Front. Energy Power Eng. China 2008, 2(3): 261–267
DOI 10.1007/s11708-008-0062-x

RESEARCH ARTICLE

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Effects of Fischer-Tropsch diesel fuel on combustion and emissions of direct injection diesel engine

Abstract Effects of Fischer-Tropsch (F-T) diesel fuel on the combustion and emission characteristics of a single-cylinder direct injection diesel engine under different fuel delivery advance angles were investigated. The experimental results show that F-T diesel fuel exhibits shorter ignition delay, lower peak values of premixed burning rate, lower combustion pressure and pressure rise rate, and higher peak value of diffusion burning rate than conventional diesel fuel when the engine remains unmodified. In addition, the unmodified engine with F-T diesel fuel has lower brake specific fuel consumption and higher effective thermal efficiency, and presents lower HC, CO, NO_x and smoke emissions than conventional diesel fuel. When fuel delivery advance angle is retarded by 3 crank angle degrees, the combustion duration is obviously shortened; the peak values of premixed burning rate, the combustion pressure and pressure rise rate are further reduced; and the peak value of diffusion burning rate is further increased for F-T diesel fuel operation. Moreover, the retardation of fuel delivery advance angle results in a further significant reduction in NO_x emissions with no penalty on specific fuel consumption and with much less penalty on HC, CO and smoke emissions.

Keywords direct injection diesel engine, F-T diesel fuel, fuel delivery advance angle, combustion, emission

1 Introduction

The particulate matter (PM) and nitrogen oxides (NO_x) emissions from diesel engines are high and difficult to reduce simultaneously because of the trade-off curve between the two emissions. Moreover, the application of exhaust aftertreatment devices to diesel engines is restricted by the high sulfur content of commercial diesel fuels. All these have become the main factors that prevent the development of diesel engines. The improvement of fuel properties is an effective way to reduce PM and NO_x emissions from diesel engines [1]. Recently, a new type of clean fuel, F-T diesel fuel produced from synthesis gas (CO and H_2) through Fischer-Tropsch synthetic processes using coal as the feedstock, has attracted increasing attention as a high quality, low emissions diesel substitute. F-T diesel fuel is characterized by a high cetane number, a near-zero sulfur content and a very low aromatic level. Some preliminary studies conducted on unmodified diesel engines have shown that the exhaust emissions are reduced significantly with the use of F-T diesel fuel [2,3]. Huang et al. [4] reported that the CO, HC, NO_x, smoke emissions from an unmodified diesel engine operating on F-T diesel fuel were reduced simultaneously when compared with those of conventional diesel fuel operation, and NO_x and smoke emissions were reduced by 16.7% and 40.3% respectively with F-T diesel fuel. Schaber et al. [5] reported that F-T diesel fuel showed reductions of 49% in HC, 33% in CO, 27% in NO_x, and 21% in PM compared with the standard federal No. 2 diesel fuel. Schaberg et al also found that proper retardation of the fuel injection timing could result in a further significant reduction in NO_x emissions with little effect on PM emission and specific fuel consumption.

The objectives of this paper are to investigate the effects of fuel delivery advance angles on the combustion and emission characteristics of the engine fueled with F-T diesel fuel on the basis of the recorded in-cylinder pressures and emission data.

2 Experimental apparatus and fuel characteristics

The engine tests were conducted on a single-cylinder, four-stroke, water-cooled, naturally aspirated, direct-injection diesel engine. The main performance and structural parameters of the engine are as follows; the bore is 100 mm; the stroke is 115 mm; and the compression ratio is 18. The

Translated from *Chinese Internal Combustion Engine Engineering*, 2007, 28(2): 19–23 [译自: 内燃机工程]

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rated speed and power of the engine are 2300 r/min and 11 kW respectively. First, the effects of F-T diesel fuel on the combustion and emission characteristics of the engine were investigated and compared with No. 0 diesel (conventional diesel fuel) when the engine remained unmodified. Then, the combustion and emission characteristics of the engine fueled with F-T diesel fuel under two different fuel delivery advance angles (24°CA and 21°CA) were studied.

