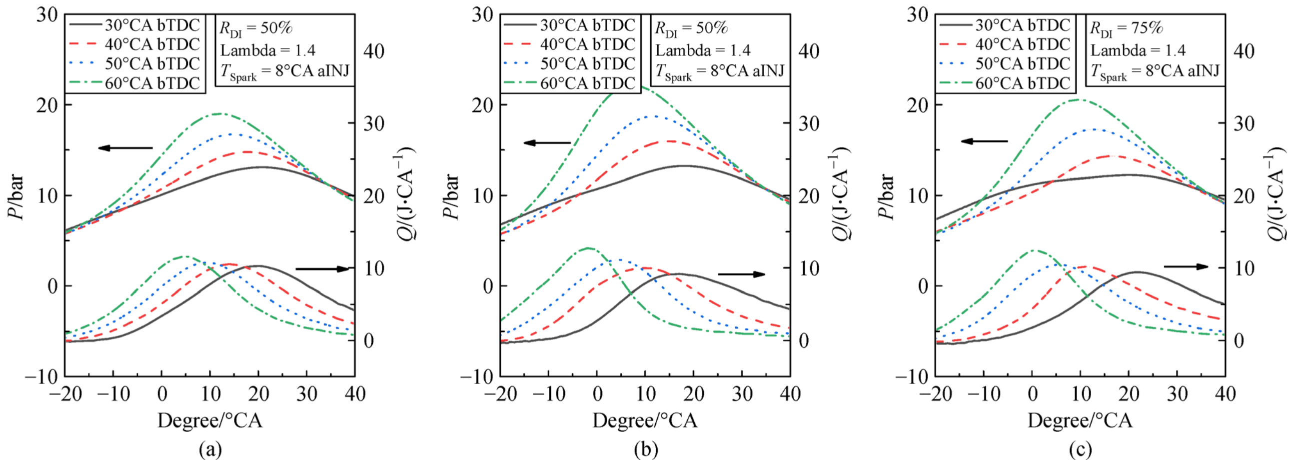


Fig. 11 Effect of DI timing on combustion pressure and combustion heat release rate at different DI ratios.
(a) 25%; (b) 50%; (c) 75%.

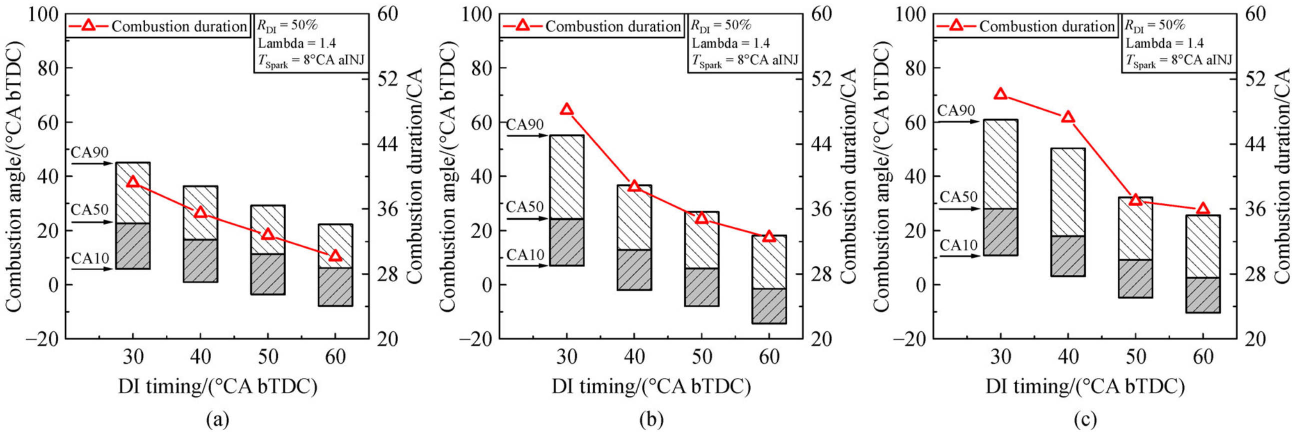


Fig. 12 Effect of DI timing on combustion phase and combustion duration at different DI ratios.
(a) 25%; (b) 50%; (c) 75%.

