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# Impact of pilot diesel ignition mode on combustion and emissions characteristics of a diesel/natural gas dual fuel heavy-duty engine

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## HIGHLIGHTS

- Combustion performance of a dual fuel heavy-duty engine was examined.
- Diesel injection timing was controlled over a wide range at a light load operation.
- Two-stage autoignition mode was observed with advancing injection timing.
- Diesel ignition mode has an obvious effect on the following combustion quality.
- Higher thermal efficiency and lower emissions can be achieved simultaneously.

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## ABSTRACT

The brake thermal efficiency and exhaust emission issues are still not fully-resolved to diesel/natural gas dual fuel engines. To better understand the effect of pilot diesel ignition mode on combustion and emissions characteristics of dual fuel engines, a detailed study concerned with diesel injection timing was conducted. The testing work was operated on a 6-cylinder turbocharged intercooler diesel/natural gas dual fuel heavy-duty engine at light load operations, and diesel injection timing was controlled over a very wide range. The investigated results show that the diesel injection timing ($T_{inj}$) has an obvious effect on pilot diesel ignition mode. A significant advancing $T_{inj}$ leads to pilot diesel ignition mode differs from traditional diesel engine compression ignition mode in the sense that it does not occur at a specific place in the spray, which is a two-stage autoignition mode. With advancing $T_{inj}$, engine combustion and emissions characteristics, including cylinder pressure, cylinder temperature, heat release rate, start of combustion (SOC), ignition delay, combustion duration, crank angle of 50% heat release (CA50), nitrogen oxides (NOx) and total hydrocarbon (THC), show completely different variation trends in different ignition modes. Overall, higher thermal efficiency and lower emissions can be achieved simultaneously in two-stage autoignition mode. Satisfactory results can be obtained with higher brake thermal efficiency (35%), lower NOx (60 ppm) and THC (0.4%) emissions, when $T_{inj}$ is 42.5 °CA BTDC.
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## 1. Introduction

As known, thermal efficiency and emission performance of internal combustion engine are still hot issues for all of research institutes and engine researchers around the world, and never stop. In return, more and more valuable research achievements emerge in an endless stream [1–3].

In term of alternative fuel, natural gas has recently emerging as a promising automotive alternative fuel because of its rich reserves and clean emissions. After long-term unremitting efforts of research, many proven techniques have been widely used [4–7]. Despite all this, a lot of further study is still in progress, one of which is about advanced diesel/natural gas combustion process and control techniques.

The development of advanced techniques of diesel engine, gasoline engine and natural gas engine brings many novel ideas to dual fuel engine [8–10]. In recent decades, the control technology of diesel engine made great progress, especially engine electronic control technology and high pressure common rail diesel fuel injection technology, which promoted the progresses of diesel/natural gas dual fuel engine combustion process and control techniques [11–13]. The latest researches show that Controlled Auto-Ignition (CAI) and

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**Nomenclature**

| | | | |
|---|---|---|---|
| BMEP | brake mean effective pressure | NOx | nitrogen oxides |
| BTDC | before top dead center | SOC | start of combustion |
| CA | crank angle | THC | total hydrocarbon |
| CA50 | crank angle of 50% heat release | $T_{inj}$ | the pilot diesel injection timing |
| CI | compression ignition | $\lambda$ | the air excess ratio |
| CNG | compressed natural gas | $\eta_e$ | the brake thermal efficiency |
| HCCI | Homogeneous Charge Compression Ignition | | |

Homogeneous Charge Compression Ignition (HCCI) combustion are radically different from the conventional spark ignition (SI) combustion in a gasoline engine and compression ignition (CI) diffusion combustion in a diesel engine. The combination of a diluted and premixed fuel and air mixture with multiple ignition sites throughout the combustion chamber eliminates the high combustion temperature zones and prevents the production of soot particles, hence producing ultra-low NOx and particulate emissions. Therefore, CAI/HCCI combustion represents for the first time a combustion technology that can simultaneously reduce both NOx and particulate emissions from a diesel engine and has the capability of achieving simultaneous reduction in fuel consumption and NOx emissions from a gasoline engine [14].

Although HCCI combustion in natural gas engines has been explored for many years, there are still some fundamental issues that remain to be resolved [15–19]. Facing the same problem as gasoline engine, it is difficult to control auto-ignition timing and combustion duration because they are controlled primarily by the chemical kinetics of fuel-air mixture, which is crucial to engine operating limit. In addition, the problem of the homogeneous mixture natural gas engine is the relatively larger cycle-by-cycle variations especially under lean combustion condition and large EGR condition [20–22]. The diesel spray ignited diesel/natural gas dual fuel engine can greatly decrease the cycle-by-cycle variations, which is a merit of operating natural gas. Amongst the numerous research papers on diesel/natural gas engine performance published over the last decade, most of them focus on the effect of boundary condition, but few concentrate on pilot diesel ignition mode e.g. CI mode and HCCI mode [23–27].

It is well known that diesel ignition process is crucial to diesel/-natural gas engine combustion process. As mentioned above, there are two kinds of diesel ignition mode. If a perfectly homogeneous mixture is created, the pressure and temperature rise during the compression stroke will lead to spontaneous ignition which differs from the classical diesel autoignition in the sense that it does not occur at a specific place in the spray, but simultaneously across the combustion chamber. Consequently, if autoignition occurs simultaneously in the whole cylinder, no high temperature flame front will appear as in the case of spark ignition engines. The absence of a high temperature flame front will lead to a practically negligible formation of nitrogen oxides, and due to the homogenized lean mixture, fuel rich zones are absent and therefore soot formation is also avoided. Reactions in HCCI engines generally involve a two-stage process, including both the so-called low temperature and high temperature reactions [28]. For diesel/natural gas dual fuel engine, two-stage process is different from those of diesel-like or gasoline-like fuel HCCI combustion and still unclear up to now. During the low temperature reactions, it may be more like that of diesel HCCI combustion for its pilot diesel ignition. But, during the high temperature reactions, the conclusion cannot be determined now for their different physical and chemical properties. It is worthwhile spending some time on researching the differences.