The smoke density, represented by the light absorption coefficient (K), was tested using a part-flow smoke opacimeter type AVL DiSmoke 4000. Gaseous emissions were measured with an AVL DiGas 4000 emissions analyzer, which was comprised of a chemiluminescence analyzer for NO_x and a non-dispersive infrared analyzer for total unburned hydrocarbon (HC) and carbon monoxide (CO). The in-cylinder pressure was obtained by a Kistler piezo-electric sensor type 6125A; the output of the pressure transducer was amplified by a Kistler charge amplifier type 5015A and then converted to digital signals and recorded by a data acquisition apparatus type CS2092H.

The F-T diesel fuel used in the test was produced by the Institute of Coal Chemistry of Chinese Academy of Sciences. Fischer-Tropsch diesel fuel is a colorless and transparent liquid which is mainly composed of normal straight chain saturated hydrocarbons and branched isomeric saturated hydrocarbons. Besides, F-T diesel fuel contains very few olefinic hydrocarbons. F-T diesel fuel can be blended with conventional diesel fuel at random proportion to improve the properties of the base fuel. A comparison between the properties of F-T diesel and those of No. 0 diesel is shown in Table 1. The engine does not need to be modified when operating on F-T diesel fuel.

Table 1 Main properties of F-T and No. 0 diesel fuels

properties	F-T diesel	No. 0 diesel
liquid density (at 20°C)/$g \cdot cm^{-3}$	0.76	0.831 2
kinematic viscosity (at 20°C)/$mm^2 \cdot s^{-1}$	3.276	3.0–8.0
flash point/°C	74	>65
freezing point/°C	−2	<0
cold filter plugging point/°C	2	<4
initial boiling point/°C	150	180
final boiling point/°C	350	370
w(sulfur)/%	<0.00005	<0.2
φ(aromatics)/%	0.1	34.68
lower heating value/$MJ \cdot kg^{-1}$	43.9	42.6
cetane number	74.8	>45
carbon mass content/%	85	86
hydrogen mass content/%	15	14

3 Experimental results and analysis

3.1 Combustion characteristics of engine

Figure 1 illustrates a comparison of the heat release rate (η_h) between the engine operating on F-T fuel and the engine operating on No. 0 diesel fuel under two different brake mean effective pressures (BMEP)(p_{me}) at an engine speed of 1200 r/min. That for an engine speed of 1800 r/min is shown in Fig. 2. The two figures can basically represent changes of heat release rate under different operating conditions. It can be seen that, in comparison with No. 0 diesel at the same fuel delivery advance angle ($\theta_{fd} = 24$°CA), the peak value of premixed burning rate for F-T diesel fuel operation is much lower and the amount of energy released in premixed combustion is smaller, while the amount of energy released in diffusion combustion is larger and the peak value of diffusion burning rate is higher. The reason for this is that F-T fuel has a shorter ignition delay owing to its higher cetane number. A shorter ignition delay results in a reduction in the mass of injected fuel and the quantity of fuel evaporation occurring before ignition, therefore resulting in a much lower burning rate and a smaller amount of energy released during the premixed combustion phase. Since the amount of energy released during the premixed combustion phase is smaller, the diffusion combustion must contain more energy. Moreover, the boiling point of F-T fuel is lower, and it readily vaporizes and mixes with air in the cylinder, resulting in a fast diffusion mixing and diffusion combustion velocity. Therefore, the peak value of diffusion burning rate with F-T fuel is higher.

As shown in Figs. 1 and 2, at $\theta_{fd} = 24$°CA, the heat release starting position (in crank angle) of F-T fuel is earlier than that of No. 0 diesel, and this phenomenon is more obvious at a low engine load. This is because low engine load has low gas temperature in the cylinder, and the ignition delay tends to be largely affected by the cetane number of the fuels. Moreover, since the compressibility of F-T diesel fuel is larger than that of No. 0 diesel, the fuel injection delay of F-T diesel fuel is longer than that of No. 0 diesel, especially at a high engine load [6]. Thus, large advance of the heat release starting position of F-T fuel appears at a low engine load.