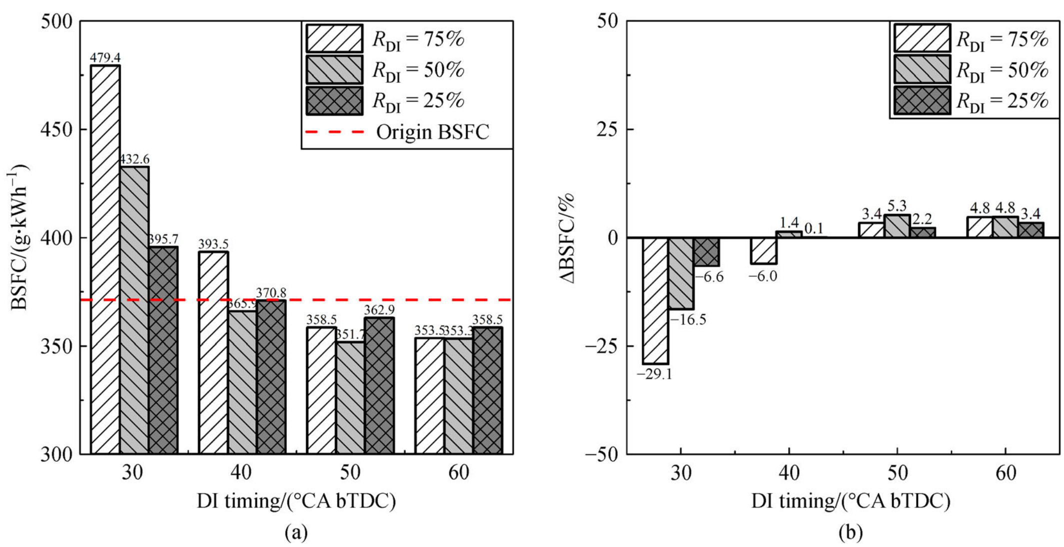


Fig. 13 Effect of DI timing on BSFC and ΔBSFC at different DI ratios.
(a) BSFC; (b) ΔBSFC.

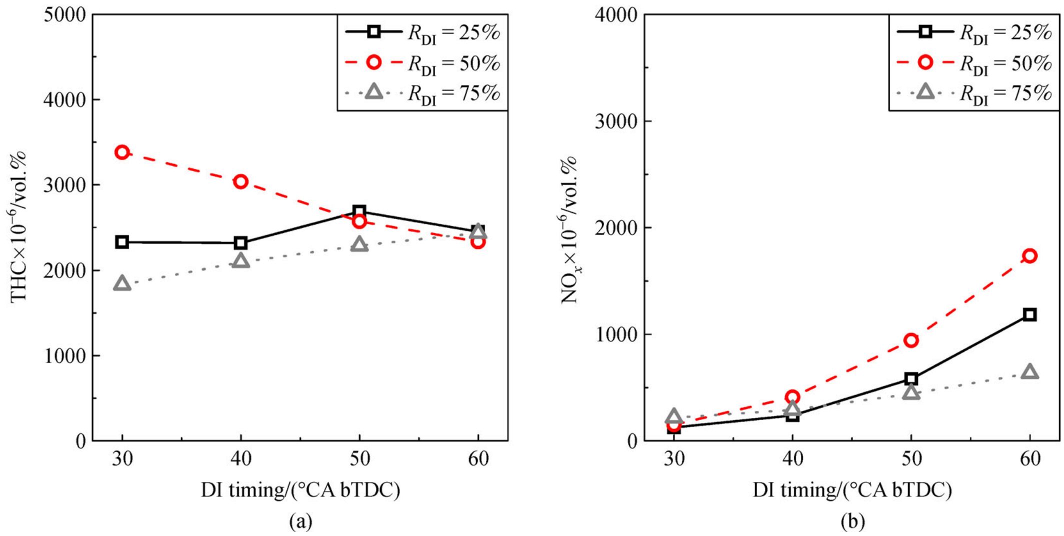


Fig. 14 Effect of DI timing on gaseous emissions at different DI ratios. (a) THC; (b) NO_x.

delay in the DI timing can improve the THC. For the minimum BSFC point, the NO_x emissions measured before TWC are 71.31% lower than that of the original engine, but the THC emissions exhibit a greater increase.

