Energy-Saving and Pollution-Reduction Potential Analysis for Diesel Engines Fueled with Fischer–Tropsch Fuel

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ABSTRACT: Alternative fuels have attracted wide attention owing to the increasing energy consumption and environmental pollution problems, which are caused by the extensive application of diesel engines for various occasions. The Fischer–Tropsch (F–T) diesel synthesized from coal is considered as one kind of ideal alternative fuel for diesel engines; however, its combustion and exhaust emission characteristics are different from those of national diesel owing to its special fuel properties. Therefore, the combustion and emission characteristics of F–T diesel and 0# diesel, which meet the China stage VI were compared in a common rail direct injection (CRDI) diesel engine. Moreover, energy-saving and pollution-reduction potential were analyzed in one CRDI diesel engine fueled with F–T diesel. The results showed that F–T diesel had an earlier ignition point, shorter ignition delay, lower cylinder pressure, and heat release rate compared with those of 0# diesel at the same operating conditions. Meanwhile, the amplitude, oscillation level, and energy of cylinder pressure were decreased to some extent, with the maximum drops of 0.98 bar, 16.4 dB, and 1.01 × 10¹² Pa, respectively. Additionally, under external characteristic conditions, maximum break thermal efficiency (BTE) and break-specific fuel consumption (BSFC) of F–T diesel were reduced by 1.1 and 2.1% on average compared with those of 0# diesel. In addition, CO, HC, NOx, and SOOT emissions of F–T diesel were found to be lower than those of 0# diesel, which were decreased by an average of 8, 3.7, 2.1, and 1.3%, respectively.

1. INTRODUCTION

Diesel engines are widely used due to their advantages of higher thermal efficiency, better fuel economy, and lower CO and HC emissions.¹–¹² However, the combustion characteristics of common rail direct injection (CRDI) diesel engines lead to high SOOT contents in the exhaust, as it is known that the SOOT emissions brought by diesel engines are 30–50 times higher than that of gasoline, which have caused great harm to the environment and human health. Meanwhile, diesel engines are undergoing new challenges with the tightening emission regulations and demand for lower carbon travel. Thus, to alleviate the environmental damage and fossil fuel depletion issues, researchers have been devoted to investigating clean alternative fuels.¹³–¹⁹ Thus, it is an extremely urgent and meaningful assignment to explore suitable alternative fuel in face of the automobile industry. However, it should be noted that the promising alternative fuels should be clean and easy to obtain, which can be used in diesel engines with little or no modifications while preserving diesel engine performances where it is required.²⁰ Currently, Fischer–Tropsch fuel has received much attention because of its capability of excellent properties and easy operation. The specifications of Fischer–Tropsch (F–T) diesel that can be synthesized from coal by indirect liquefaction technology are similar to 0# diesel, which can be directly used in the diesel engine without any modification. Owing to its excellent properties, it is considered to be one kind of clean and promising alternative fuel.¹¹–¹³ Many useful research studies about F–T diesel have been conducted in the past decades. Atkinson et al.¹⁴ examined the emission characteristics of NOx, PM, HC, and CO under 12 working points in an original engine fueled with F–T diesel. The results showed that compared with 0# diesel, NOx decreased by 20% on average, PM decreased by 31%, and HC, CO, and CO₂ reduced by 14, 21, and 5%, respectively. Amras et al.¹⁵,¹⁶ concluded that the fuel consumption rate and emission decreased at different degrees when a four-cylinder supercharged direct injection (DI) diesel engine was fueled with F–T diesel, and engine power had a downward trend. Shi et al.¹⁷–¹⁹ conducted a series of F–T diesel tests in an
intercooling and turbocharged engine. They examined that F–T diesel burned early, it had short ignition delay, and meanwhile, it presented soft vibration intensity. Vibration events are very common in CRDI diesel engines with higher combustion pressure and compression ratio, which accelerate the wear of components, reduce the durability of parts, and, ultimately, decrease the fatigue life of the engine. Fuel quantity and fuel type in diesel engines make significant influences on the vibration of the engine. Yang et al.\textsuperscript{21} researched the decreased by 14.5 and 35.6% on average compared with # diesel, respectively. Yong-chen et al.\textsuperscript{21} researched the combustion and emission regulation of F–T diesel. They concluded that the ignition delay duration time was at the same level. Fourie transform (FFT) and continuous wavelet transformation analysis. The results showed that vibration energy motivated by combustion was weak for different frequency bands when the engine was fueled with F–T diesel, and the F–T diesel had the potential to decrease the vibration of the diesel engine. Meanwhile, NOx and SOOT emissions were decreased by 14.5 and 35.6% on average compared with # diesel, respectively. Yong-chen et al.\textsuperscript{21} researched the combustion and emission regulation of F–T diesel in a direct injection diesel engine. They concluded that the ignition delay period was reduced by an average of 18.7%, the premixed injection diesel engine. They examined that F–T diesel showed a good ability to reduce CO, HC, and NOx emissions. However, research studies are confined for now to limited operating points and the evaluation still does not seem to be comprehensive and sufficient. Therefore, more studies will be required and performance evaluation between F–T and # diesel under external characteristic conditions is necessary, which is of great significance for bringing the benefits of F–T into full play and optimizing engine performance.