It can also be seen from Figs. 1 and 2 that when fuel delivery advance angle is retarded by 3°CA, the beginning crank angle of heat release is retarded, the peak value of premixed burning rate is further reduced, the peak value of diffusion burning rate is further increased, and the combustion duration is obviously shortened for F-T diesel fuel operation.

Figure 3 illustrates the effects of fuel delivery advance angle and F-T fuel on the combustion duration (θ) of the engine at load characteristics. It can be seen that, in comparison with No. 0 diesel at $\theta_{fd} = 24$°CA, F-T diesel fuel exhibits a slightly longer combustion duration than No. 0 diesel since the heat release of F-T diesel fuel starts earlier and its late combustion phase shows very little difference from that of No. 0 diesel (see Figs. 1 and 2). When fuel delivery advance angle is retarded by 3°CA, the combustion duration of the engine operating on F-T fuel is shortened by 2°CA, which is shorter than that of the original diesel engine.

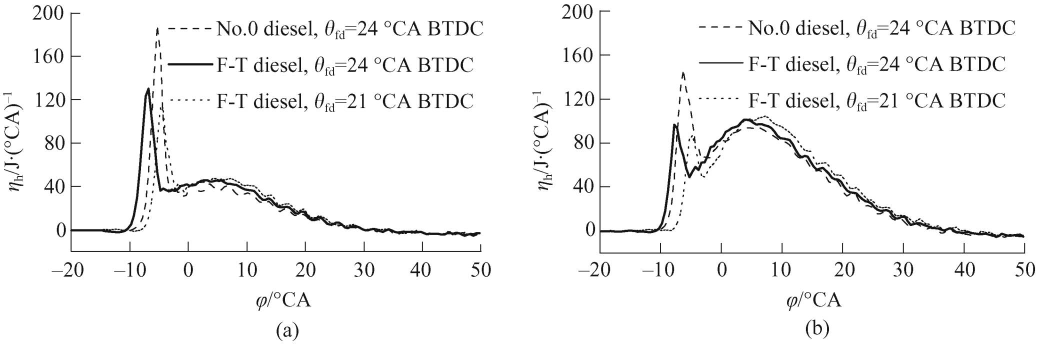

Fig. 1 Comparison of heat release rate between F-T and No. 0 diesel fuels at $n = 1200$ r/min (a) $p_{me} = 0.28$ MPa; (b) $p_{me} = 0.70$ MPa

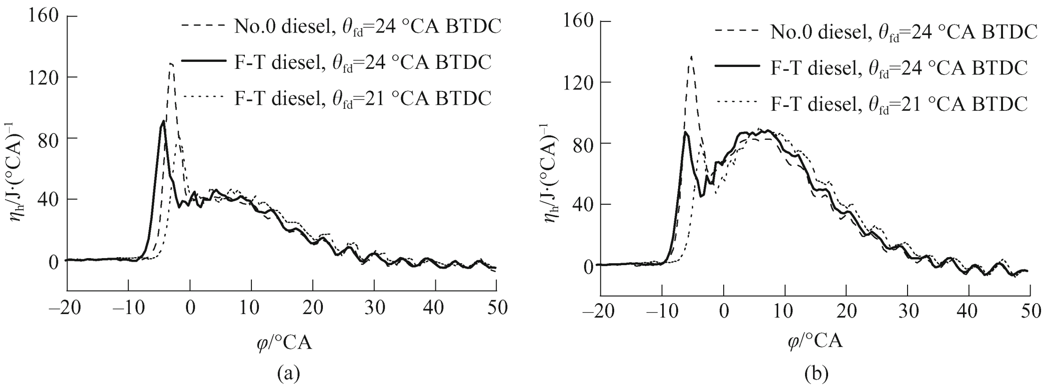

Fig. 2 Comparison of heat release rate between F-T and No. 0 diesel fuels at $n = 1800$ r/min (a) $p_{me} = 0.28$ MPa; (b) $p_{me} = 0.70$ MPa

