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Full Length Article

Effects of retarded fuel injection timing on combustion and emissions of a diesel engine fueled with canola biodiesel

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ABSTRACT

In this study, retarded injection timing was investigated to overcome higher NO_x emissions of a direct injection diesel engine fueled with diesel fuel (90%)/canola biodiesel (10%) blend. The experiments were performed at the maximum torque speed (2200 rpm) under four loads (3.75 Nm, 7.5 Nm, 11.25 Nm and 15 Nm) and three injection timings (original –28 °CA, 26 °CA and 24 °CA bTDC). The effects of the retarded fuel injection timings on the performance, the emissions and the combustion were examined in detail. The changes in the cylinder pressure, the combustion timing, the heat release rate, the fractions and the durations of premixed and diffusion combustion phases, the injection and the ignition delays, NO_x, total hydrocarbon, CO, CO_2, smoke, break specific fuel consumption and break thermal efficiency were determined and presented in the paper. The experimental results showed that the injection retardation of 2 °CA could satisfy lower NO_x and break specific fuel consumptions without significant adverse effects on the other engine parameters, while the further retarding of the injection timing deteriorated the parameters. The retarded injection timing of 2 °CA provided decreases in NO_x up to 11% and in BSFC up to 2.7%.

1. Introduction

Many studies have been worked on alternative diesel fuels to reduce the greenhouse gas emissions and replace some of fossil fuels [1,2]. As an alternative fuel, biodiesel has been widely used without any modification in all segment vehicles with old and new technology compression ignition engine types [3]. Although the biodiesel can be used purely, engine manufacturers warrant the usage of biodiesel blends which contain up to 20% in volume [4,5]. In the literature, it was reported that the biodiesel addition changes the parameters of engine performance, combustion and emissions even at 5% blending ratio depending on the properties of the biodiesel and the engine specifications [6–8]. The biodiesel fuel compatible with biodiesel standards EN 14214 and ASTM D6751 was crucially affected from the raw material properties [9]. There are approximately 300 feedstocks for biodiesel production [10]. Among them, canola (rapeseed) is commonly preferred in European countries due to suitable properties for meeting the standards, easy growing in different climate conditions and high oil content [11]. Although the positive effects of high quality biodiesel are also taken into account, such as lesser engine oil consumption, lower engine wear, emission reduction and carbon-neutrality, the increments in NO_x emissions and break specific fuel consumption (BSFC) are the most notable negations [12–14].

NO_x formation mainly occurs with three mechanisms, namely thermal, prompt and fuel. Thermal NO_x is more dominant than others in diesel engines [15]. However, NO_x formation with biodiesel fuel blends is linked on a number of related effects such as the changing of combustion timing due to fuel injection system response from different speed of sound and bulk modulus of biodiesel; biodiesel fuel characteristics as oxygen content, cetane number, double bound or saturated fatty acid content; engine running conditions especially the engine load [16,17]. Different strategies such as ethanol or methanol additions, exhaust gas recirculation, changing injection timing and pressure, water emulsion, using of fuel additives and dual biodiesel blends were used to control the NO_x formation [18–20]. Among them, the changing the injection timing was reported as an important technique for effective NO_x emissions control and it can be applied easily in all engine types without significant modifications [21]. It is known that NO_x formations mainly occur in the lean flame region during the

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Nomenclature

A	Surface area of cylinder wall
aBDC	After bottom dead center
aTDC	After top dead center
bBDC	Before bottom dead center
bTDC	Before top dead center
B10	Blend of the canola biodiesel (10%) and No.2 diesel fuel
BSFC	Break specific fuel consumption
BTE	Break thermal efficiency
CA	Crank angle
CA50	50% of accumulated heat release
CA90	90% of accumulated heat release
COV_{imep}	Cyclic variations of indicated mean effective pressure
c_p	Constant pressure specific heat ratio
D	No. 2 diesel fuel
EOI	End of fuel injection
EOP	End of the premixed combustion stage
EVC	Exhaust valve closing
EVO	Exhaust valve opening
FMEP	Friction mean effective pressure
h	Heat transfer coefficient
h_f	Fuel enthalpy
HRR	Heat release rate
IA24	Injection advance of 24 °CA bTDC
IA26	Injection advance of 26 °CA bTDC
ID	Ignition delay
IMEP	Indicated mean effective pressure
IVC	Intake valve closing
IVO	Intake valve opening
m	Total mass in cylinder
m_f	Injected fuel mass
n	Engine speed
p	In-cylinder pressure
Q_{gr}	Gross heat release
Q_{ht}	Transferred heat from cylinder walls
R	Universal gas constant
SOC	Start of combustion
SOI	Start of fuel injection
TDC	Top dead center
T_g	In-cylinder gas temperature
T_w	Mean temperature in cylinder walls
V	Total in-cylinder volume

premixed combustion phase in compression ignition engines. Therefore, the maintaining optimal combustion timing with controlling the amount of fuel burned in the premixed phase can significantly reduce the NO_x emissions [22].

Gnanasekaran et al. [23] investigated the influence of injection timing on the fish oil biodiesel/diesel blend ratio ranged from 0:100 to 100:0 in a DI compression ignition engine. The tests were carried out with the retarded and advanced injection timings of 3 °CA with respect to the original timing. It was evaluated that the retardation of injection timing caused lower NO_x and higher break thermal efficiency. In another study, mahua oil biodiesel and its blends at different ratios with diesel fuel were tested in a Richardo research engine for three injection timings at various compression ratios by Raheman and Gahdge [24]. It was found that the decreasing engine performance due to biodiesel addition was enhanced with the advanced injection timing. In addition, a study including the more sensitive injection timing was performed for 20% castor biodiesel blend by Deep et al. [21]. The lowest NO_x emission was obtained with retarded injection of 2 °CA, reasons of decrease in the break thermal efficiency were remarked as the increased mechanical and heat losses for the advanced injection timing, and reduction of peak cylinder pressure for retarded injection timing.