At the same time, some research studies have shown that the emission and combustion performances of diesel engines are closely related to combustion pressure oscillation. Pressure oscillation made significant effects on the diesel engine performances.\textsuperscript{22,23} Zhang et al.\textsuperscript{24} reported the research of cylinder pressure oscillation in a direct injection diesel engine. The results showed that the energy of pressure oscillation sharply increased when the frequency reached a threshold of 4 kHz at an engine speed of 1400 rpm. For instance, Shu et al.\textsuperscript{25} stressed on the high-frequency pressure oscillation signals, which can be used to diagnose and predict combustion state in a cylinder. Fan et al.\textsuperscript{26} showed that with the increase in F–T proportion in diesel, the oscillation intensity increased, the attenuation speed slowed down, and the combustion process became rougher. It is worth noting that how important the combustion pressure oscillation is to engine performance. At present, a lack of understanding of the nature of combustion pressure oscillation and then how to improve engine performance by reducing oscillation energy are still a difficult problem.

To sum up, all of the research studies indicated that F–T diesel had the potential to reduce engine exhaust emissions and improve combustion performances. Although, few researchers have delved into the pressure oscillation and pressure frequency spectrum of F–T diesel under overall conditions, it should be noted that the pressure oscillation was directly related to the engine working conditions, which was one kind of important technology to improve the engine performance. For this reason, the combustion, emission, and pressure oscillation were compared in detail in a CRDI diesel engine fueled with # diesel and F–T diesel, respectively, which can establish the foundation to make full use of the excellent characteristics of F–T diesel. It is good to promote engine fuel pluralism and improve diesel engine performance.

2. EXPERIMENTAL SETUP AND TEST FUELS

2.1. Engine Test Bench. The main technical specifications of the CRDI diesel engine that was used in this research are displayed in Table 1; multiple injection coupling with EGR is used in the engine to reduce emissions. The view of the test bench is shown in Figure 1.

| Table 1. Main Technical Specifications of the Test Engine |  |
|---|---|
| specifications |  |
| engine displacement (L) | 3.298 |
| bore × stroke (mm × mm) | 100 × 105 |
| fuel injection system | common rail direct injection |
| compression ratio | 17:1 |
| calibration power (kW)/speed (rpm) | 85/3300 |
| maximum torque (kW)/speed (rpm) | 315/1600-2400 |