In addition, mixture composition, in-cylinder temperature and pressure have a dominant effect on ignition process of pilot diesel. If no other engine setting is modified, autoignition takes place at a given crank angle position independently of the start of injection, provided that enough time is made available to create a more or less homogeneous mixture [29]. But, too early injection certainly will lead to diesel impingement and poor atomization quality for lower in-cylinder temperature. On the contrary, too late injection will lead to traditional compression ignition [30]. At the same time, the effects of pilot diesel fuel ignition mode on the following mixture combustion need further investigation. So, further detailed investigation about impact of diesel injection timing should be carried out. To better understand the effect of pilot diesel ignition mode on combustion and emissions characteristics of dual fuel engines, a detailed study concerned with diesel injection timing was conducted. The testing work was operated on a 6-cylinder turbocharged intercooler diesel/natural gas dual fuel heavy-duty engine at light load operation, and diesel injection timing was controlled over a very wide range.

## 2. Experiment

### 2.1. Experimental setup

The research engine used for this study is an 8.6 L, 6-cylinder, turbocharged, intercooler, heavy-duty and diesel/natural gas dual fuel engine. The technical specifications of the engine were given in Table 1.

A schematic of the engine experimental setup is shown in Fig. 1. A dual fuel electronic-controlled system was developed with functions such as direct diesel injection, multi-point natural gas injection, electronic throttle control, idle flashover, fault diagnosis and communication. Natural gas was charged in the compressed vessels around 20 MPa and decompressed to 0.8 MPa through a regulator warmed by engine cooling water before injected into the intake port. The diesel and natural gas injection timing, pressure and pulse width were controlled with computer. In this research, pilot diesel was injected into cylinder during compression stroke with a common rail fuel injection system with different injection timing, and compressed natural gas (CNG) was injected into the intake port at a constant crank angle. Natural gas was selected as

**Table 1**
Detailed technical specifications of the test engine.

| Engine parameters | Specifications |
|---|---|
| Bore × stroke | 112 × 145 mm |
| Number of cylinders | 6 |
| Displacement | 8.6 L |
| Rated power/speed | 260 kW@2100 r/min |
| Compression ratio | 17.2:1 |
| Number of injector nozzle holes | 8 |

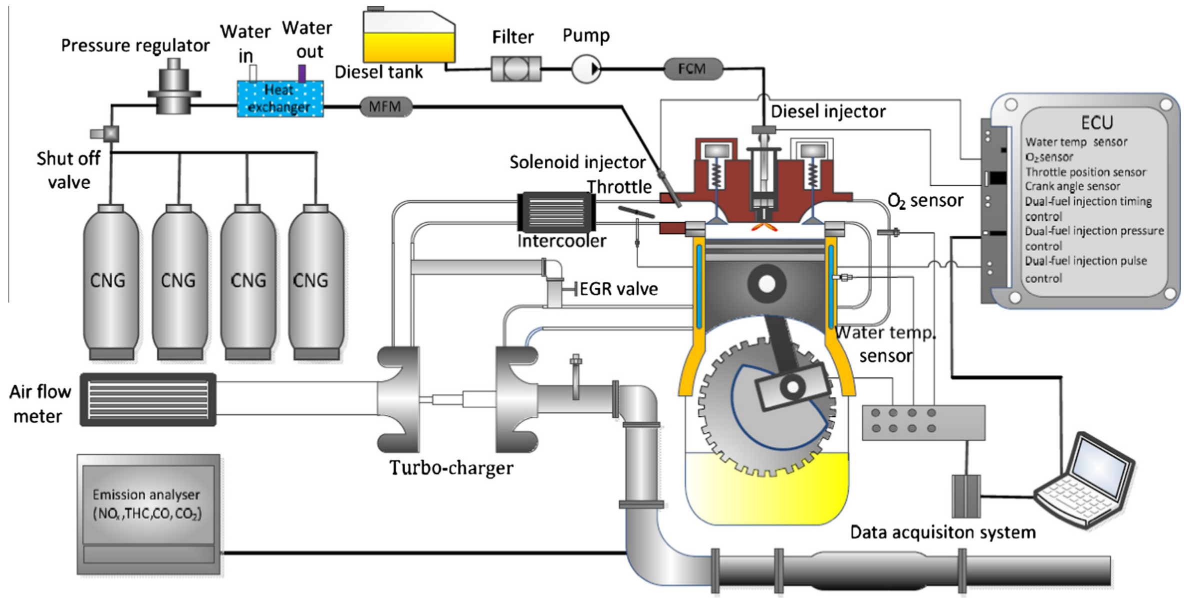



**Fig. 1.** Schematic diagram of the engine test rig.

the main fuel, which was used in Jilin province of China. Tables 2 and 3 show the properties of diesel and natural gas.

During the test, engine loads and speeds were controlled manually by an eddy current engine dynamometer (CW260, CAMA Luoyang, China) of 260 kW. Engine oil temperature, coolant temperature, exhaust temperature, and inlet air temperature were measured using K type thermocouples (within an accuracy of ±0.75%FS). The flow rate of diesel was measured by a Toceil CMFD015 mass flow meter (MFM) (with an accuracy of ±0.1%FS). The flow rate of CNG was measured by a MFM (with an accuracy of ±0.35%FS). The flow rate of air was measured by a laminar flow meter with an accuracy of ±1%. For each test point, the cylinder pressure was measured with a Kistler 6125B piezoelectric transducer (with an accuracy of ±0.4%FSO) and a 5011B charge amplifier, which sampled over 200 cycles at intervals of 1 degree crank angle. The exhaust emissions were measured by a Horiba MEXA-7100 DEGR gas analyzer system (with an accuracy of ±1.0%FS).