A research group investigated effects of fuel injection timing on emissions, injection, combustion, and performance characteristics of a direct-injection diesel engine fueled with canola oil methyl ester-diesel fuel blends [8,25]. They reported that the retarding the injection timing by 5 °CA (from 20 to 15 °CA bTDC at 20 Nm the engine load) resulted in decreasing of the nitrogen oxide and carbon dioxide, and increasing of the smoke opacity, the hydrocarbon and the carbon monoxide for all test conditions. In addition, it was pointed out that the advanced and retarded injection timings caused negative effects on the brake-specific fuel consumption and the brake thermal efficiency.

The effects of injection pressure and timing were examined by Kannan and Anand [26]. In order to compare the waste cooking oil biodiesel and diesel fuels, the advanced injection timing was performed at 1.5°CA and 3.0°CA with regard to the original value. It was stated that the ignition delay was mainly influenced with the injection timing rather than injection pressure. Moreover, they indicated that the advanced injection timing and high injection pressures reduced the heat release rates. Jaichandar et al. [27] investigated the combined effects of combustion chamber geometry and injection timing for an engine fueled with pongamia oil methyl ester. The fuel injection timing was sequentially retarded with 1 °CA increment from the standard injection timing (23 °CA bTDC) to 20 °CA bTDC, and advanced to 24 °CA bTDC at the five different loads. It was found that the low retardations in the injection timing improved the engine performance while the more retardation adversely affected. The advanced injection timing caused increase in NO_x similar to Hwang et al. [28]. Furthermore, Ganapathy et al. [29] investigated the effects of injection timing on jatropha biodiesel usage. It was found that the modulation of injection timing for biodiesel blended fuels can satisfy benefits in terms of performance and emissions.

The literature studies mentioned above reported that one of the most significant engine parameters affecting the NO_x emissions is the retarding injection timing. Also, these literature outcomes reveal that the investigation of the effects of injection timings has an importance in the adaptation of biodiesel/diesel blends to existing engines even at the low blend ratios [30]. In addition, even small changes in the injection timings have some effects on the engine characteristics [31]. Previous studies mainly involve the effects of injection timing on performance and emission parameters, however combustion and injection parameters such as heat release rate, combustion duration, fuel line pressure, ignition delay were investigated in minor parts of these works [21,23,25–28,32]. In addition, other combustion and injection parameters such as fractions and durations of combustion phases, injection and ignition delays and center of accumulated heat release, should be considered for NO_x formation without penalty in BSFC [24–26]. Also, these parameters are useful information in the combustion modelling and the determination of the strategies for the internal combustion engines [33].

A comprehensive study including the parameters mentioned above has not been found in the literature on canola biodiesel within the knowledge of the authors [34–36]. In this present study, detailed engine tests were performed with the canola biodiesel/diesel blend to investigate the effects of fuel injection timing on the performance, the emissions and the combustion. The changes in the break specific fuel consumption and the break thermal efficiency, the cylinder pressure, the combustion timing, the injection and the ignition delays, the heat release rate, the fractions and the durations of premixed and diffusion combustion phases, NO_x, total hydrocarbon, CO, CO_2 and smoke, were found and compared in the paper.

2. Material and methods

2.1. Test fuels

Canola oil, which is assumed to be commonly used in the future as a biodiesel feedstock, was chosen for the biodiesel production in this study. Canola can be easily grown in different climate conditions and has high oil content and canola oil is one of the most suitable feedstock for biodiesel production due to suitable properties for meeting the standards. The canola oil contains mainly unsaturated fatty acid compositions (58.9% oleic and 20.57% linoleic). The amount of free fatty acid in canola oil was below 0.5%. Therefore, methanol/oil molar ratio and amount of sodium-hydroxide per liter of the oil were applied in the transesterification process as 6:1 and 3.5 g, respectively. Reaction temperature was specified as 60 °C. After separation of glycerol and three times washing with distilled water, the ester was heated up to 100 °C in order to eliminate the residuals. Conversion ratio to methyl esters of 97% was achieved by using this procedure. The important properties of the biodiesel were given in comparison with the standard limits in Table 1. Among the properties, the oxidation stability of the biodiesel was not within the limits. The other properties of canola biodiesel were within the EN 14214 limits. The oxidation stability was improved by adding 500 ppm pyrogallol as an antioxidant [37]. The biodiesel fuel produced by using the canola oil was blended with a commercial diesel fuel (D) with 10% mixing ratio by volumetric (B10), which is suitable for real life applications.

2.2. Test system and method

All the experiments were performed at the maximum torque speed (2200 rpm) under low (3.75 Nm), partial (7.5 Nm), medium (11.25 Nm) and high (15 Nm) loads in a DI compression ignition engine. The tests examined under these four loads were performed for diesel fuel (D) and diesel–biodiesel blend fuel (B10) with standard injection, retarded injections of 2 (IA26) and 4 (IA24) °CA. During the experimental procedures, the oil temperature was kept at 82 ± 2 °C. To prevent the variations at the inlet temperature, the inlet air temperature was kept constant at 30 ± 1 °C by using an air conditioning device.

The specifications of the test engine and a visual description of the experimental set-up are given in Table 2 and Fig. 1, respectively. A DC dynamometer (10 kW @4000 rpm) was used for the engine loading on Cussons P8160 test bed. Measurements of inlet air flow rates were made by using a laminar flow element (Merriam Z50MC2-4F) and a mass flow computer (Merriam LFS-1). The mass of injected fuel during test period was measured by using an electronic balance with high precision. NiCr-Ni type thermocouples were used in the measurements of the lubrication oil, the inlet air and the exhaust gas temperatures. The determinations of CO, CO_2, THC and NO_x emissions were performed with Environnement S.A. brand EGAS 2 M model exhaust gas analyzer. AVL 4000 DiSmoke model opacimeter was used to measure the smoke emissions. Koyo brand TRD J1000-RZ model optic encoder with 1000 pulse incremental was placed to the end of pulley in order to determine of the crank angle and TDC.

