



# Article Cycle-to-Cycle Variation of a Diesel Engine Fueled with Fischer–Tropsch Fuel Synthesized from Coal

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Abstract: Cycle-to-cycle variations during the combustion phase should be comprehensively investigated because these variations are among the most serious causes of higher emissions and lower efficiency. The main objective of this study was to evaluate the relationship between cyclic variations and combustion parameters. The combustion and cyclic variation characteristics were investigated using a diesel engine operating on Fischer–Tropsch (F–T) fuel synthesized from coal. Experiments were conducted under full load conditions at three engine speeds of 1200, 2000, and 2800 rpm. The results revealed that cyclic variations of F–T diesel were lower than those of 0# diesel, acquired the minimum value at the speed of 2000 rpm, and reached the maximum at the speed of 2800 rpm. The mean fluctuation intensity of F–T diesel was 0.185, 0.189, and 0.205 at speeds of 1200, 2000, and 2800 rpm, respectively, smaller than that of 0# diesel under the corresponding conditions. The relationships between cyclic variations and combustion parameters were analyzed by correlation methods. Maximum in-cylinder pressure (Pmax) increased linearly with increased ignition delay, while it decreased linearly with increased combustion duration. The Pearson's correlations between  $P_{max}$  and ignition delay were 0.75, 0.78, and 0.73; however, the corresponding values between  $P_{max}$ and combustion duration were 0.61, 067, and 0.65 when fueled with F–T diesel at speeds of 1200, 2000, and 2800 rpm, respectively. Moreover, the Pearson's correlations of 0# diesel were higher than those of F-T diesel at the same operating loads. Compared with combustion duration, the ignition delay had more important effects on cyclic variations with a higher Pearson's correlation. Furthermore, the ignition delay significantly influenced cyclic variation under a high speed load, while the combustion duration had a marked effect under low speed conditions. Overall, the results revealed the importance of combustion parameters on cyclic variation, which has great significance for controlled cyclic variation in diesel engines.

**Keywords:** cycle-to-cycle variation; fluctuation intensity; diesel engine; combustion parameters; F–T diesel

# 1. Introduction

With the increasing drive towards energy savings and low-carbon living, greater demand for stringent emissions regulations, and huge fuel consumption, researchers have progressively paid more attention to exploring clean alternative fuels for internal combustion engines in recent years [1,2]. The Fischer–Tropsch (F–T) catalytic conversion process can be used to synthesize diesel fuels from a variety of feedstocks including coal, natural gas, and biomass. Moreover, liquid fuels produced via the F–T process promise an attractive, clean, carbon-neutral, and sustainable energy source for the

transportation sector. In particular, the F–T diesel synthesized from coal by the F–T method has broad application prospects, as it has matured and entered the industrial stage [3,4].

Owing to the excellent specifications [5], F–T diesel can be used in unmodified diesel engines; therefore, many diesel engines powering large trucks, buses, and farm, railroad, marine, and construction equipment have been investigated to date [6–13].

Cycle-to-cycle variations were first analyzed in Spark engines, which are less significant in diesel engines due to the non-premixed combustion. However, this has been a research area of increased interest in the recent past since the two-stage injection strategies on the mixture formation were used in common rail diesel engines. These cyclic variations were considered to cause combustion instability and are detrimental to engine performance [14]. Koizumi et al. [15] found that variations in in-cylinder pressure were caused by increased ignition delay in cold-start engines. Zhong et al. [16] observed cyclic variations in the fuel path of diesel engines. Furthermore, Schmillen [17] reported that the in-cylinder pressure variations could not be explained in terms of injection variation. Later, a change in the intake pressure and intake temperature resulted in longer ignition delay, which led to an increase in the cycle-to-cycle variations [18,19]. Moreover, literature reports [18,20] showed that the variation of in-cylinder pressure decreased in fuels with high cetane number (CN). Furthermore, Panagiotis [21] predicted the influence of cyclic variations on NOx exhaust emissions, and the results showed that the NOx emissions increased by 5% when a point with fluctuating injection timing compared to a stable state. Jakob et al. [19] found that cyclic variation in soot was caused by combustion instabilities. Previous studies showed that the cyclic variability, together with in-cylinder pressure fluctuations, had a similar influence, increasing energy release during the combustion phase observed in homogeneous charge compression ignition engines [22-24].