**Table 2**
Test fuel properties.

| Fuel | Diesel | Natural gas |
|---|---|---|
| Boiling point (°C) | 180–360 | −162 |
| Lower heating value (MJ/kg) | 43.5 | 50.0 |
| Cetane number | 52.5 | – |
| Octane number | – | 130 |
| Evaporation heat (kJ/kg) | 250 | – |
| Auto-ignition temperature (°C) | 250 | 650 |
| Stoichiometric air–fuel ratio (kg/kg) | 14.3 | 16.4 |
| Volume ratio of rich combustion limits in air (%) | 7.6 | 13.9 |
| Volume ratio of lean combustion limits in air (%) | 1.4 | 5.0 |

**Table 3**
Compositions of natural gas.

| Constituent | Volume percent |
|---|---|
| Methane | 88.70 |
| Ethane | 5.45 |
| Propane | 2.32 |
| Butane and heavier | 1.82 |
| Nitrogen | 1.08 |
| Carbon dioxide | 0.63 |

*2.2. Experimental conditions*

To better understand the effect of pilot diesel ignition timing on ignition mode and following combustion quality of dual fuel engine at light load operating condition, a detailed study concerned with diesel injection timing was conducted.

The results with different engine speed operating conditions showed that similar conclusions were obtained with engine speeds 1335, 1655 and 1975 r/min. Here, only one engine speed 1335 r/min was discussed in detail. In addition, lower brake thermal efficiency and higher unburned hydrocarbon emissions are the main issues to diesel/natural gas dual fuel engines, especially under light load operating conditions. The test results showed that controlling pilot diesel ignition mode is a valid way to resolve the above-mentioned issues. Instead, knock is the major issue to diesel/natural gas dual fuel engines under high load operating conditions, and exhaust gas recirculation may be the most effective way to resolve this issue [20–22]. So, only one typical light load operating condition was tested in this study. In order to improve the engine economy in use, as high as possible of natural percentage energy substitution was selected for its low price, and the related investigation results showed that better combustion stability can be observed with the natural gas energy accounted for 90%, and higher natural gas percentage energy substitution will lead to dramatically rise of unburned hydrocarbon emissions. Therefore, only one percentage energy substitution of dual fuel was tested during this research. The pilot diesel energy accounted for 10% of the total fuel energy and the natural gas energy accounted for 90%. Here, the energy was calculated by fuel lower calorific value as showed in Table 2. The total fuel energy in dual fuel mode was determined when engine outputted the equal power to the original single diesel fuel engine at 25% load (375 N m) with pilot diesel injection timing 20 °CA BTDC. The results showed that pilot diesel and natural gas mass flow rates were 1.407 kg/h and 11.02 kg/h, and the energy ratio was 10:90. No obvious changes were observed with different natural gas injection timings, and the natural gas injection pressure was set at constant value 0.8 MPa with constant injection timing 315 °CA BTDC. The related test results showed that higher thermal efficiency and lower emissions can be achieved with diesel injection pressure 70 MPa than 90 MPa, 110 MPa and 140 MPa.

Therefore, the diesel injection pressure was set at a constant value 70 MPa in this study.

## 3. Results and discussion

### *3.1. Performance characteristics*

The investigated results show that the pilot diesel injection timing ($T_{inj}$) has an obvious effect on engine performance at light load operating condition. Fig. 2 indicates the relationship between $T_{inj}$ and the brake mean effective pressure (BMEP). BMEP quickly rises with $T_{inj}$ changing from −5 to 27.5 °CA BTDC. BMEP reaches the maximum and almost keeps unchanged with variation from 27.5 to 42.5 °CA BTDC; however, it drops dramatically while $T_{inj}$ more than 45 °CA BTDC. The main reason is that pilot diesel ignition mode changed when $T_{inj}$ varied over a very wide range, and the detailed discussion will be given in Section 3.2. Fig. 3 shows the effect of $T_{inj}$ on the brake thermal efficiency ($\eta_e$). The relationship between $T_{inj}$ and $\eta_e$ indicates a very similar trend with BMEP. $\eta_e$ reaches approximately 35% with $T_{inj}$ changing from 27.5 to 42.5 °CA BTDC, which is near to original diesel engine the brake thermal efficiency 36.8%. This study focuses on the effect of $T_{inj}$, so with optimizing of other boundary conditions there has potential to achieve higher $\eta_e$ than original diesel engine.

Fig. 4 indicates the relationship between $T_{inj}$ and the intake pressure. The tested engine is a turbocharged dual fuel engine, and the change of combustion quality leads to the difference of exhaust gas condition by varying $T_{inj}$, which will affect turbocharger efficiency and the intake condition. Fig. 5 shows the effect of $T_{inj}$ on the air excess ratio ($\lambda$). In this study, the air excess ratio ($\lambda$) is defined as follows:

$$\lambda = \frac{\dot{m}_{air}}{\dot{m}_{CNG} \times AFR^{th}_{CNG} + \dot{m}_{Diesel} \times AFR^{th}_{Diesel}} \quad (1)$$

where $\dot{m}_{Diesel}$, $\dot{m}_{CNG}$ and $\dot{m}_{air}$ are the mass flow rates of diesel, CNG and air respectively. $AFR^{th}_{Diesel}$ and $AFR^{th}_{CNG}$ are the theoretical air fuel ratios of diesel and natural gas, respectively. During the test, the diesel and natural gas flow rates are constant, and the throttle value position keeps to the fully open position. The variation of intake pressure leads to the change of intake mass and $\lambda$. While $T_{inj}$ more than 45 °CA BTDC, $\lambda$ drops dramatically with the advancing $T_{inj}$.

### *3.2. Combustion characteristics*

Here, the variation trends of engine combustion and emissions characteristics with advancing $T_{inj}$ were discussed in detail, including cylinder pressure, cylinder temperature, heat release rate, start of combustion (SOC), ignition delay, combustion duration, crank angle of 50% heat release (CA50), nitrogen oxides (NOx) and total hydrocarbon (THC) emissions.