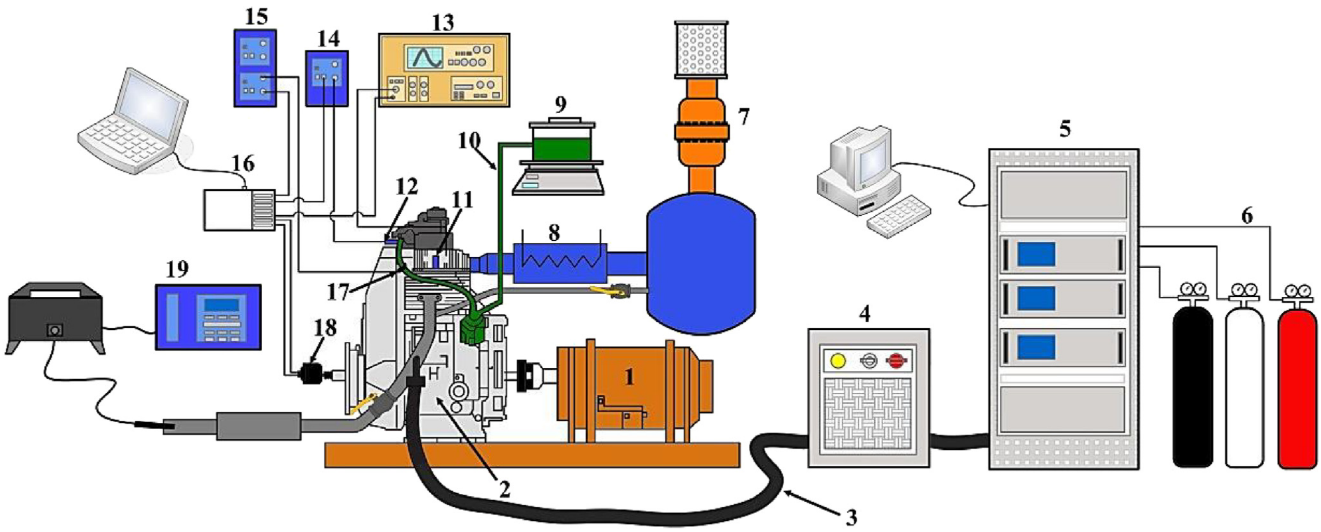
The definitions of combustion and injection characteristics were shown in Fig. 2. The static injection advance of the test engine having a Jerk-type fuel pump was adjusted by modifying the thickness of shim between pump and engine block and measured with Sincro brand DS-88n model stroboscope based on the piezo resistor placed on fuel line. However, the start of fuel injection (SOI) was regarded as the dynamic injection timing. Dynamic injection timing was measured using a needle lift sensor (Wolff brand proximity sensor based on hall-effect) and signal conditioner. The needle lift threshold value of 0.01 was found enough for the determination of SOI and end of fuel injection (EOI). The injection durations were obtained from the angle difference between SOI and EOI. Ignition delay (ID) is equal to the angular spacing between the SOI and the start of combustion (SOC). Second derivation of cylinder pressure was employed to obtain the SOC. The corresponding crank angle to the zero point proceeding the peak value of the second derivation was assigned as SOC. End of premixed combustion stage (EOP) is the crank angle in the local minimum just after the maximum value of heat release rate (HRR). The premixed combustion duration in terms of the crank angle was determined from the interval time between SOC and EOP, meanwhile the diffusion combustion duration was calculated with the interval time between EOP and 90% of accumulated heat release (CA90). The total combustion duration is the summation both the premixed and the diffusion combustion durations. The crank angle which is

Table 1
Properties of the Canola Oil Methyl Ester.

Property	Analysis result	EN 14214 Limits Min/Max	Test method
Ester content (% m/m)	97.2	96.5/-	EN 14103
Density @ 15 °C (kg/m^3)	884.4	860/900	EN ISO 12185
Viscosity @ 40 °C (mm^2/s)	4.526	3.5/5	EN ISO 3104
Flash point (°C)	177.6	120/-	EN ISO 3679
Sulphur content (mg/kg)	2.5	-/10	EN ISO 20846
Cetane number	54.3	51/-	EN ISO 5165
Oxidation stability at 110 °C (h)	**2.6**	6/-	EN 14112
Acid value (mg KOH/g)	0.48	-/0.50	EN 14104
Iodine value g (iodine/100 g)	120	-/120	EN 14111
Linolenic acid methyl ester (% m/m)	9.3	-/12	EN 14103
Polyunsaturated methyl esters ($\geq$4 double bonds) (% m/m)	0.0	-/1	EN 14103
Methanol content (% m/m)	0.001	-/0.20	EN 14110
Cold filter plugging point (°C)	−8	-/−15 (win.) +5 (sum.)	EN 116

Table 2
Technical specifications of the test engine.

Engine type		DI-Diesel engine, natural aspirated, air cooled
Make/Model		Antor/6LD400
Cylinder number		1
Compression ratio		18:1
Bore × stroke		86 × 68 mm
Displacement		395 cm^3
Max. power		5.4 kW@3000 rpm
Max. torque		19.6 Nm@2200 rpm
Fuel injection system		PF Jerk-type fuel pump
Combustion chamber geometry		ω type
Nozzle opening pressure		180 bar
Injection nozzle		0.24 mm × 4 holes × 160°
Static fuel injection timing		28 bTDC °CA
Valve timings	IVO/IVC	7.5 bTDC/25.5 aBDC °CA
	EVO/EVC	21 bBDC/3 aTDC °CA



1. DC dynamometer, **2.** Test engine, **3.** Heated emission sampling line, **4.** Emission sampling system, **5.** Emission analyzers rack cabin, **6.** Function and span gases, **7.** Laminar air flow element, **8.** Inlet air heater, **9.** Balance, **10**. Fuel line, **11.** In-Cylinder Pressure Sensor, **12.** Needle-lift sensor, **13.** Combustion analyzer, **14.** Needle-lift sensor amplifier, **15.** Fuel line pressure sensor amplifier, **16.** Data acquisition system, **17.** Fuel line pressure sensor, **18.** Encoder, **19.** Opacimeter.

Fig. 1. The experimental setup.