It is well known that cycle-to-cycle variation in diesel engines can contribute to lower thermal efficiency and output power, as well as higher exhaust emissions [25,26]. The cycle-to-cycle variation is unexpected, which is harmful to diesel engine performance. Moreover, fuels with high CN and thus short ignition delay exhibited beneficial effects, decreasing cyclic variations of in-cylinder pressure. However, literature reviews showed that the research has been conducted on F–T diesel or F–T diesel blends, and mainly focused on emissions and combustion characteristics. Cycle-to-cycle variation in diesel engine fueled with F–T diesel has rarely been investigated. The main objective of this study was to investigate the variation performance in diesel engines with short ignition delay and long combustion duration conditions. For this purpose, F–T diesel and 0# diesel were used in a common rail engine for comparative analysis. Furthermore, the combustion parameters such as ignition delay and combustion duration coefficient under in-cylinder pressure condition and intensity of fluctuation under different ignition delay and combustion duration conditions were comprehensively analyzed. Finally, the effects of ignition delay and combustion duration on cyclic variation were presented through Pearson's correlation under varying speed conditions.

# 2. Materials and Methods

## 2.1. Research Engine Test Bench

Experiments were conducted in an inter-cooling, turbocharged, common rail diesel engine (Yun nei Group, Kunming, China) that was coupled to an electrical eddy current dynamometer and an external water circulation system. The detailed specifications of the diesel engine are listed in Table 1 and the test bench is schematically illustrated in Figure 1. A pressure sensor (Kistler 6050A41, Winterthur, Switzerland, sensitivity: 17 pc/bar) was installed to measure the in-cylinder pressure. An angle encoder (Kistler2613B1, Winterthur, Switzerland) was used to collect the top dead center (TDC) signal as well as crank angles. The in-cylinder pressure signal was collected for no less than 120 cycles under each load condition. Then the combustion parameters and combustion processes were derived by using MATLAB software (MathWorks, Natick, MA, USA) based on the data obtained from the in-cylinder pressure

sensor and angle encoder. The final in-cylinder pressure data were cycle-averaged for 100 cycles. Fuel consumption was obtained by calculating the fuel mass (Chengbang Science & Technology, Chengdu, China) during the engine operation under different conditions. Under all the test conditions, the intake air temperature was kept at  $50 \pm 3$  °C and the engine cooling water temperature was maintained in the range of  $80 \pm 5$  °C for reliable comparison. The test data were collected when the engine was operating at steady state. The uncertainty and accuracy of the key equipment used in this research are listed in Table 2.



Figure 1. Research engine test bench. ECU: electronic control unit.

Table 2. U	Jncertainty	and a	accuracy	of the	measuring	equipmen	t.
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Instrument	Parameters	Range	Accuracy	Uncertainty (%)
Electric eddy current	Torque Speed	0–600 (nm)	0.4%	
dynamometer	Torque Speed	0–8000 (rpm)	0.1%	-
Fuel consumption meter	Fuel mass	0–20 (kg)	0.4%	-
Pressure sensor	In-cylinder pressure	0–250 (MPa)	0.05 (MPa)	0.3

# 2.2. Test Fuels

F–T diesel and commercial 0# diesel that meets the China Stage VI standards were used in this study. The main specifications of the fuel are listed in Table 3, which were provided by the F–T diesel manufacturer. The F–T diesel is a type of industrial chemical and liquid fuel produced from coal as a raw material. It is an indirectly synthesized diesel with negligible sulfur and aromatic content. Compared to 0# diesel, F–T diesel has the properties of lower density and boiling point; however, its heating value and CN number are higher.

Specifications	0#-Diesel	FT-Diesel
Density (g/cm <sup>3</sup> )	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
Heating value (J/g)	42,652	47,128
CN	55.8	62.5

Table 3. The main specifications of the fuel.