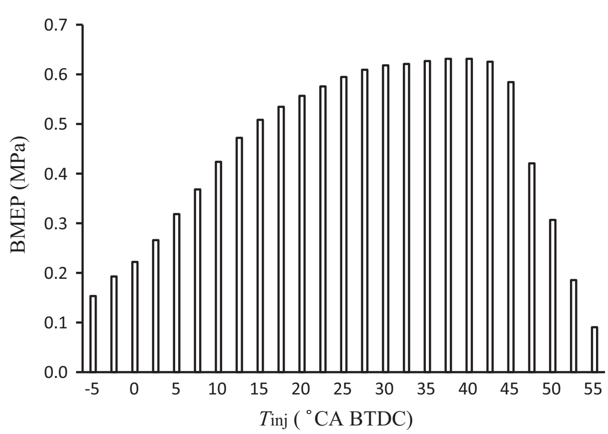



**Fig. 2.** Effect of $T_{inj}$ on engine output at 1335 r/min.

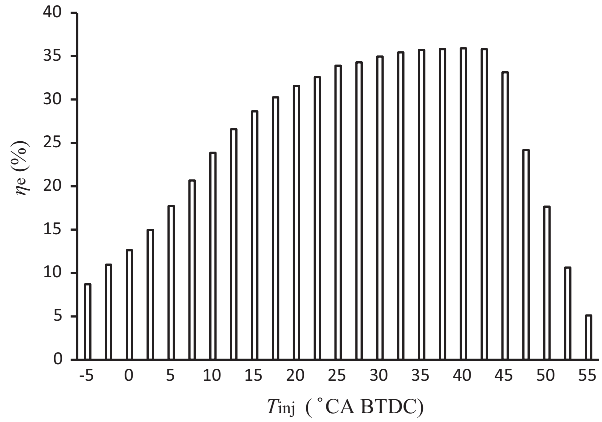



**Fig. 3.** Effect of $T_{inj}$ on engine thermal efficiency at 1335 r/min.

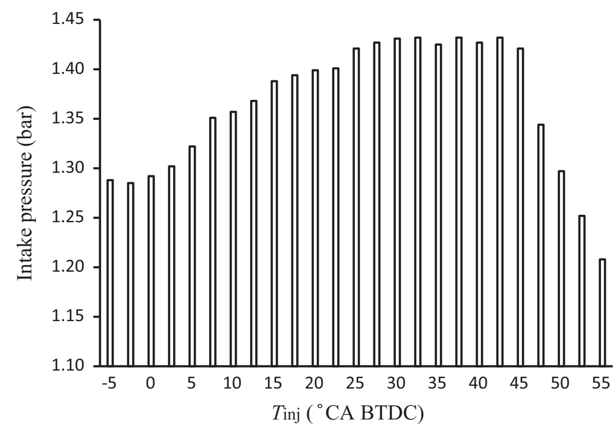



**Fig. 4.** Effect of $T_{inj}$ on engine intake pressure at 1335 r/min.

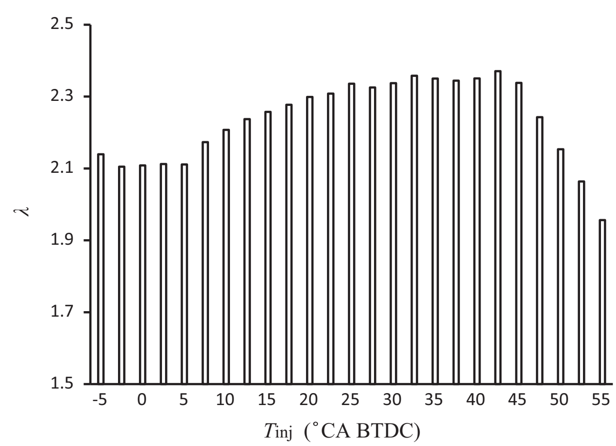
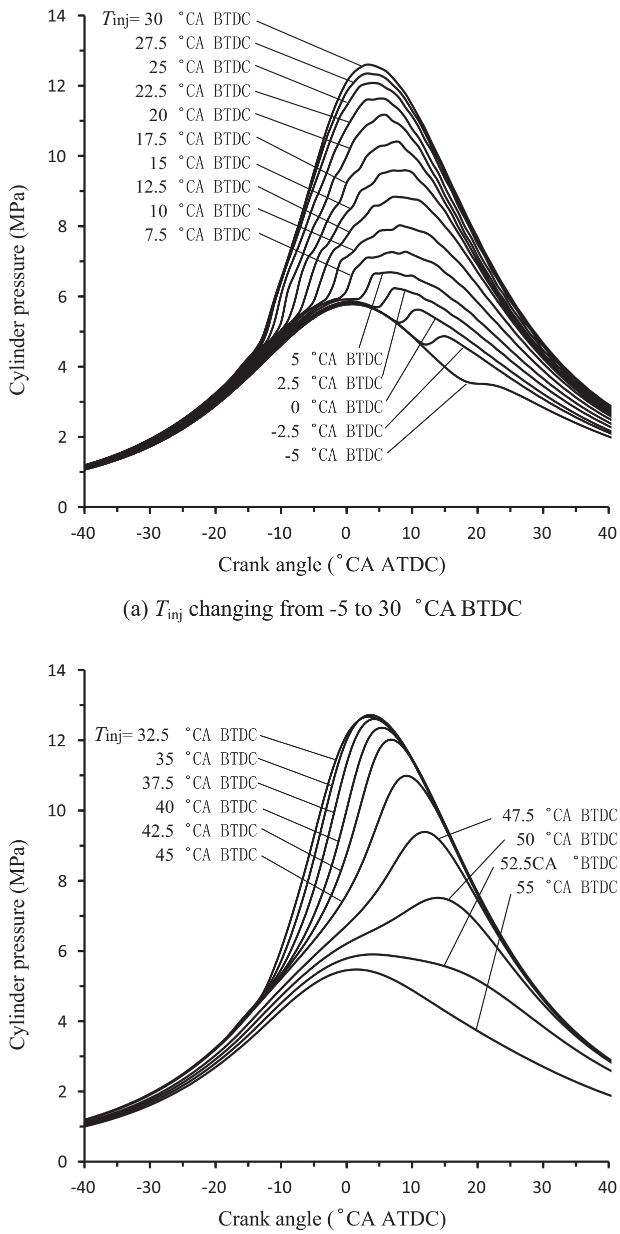