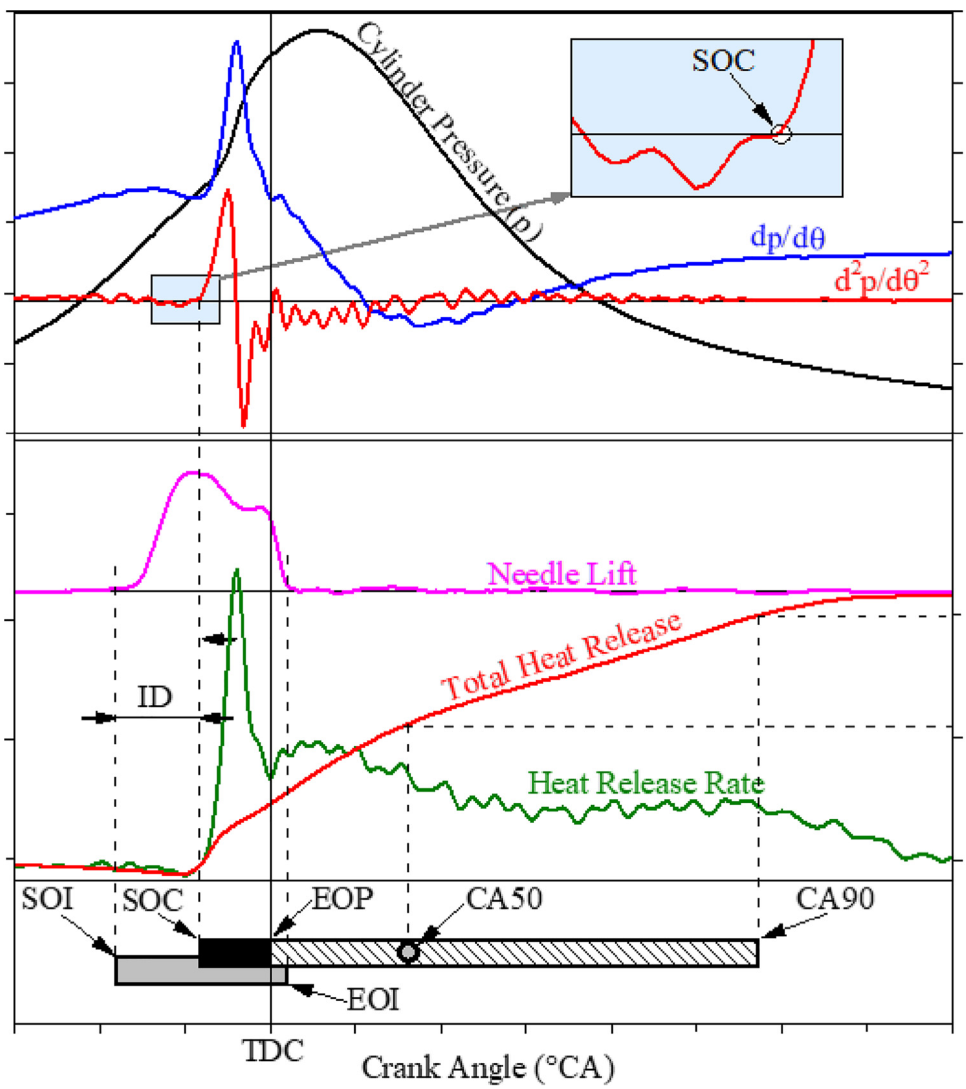


Fig. 2. The definitions of combustion and injection characteristics.

equal to 50% of the accumulated heat release is abbreviated as "CA50".

The values of in-cylinder gas pressures were obtained from Cussons P4110 combustion analyzer with an AVL 8QP500c model pressure sensor. The data collected from the sensors using NI USB 6259 DAQ card with 0.36°CA resolution were averaged for 50 consecutive cycles after the filtering procedure. Based on the Thermodynamics' First Law, the single-zone combustion model was used in the calculation of the HRR. The single-zone model equations including both the heat transfer from cylinder walls and the internal energy change with temperature are given in following equations [38].

$$\frac{dQ_{gr}}{d\theta} = \left(\frac{c_p}{R} - 1\right)\left(P\frac{dV}{d\theta} + V\frac{dP}{d\theta} - \frac{PV}{m}\frac{dm}{d\theta}\right) + P\frac{dV}{d\theta} - h_f\frac{dm_f}{d\theta} + \frac{dQ_{ht}}{d\theta} \quad (1)$$

$$\frac{dQ_{ht}}{d\theta} = \frac{\bar{h}A}{6n}\left(T_g - T_w\right) \quad (2)$$

The constant pressure specific heat ratio, universal gas constant, fuel enthalpy and heat transfer coefficient are symbolized as c_p, R, h_f and h, respectively. The changes in gas composition were determined according to mass flow rates of the injected fuel (m_f) and burned fuel. Variations of c_p were computed by using the mean in-cylinder gas temperature (T_g) and JANAF coefficients. The heat transfer was calculated with the Hohenberg correlation and the mean wall temperature (T_w) was regarded as 450 °C (723 K) [39]. The accuracies and uncertainties for measurements and calculations are shown in Table 3.

3. Result and discussion

3.1. Effects of the biodiesel blend and the injection timing on the combustion and the injection parameters

The effects of the fuel injection timing with the biodiesel addition on the combustion/injection parameters and the premixed/diffusion combustion phase fractions are given in Fig. 3 and Fig. 4, respectively. The presented results in the figures will be discussed together for all engine loads by considering the effect of each parameter.

The biodiesel blend generally resulted in low peak HRR and thus prominently low in-cylinder pressure at especially the medium and the high loads. The percent reductions in HRR values with B10 are 14.47, 16.10, 21.83 and 21.61% from the low load to the high load. These reductions can be explained mostly due to the lower heating value of the biodiesel and earlier SOI leading to the short ID. As can be seen in Fig. 3, SOI was advanced with the biodiesel addition, except the low load. Advancement up to 1.65 °CA in SOI is seen at the high load, while there is a little retardation of 0.13 °CA at the low load. Besides of the lower vapor pressure of the biodiesel, its high viscosity, density and bulk modulus are important factors leading to rapid pressure wave propagation in the fuel rail and so the earlier SOI [40].

Table 3
Measurement accuracies and calculation uncertainties.