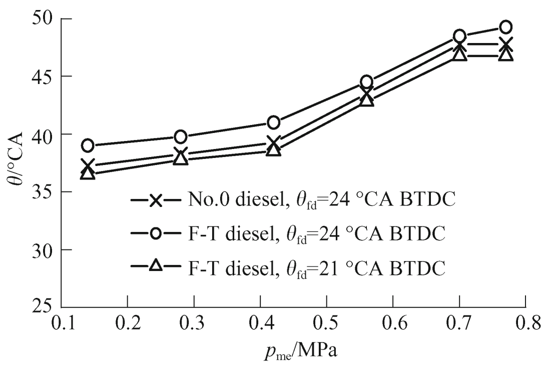

Fig. 3 Comparison of combustion duration between F-T fuel and No. 0 diesel fuel at $n = 1800$ r/min

Figure 4 shows the in-cylinder pressures (p_i) and rates of pressure rise (η_p) of the engine fueled with F-T and No. 0 diesel fuels versus the crank angle. In comparison with No. 0 diesel at $\theta_{fd} = 24°$CA, the peak combustion pressure of F-T diesel fuel is lower and occurs at a later crank angle, and the maximum rate of pressure rise is much lower and occurs at an earlier crank angle. When fuel delivery advance angle is retarded by 3°CA, the peak values of combustion pressure and pressure rise rate are further reduced and their corresponding crank angles are retarded. The lower maximum pressure and rate of pressure rise with F-T diesel fuel result in a reduction in mechanical load and combustion noise.

3.2 Fuel economy of engine

A comparison of the brake specific fuel consumption and the effective thermal efficiency between the engine fueled with F-T fuel and the engine fueled with No. 0 diesel fuel at load characteristics is shown in Fig. 5. The effective thermal efficiency was calculated by

$$\eta_{et} = \frac{3.6 \times 10^6}{H_l \cdot C_b} \times 100\%, \tag{1}$$

where H_l is the lower heating value of the fuel, C_b is the brake specific fuel consumption.

In comparison with No. 0 diesel at $\theta_{fd} = 24°$CA, the brake specific fuel consumption with F-T diesel fuel is reduced by an average of about 6.6%, and the effective thermal efficiency with F-T diesel fuel is increased by an average of about 4.5%. Since the lower heating value of F-T diesel fuel is higher than that of No. 0 diesel (see Table 1), the percentage of increase in the effective thermal efficiency is smaller than the percentage of decrease in

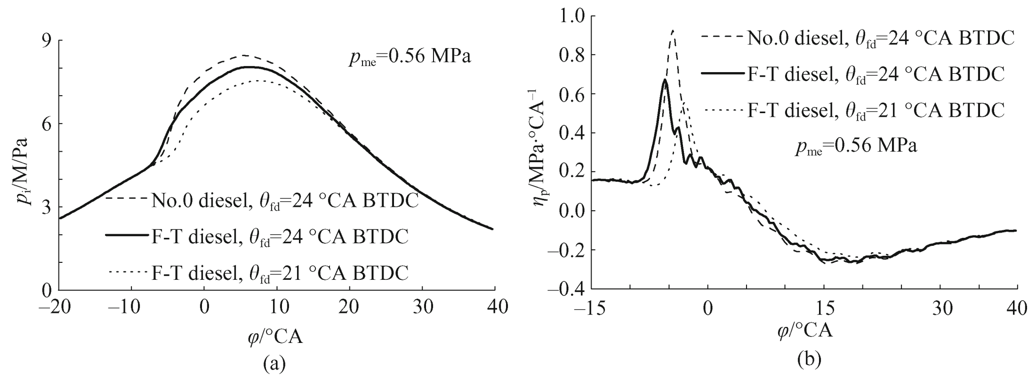

Fig. 4 Comparison of in-cylinder pressure and rate of pressure rise between F-T fuel and No. 0 diesel fuel at n = 1800 r/min
(a) In-cylinder pressure; (b) rate of pressure rise