Figures 15, 16, and 17 show the effects of the DI timing on the particle size distribution, the total particle number and mode ratio, and the PM and GMD at different DI ratios, respectively. The influence of the DI timing on the mode ratio at a low DI ratio is not consistent with that of other ratios. When RDI⩾50%, the accumulation mode ratio decreases with the advancement of the DI timing, mainly because the advancement of the DI timing improves the quality of the air-fuel mixture. The advancement in the combustion phase and the shortening of the combustion duration inhibit the generation process of the accumulation mode. When RDI = 50%, the PN greatly decreases with the advancement of the DI timing, while the accumulation mode ratio increases. Therefore, the trends of GMD and PM are different from those of the other DI ratios. The trend of GMD is the same as that of the accumulation ratio, which greatly increases with an advancement in DI timing, while PM decreases first and then increases. The PN emission at the minimum BSFC point is 81.45% lower than those of the original engine, which means that the dual injection stratified lean burn can greatly reduce both fuel consumption and PN emissions.

3.3 Optimal control strategy for dual injection stratified lean burn

Based on the above analysis, the effects of the different control parameters of the dual injection stratified lean burn on combustion, fuel consumption and emissions are summarized, yielding the optimized control strategy road map shown in Fig. 18. First, an appropriate Lambda should be chosen. Too low a Lambda means a greater pumping

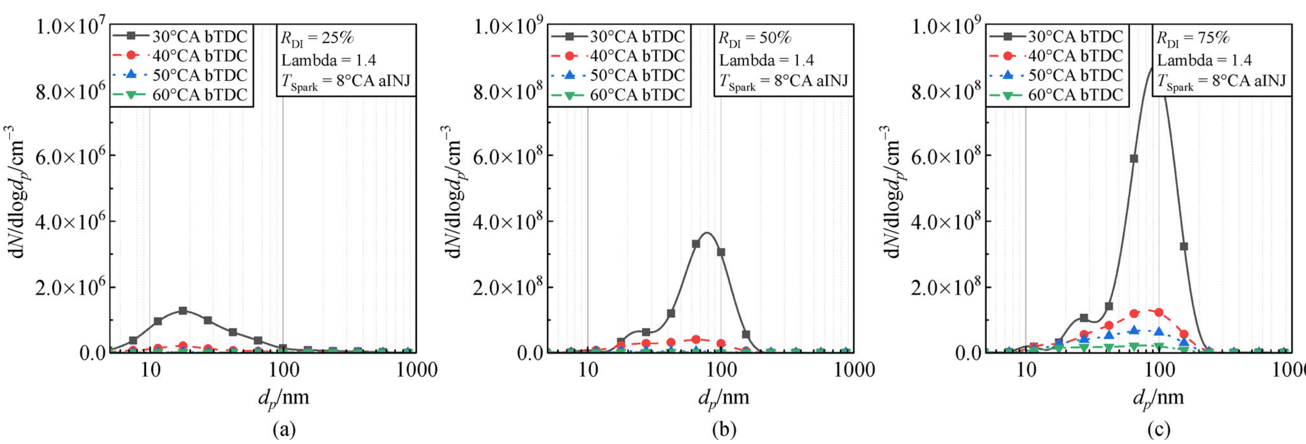


Fig. 15 Effect of DI timing on particle size distribution at different DI ratios. (a) 25%; (b) 50%; (c)75%.