In the whole process, the intake air temperature and engine cooling water temperature were kept in the same state through the external supportive and safeguard systems to ensure the reliability of the results. All of the modules began to collect data of each control appliance after stabilizing the engine operating parameters. Equation 1 defines the heat release rate, which is based on the first law of the thermodynamics.\textsuperscript{27}

\[
\frac{dQ}{d\theta} = \frac{k}{k-1} \frac{dV}{d\theta} + \frac{1}{k-1} V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{1}
\]

Heat transfer is calculated according to Newton’s cooling law, as \textsuperscript{27} and heat transfer coefficient can be calculated according to eqs \textsuperscript{3–5.28–31}

\[
\frac{dQ_{ht}}{d\theta} = \frac{1}{6n} \frac{h_g A (T_g - T_w)}{n} \tag{2}
\]

\[
h_g = 130V^{-0.06}P^{0.8}T_g^{-0.4} \omega^{0.8} \tag{3}
\]

\[
T_{6+1} = T_e \left( \frac{V}{V_{6+1}} \right)^{n_e} \tag{4}
\]

\[
T_w = 100 + 40 \times p_e + 273 \tag{5}
\]

where \(k\) and \(V\) are the adiabatic exponent and the working volume. In addition, \(P, Q_{ht}, h_g, A, T_g, T_w, \omega, n_e\) are the cylinder pressure, the heat transfer coefficient, the area of the combustion chamber, gas temperature, the mean piston velocity, in-cylinder wall temperature, and the polytrophic index, respectively.

2.2. Test Fuels. The specifications of the test fuels are shown in Table 2. It can be seen that compared with # diesel, F–T diesel has a lower density and boiling-point temperature. Moreover, F–T diesel has no sulfur content and the aromatic content is much lower than # diesel, which is helpful to reduce SOOT emissions. Furthermore, it should be noted that the CN of F–T diesel is significantly higher than that of # diesel, which can promote engine performance effectively.

2.3. Cost Analysis of the Test Fuels. A comprehensive cost analysis between F–T diesel and # diesel is helpful to attain competitive advantage and improve applicable prospects. Fischer–Tropsch fuel is primarily synthesized from coal in South Africa; the essential characteristic of the F–T diesel is
the coal chemical industry, so a large amount of money needs to be poured into infrastructure construction. In the coal-to-liquid plants, manufacturing cost, operating cost, management cost, tax cost, and financial cost make a major rate in the cost of the manufacture. In the current conditions, based on Porter's model, the per ton of F–T diesel production cost is about ¥5451∼6087, which has a potential advantage as long as oil price remains above $50 per barrel. So far, this year, oil price stands around $65 per barrel; the competitive advantage of F–T diesel further underscoring, moreover, this competitive advantage would increase along with the improvement of manufacturing technology and the production value size growth.

3. RESULTS AND DISCUSSION

3.1. Combustion Analysis. By comparing the differences of the cumulative heat release rate, the cylinder pressure, the heat release rate, cylinder pressure oscillation, and cylinder temperature between 0# diesel and F–T diesel, the potential of F–T diesel in improving combustion performances was researched in this section; further, 2000 rpm and full load conditions were used to serve as the data being analyzed, which are the typical operating cases of the test engine.

| specifications     | 0# diesel | F–T diesel |
|--------------------|-----------|------------|
| density (g/cm³)    | 0.81      | 0.76       |
| initial boiling point (°C) | 200      | 180.5      |
| end boiling point (°C) | 375      | 311.5      |
| sulfur content (ppm) | 10       | 0          |
| aromatic content (%) | ≤7       | 0.009      |
| calorific value (J/g) | 42.652   | 47.128     |
| CN                 | 55.82     | 62.54      |

Table 2. Main Specifications of the Test Fuel

Figure 1. View of the engine test bench.