**Fig. 5.** Effect of $T_{inj}$ on $\lambda$ at 1335 r/min.

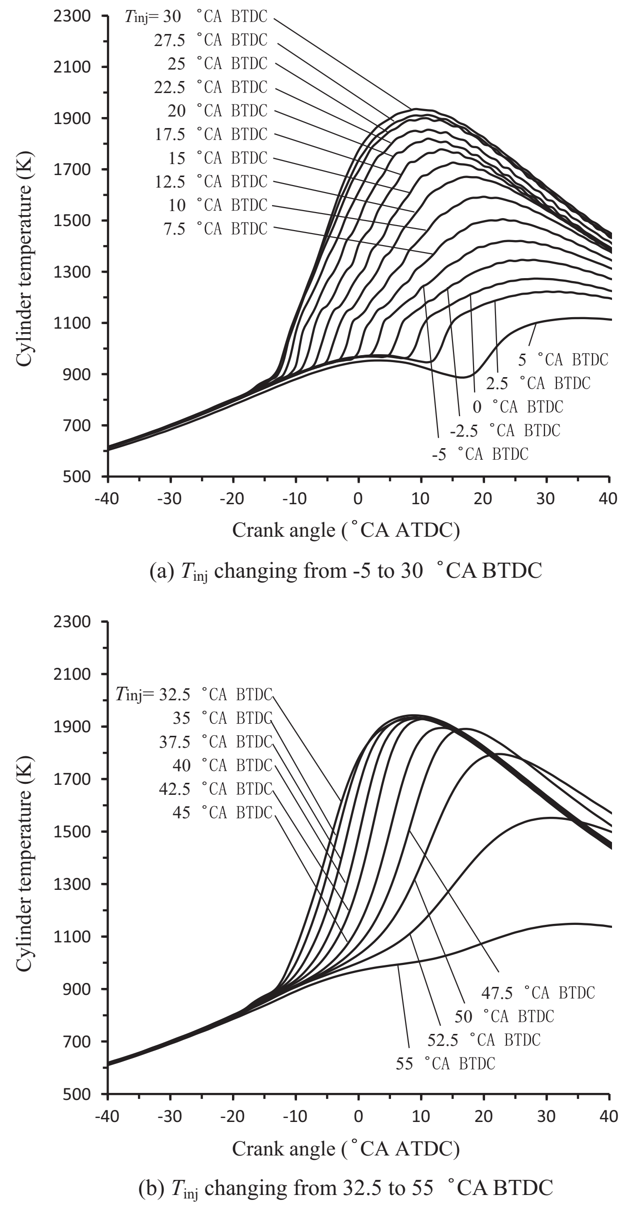
Fig. 6 indicates the variations of cylinder pressure profile. The results show completely different behaviors between $T_{inj}$ being more or less than 30 °CA BTDC, as shown in Fig. 6. While $T_{inj}$ less than 30 °CA BTDC, as Fig. 6(a) shown, with advancing $T_{inj}$ the crank angle of peak pressure is advanced and the value of maximum




(a) $T_{inj}$ changing from -5 to 30 °CA BTDC

(b) $T_{inj}$ changing from 32.5 to 55 °CA BTDC

**Fig. 6.** Pressure profiles with $T_{inj}$ at 1335 r/min.




(a) $T_{inj}$ changing from -5 to 30 °CA BTDC

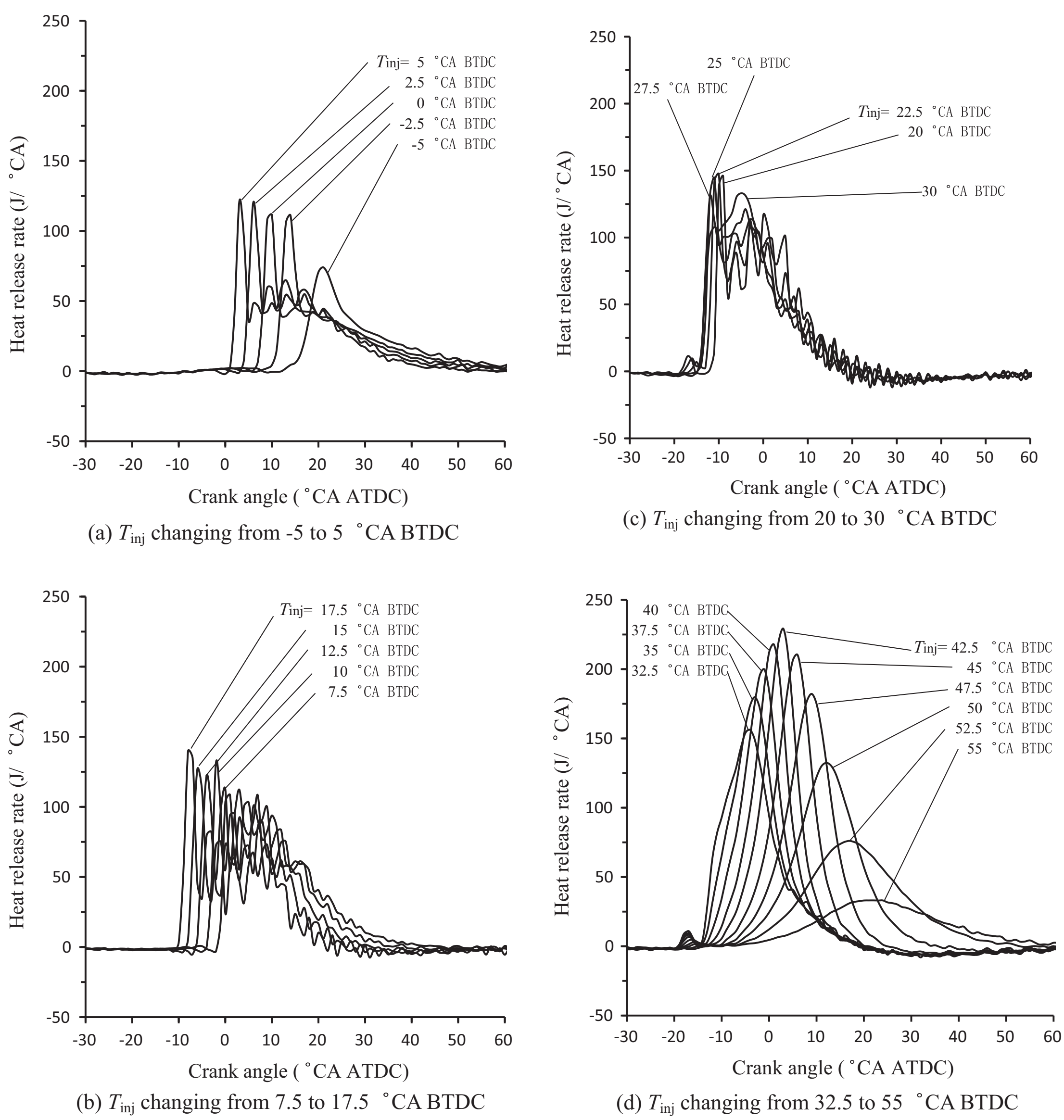
(b) $T_{inj}$ changing from 32.5 to 55 °CA BTDC