# 3. Results and Discussion

In order to enhance the contrast effects and analyze the engine performance under severe operating conditions, a series of experiments were conducted under full load conditions. The engine used had a low speed of 1200 rpm and a high speed of 2800 rpm; in addition, a maximum break torque speed of 2000 rpm was selected. For more accurate assessment, the in-cylinder pressure, in-cylinder pressure increase rate, and combustion parameters such as HRR, ignition point, ignition delay, combustion duration, and CA50 were analyzed. The cyclic variation characteristics of in-cylinder pressure were also studied. The combustion characteristics of different fuel samples under various conditions were estimated in the following sections.

## 3.1. In-Cylinder Pressure

The variations of in-cylinder pressure for different fuels at various engine speeds are shown in Figure 2. Clearly, the F–T diesel leads to lower maximum in-cylinder pressure and the peak position is slightly advanced compared with 0# diesel under the same load conditions. This was genetically determined based on fuel properties, such as the CN and low heating value. Less combustible mixtures are formed due to the higher CN of F–T diesel, which leads to an obvious shortening of the ignition delay. The two fuels showed a similar trend in that the maximum in-cylinder pressure increased with increasing speed, and it was a downward trend when the engine was at the speed of 2800 rpm. This was attributed to the fact that the cycle time shortened and the residual exhaust gas rate increased with the further increase in the engine speed; thus, the dilution effect of residual exhaust gas became more obvious. The maximum in-cylinder pressure decreased and the corresponding peak position was retarded. Furthermore, visible roughness was observed at the speed of 2800 rpm, compared to that at other speeds. This was in accordance with the intensity of pressure oscillation; the combustion instabilities significantly increased and in-cylinder pressure fluctuation obviously enhanced when the engine was operated at the speed of 2800 rpm.



Figure 2. Variations of in-cylinder pressure.

#### 3.2. Heat Release Rate

HRR is an important combustion indicator and is calculated by using a single zone model based on the first law of thermodynamics, which has been widely used in previous studies [27–29]. HRR is defined by utilizing Equations (1) and (2), where the heat ratio is expressed in k, and A, h, n, Tg, and Tw represent the area of combustion chamber, the heat transfer coefficient, engine speed, gas temperature, and in-cylinder wall temperature, respectively [30–33].

$$\frac{\mathrm{d}Q}{\mathrm{d}\theta} = \frac{k}{k-1} P \frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{1}{k-1} V \frac{\mathrm{d}p}{\mathrm{d}\theta} + \frac{\mathrm{d}Q_{\mathrm{heat}}}{\mathrm{d}\theta} \tag{1}$$

$$\frac{dQ_{heat}}{d\theta} = \frac{Ah}{6n} (T_g - T_w)$$
<sup>(2)</sup>

Figure 3 shows the HRR under different speed conditions when the engine was fueled with the test fuels. As indicated in Section 3.1, the HRR variation was similar to the in-cylinder pressure. The maximum HRR increased with increasing speed, and it showed a downward trend when the engine operated at the speed of 2800 rpm. The peak HRR of premixed combustion and diffusion combustion can be obviously observed at the speeds of 1200 and 2000 rpm; however, the engine maintains the main injection only at the speed of 2800 rpm, and the HRR evolves from two peaks to a single peak. The F–T diesel was mainly mixed with the straight-chain paraffin with higher CN and burning rate. The maximum HRR phase decreased, and corresponding peak position was advanced under the fast combustion speed and less pre-mixed fuel.



Figure 3. Variations of heat release rate.

#### 3.3. In-Cylinder Pressure Increase Rate

The dp/d $\phi$  is related to the combustion noise of the engine. When the dp/d $\phi$  increases, unacceptable noise may be caused, which eventually leads to damage to the engine [34].

