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Article

Effects of Canola Oil Biodiesel Fuel Blends on Combustion, Performance, and Emissions Reduction in a Common Rail Diesel Engine

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Abstract: In this study, we investigated the effects of canola oil biodiesel (BD) to improve combustion and exhaust emissions in a common rail direct injection (DI) diesel engine using BD fuel blended with diesel. Experiments were conducted with BD blend amounts of 10%, 20%, and 30% on a volume basis under various engine speeds. As the BD blend ratio increased, the combustion pressure and indicated mean effective pressure (IMEP) decreased slightly at the low engine speed of 1500 rpm, while they increased at the middle engine speed of 2500 rpm. The brake specific fuel consumption (BSFC) increased at all engine speeds while the carbon monoxide (CO) and particulate matter (PM) emissions were considerably reduced. On the other hand, the nitrogen oxide (NO_x) emissions only increased slightly. When increasing the BD blend ratio at an engine speed of 2000 rpm with exhaust gas recirculation (EGR) rates of 0%, 10%, 20%, and 30%, the combustion pressure and IMEP tended to decrease. The CO and PM emissions decreased in proportion to the BD blend ratio. Also, the NO_x emissions decreased considerably as the EGR rate increased whereas the BD blend ratio only slightly influenced the NO_x emissions.

Keywords: canola oil biodiesel blends; exhaust gas recirculation; combustion characteristics; exhaust emissions; diesel engine

1. Introduction

Over the past few decades, global warming has intensified due to the drastic increase in greenhouse gases (GHG) produced by fossil fuels. Abnormal climate change resulting from global warming has caused phenomena such as floods, sudden rainstorms, intense heat, and typhoons. Climate change and the mitigation of GHG emissions has become a primary motivation for biofuels research. Furthermore, growing concerns of environmental pollution caused by the extensive use of conventional fossil fuels has led to the search for more environmentally friendly and renewable fuels. Therefore, biofuels such as alcohols and biodiesel have been proposed as alternatives for internal combustion engines [1,2]. In particular, biodiesels derived from vegetable oils have received wide attention as a replacement for diesel fuel because they emit less GHG and other pollutant emissions. Diesel engines are mostly used in industrial transportation, passenger car, and agricultural applications. Despite their disadvantages including noise and vibration, they have a high thermal efficiency, large power output, and are highly reliable. However, a diesel engine emits relatively more particulate matter (PM) and nitrogen oxide (NO_x) than a gasoline engine [3–5]. Furthermore, the regulations for PM and NO_x emissions from diesel have strengthened and their emissions are an important environmental issue [6]. Therefore, many researchers have studied ways to reduce exhaust emissions such as PM and NO_x by the use of diesel particulate filters (DPF), selective catalytic reduction (SCR) [7,8], and alternative fuels which can be used without requiring modification of the diesel engine [9–12]. Biodiesel fuels can be produced from various vegetable oils, waste cooking oils, and animal fats. The fuel properties of biodiesel may change when different feed stocks are used [13–15]. In general, if the fuel properties of biodiesel are compared to petroleum diesel fuel, biodiesel has a higher viscosity, density, and cetane number. Moreover, it has been reported that biodiesel can reduce tailpipe emissions because it has a sufficient amount of oxygen in its molecular structure [16–19]. However, biodiesel must be blended with pure diesel because the net calorific value of biodiesel is less than that of conventional diesel fuel. Ileri *et al.* [20] experimentally analyzed the effect of antioxidants on the engine performance and exhaust emissions of a diesel engine fueled with a canola oil methyl ester (COME) and diesel blend. They reported that the addition of antioxidants did not cause any negative effect on the basic fuel properties of COME blended with diesel. Ozsezen *et al*. [15] investigated the performance and combustion characteristics of a direct injection diesel engine with neat biodiesel such as palm oil and canola oil where they demonstrated that the engine performance was slightly weakened and the combustion characteristics such as carbon monoxide (CO), unburned hydrocarbon (HC), and smoke capacity changed slightly compared to pure diesel fuel.