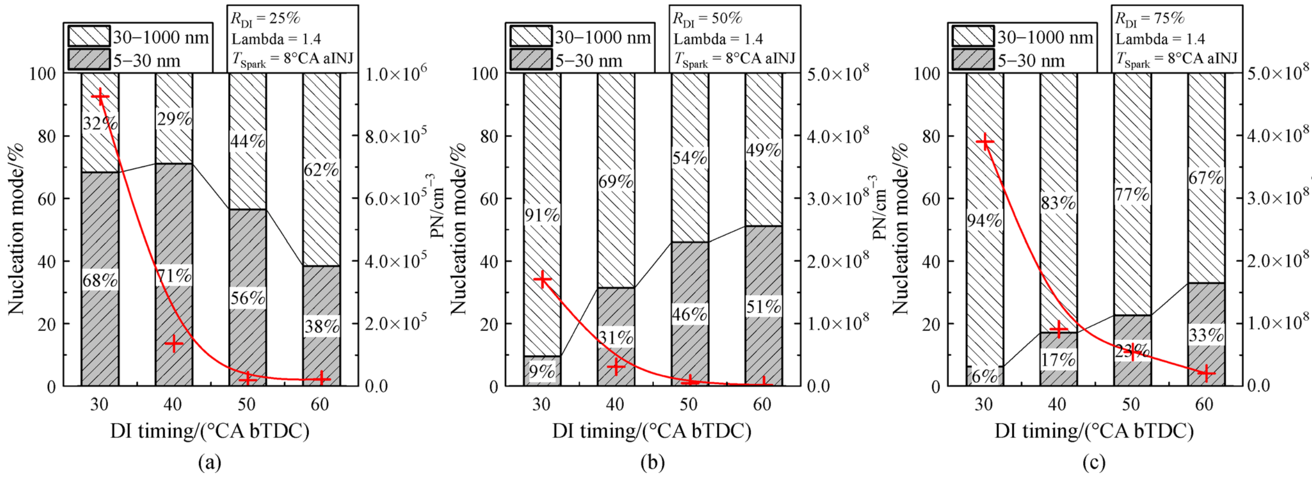


Fig. 16 Effect of DI timing on mode ratio and PN at different DI ratios.
(a) 25%; (b) 50%; (c) 75%.

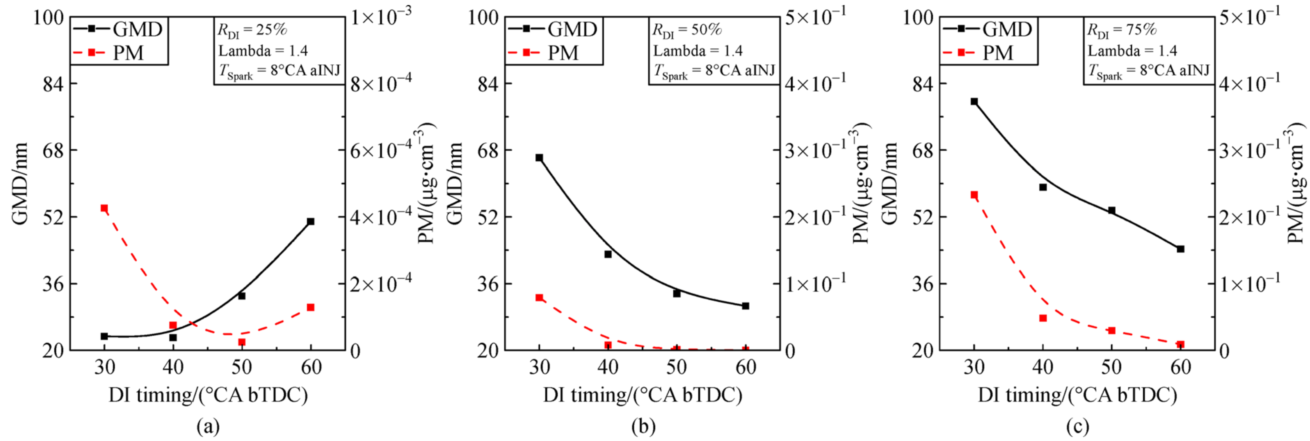


Fig. 17 Effect of DI timing on GMD and PM at different DI ratios.
(a) 25%; (b) 50%; (c) 75%.