Accuracy		Uncertainty	
CO	2%	Torque	±0.25% (Nm)
CO_2	2%	Fuel flow rate	±0.72% (g/min)
NO_x	0.1 (ppm)	Intake air flow	±1.2% (L/min)
THC	0.05 (ppm)	Thermal efficiency	±1.16%
Smoke	0.01 (m^{-1})	Specific fuel consumption	±1.26% (g/kWh)
Time	±0.5% (s)		
Load	±0.25% (N)		
Fuel mass	±0.1 (g)		
Temperature	±1 (°C)		
Engine speed	±1% (1/min)		
Cylinder pressure	±0.6 (bar)		

Injection duration increased up to 2 °CA with the biodiesel addition at the high load, also a slightly increment was observed at the other loads. It is thought that the increase in the injection duration at high load can be mainly impressed by the more fuel requirements to provide the same engine torque with the biodiesel fuel blend having lower calorific value. The SOC occurred earlier with the biodiesel blend with respect to the diesel fuel under the all loads. These advancements in SOC from the low load to the high load are 0.44, 0.94, 1.07 and 1.14 °CA, respectively. Besides the advancement on SOI as mentioned above, higher cetane number and inherent oxygen content of the biodiesel also caused the early SOC. However, the further decrease of ID in parallel with the load increase can be explained by the higher in-cylinder temperature.

The biodiesel blend prompted the increase in the combustion duration at the low load. Also, this increase was not appeared in the other loads due to the higher in-cylinder temperature. Except an increase of 3 °CA in diffusion combustion duration at the low load, the premixed and diffusion combustion durations were not changed significantly. The CA50 at the low and the high loads moved away from TDC for the biodiesel blend, while it was very similar with the diesel fuel at the partial and the medium loads. It is thought that this phasing in CA50 was caused from the low combustion temperature at the low load and the increase of injection duration at the high load. For the all loads, the biodiesel fuel blend led to the reduction of the total heat release in the premixed combustion phase as can be seen in Fig. 4. This decrease in the premixed combustion fraction is a result of the earlier SOC and the lower heating value of biodiesel blend. All diffusion combustion fractions increased as a consequence of the continued fuel injection parallel with the engine load.

The static fuel injection timing of the test engine is 28 °CA bTDC. The results of different injection timings on the biodiesel blend fuel are given in Fig. 3 and Fig. 4. It can be said that the injection parameters have considerably effects on the combustion parameters. The SOC generally delayed in parallel with the retarded injection timing. These retardations in SOC generally resulted in the low HRR in the premixed combustion phase and combustion pressures. While it was not observed a significant difference in the ID at the low and the partial loads, ID was decreased with retarded injection at the other loads, except with IA24 at the medium load. The increase in ID based on further retardation (IA24) at the medium load was attributed to the low temperature environment in the combustion chamber [41,42]. Even though it was not clearly observed in IA26, a significant increase in the premixed combustion durations was determined for IA24, except the low load.

With the retardation of injection timing, CA50 values were generally shifted beyond TDC as based on the late SOC. The premixed and the diffusion combustion fractions were not significantly affected from the retarded injection timing at the medium and the high loads, while there were the steadily declining in the premixed combustion fractions at the low and the partial loads. The injected fuel masses before SOC were very close at these engine loads due to identical ID durations. Moreover, the in-cylinder temperatures decreased due to lower HRR with the retarded injection. The low temperature resulted in the low burning speed which caused the shifting of combustion towards EOI and decrease in the premixed combustion fractions at the low and the partial loads, while the rising temperature with load increment compensated the decreasing of ID at the medium and the high loads.

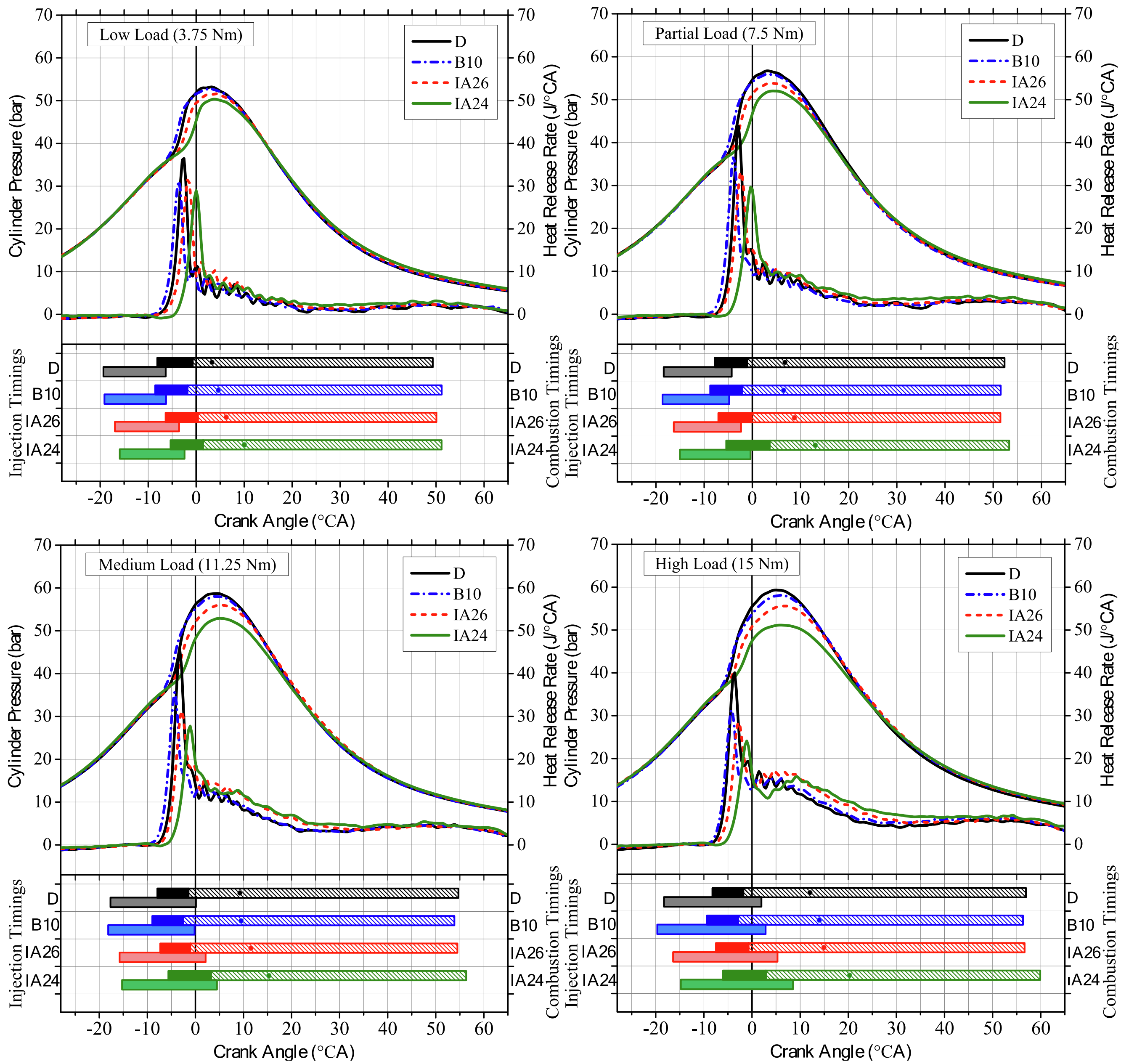


Fig. 3. The effects on combustion and injection parameters.