Figure 4 presents the relationship between  $dp/d\phi$  versus crank angle under different working conditions with the test fuels. The maximum  $dp/d\phi$  of F–T diesel was lower than that of 0# diesel, and the peak position is slightly advanced compared to 0# diesel. The maximum  $dp/d\phi$  decreased with increasing operating speed using the test fuels. This was mainly attributed to the fact that the ignition delay of F–T diesel was shorter than that of 0# diesel due to its higher CN. Thus, the gas and cylinder temperatures were lower for shorter air–fuel mixing process. Moreover, the constant volume combustion near TDC was decreased. The residual gas significantly affected the air–fuel mixture, as the speed increased for increasing residual exhaust gas rate. Thus, the  $dp/d\phi$  was slightly decreased and the peak value was advanced with increasing engine speed.



Figure 4. Variations of in-cylinder pressure increase rate.

## 3.4. Combustion Parameters

The ignition point is the crank angle that forms the first flame kernel. Ignition delay is calculated as the crank angle between fuel injection timing and the ignition point. CA10, CA50, and CA90 are defined as the crank angles where 10%, 50%, and 90% of mixture undergoes combustion, respectively [35].

The combustion parameters mentioned above are shown in Figure 5a. For the F–T diesel, the ignition point was advanced and the ignition delay was shortened due to its higher CN. The ignition delay was advanced by 1.2, 0.9, and 0.7 deg CA, respectively, for the speeds of 1200, 2000, and 2800 rpm. Furthermore, the F–T diesel vaporized at a much faster rate because it has a lower boiling point. CA50 was basically advanced by 2 deg CA with the increase in the engine speed. The increase of engine speed caused the system to prepare more mixture, so the ignition point was retarded and ignition delay was increased. However, the air–fuel mixture uniformity and combustion processes were modified when the engine was operated at higher speed, and the impact of fuel properties on combustion was reduced.



**Figure 5.** Comparison of combustion parameters: (**a**) Ignition point, ignition delay, and CA50; (**b**) combustion duration and break thermal efficiency (BTE).

The combustion duration refers to the crank angle between CA90 and CA10. The BTEis one of the key parameters that indicates the fuel conversion efficiency. Figure 5b shows the variations of combustion duration and BTE at different engine speeds.

Under different test conditions, F–T diesel showed a longer combustion duration than 0# diesel, and showed an opposite trend compared to ignition delay, as shown in Figure 5a. The combustion duration decreased with increasing engine speed using the test fuels. Moreover, the combustion duration was extended by about 5.8, 6.4, and 4.2 deg CA, respectively, for the speeds of 1200, 2000, and

2800 rpm compared to 0# diesel. Furthermore, the combustion duration decreased with the increase in engine speed. This is attributed to the fact that the shorter ignition delay resulted in a longer air–fuel mixing process to achieve a uniform result, thus leading to a slower premixed burning rate. The air–fuel mixing process was improved with the increase in the engine speed. The combustion speed was quicker, the combustion rate was higher, and the combustion process was shorter. Compared to 0# diesel, the BTE of F–T diesel was increased by 1.5%, 1.4%, and 0.7%, respectively, for the speeds of 1200, 2000, and 2800 rpm. The BTE exhibited great relevance at CA50. The BTE was increased as CA50 was advanced for the improved constant volume combustion near TDC [34,36–39]. Figure 5a,b exhibit good agreement between BTE and CA50. More heat leaked between the cylinder wall and the cooling water as CA50 was retarded, and the corresponding BTE was lower.

# 3.5. Cyclic Variation Characteristics

The inequality of fuels injected into the engine caused cycle-to-cycle variations, which significantly affected the combustion stability. The  $P_{max}$  has "easy to measure" and "sensitive to cyclic variations" characteristics, so the coefficient of variation (COV) was analyzed to characterize the difference based on the published research [40,41]. COV was defined by using Equations (3)–(5); the linear dependence of relativity on peak in-cylinder pressure and combustion parameters ( $R(P_{max}, y)$ ) was measured in terms of the Pearson's correlation, as presented in Equation (6), where the mean value of in-cylinder pressure is expressed in  $\overline{P_{max}}$ , N is the total cycle number, i represents the cycle index, and  $\sigma_P$  is the standard deviation [42–44]. A significant correlation was observed among parameters when R was in the range of 0.5–1.0.