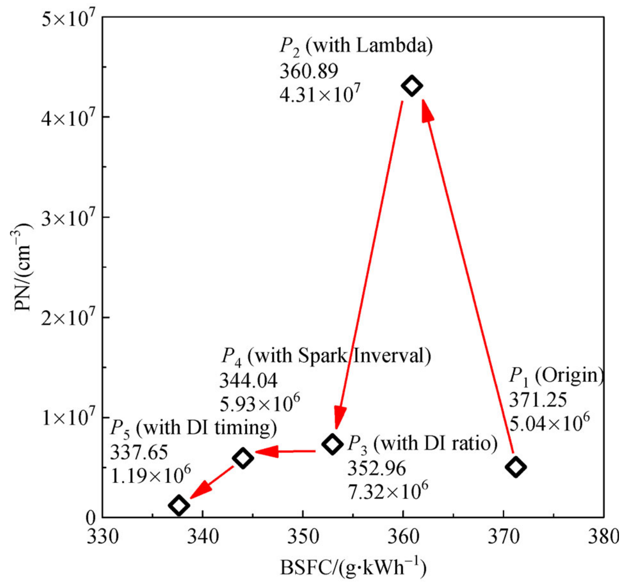
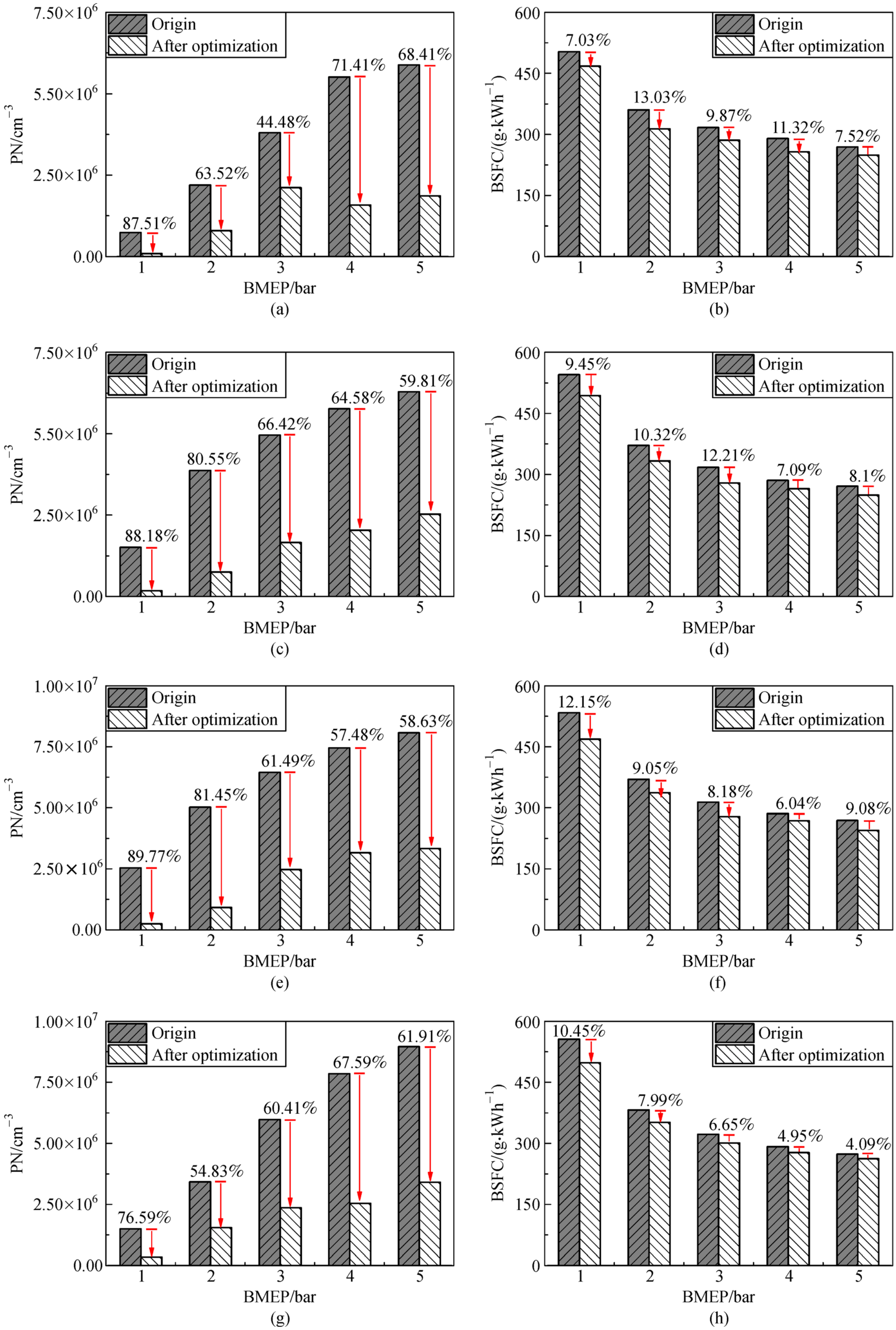