3.2. Effects of biodiesel blend and injection timing on the performance

The BSFC and break thermal efficiencies (BTE) are given in Fig. 5. The trends of the performance parameters were generally similar at almost all loads. The lower heating value of B10 resulted in the rise of BSFC and the decrease of BTE with comparing to the diesel fuel [4,12–14,30,37,43,44]. To provide the same power of a fuel with lower heat capacity, the more fuel requirement was the main factor of increasing in the BSFC. The other factors affecting the BSFC and the BTE are the increase of viscosity, density and surface tension, which provide the more fuel injection in mass by increasing the line pressure [40].

The retarded injection up to 2 °CA (IA26) provided to largely overlap the performance parameters of the biodiesel blend and the diesel fuel as can be seen in Fig. 5. The indicated mean effective pressure (IMEP) values for B10 (7.78 bar) and IA26 (7.76 bar) were higher than those of D (7.62 bar) at high loads, while vice versa for IA24 (7.57 bar). These increments in the IMEP value were also seen at the other loads with IA26 caused to the enhancement in the BTE. However, the decrease of in-cylinder pressure resulting with the lower friction mean effective pressure (FMEP) can be thought as another reason of this performance enhancement. The studies on engine friction reveal that the frictions in the piston and main bearings increase with the rise of maximum cylinder pressure [45–47]. A decrease in the BTE with the retardation of CA50 is the expected situation. The CA50 values shifted beyond TDC with the retarded fuel injections with respect to B10, as seen from Fig. 3. Also, the decrease in the combustion durations for IA26 at all loads prevented this negative situation by increasing the degree of constant volume combustion [48,49]. The decrease of the BTE value with retardation of 4 °CA (IA24) at the high load can be attributed to the excessive shifting of CA50 as well as the increase in combustion duration.

Coefficient of variations of IMEP (COV_{imep}) generally increased with B10 in reference to the diesel, except the high load. Inherent oxygen content of biodiesel can be seen as a reason of the enhance-

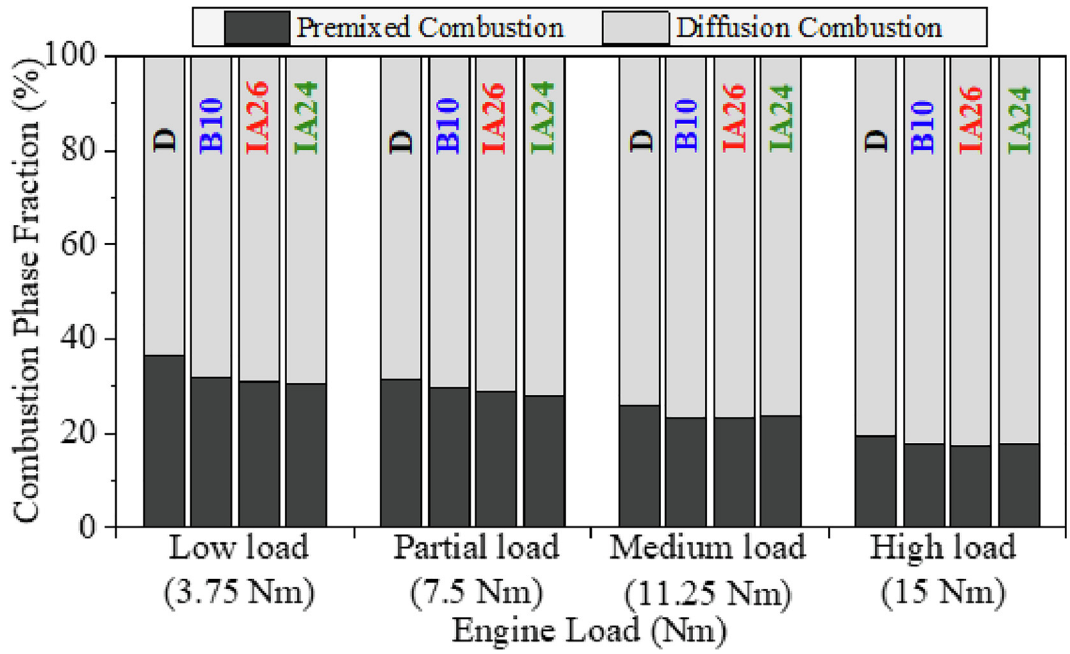


Fig. 4. The effects on premixed and diffusion combustion fractions.