$$\overline{P_{max}} = \frac{1}{N} \sum_{i=1}^{N} P_i$$
(3)

$$\sigma_{\rm P} = \sqrt{\frac{\sum_{i=1}^{\rm N} (P_i - \overline{P_{max}})^2}{{\rm N} - 1}} \tag{4}$$

$$Cov_{P} = \frac{\sigma_{P}}{\overline{P_{max}}} \times 100\%$$
(5)

$$R(P_{max}, y) = \frac{\sum_{i=1}^{N} (P_i - \overline{P})(y_i - \overline{y})}{(N-1)(\sigma_P \cdot \sigma_y)}$$
(6)

 $P_{max}$  is an important mechanical indicator in modern compression engine [34,45]. Figure 6 demonstrates the variations of  $P_{max}$  for 100 consecutive cycles under three speed conditions with the test fuels. Furthermore, the differences between the mean  $P_{max}$  and COV under 100 cycles were calculated. P<sub>max</sub> changed stochastically under different speed conditions. However, the mean P<sub>max</sub> increased with the increase in the speed, and the F–T diesel showed a lower mean value than 0# diesel at the same speed, which was in line with the results presented in Section 3.1. Notably, values of COV of F–T diesel are 1.72, 1.32, and 2.02%, respectively, for the speeds of 1200, 2000, and 2800 rpm, which are lower than 2.49, 2.16, and 4.01% of 0# diesel. The F–T diesel with higher CN and lower viscosity was found to be beneficial to improve air-fuel mixture. Moreover, fuel atomization was in accordance with gas turbulence intensity during the kernel formation phase owing to its advanced combustion. Ascribed to these factors mentioned above, the F-T diesel exhibited excellent properties of combustion stability with little variation. On the other hand, cyclic variations at the speed of 2000 rpm were minimal and the maximum was obtained at the speed of 2800 rpm. This was mainly because the quality of air-fuel mixtures was further improved with a higher in-cylinder temperature and airflow motion as the engine speed increased. Thus, the combustion stability increased, and finally the COV decreased at a speed of 2000 rpm compared to that at 1200 rpm. At the speed of 2800 rpm, the time for proper combustion phasing was reduced; moreover, the residual exhaust gas was increased and therefore it was easier to cause combustion instability, so the cyclic variation sharply increased.



Figure 6. COV of P<sub>max</sub> under different speeds with the test fuels. (a) F-T diesel, (b) 0# diesel.

Figure 7 displays the fluctuation intensity of 100 consecutive cycles versus ignition delay under different conditions with the test fuels. Correspondingly, the intensity of fluctuation versus combustion duration is shown in Figure 8. Notably, the intensity of fluctuation follows a trend similar to that of the COV, which acquired the minimum average value at the engine speed of 2000 rpm and the maximum value at the speed of 2800 rpm. The F–T diesel presents a lower mean fluctuation intensity than 0# diesel; moreover, the mean intensity of fluctuation was 0.185, 0.189, and 0.205 at the speeds of 1200, 2000, and 2800 rpm during the entire test conditions, respectively. Furthermore, the mean intensity of fluctuation of 0# diesel was 0.187, 0.191, and 0.211, respectively, under the corresponding speed conditions. As mentioned above, F–T diesel showed excellent properties of combustion stability, so the mean and variance of in-cylinder pressure were small, which resulted in lower fluctuation intensity. When the engine was operated at the speed of 2800 rpm, the in-cylinder pressure increased and the intensity of fluctuation was enhanced with the increase in combustion instability.



Figure 7. Intensity of fluctuation versus ignition delay under different speeds with the test fuels.



Figure 8. Intensity of fluctuation versus combustion duration under different speeds with the test fuels.