Fig. 18 Roadmap of optimal control strategy for dual injection stratified lean burn.

loss, while too high a Lambda increases the fuel consumption. The suitable Lambda values are in the range of 1.2–1.4 without changing the combustion chamber structure. Next, an appropriate DI ratio should be chosen to reduce the particulate emissions. A higher DI ratio could cause a significant increase in the particulate emissions, while a lower DI ratio could not increase the combustion flame propagation speed. The suitable DI ratio is 30%–50%. Then, an appropriate spark interval between 8 and 10°CA aINJ should be chosen. Finally, an appropriate DI timing should be chosen, which can greatly reduce the fuel consumption while minimizing the particulate emissions. The suitable DI timing is approximately 50°CA bTDC.

3.4 Application of the optimal control strategy at different speeds and loads

As shown in Fig. 19, the optimized control strategy with engine speed and load varying from 1200 r/min to 3200 r/min and 1 bar to 5 bar respectively have been



Origin
After optimization
PN/cm^-3
BSFC/(g.kWh^-1)
BMEP/bar
87.51%
63.52%
44.48%
71.41%
68.41%
(a)
7.03%
13.03%
9.87%
11.32%
7.52%
(b)
88.18%
80.55%
66.42%
64.58%
59.81%
(c)
9.45%
10.32%
12.21%
7.09%
8.1%
(d)
89.77%
81.45%
61.49%
57.48%
58.63%
(e)
12.15%
9.05%
8.18%
6.04%
9.08%
(f)
76.59%
54.83%
60.41%
67.59%
61.91%
(g)
10.45%
7.99%
6.65%
4.95%
4.09%
(h)