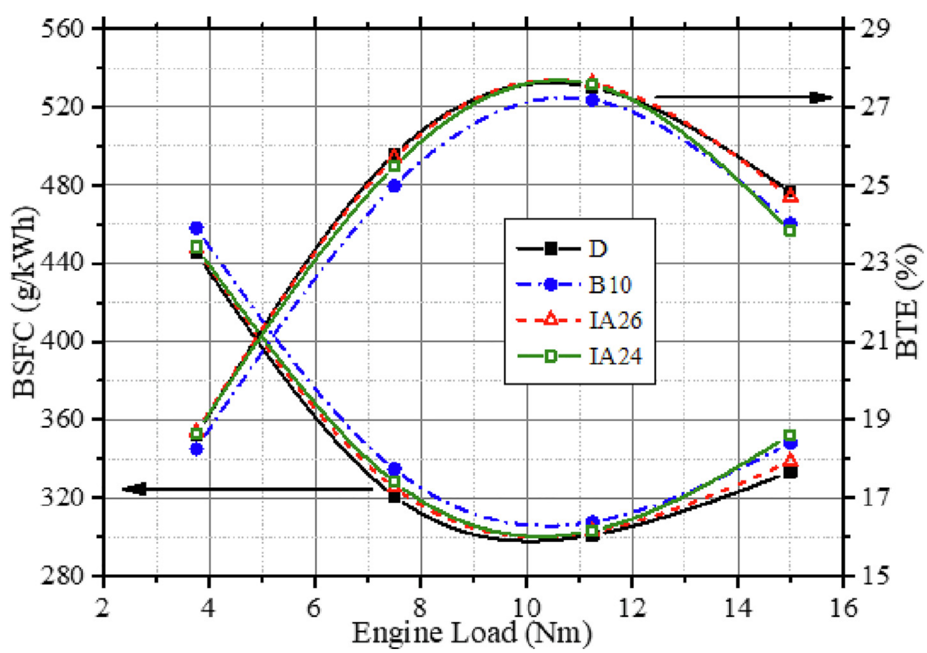


Fig. 5. The effects on performance parameters.

ment in COV_{imep} at the high load observing lack of oxygen. COV_{imep} values decreased up to 1.26% with IA26 compared to B10. A deterioration in COV_{imep} value of 1.01 was determined at the high load with IA24 with respect to B10. The more shifting of CA50 away from TDC and decrease in cylinder temperature can be seen as a reason for this situation.

3.3. Effects of biodiesel blend and injection timing on the emissions

THC, CO and CO_2 emissions are given in Fig. 6. When comparing with the diesel, generally B10 gave better results in terms of THC and CO while it had a bit more CO_2 emissions [4,30,37,40,43,44]. The retarded injection also showed a positive effect, except CO_2. The lowest THC values were obtained with IA26. At the medium load, a decrease of 18% in THC was seen with IA26 compared to D. As it is seen from the results presented in Fig. 3, ID periods were found slightly shorter for B10 and decreased significantly with the retarded injection timings when compared with D fuel. Shorter ID period can reduce the amount of fuel burning the premixed combustion phase. In general, THC emissions were mostly eliminated by decreasing the possibility of overmixing and over-lean regions in the combustion chamber due to shorter ID periods of B10 with IA24 and IA26. Also, the oxygen content of B10 fuel improved the oxidation of THC and CO sources at under-mixing conditions during the longer CD when increased the engine load. However, the more retarding of injection deteriorated the THC and CO emissions

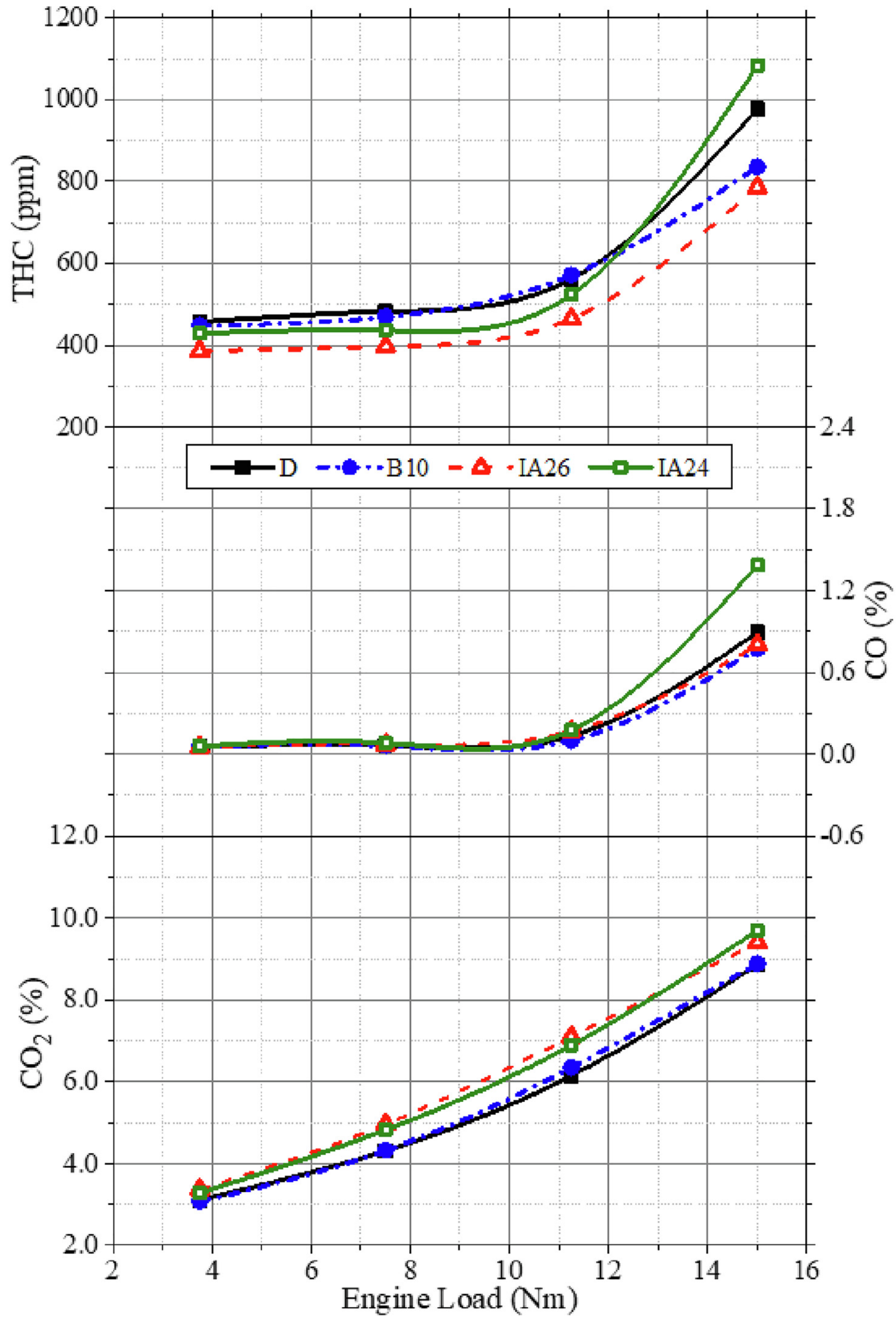


Fig. 6. The effects on CO, THC and CO_2 emissions.

at the high load up to 30% and 79%, respectively. These deteriorations for IA24 resulted from under-mixed fuel in the over delay in the end of combustion (as also seen from more retarded CA50 location) and slower thermal oxidation reactions at the high engine load.