Figure 2. Cumulative heat release versus the crank angle.
CA10, CA50, and CA90 of 0# diesel and F−T diesel are displayed in Figure 2, which are defined as the corresponding crank angle of 10, 50, and 90% cumulative heat release rates, respectively. Additionally, flame development duration was calculated from the ignition point to CA10, and flame propagation duration is defined from CA10 to CA90. Table 3 compares the ignition point, ignition delay, CA10, CA50, and CA90 of the two fuels. It could be concluded that under the test conditions, the ignition point of F−T diesel was 0.6 crank angle (CA) ahead of which of 0# diesel, and the ignition delay period was also shortened by 0.6 CA. The ignition point of F−T diesel was 4.9 crank angle before top death center (CA BTDC), which was advanced by 6.1% compared with that of 0# diesel. However, CA10, CA50, and CA90 of F−T diesel were delayed by 1.5, 6.7, and 12.6 CA, respectively.

These characteristics of higher CN (cetane number) of F−T diesel reduced specific gravity in the premixed combustion stage, which affected the combustion rate in the diffusion combustion stage. Meanwhile, its boiling point was lower than that of 0# diesel, larger fuel evaporation heat absorption existed in the flame diffusion stage, and then the cumulative release rate increased slowly, thus the flame development duration and propagation duration of F−T diesel were 17.4 and 75.4 CA, which are longer than that of 0# diesel.

The cylinder pressure and the heat release rate for different fuels are compared in Figure 3. It is obvious that the maximum cylinder pressure and heat release rate of F−T diesel are lower than that of 0# diesel, which are 105.3 bar and 44930 J/deg CA, respectively. Additionally, the ignition point, CA10, CA50, and CA90 of F−T diesel are advanced, which is related to its high cetane number and low density. Moreover, the combustion process of F−T diesel was similar to that of 0# diesel, which was consisted of premixing combustion and diffusion combustion. The premixing combustion phase was mainly influenced by the fuel properties, which caused different ignition delay and spray characteristics. Furthermore, the differences in fuel characteristics were mainly caused by the cetane number that made significant effects on ignition delay and the ignition point; in addition, viscosity and density played an important role in fuel atomization and combustion performance. Due to its higher cetane number, lower viscosity, and density, F−T diesel had shorter ignition delay, which reduced the proportion of premixing combustion; on the other hand, a less mixture was formed during the relatively shorter ignition delay period, which caused a lower cylinder pressure and heat release rate when the CRDI diesel engine was fueled with F−T diesel. Meanwhile, F−T diesel had a lower density and longer flame development duration; the heat exchanged with the cylinder wall was much higher, which made the diesel engine run smoothly.

Figure 4 shows the pressure oscillation, preinjection combustion (PIC) energy, and main injection combustion (MIC) energy under 2000 rpm and full load conditions when the engine was fueled with F−T diesel and 0# diesel. PIC is defined as the period from the first significant rise of the heat release rate, which indicates the start of preinjection combustion (SPIC) to the second rise of the heat release rate, which is the ending of preinjection combustion (EPIC), while MIC is defined as the period from the crank angle after EPIC that is the start of main injection combustion (SMIC) to CA90, which is the ending of main injection combustion (EMIC). Equations 6 and 7 were used to calculate the pressure oscillation

$$E_{\text{PIC}} = \int_{\theta_{\text{PIC}}}^{\theta_{\text{SPIC}}} P_{\text{osc}}^2 d\theta$$

(6)

$$E_{\text{MIC}} = \int_{\theta_{\text{SPIC}}}^{\theta_{\text{MIC}}} P_{\text{osc}}^2 d\theta$$

(7)

where $E_{\text{PIC}}$ and $E_{\text{MIC}}$ are the oscillation energies of PIC and MIC, respectively. The integral interval is the corresponding period of PIC and MIC; $P_{\text{osc}}$ is the pressure oscillation in the corresponding interval and $\theta$ indicates the crank angle.