**Fig. 7.** Cylinder temperature profiles with $T_{inj}$ at 1335 r/min.

combustion pressure is higher, and the positions of combustion pressure curve break away from compression curve show an advanced trend. On the contrary, while $T_{inj}$ more than 32.5 °CA BTDC, as Fig. 6(b) shown, the crank angle of peak pressure is retarded and the maximum cylinder pressure is lower with advancing $T_{inj}$. Less intake mass leads to lower compression pressure while $T_{inj}$ changing from 47.5 to 55 °CA BTDC. In addition, the positions of combustion pressure curve break away from compression curve keep are almost unchanged while $T_{inj}$ changing from 32.5 to 45 °CA BTDC, which indicates that the pilot diesel ignition modes are different. At the same time, by comparing Fig. 6(a) and (b), the combustion pressure curve of the later are smoother than the former. From what has been discussed above, the pilot diesel ignition mode is similar to traditional diesel engine compression ignition mode when $T_{inj}$ is less than 30 °CA BTDC, and it is similar to two-stage autoignition mode while $T_{inj}$ more than 32.5 °CA BTDC.

Fig. 7 indicates the profiles of cylinder temperature. Similarly, the results show different behaviors between $T_{inj}$ being more or less than 30 °CA BTDC, as shown in Fig. 7. While $T_{inj}$ less than 30 °CA BTDC, as Fig. 7(a) shown, with advancing $T_{inj}$ the crank angle of peak temperature is advanced and the value of maximum combustion temperature is higher, and the positions of combustion temperature curve break away from compression curve show an advanced trend. On the contrary, while $T_{inj}$ more than 32.5 °CA BTDC, as Fig. 7(b) shown, the crank angle of peak temperature is retarded and the maximum temperature is lower with advancing $T_{inj}$. In addition, the positions of combustion temperature curve break away from compression curve keep are almost unchanged with $T_{inj}$ changing from 32.5 to 52.5 °CA BTDC. The pilot diesel ignition mode is more similar to two-stage autoignition mode when $T_{inj}$ is more than 32.5 °CA BTDC, and this result is in consistence with Jun, who reported that the low temperature oxidation reaction process becomes dominant at the temperature from 700 K to 900 K [31]. From Fig. 7(b), it can be seen clearly that the start temperature of low temperature reaction is near 800 K. As mentioned above, in-cylinder ambient temperature is the key parameter which decides the mode of ignition and the following combustion process. If the ambient temperature is more than 800 K when the pilot diesel was injected into in-cylinder, the ignition mode is similar to traditional diesel engine compression ignition mode. On the contrary, if the ambient temperature is lower than 800 K, the pilot diesel will experience a longer atomization and evaporation process until the ambient temperature is more than 800 K, and the ignition mode is more similar to two-stage autoignition mode.

Fig. 8 shows the variations of heat release rate profile under different $T_{inj}$. In order to show the variation clearly, each from Fig. 8 (a)–(d) just show five profiles under continuously variable $T_{inj}$. The heat release rate profiles when $T_{inj}$ is no more than 30 °CA BTDC, as shown in Fig. 8(a)–(c), are similar to that of conventional diesel ignition mode, and with advancing $T_{inj}$ higher first heat release is observed and the peak value is near to 150 J/°CA. In addition, the crank angle of first heat release is advanced. A very different variation trend can be seen in Fig. 8(d) when $T_{inj}$ changing from 32.5 to 55 °CA BTDC. A two-stage autoignition mode or spontaneous ignition mode can be seen clearly. In all relevant studies performed with either diesel fuel or surrogates, it has been verified that autoignition process consists of two stages. The two-stage autoignition chemistry is also reflected in the heat release patterns observed for combustion of homogeneous mixtures in engine situations [28]. As shown by Gen and Tomonori, cool flames arising from the first stage of the autoignition process provide a significant heat release, followed by a period with no apparent heat release corresponding to the so-called negative temperature coefficient (NTC) which lasts up to the high temperature stage leading to the main heat release which will control engine performance [32]. At the same time, the first stage of the autoignition process can also be observed as Fig. 8(c) shown, and this result is in consistence with Zhao [14]. If the mixture is induced by a spray-type flow, i.e. if the mixture is not homogeneous but a spray structure remains, the problem becomes more complicated: physical phenomena may play an important role, and combustion development follows a pattern closer to that described for the conventional diesel combustion. However, the two-stage autoignition pattern is maintained. In addition, it can be seen from Fig. 8(d) that the start of the first stage of the autoignition process is nearly at the same position, and which is decided by mixture temperature as shown in Fig. 7(b). The first stage of the autoignition processes is different



(a) $T_{inj}$ changing from -5 to 5 °CA BTDC

(c) $T_{inj}$ changing from 20 to 30 °CA BTDC

(b) $T_{inj}$ changing from 7.5 to 17.5 °CA BTDC

(d) $T_{inj}$ changing from 32.5 to 55 °CA BTDC

**Fig. 8.** Heat release rate profiles with $T_{inj}$ at 1335 r/min.

with variation of $T_{inj}$ and has an obvious effect on the following combustion process. The main heat release rate curves become symmetrical about the crank angle of the maximum heat release rate. The peaks of heat release rate show an initial increase and then decrease trend, and the maximum value is close to 225 J/°CA. Many factors led to the differences between them, including mixture pressure, mixture temperature, mixture homogeneity, and so on. In this study, $T_{inj}$ will directly affect diesel atomization and evaporation processes. The mixture pressure and temperature are different with variation of $T_{inj}$, which will lead the changing of diesel atomization and evaporation processes, as well as impingement against the engine walls. There exists an optimized $T_{inj}$ for pilot diesel finishing its atomization and evaporation processes. The study shows that the diesel injection timing ($T_{inj}$) has an obvious effect on pilot diesel ignition mode. A significant advancing $T_{inj}$ leads to pilot diesel ignition mode differs from traditional diesel engine compression ignition mode in the sense that it does not occur at a specific place in the spray, which is a two-stage autoignition mode.