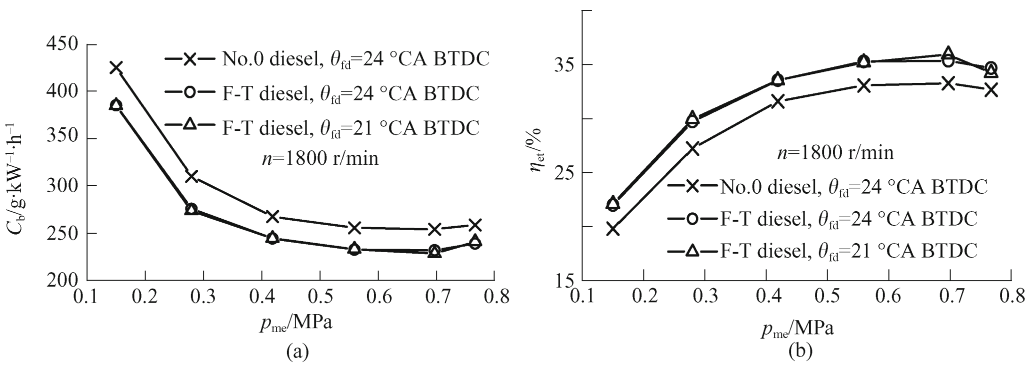

Fig. 5 Brake specific fuel consumption and effective thermal efficiency of F-T and No. 0 diesel fuels
(a) Brake specific fuel consumption; (b) effective thermal efficiency

fuel consumption. The reason for the improvement in fuel economy is that some chemical and physical properties of F-T diesel fuel - such as the absence of aromatics and composition based on the paraffin, lower boiling point - seem to improve the combustion process. In addition, the engine fueled with F-T diesel fuel has a lower peak value of in-cylinder pressure and a much lower rate of pressure rise, which reduces the mechanical load and realizes smooth combustion, thus improving the thermal efficiency.

It can also be seen from Fig. 5 that when fuel delivery advance angle is retarded by 3°CA, i.e., θ_{fd} = 21°CA, the brake specific fuel consumption and the effective thermal efficiency of the engine operating on F-T fuel vary little in comparison with those at θ_{fd} = 24°CA. This is because the delay of fuel delivery advance angle retards the beginning of combustion but obviously shortens the combustion duration.

3.3 Emission characteristics of engine

3.3.1 HC emissions

Figure 6 compares the HC emissions of the engine fueled with F-T and No. 0 diesel fuels under two different engine speeds at load characteristics. The results show that, in comparison with No. 0 diesel at θ_{fd} = 24°CA, HC emissions for F-T diesel fuel operation are reduced. In general, there are three major causes of HC emissions in diesel engines: ① fuel mixed to be leaner than the lean combustion limit during the ignition delay period; ② undermixing of fuel near the spray core during the combustion process; and ③ wall quenching of flame due to the fuel spray impinging on the combustion chamber walls. The ignition delay of F-T diesel fuel is shorter because of its higher cetane number, so the overlean region formed during the delay period is less for F-T fuel. Moreover, F-T diesel fuel has a lower boiling point than No. 0 diesel. The evaporation process occurs faster for F-T fuel, so wall impingement quenching and undermixing of fuel are less probable for F-T fuel. All these factors favor a reduction in HC emissions for F-T diesel fuel operation in comparison with No. 0 diesel at the same θ_{fd}.

It can also be seen from Fig. 6 that when θ_{fd} is retarded by 3°CA, HC emissions for F-T diesel fuel operation increase slightly, which are comparable with those of the original diesel engine. The reason for this is that the retardation of θ_{fd} can shortens the ignition delay, and consequently results in a reduction in the overlean region, but the retardation of θ_{fd} will shorten the air-fuel mixing