To calculate the pressure oscillation, a band-pass filter was used to filter out the interference caused by the pressure baseline and high-frequency random noise in the cylinder pressure, which was in a range of 2kHz $\sim$ fs/2 (fs is the sampling frequency). Obviously, the pressure oscillation occurred near the area where the cylinder pressure increased rapidly, while the oscillation energy of PIC occupied a low proportion in the whole processing. The peak value of pressure oscillation occurred in a range of 2−4° CA and almost disappeared at 20 crank angle after top death center (CA ATDC); the pressure oscillation occurred in the main combustion period, and then decreased exponentially and disappeared in the slow-burning period. Moreover, compared with 0# diesel, F−T diesel could decrease the pressure oscillation of the diesel engine to a certain degree, and the corresponding angle also advanced, which is mainly due to its shorter ignition delay period and low cycle-to-cycle variation of maximum combustion pressure.

When the piston moved close to top death center (TDC), a high-temperature and high-pressure mixture led to an increase in pressure oscillation, while the pressure oscillation began to

| Table 3. Combustion Parameters of Test Fuel at 2000 rpm and Full Load Conditions |
|-----------------------------------------------|
| fuel       | ignition point (deg CA) | ignition delay (deg CA) | CA10 (deg CA) | CA50 (deg CA) | CA90 (deg CA) |
|------------|-------------------------|-------------------------|---------------|---------------|---------------|
| 0# diesel  | −7.1                    | 9.9                     | 8.2           | 24.7          | 72.5          |
| F−T diesel | −7.7                    | 9.3                     | 8.2           | 24.7          | 72.5          |

Figure 3. Cylinder pressure and heat release versus the crank angle.
decrease as the piston moved downward. F−T diesel had an earlier ignition point, shorter ignition delay, and lower premixed combustion proportion due to its higher cetane number, which resulted in a lower combustion rate in the main combustion phase, and consequently caused smaller pressure oscillation amplitude.

Spectrum signals of cylinder pressure contain abundant information that are related to the combustion state of diesel engine; FFT was selected to analyze the pressure oscillation level of F−T diesel and 0# diesel in the frequency domain, as shown in Figure 5. It is obvious that the two fuels have similar but not identical characteristics; the energy of pressure oscillation was mainly concentrated in the range of \( f > 4 \) kHz. Pressure oscillation decreased with the increase in frequency when the frequency was less than 4 kHz, while it increased with the continuous increase in frequency when the frequency exceeded 4 kHz. Obviously, under 2000 rpm and full load conditions, the critical frequency of pressure oscillation was 4 kHz. Moreover, it can be concluded from the comparisons that the amplitudes of the pressure oscillation level and the corresponding frequency of F−T diesel are 47.7 dB and 6.4 kHz, while which are 64.1 dB and 7.3 kHz for 0# diesel, thus the effectiveness of restraining pressure oscillation was verified when the engine was fueled with F−T diesel.

Cylinder temperature and fuel injection pressure are compared in Figure 6. In this phase, the injection pressure of F−T diesel was slightly lower than that of 0# diesel due to its lower density. The cylinder temperature decreased to 134 K,
meanwhile, the peak phase advanced 5° CA. As it is known that the cylinder temperature creates an appropriate environment to promote the rapid combustion of fuel and makes significant influences on the combustion process and exhaust emissions, so the excellent characteristics of F−T diesel and its lower cylinder temperature are inseparable, which was closely related to its high cetane number and calorific value. Thus, the engine fueled with F−T has the advantage of a low mechanical load and long service life.

3.2. Exhaust Emission Reduction Analysis. In this section, the research focused on the emission performances when the engine was fueled with 0# diesel and F−T diesel under different speeds varying from 1200 to 2800 rpm and full load. The characteristics of NOx and soot emissions related to engine operating conditions were compared and analyzed, as shown in Figure 7. The results show that the NOx and Soot emissions were extremely sensitive to engine speeds. The exhaust emissions of F−T diesel were lower than that of 0# diesel at the same operating conditions. NOx emissions decreased to 4.70, 0.28, −1.85, 3.22, 1.64, 0.19, 5.15, 2.77, and 3.39%, respectively, while soot emissions decreased to 1.30 mg/m³ averagely. Moreover, the soot emissions were significantly decreased, especially at higher-speed zones.