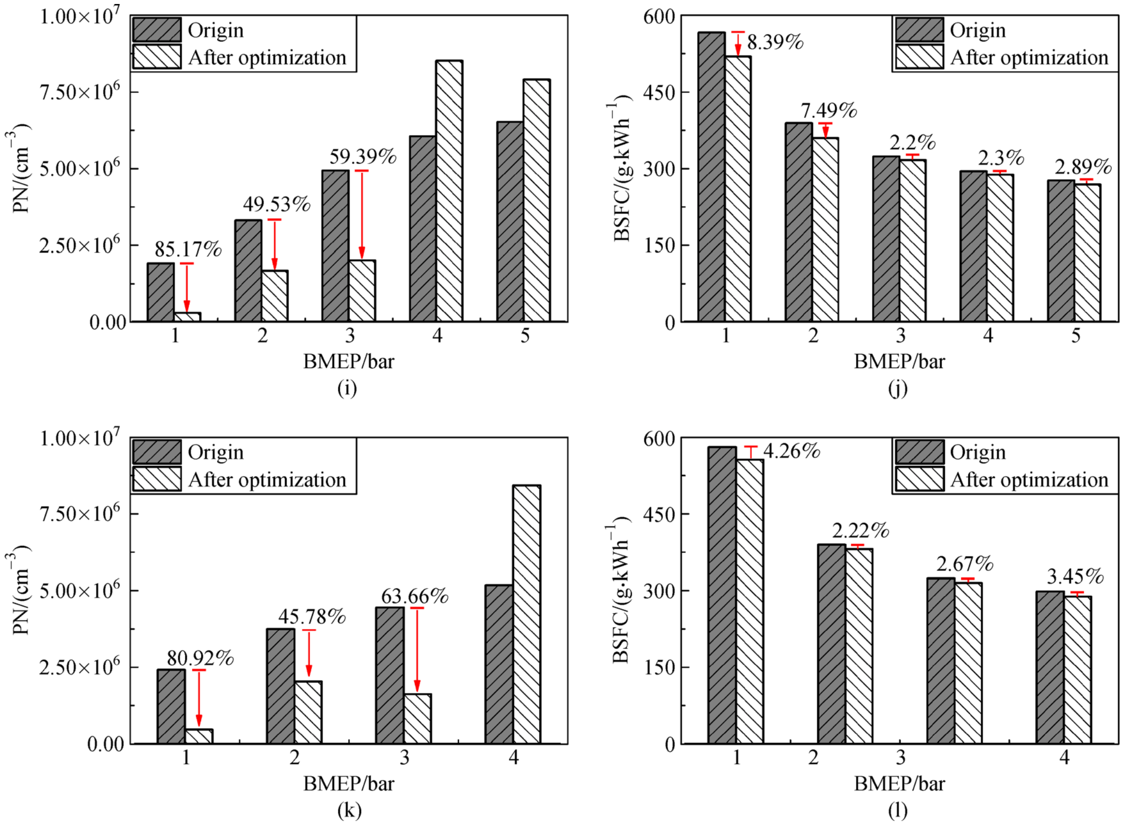


Fig. 19 Optimization of PN and BSFC at different speeds and loads.
(a) PN/Speed = 1200 r/min; (b) BSFC/Speed = 1200 r/min; (c) PN/Speed = 1600 r/min; (d) BSFC/Speed = 1600 r/min; (e) PN/Speed = 2000 r/min; (f) BSFC/Speed = 2000 r/min; (g) PN/Speed = 2400 r/min; (h) BSFC/Speed = 2400 r/min; (i) PN/Speed = 2800 r/min; (j) BSFC/Speed = 2800 r/min; (k) PN/Speed = 3200 r/min; (l) BSFC/Speed = 3200 r/min.

applied for dual injection stratified lean burn. It should be pointed out that, due to the limitation of the supercharger, the intake air is not sufficient when the load is higher than 5 bar, which restricts further applications at higher loads. It can be seen from the Fig. 19 that, with the increase of load at different speeds, the BSFC reduction rate is reduced. When the speed is lower than 2400 r/min with loads below 5 bar, the BSFC reduction rate is higher than 5%, and the PN reduction rate is more than 50%. However, when the speed is higher than 2800 r/min, the BSFC reduction rate is decreased. Thus, the dual injection stratified lean burn mode could effectively reduce fuel consumption and particulate emissions in a wide range of engine conditions.

4 Conclusions

This paper analyzes the effects of Lambda, the spark interval, and the DI timing on dual injection stratified lean burn at different DI ratios. The effects of different control parameters on the combustion, gaseous emissions and particulate emissions are analyzed in detail. The followings conclusions can be drawn:

Based on the adjustment of the dual injection strategy, a stratified mixture is formed in the cylinder. The lean burn limit can be extended to Lambda = 1.8 at a low compression ratio of 10. Different from other combustion modes, there is a spark window for dual injection stratified lean burns. Within the spark window, the spark interval has a very weak influence on combustion.

For dual injection stratified lean burn, many intermediate combustion products are derived from the local high fuel concentration zone near the spark plug and converted into the accumulation mode through the later stage of combustion. Therefore, based on the adjustment of the DI ratio and DI timing, the particulate emissions can be greatly reduced.

When the dual injection stratified lean burn is optimized, the appropriate Lambda should be selected first. Suitable Lambda values are in the range of 1.2–1.4 without changing the combustion chamber structure. The suitable DI ratio should be 30%–50%, and the appropriate spark interval should be 8–10°CA aINJ. The DI timing has a greater impact on the fuel consumption and particulate emissions, and the appropriate DI timing of approximately 50°CA bTDC should be selected.

The optimized control strategy has been applied for the dual injection stratified lean burn mode, which could effectively reduce fuel consumption and particulate emissions in a wide range of engine conditions.

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Notations

SI	Spark ignition
PFI	Port fuel injection
GDI	Gasoline direct injection
BMEP	Brake mean effective pressure
TDC	Top dead center
bTDC	Before TDC
aINJ	After injection timing
CA10	Crank angle at 10% of total heat release
CA50	Crank angle at 50% of total heat release
CA90	Crank angle at 90% of total heat release
BSFC	Brake specific fuel consumption
TWC	Three way catalyst
GMD	Geometric mean diameter
PM	Particle mass

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