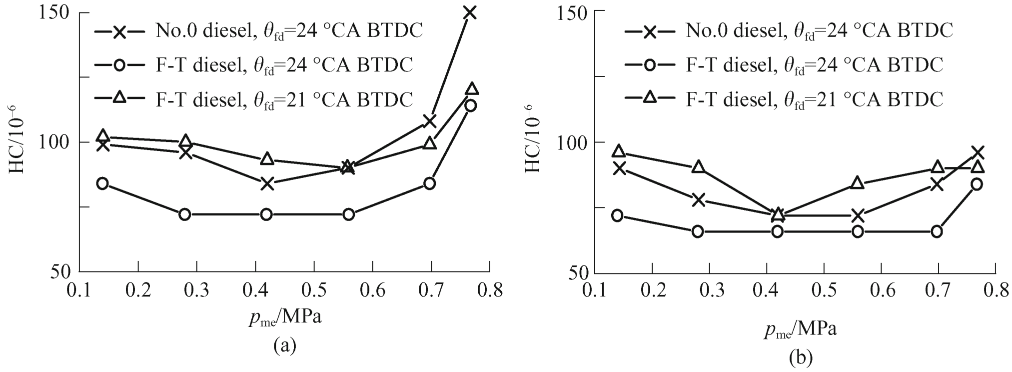

Fig. 6 Comparison of HC emissions between F-T fuel and No. 0 diesel fuel
(a) n = 1200 r/min; (b) n = 1800 r/min

time, and consequently result in an increase in the overrich region, thus resulting in an increase in HC emissions.

3.3.2 CO emission

A comparison of the CO emission between the engine fueled with F-T fuel and the engine fueled with No. 0 diesel fuel at load characteristics is shown in Fig. 7. It can be seen that at $\theta_{fd} = 24°CA$, the difference of CO emission between the two fuels is very little at low and medium loads due to the large relative air-fuel ratio. However, F-T fuel can reduce the CO emission significantly at high loads compared with No. 0 diesel at the same θ_{fd}. At full load, the CO emission with F-T fuel is reduced by 58% at low engine speed (1200 r/min) and 41% at high engine speed (1800 r/min) in comparison with No. 0 diesel at $\theta_{fd} = 24°CA$. The causes of the reduction in CO emission with F-T fuel can be explained by the mechanisms parallel to that described for HC emissions. The formation of fuel overlean region, fuel undermixing region and wall impingement quenching region is less probable for F-T fuel, resulting in a reduction in CO emission. Moreover, the engine operating on F-T diesel has a higher thermal efficiency. This will result in a reduction in injected fuel mass at the same load and air mass, thus resulting in a larger relative air-fuel ratio. This is the other main reason for the reduction in CO emission for F-T fuel operation. It can also be seen that when θ_{fd} is retarded by 3°CA, the CO emission with F-T fuel increases slightly, and the increase is slightly higher at low engine speed. However, the CO emission is still far lower than that of the original diesel engine.

3.3.3 NO_x emissions

Figure 8 compares the NO_x emissions of the engine fueled with F-T and No. 0 diesel fuels under two different engine speeds at load characteristics. The results show that, in comparison with No. 0 diesel at $\theta_{fd} = 24°CA$, NO_x emissions with F-T diesel fuel show an obvious reduction which increases with the increase of engine load. At full load, NO_x emissions with F-T fuel are reduced by 450×10^{-6} (17.3% reduction) at 1200 r/min and 389×10^{-6} (16.5% reduction) at 1800 r/min in comparison with No. 0 diesel at $\theta_{fd} = 24°CA$. The reasons for this reduction are as follows: the shorter ignition delay of F-T diesel fuel owing to its higher cetane number can considerably reduce the amount of energy released during the premixed combustion phase and result in a reduction in the maximum combustion pressure and temperature in the cylinder, thus resulting in a reduction in NO_x emissions. Moreover, the extremely low aromatic content of F-T diesel fuel can reduce local