Fig. 9 shows the variations of start of combustion (SOC). The start of combustion (SOC) was defined as the crank angle of 2% accumulated heat release. The results indicate that SOC shows an advancing trend when $T_{inj}$ is no more than 30 °CA BTDC and a retarding trend when $T_{inj}$ is more than 32.5 °CA BTDC. The main reason is that variation of $T_{inj}$ leads to the change of pilot diesel fuel ignition mode. $T_{inj}$ itself is the main factor for SOC of traditional diesel combustion mode, but mixture pressure and temperature are more important than $T_{inj}$ for SOC under two-stage autoignition mode. The ignition delay is defined as the interval of crank angle from the diesel injection timing to the start of combustion. The change of ignition delay is minimal when $T_{inj}$ is no more than 17.5 °CA BTDC as shown in Fig. 10, and coincidentally, there are on any signs of NTC can be observed as shown in Fig. 8(a) and (b). When $T_{inj}$ is more than 20 °CA BTDC, the ignition delay is prolonged with advancing $T_{inj}$. It can be seen that ignition delay increases linearly while $T_{inj}$ more than 32.5 °CA BTDC. Fig. 11 indicates the relationship between $T_{inj}$ and the combustion duration. The combustion duration is defined as the interval of crank angle from 2% to 98% accumulated heat release. It can be seen clearly that combustion duration is shortened with advancing $T_{inj}$ under traditional compression ignition mode. Under the two-stage autoignition mode, combustion duration decreases slightly when $T_{inj}$ is no more than 45 °CA BTDC, but it increases dramatically with further advancing $T_{inj}$. As shown in Fig. 8(d), lower mixture temperature and more pilot diesel impingement lead to deterioration of the combustion and longer combustion duration. Fig. 12 shows the effect of $T_{inj}$ on the crank angle of 50% heat release (CA50). When $T_{inj}$ is no more than 30 °CA BTDC, CA50 advances with the $T_{inj}$, but an opposite trend is observed when $T_{inj}$ is more than 30 °CA BTDC. In addition, it can be seen clearly that the CA50 is prior to TDC while $T_{inj}$ changing from 20 to 40 °CA BTDC. By comparing Figs. 8(d) and 3, it can be concluded that CA50 near to TDC corresponds to higher peak of heat release rate and higher brake thermal efficiency.