Smoke and NO_x emissions are presented in Fig. 7. The literature results indicated that the smoke emissions can be improved with the biodiesel addition while the NO_x increase [4,30,37,40,43,44]. Lower sulphur and inherent oxygen contents of the biodiesel blend can be seen as a reason of the decrease in the smoke. In other respect, the oxygen content increased the thermal NO_x formation with high local flame temperatures by enabling the conversion reactions at especially local regions where the equivalence ratio is high. Consequently, NO_x formation rates especially during the initial and intermediate of diffusion combustion phase were provoked by the combustion of blended biodiesel fuel. This can also be seen from increasing combustion rates and longer duration for the canola biodiesel blends during mixing controlled diffusion combustion phase in HRR diagrams (Fig. 3. and Fig. 4), especially at the high and the medium loads. Retardation of injection timing can be used to reduce the NO_x without any noticeable increase in smoke emissions [50].

As discussed before, ID period have a strong function on in-cylinder gas temperatures and the SOC generally delayed in parallel with the retarded injection timing. Therefore, more fuel burns in the expansion stroke with the retarded fuel injection timing, which causes lower entire in-cylinder temperature history and increases soot generating precursors due to NO_x-smoke trade-off character-

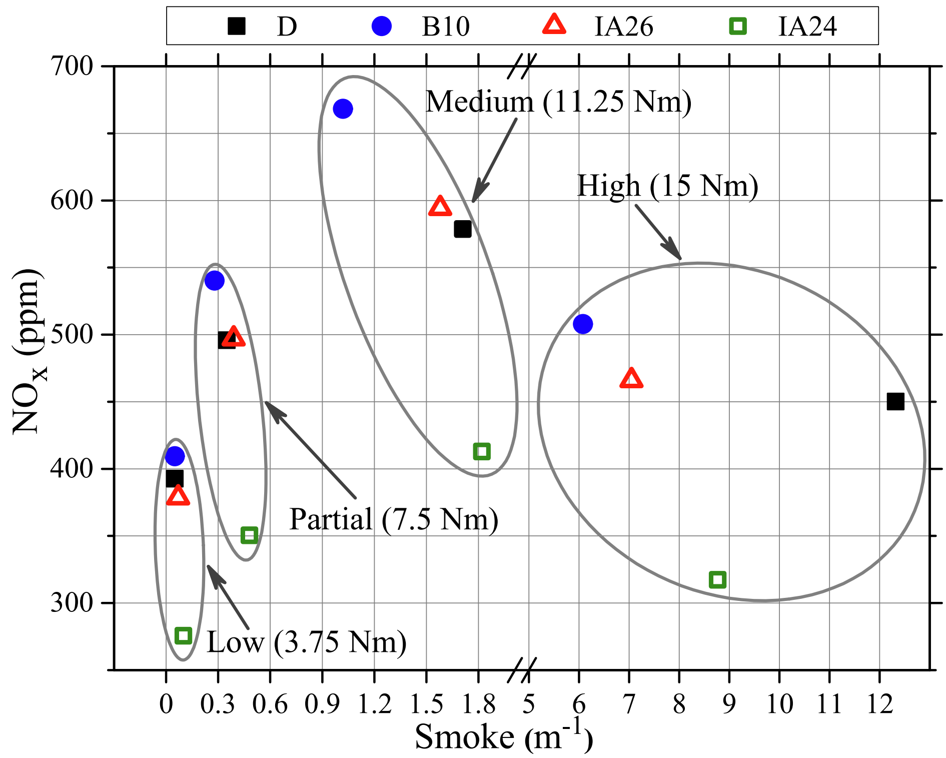


Fig. 7. The effects on smoke and NO_x emissions.

istics. IA26 showed the almost same NO_x and smoke values with the diesel fuel except the high load and much better NO_x emissions than those of B10 at all load conditions. The decrease of in-cylinder temperature history in parallel with HRR is seen as a reason for both the increase in smoke emissions up to 54% and the decrease in NO_x emissions up to 11% with IA26 in comparison with B10. However, smoke emissions with B10 fuel and retarded fuel injection timings were still lower than diesel fuel for the high load.

4. Conclusions

In this study, canola biodiesel/No. 2 diesel fuel blend was used in a compression ignition engine. Although the canola biodiesel meets to the biodiesel standards, there is an accepted opinion in the literature on the decrease in the engine performance and the increase in NO_x emissions. In this study, the retardation of injection timing was examined in order to overcome these adverse effects. The experimental results showed that:

- The maximum HRR values decreased up to 22% with the biodiesel blending at all engine loads, while the maximum cylinder pressures were close to those of the diesel fuel. Retarded injection generally decreased both the maximum HRR and the cylinder pressure.
- The starts of combustion occurred earlier for the biodiesel/diesel blend at all loads, while the starts of injection noticeably advanced, except the low load.
- Although the biodiesel addition hardly ever influenced the premixed and diffusion combustion durations, the premixed combustion fractions decreased. While the retarded injection caused some decreases in the premixed combustion fractions, the injection advance above 2 °CA generally increased the premixed combustion durations up to 2.5 °CA.
- The biodiesel/diesel blend having the lower heating value caused to high BSFC and low BTE. These adverse effects were not observed in the retardation of injection timing.
- The biodiesel/diesel blend resulted in decrease in CO, THC and smoke emissions and increase in NO_x emissions. The more reductions in THC (up to 30%) and CO (up to 79%) as well as NO_x emissions (up to 11%) were obtained with 2 °CA retarded injection. The retardation of injection timing of 4 °CA caused to deterioration of THC and CO emissions at high load, and of smoke emissions for all loads.

These results can be summarized as that the retardation of injection timing up to a certain level is a suitable approach with regards to the engine performance and emission, without a significant alteration in the combustion parameters.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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