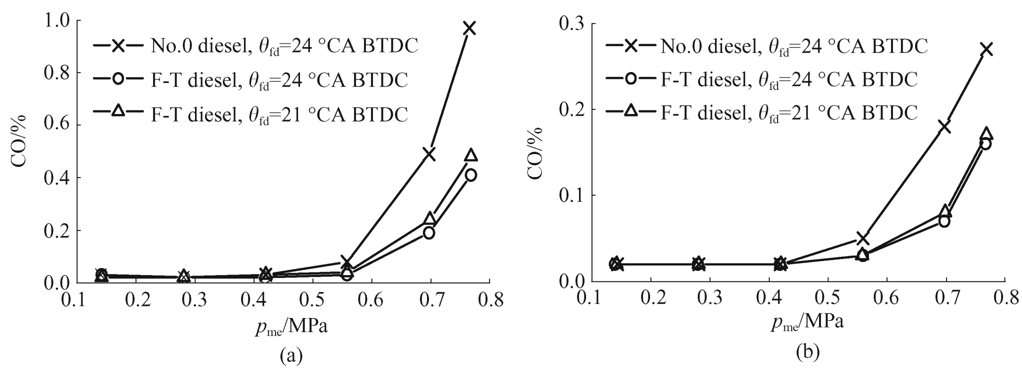

Fig. 7 Comparison of CO emission between F-T fuel and No. 0 diesel fuel
(a) n = 1200 r/min; (b) n = 1800 r/min

adiabatic flame temperature, and consequently reduce NO_x emissions because fuel with ring structures tends to have a higher adiabatic temperature [7].

It can also be seen from Fig. 8 that when θ_{fd} is retarded by 3°CA, NO_x emissions with F-T fuel show a further significant reduction. At full load, NO_x emissions of F-T fuel at $\theta_{fd} = 21$°CA are reduced by 350×10^{-6} (16.3% reduction) at 1200 r/min and 456×10^{-6} (23.1% reduction) at 1800 r/min in comparison with those of F-T fuel at $\theta_{fd} = 24$°CA, and reduced by 800×10^{-6} (30.8% reduction) at 1200 r/min and 845×10^{-6} (35.7% reduction) at 1800 r/min in comparison with No.0 diesel at $\theta_{fd} = 24$°CA.

3.3.4 Smoke emissions

The light absorption coefficient (K) of smoke of the engine operating on F-T and No. 0 diesel fuels versus BMEP is illustrated in Fig. 9. In comparison with No. 0 diesel at $\theta_{fd} = 24$°CA, smoke emissions for F-T diesel fuel operation are reduced. This reduction is more obvious at medium and high loads. At full load, the light absorption coefficient of smoke for F-T fuel operation is reduced by 0.83 m^{-1} (48.5% reduction) at 1200 r/min and 0.73 m^{-1} (43.7% reduction) at 1800 r/min in comparison with No. 0 diesel. The main causes of this reduction are as follows: first, the chemical composition of F-T diesel fuel, with near zero sulfur content, a very low aromatic level and a higher H/C ratio, should decrease the formation of particulate precursors. Second, the improvement in thermal efficiency results in a reduction in injected fuel mass at the same load and air mass, and subsequently results in an increase of relative air-fuel ratio, which goes with the rise of exhaust O_2 concentration, thus resulting in a great reduction in smoke. Moreover, the fast diffusion combustion of F-T diesel fuel should suppress smoke formation.

When θ_{fd} is retarded by 3°CA, smoke emissions with F-T fuel increase slightly, and the increase is slightly higher at low engine speed. However, smoke emissions with F-T fuel at $\theta_{fd} = 21$°CA are still far lower than those of the original diesel engine. At full load, the light absorption coefficient of smoke for F-T fuel operation at $\theta_{fd} = 21$°CA is increased by 0.17 m^{-1} at 1200 r/min and 0.16 m^{-1} at 1800 r/min in comparison with those for F-T fuel operation at $\theta_{fd} = 24$°CA.

4 Conclusions

1) When the diesel engine remains unmodified, F-T diesel fuel shows a shorter ignition delay and an earlier heat release starting position (more obvious at low loads) in comparison with No. 0 diesel. The peak value of premixed burning rate for F-T diesel fuel operation is much lower