The formation conditions for NOx are high temperature, rich oxygen, and long reaction time. The temperature in the cylinder increased with the increase in the speeds; this was primarily because of the increase in volumetric efficiency and gas flow activities, which led to faster mixing of fuel and air in the cylinder at higher speeds. As the engine speed increased, the quantity of the mixture was also increased, and then it gave out enormous heat when the mixture was burned. Although there was a significant reduction of the reaction time for each cycle and the residence time of the mixture in the cylinder, high temperature had crucial impacts on NOx formation. As it is shown in Figure 6, F−T diesel had lower cylinder temperature due to its fuel properties, which provided help in reducing the NOx emissions. In addition, the much lower soot emissions of F−T diesel were caused by its excellent fuel
properties of a lower aromatics content and almost no sulfur content. At the same time, it can be seen that the “trade-off” relationship between NOx and soot emissions is improved to a certain extent, and both of them can reach a lower level when the diesel engine is fueled with F−T diesel.

Figure 8 displays the CO and HC emission variations with different operating conditions when the engine burned with F−T diesel and 0# diesel. It can be found that engine speeds make a larger influence on CO emissions than HC emissions. The CO emissions were at the lowest level at the medium speed range, while they remained at a higher state at lower-and higher-speed conditions. Furthermore, the HC emissions were not sensitive to the engine speed and they varied in a small range under the test conditions. It is worth noting that the CO and HC emissions of F−T diesel were obviously lower than that of 0# diesel, which decreased by 4.0−17.1 and 0.7−7.5%, respectively.

The engine speeds are closely related to the process of mixture flow and mixing. The CO and HC emissions were obviously increased at lower speed conditions because of poor mixture formation and a low-effective combustion process; however, as the speed increased, the increase in volumetric efficiency and gas flow activities made the mixing of air and fuel more uniform, which led to the increase in the cylinder temperature. Meanwhile, the reaction time and the residence time of the mixture reduced in the cylinder, so the CO emissions increased with increasing engine speeds. F−T diesel has a lower end boiling point and a faster evaporation process, as compared in Section 2.2. Therefore, the atomization was better when the F−T diesel was injected into the cylinder, and the mixed gas excessive area can be reduced. Due to the less possibility of excessive area, F−T diesel had lower CO emissions. As it can be seen that there are more HC emissions especially in the low- and high-speed areas when fueled with
F−T diesel and the maximum difference is 13 ppm, which depended on the oxidation reaction.

3.3. Engine Energy-Saving Analysis. Figure 9 shows the engine torque and power performances under the external characteristic conditions. It can be seen that the engine power increased with increasing speed, but there were no distinct differences between 0# diesel and F−T diesel on the power performance. Moreover, the power of F−T diesel was slightly lower than that of 0# diesel when the engine speed was larger than 2000 rpm, which decreased by an average of 2 kW. Furthermore, maximum torques of 0# diesel and F−T diesel were 322.9 and 318.2 Nm at 2000 rpm, respectively, while the minimum torques were 260.1 and 248.0 Nm at 1400 rpm.

The higher the speed, the more the mixture, the more heat released, and then the more the power. However, when the engine speed was larger than 2000 rpm, the rate of power growth became slow due to the reduction of the reaction time; this was why the maximum torque speed occurred at 2000 rpm. Furthermore, the engine power and torque of F−T diesel were lower than those of 0# diesel due to the shorter ignition delay, which reduced the proportion of premixed combustion; on the other hand, decreased accumulation of fuels in the combustion process during the relatively shorter delay period caused a lower cylinder pressure and heat release rate when the engine was fueled with F−T diesel compared with 0# diesel.