The high oxygen content in biodiesel results in improvement of its combustion efficiency as well as reductions of PM, CO, and HC emissions but at the same time, produces higher NO_x emissions [21–25]. There are various problems associated with vegetable oils being used as fuel in diesel engines due to the high viscosity, density, iodine value, and poor volatility of the vegetable oil. Researchers have clearly proven that trans-esterification is the best way to use vegetable oil as a fuel in existing diesel engines [26,27]. The high viscosity and surface tension of biodiesel affect atomization by increasing the mean droplet size which, in turn, increases the spray tip penetration. The higher mean droplet size of biodiesel is due to the lower Weber number, which is due to the high surface tension [28]. Jindal *et al.* [29] researched the effects of the injection pressure on engine performance with regard to the brake specific fuel consumption (BSFC) and break thermal efficiency (BTE) when *Jatropha* methyl ester was used as a fuel. It was found that increasing the injection pressure increases the BTE and reduces the BSFC. Canola is a popular biodiesel feedstock because it produces more oil per unit of land area compared to other oil sources. Therefore, many studies have investigated the trans-esterification reaction of canola oil for biodiesel production and evaluated its ester in diesel engines with regard to engine performance and combustion characteristics [5,20,30]. Roy *et al.* [31] tested canola oil biodiesel (BD)-diesel blends at high idling operations. The results showed that the CO and HC emissions were significantly reduced compared to pure diesel fuel. In addition, with up to 5% BD blends, the NO_x emissions did not increase whereas NO_2 production at high idling was more than 50% of the total NO_x emissions. Lee *et al.* [28] reported a method of injection when using ultralow sulfur diesel (ULSD) and biodiesel on a common rail diesel engine. They showed that with a higher biodiesel blend ratio, a higher injection pressure is required due to the larger surface tension of biodiesel. Al-Dawody *et al.* [32] suggested various strategies such as cooling the air temperature, retarding the injection timing, optimizing the piston bowl design, exhaust gas recirculation (EGR), and increasing the nozzle diameter as methods to reduce the soybean biodiesel NO_x effect in a diesel engine. Among the various strategies, they found that the most significant reduction of NO_x emissions resulted from retarding the injection timing and increasing the nozzle diameter. On the other hand, the decreasing level of NO_x was accompanied by considerable increases of the smoke emissions and fuel consumption. From the literature review, the effects of canola oil biodiesel fuel blends on the combustion characteristics and exhaust emissions reduction in a direct injection (DI) diesel engine injected under a high pressure has not been clearly studied when using canola oil fuels blended with diesel. Therefore, these topics need to be investigated to address this deficiency in the literature. For this reason, in present study, the effects of BD fuel blends and the EGR rate on the combustion and exhaust emissions characteristics in a common rail diesel engine were experimentally investigated.

2. Experimental Materials and Methods

2.1. Test Fuels and Operating Conditions

BD was blended with pure diesel at 10%, 20%, and 30% on a volume basis. The fuels were characterized by determining their viscosity, density, pour point, distillation temperature, flash point, acid number, ester content, total free glycerin, and calculated index. In order to measure the fuel properties of pure diesel, neat BD, and the BD blends, the ASTM-D6751 and EN-14214 standard test methods were used. The fuel properties of the pure diesel, neat BD, and BD blended fuels are presented in Table 1.

Table 1. Properties of pure diesel, neat biodiesel and biodiesel (BD) blends.

Properties (units)	Pure diesel	Neat BD	BD 10	BD 20	BD 30	Test method
Density (kg/mm^3 at 15 °C)	836.8	880	842	846	850	ASTM D941
Viscosity (mm^2/s at 40 °C)	2.719	4.290	2.818	2.991	3.172	ASTM D445
Calorific value (MJ/kg)	43.96	39.49	43.29	42.71	42.12	ASTM D4809
Cetane index	55.8	61.5	-	-	-	ASTM D4737
Flash point (°C)	55	182	-	-	-	ASTM D93
Pour point (°C)	−21	−8	-	-	-	ASTM D97
Oxidation stability (h/110 °C)	25	15	-	-	-	EN 14112
Ester content (%)	-	98.9	-	-	-	EN 14103
Oxygen (%)	0	10.8	-	-	-	-

In this study, BD was used along with BD blends and ULSD, which had a sulfur content of 0.005%, for comparison. In order to investigate the characteristics of combustion and exhaust emissions depending on the BD blend ratio, tests were carried out at the warmed up condition of the engine under four different engine speeds: 1000, 1500, 2000, and 2500 rpm. In addition, the coolant temperature was held constant at 70 ± 3 °C, the intake air temperature was 20 ± 3 °C, and a constant load of 30-N·m torque from the engine dynamometer was applied to the test engine at each speed in order to ensure consistent test conditions. The experimental and operating conditions are summarized in Table 2.