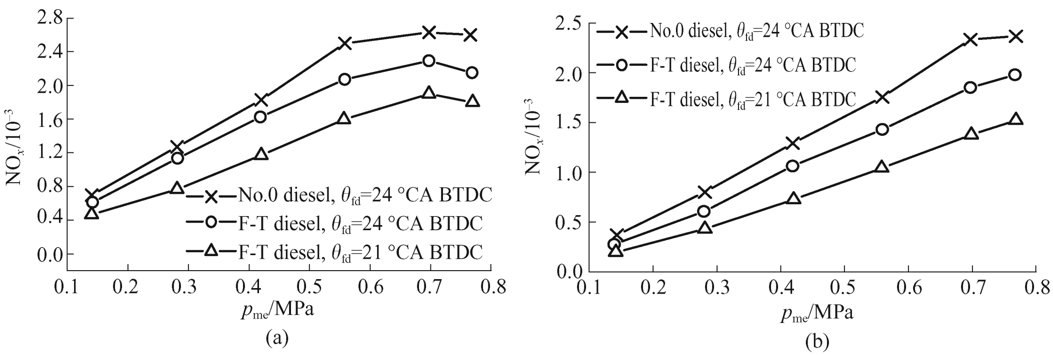

Fig. 8 Comparison of NO_x emissions between F-T fuel and No. 0 diesel fuel
(a) $n = 1200$ r/min; (b) $n = 1800$ r/min

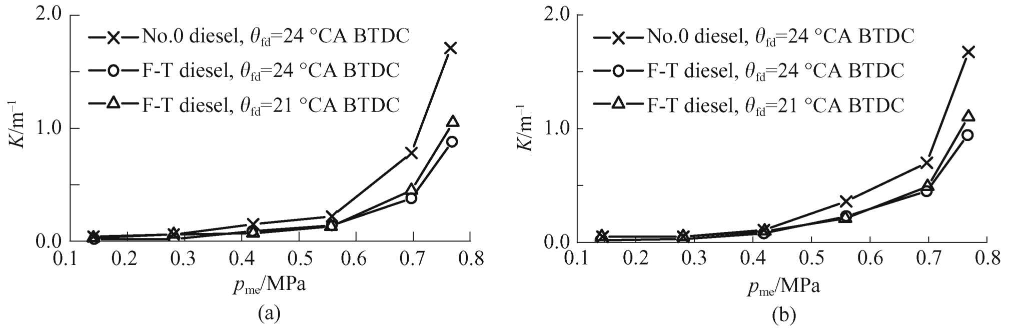

Fig. 9 Comparison of smoke emissions between F-T fuel and No. 0 diesel fuel
(a) $n = 1200$ r/min; (b) $n = 1800$ r/min

and the amount of energy released in premixed combustion is smaller, while the amount of energy released in diffusion combustion is larger and the peak value of diffusion burning rate is higher compared with those of No. 0 diesel. Meanwhile, F-T diesel fuel exhibits a slightly longer combustion duration than No. 0 diesel. In addition, F-T diesel fuel shows a lower peak combustion pressure and a far lower rate of pressure rise than No. 0 diesel, and thus the engine with F-T diesel fuel has a lower mechanical load and combustion noise.

2) When fuel delivery advance angle is retarded by 3°CA, the peak values of premixed burning rate, combustion pressure and pressure rise rate are further reduced, while the peak value of diffusion burning rate is further increased for F-T diesel fuel operation. Meanwhile, the combustion duration for F-T fuel operation is obviously shortened, which is shorter than that of the original diesel engine.

3) The unmodified engine with F-T diesel fuel has lower brake specific fuel consumption and a higher effective thermal efficiency compared with No.0 diesel. The retardation of fuel delivery advance angle by 3°CA has very little influence on the fuel economy of the engine operating on F-T fuel.

4) When the diesel engine remains unmodified, the CO, HC, NO_x, and smoke emissions for F-T diesel fuel operation were reduced simultaneously in comparison with those for No. 0 diesel, and the reduction of CO and smoke emissions is more obvious at medium and high loads. When fuel delivery advance angle is retarded by 3°CA, NO_x emissions for F-T fuel operation are further reduced significantly, while HC, CO and smoke emissions are increased slightly. Meanwhile, HC emissions are comparable, and CO and smoke emissions are still far lower than those of the original diesel engine.

Acknowledgements This work was sponsored by the National Natural Science Foundation of China (Grant No. 50306020).

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