Figure 10 shows the break-specific fuel consumption (BSFC) and the break thermal efficiency (BTE) variations when the engine is fueled with F−T diesel and 0# diesel under the external characteristic conditions. The minimum BSFC of F−T and 0# diesel were 198.8 g/kWh and 203 g/kWh, while the maximum BTE were 40.1 and 41.4%, respectively. It is worth noting that BSFC of F−T diesel was lower than that of 0# diesel at different levels at the same working condition, and then the maximum BTE of F−T diesel was 40.9%, which was 1.1% lower than that of 0# diesel on average.

As previously mentioned, F−T has higher CN and lower distillation temperature, which are found to improve the combustion process. In addition, the calorific value of F−T is slightly higher than that of 0# diesel, so it releases more energy with less fuel injected to achieve the specified conditions.

When these factors are added together, the BSFC of F−T is slightly lower than that of 0# diesel. Obviously, there are many aspects for the influence of BTE and there is no doubt at all that the combustion process is the most important key factor. It could be seen from the previous analysis that the center of the heat release rate of F−T diesel is remote from TDC and the constant volume of combustion decreased, which increase the amount of heat exchange between the combustion wall and cooling water. Moreover, thermal losses in the combustion process increased, which led to a decrease in BTE by an average of 1.1% compared with 0# diesel.

4. CONCLUSIONS

The research analyzed fuel properties of F−T diesel and national 0# diesel, discussed the influence of these fuel properties on the combustion process and exhaust pollution, and made a comparison related to pollution and combustion characteristics between F−T diesel and 0# diesel in a CRDI diesel engine. The results indicated that F−T diesel can be considered as one kind of potential alternative fuel in terms of environmental friendliness and has been fully certified to be efficiently used in diesel engines without modification. After comparison, it has been found that the combustion performance of tested diesel engines with F−T is improved to some extent. The cylinder pressure, heat release rate, and combustion energy decreased owing to its excellent characteristics. Meanwhile, the crank angle of the peak value occurs early, which is helpful to reduce cylinder pressure oscillation. The emission pollution level of the diesel engine is very important in terms of environmental issues. NOx emission decreased and maximum reduction was 5.15%, while SOOT emission also decreased by 1.30 mg/m³ on average, which improved the trade-off relation between NOx and SOOT. However, it should be noted that when the diesel engine was fueled with F−T, it resulted in a decline in engine dynamic performances and a decrease in break thermal efficiency. As an alternative fuel for diesel engines, Fischer−Tropsch fuel is promising. Currently, there exist two main problems in Fischer−Tropsch research: (1) the poor lubrication and low viscosity result in wear failure and (2) the cost of Fischer−Tropsch remains at a relatively high level, which cannot meet
the need of large-scale commercial applications. Thus, it is necessary to provide further studies and develop additional fuel performance measurements so as to fully bring its excellent properties into play.

The comprehensive economic-technical analysis indicates that cost and performance play a key role in the application of the F−T fuel. The future system optimization could focus on reducing cost and improving its energy-saving and pollution-reduction potential. The analysis results can be combined with injection parameter adjustment to bring the fuel performance into play. In addition to F−T diesel, methanol and diesel can be used as a blending component and chemical feedstock to improve energy-saving and pollution-reduction performance on the diesel engine.

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■ ABBREVIATIONS

F−T, Fischer−Tropsch; CRDI, common rail direct injection; FFT, fast fourier transform; BTE, break thermal efficiency; BSFC, break-specific fuel consumption; DI, direct injection; CA, crank angle; CA BTDC, crank angle before top dead center; CA ATDC, crank angle after top dead center; TDC, top dead center; PIC, preinjection combustion; MIC, main injection combustion; SPIC, start of preinjection combustion; EPIC, ending of preinjection combustion; SMIC, start of main injection combustion; EMIC, ending of main injection combustion

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