To explore the trend of peak in-cylinder pressure with ignition delay, the relationship between  $P_{max}$  and ignition delay was established as shown in Figure 9. Figure 9 shows that the  $P_{max}$  increases linearly with a longer ignition delay. The relationship can be described by the equation in the figure and R was 0.75, 0.78, and 0.73 when the engine was fueled with F–T diesel at the speeds of 1200, 2000, and 2800 rpm, respectively. A similar method was used to state the relationship between  $P_{max}$  and ignition delay under different working conditions using 0# diesel. The results are displayed in the corresponding figure and the R was 0.81, 0.86, and 0.83, respectively. It can be concluded that  $P_{max}$  and ignition delay were highly correlated, and a longer ignition delay was useful in increasing in-cylinder pressure. However, the cyclic variation enhanced and the tendency of combustion instability increased. The results indicated that cyclic variability rapidly increased with ignition delay under high speed, which was relatively small under lower speed conditions according to the fitting equations between  $P_{max}$  and ignition delay. Compared to 0# diesel, ignition delay had relatively little effect on the cyclic variation since the R of F–T diesel was smaller than that of 0# diesel at the corresponding speed conditions.



**Figure 9.** The correlation between peak in-cylinder pressure and ignition delay with (**a**) F–T diesel, (**b**) 0# diesel.

The relationship between  $P_{max}$  and combustion duration is shown in Figure 10. In contrast to the results presented in Figure 9, the  $P_{max}$  decreases linearly with increased combustion duration. The relationship can be described in terms of the equations displayed in the figure and R was 0.61, 0.67, and 0.65 when the engine was fueled with F–T diesel at the speeds of 1200, 2000, and 2800 rpm, while for 0# diesel R was 0.73, 0.79, and 0.75 under the corresponding speed conditions, respectively. Similar findings are displayed under other operating conditions. This indicates that  $P_{max}$  increases with short combustion duration, which is correct since the higher constant volume combustion leads to a shorter combustion duration. Within a certain range, combustion stability might be improved

by decreasing the combustion duration. According to the fitting equations, the combustion duration significantly influenced the  $P_{max}$  under the lower speed condition. Similarly, the correlation of F–T diesel was smaller than that of 0# diesel. Moreover, combustion duration had a more obvious effect on cyclic variation when fueling with F–T diesel compared to 0# diesel, because the absolute slope of linear regression was greater than that of 0# diesel at the same speed condition due to its higher CN and shorter combustion duration.



**Figure 10.** The correlation between peak in-cylinder pressure and combustion duration with (**a**) F–T diesel, (**b**) 0# diesel.

## 4. Conclusions

The main objective of this study was to research the performance characteristics when using Fischer–Tropsch (F–T) fuel in diesel engines and highlight the importance of a short ignition delay and a long combustion duration on cyclic variation. The following conclusions can be drawn compared to 0# diesel.

The F–T diesel has the properties of high CN and low boiling point, which led to a short ignition delay and a long combustion duration. Compared to 0# diesel, owing to these superior characteristics that are beneficial to improve the air–fuel mixture and reduce the gas turbulence during the kernel formation phase, the cyclic variation and fluctuation intensity of F–T diesel were smaller. To a certain extent, cyclic variation was improved with increased speed, and then the average fluctuation intensity decreased when the engine was fueled with the test fuels. A strong linear relationship was observed between cyclic variation and ignition delay. The cyclic variation linearly increased with ignition delay because pre-mixed reactivity increased with long ignition delay. Moreover, the influence was more obvious under high speed conditions. Compared to ignition delay, the combustion duration made less impact on cyclic variation, which resulted in a smaller value of R at the same speed. Furthermore, the cyclic variation linearly decreased with long combustion duration; the F–T diesel showed lower R than 0# diesel under all the test conditions. Moreover, the combustion duration had a significant influence on cyclic variation at low speeds due to the large absolute slope of linear regression.

In all, in-cylinder pressure fluctuations and the effect of short ignition delay and long combustion duration on cycle-to-cycle variations were further analyzed. These effects can be used to improve combustion stability and control the harmful damage caused by cyclic variation. Furthermore, the diesel engine can run stably and effectively without any modifications with F–T diesel.

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# Nomenclature

crank angle
crank angle at which 50% of the fuel has burnt
break thermal efficiency
electronic control unit
maximum in-cylinder pressure
exhaust gas recirculation
top dead center
Pearson's correlation efficiency
cetane number
heat release rate
ignition delay
combustion duration
coefficient of variation of maximum in-cylinder pressure

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