Table 2. Experiment and operating conditions.

Test parameters	Unit	Operating condition
Engine speed	rpm	1000, 1500, 2000, 2500
Torque	N.m	30
Test fuels	-	BD blended rate with diesel (vol. %)
0	-	Diesel 100% + BD 0%
BD 10	-	Diesel 90% + BD 10%
BD 20	-	Diesel 80% + BD 20%
BD 30	-	Diesel 70% + BD 30%
Cooling water temp.	°C	70 ± 3
Intake air temp.	°C	20 ± 3
Gas recirculation (EGR) rate	%	0, 10, 20, 30
Injection pressure	MPa/rpm	30/1000, 37/1500, 45/2000, 60/2500

2.2. Test Engine and Experimental Procedure

In this study, the experimental apparatus consisted of the components shown in Figure 1. This apparatus was used to investigate the combustion and exhaust emission characteristics as functions of the BD blend ratio and EGR rate in a 4-cylinder common rail diesel engine. The experimental equipment consisted of a 4-cylinder electronic common rail diesel engine equipped with a turbocharger, a fuel consumption rate tester with a fuel pump driven by an electrical voltage of 220 V, a control unit connected to an electronic control unit (ECU) to control the injection timing, and an eddy current type EC dynamometer (DY-230kW, Hwanwoong, Korea) to control the engine speed. A piezoelectric pressure sensor (6056a, Kistler, Swiss) was mounted onto the position of the glow plug to measure the combustion pressure. Data was then acquired using a data acquisition board (PCI6040E, National Instrument, Austin, TX, USA). The combustion pressure in the cylinder was analyzed using a combustion analyzer. The main specifications of the 4-cylinder common rail diesel engine used in this study are summarized in Table 3. The EGR rate (%) is defined as the difference between the quantity of fresh air induced without EGR (Q_0) and that of air with EGR (Q_{EGR}) divided by the quantity of fresh air induced without EGR (Q_0), as shown below:

$$EGR(\%) = \frac{Q_0 - Q_{\mathrm{EGR}}}{Q_0} \times 100 \quad (1)$$

Figure 1. Schematic diagram of the experimental apparatus. HC: hydrocarbon; ECU: electronic control unit; ESR: gas recirculation; CO: carbon monoxide.