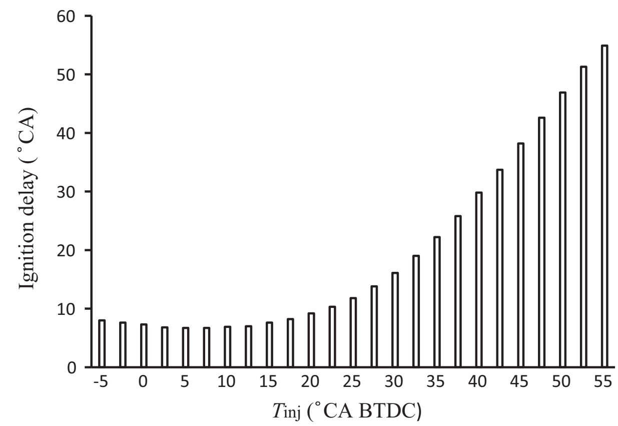



**Fig. 10.** Effect of $T_{inj}$ on ignition delay at 1335 r/min.

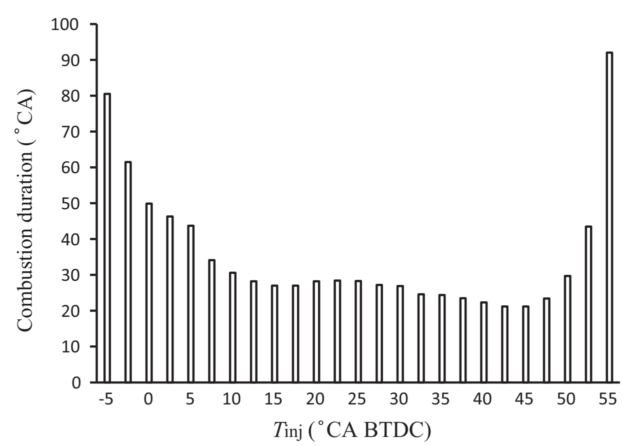



**Fig. 11.** Effect of $T_{inj}$ on combustion duration at 1335 r/min.

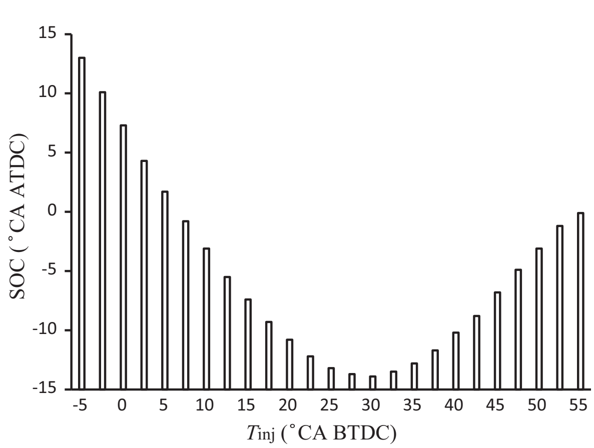



**Fig. 9.** Effect of $T_{inj}$ on SOC at 1335 r/min.

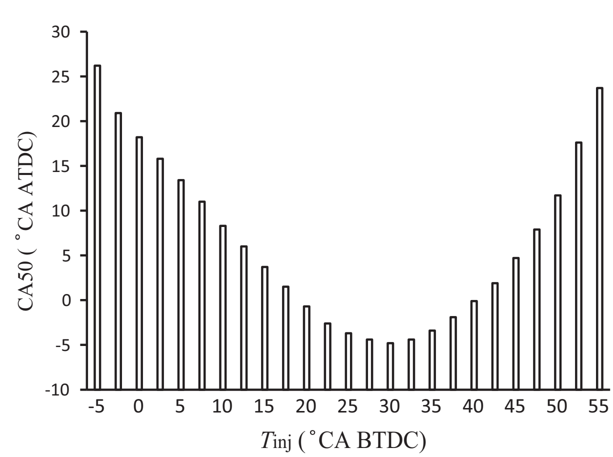



**Fig. 12.** Effect of $T_{inj}$ on CA50 at 1335 r/min.

### 3.3. Exhaust emissions

Fig. 13 indicates the relationship between $T_{inj}$ and the NOx emission. As known, NOx formation mechanism is predominantly controlled by the cylinder mixture temperature and the local oxygen excess ratio. As observed, the NOx emission is higher in transitional period from traditional compression ignition mode to two-stage autoignition mode, and with advancing $T_{inj}$, it increases in the traditional compression ignition mode and decreases in two-stage autoignition mode. In the traditional compression ignition mode, with advancing $T_{inj}$, CA50 shows an advancing trend and $\lambda$ simultaneously shows a trend of rising, which contribute to NOx emission dramatic increase. In the transitional period, NOx emission reaches the maximum. In the case of two-stage autoignition mode, the combustion reactions take place with the whole locations within the combustion chamber, and temperature field is more uniform and less local high temperature zone than traditional compression ignition mode. At the same time, with advancing $T_{inj}$, CA50 shows a retarding trend and $\lambda$ simultaneously shows a trend of decrease when $T_{inj}$ is more than 45 °CA BTDC, which contribute to NOx emission dramatic reduction.

Fig. 14 shows the relationship between $T_{inj}$ and the THC emissions. In the traditional compression ignition mode, THC emissions decease with advancing $T_{inj}$. As expected, higher combustion temperature and $\lambda$ lead to lower THC emissions. Simultaneously, lower THC emissions can be seen in two-stage autoignition mode and a small decrease in THC emissions due to the result of more complete fuel oxidation in the boundary layers and bulk gas. An interesting result of THC emissions can be observed in two-stage autoignition mode. THC emissions remain unchanged when $T_{inj}$ is no more than 42.5 °CA BTDC, which make it possible to optimize engine economic performance and NOx emission while maintaining relatively low THC emissions. When $T_{inj}$ is more than 45 °CA BTDC, there is a high probability of incomplete combustion and misfiring which lead to very high THC emissions.

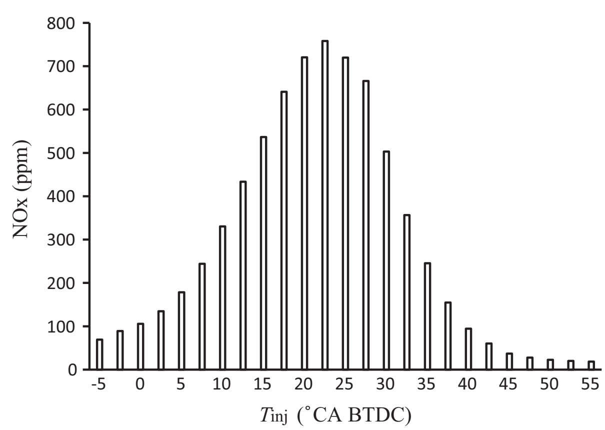



**Fig. 13.** Effect of $T_{inj}$ on NOx emissions at 1335 r/min.

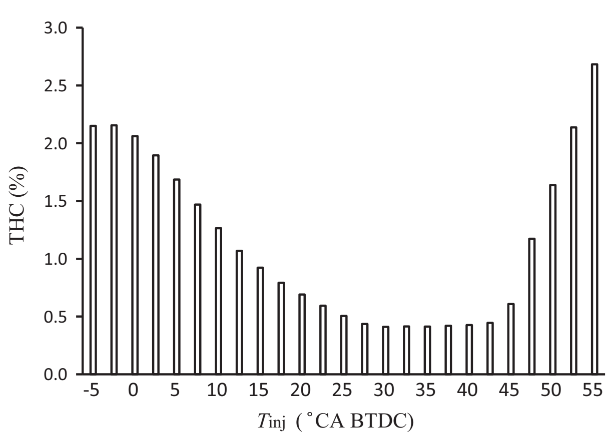



**Fig. 14.** Effect of $T_{inj}$ on THC emissions at 1335 r/min.

## 4. Conclusions

In the present work, pilot diesel ignition mode and engine combustion and exhaust emissions characteristics were investigated at light load operating condition. A detailed investigation with a wide range of diesel injection timing, varying from −5 to 55 °CA BTDC, was conducted at a constant flow rate of pilot diesel and CNG during the test.

The major findings from this research were summarized as follows. Pilot diesel injection timing is a sensitive parameter to diesel/natural gas dual fuel engines, which decides the pilot diesel ignition mode and quality. Different ignition modes can be observed with advancing pilot diesel injection timing. With normal pilot diesel injection timing, the ignition mode is similar to traditional diesel engine compression ignition mode, and two-stage autoignition mode can be achieved when advancing diesel injection timing over one fixed value which is determined mainly by mixture temperature. In general, engine performance, combustion and emissions characteristics in two-stage autoignition mode are better than those in CI mode. In two-stage autoignition mode, the brake thermal efficiency and THC emissions almost keep unchanged, and NOx emission drops dramatically with advancing diesel injection timing. But, too early injection certainly will lead to diesel impingement and poor atomization quality for lower in-cylinder temperature. On the contrary, too late injection will lead to traditional compression ignition. Optimized result in two-stage autoignition mode can be achieved with higher brake thermal efficiency (35%), lower NOx (60 ppm) and THC (0.4%) emissions, when diesel injection timing is fixed at 42.5 °CA BTDC.

In short, high thermal efficiency and clean combustion of diesel/natural gas dual fuel engines at light loads can be achieved with two-stage autoignition mode, and which can be obtained by optimizing pilot diesel injection timing.

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