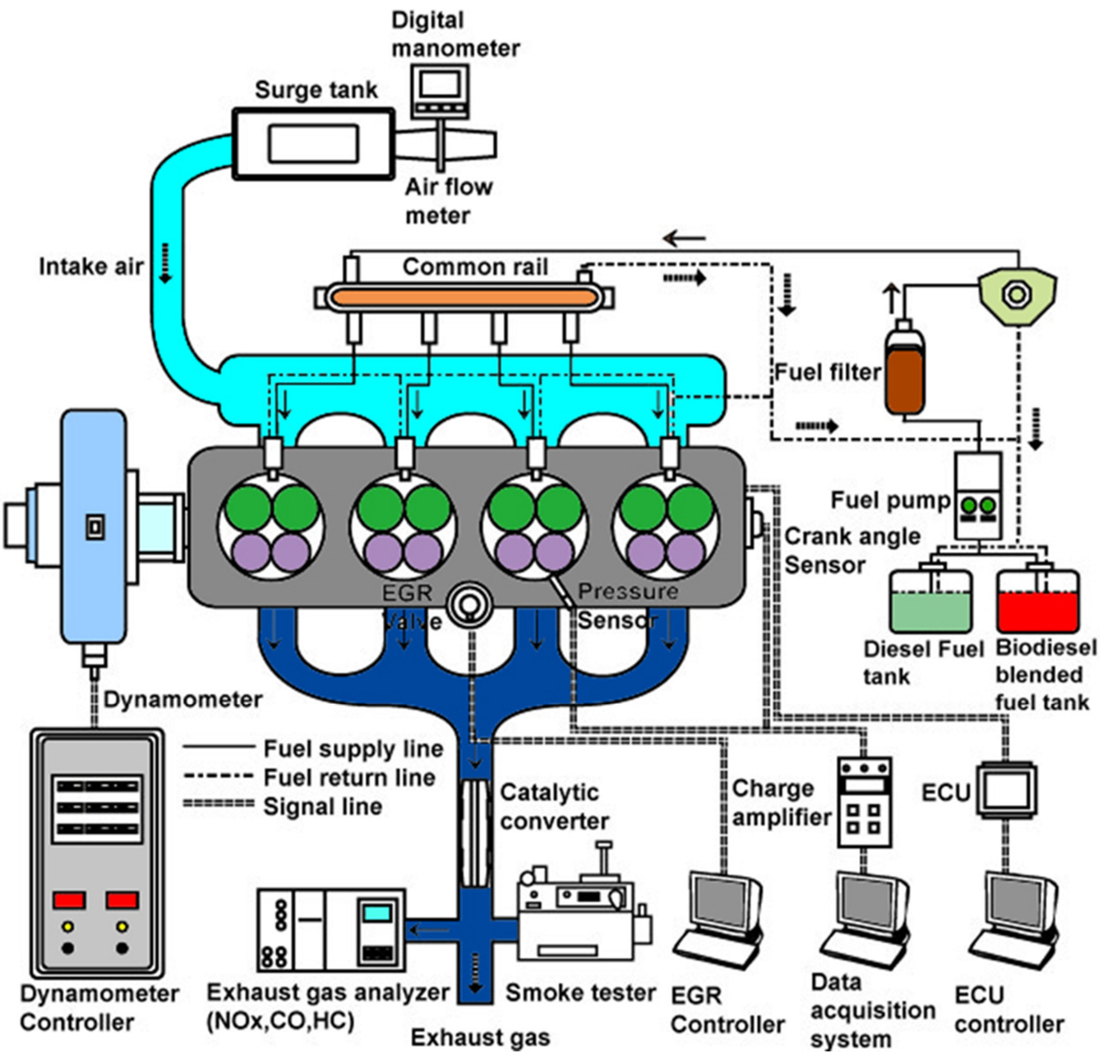


Table 3. Specifications of the test engine.

Test model	Parameter (units)	Specification
Engine type	Engine type	4-cylinder
	Bore (mm)	81
	Stroke (mm)	96
	Displacement (cm^3)	1979
	Combustion type	Direct injection
	Injection procedure	1-3-4-2
	Compression ratio	17.7:1
	Maximum power (kW/rpm)	82/at 4000
	Maximum torque (N·m/rpm)	260/at 2000
	Maximum engine speed (rpm)	4500
Fuel injection system	Fuel control	ECU control
	Maximum fuel pressure (MPa)	145
	Number of injector nozzle holes	5
	Injector spray angle (degree)	150
	Injector hole diameter (mm)	0.17

The exhaust gas was delivered to the intake manifold through a water-cooled unit by an EGR valve and the gas flow rate was regulated by controlling the EGR duty ratio using a computer. The NO_x emissions were monitored in real time using an exhaust analyzer. Exhaust measuring equipment was used for the exhaust component analysis. A multi-gas analyzer (MK2, Eurotron, Italy) was used to measure the O_2, CO, NO, NO_2, and HC contents of the exhaust gases. In order to detect PM, an opacity smoke meter (OPA-102, Qurotech, Korea) was also utilized using the partial flow sampling method. The gas analyzer specifications along with the resolution, range, and accuracy are summarized in Table 4. In this work, the heat release rate (HRR) of BD combustion in the engine was calculated using the following formula [33]:

$$\frac{\mathrm{d}Q}{\mathrm{d}\theta} = \frac{k}{k-1} P \frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{1}{k-1} V \frac{\mathrm{d}P}{\mathrm{d}\theta} \tag{2}$$

Here, $\mathrm{d}Q/\mathrm{d}\theta$ is the HRR, k is the specific heat ratio which was assumed to be 1.35, $\mathrm{d}P/\mathrm{d}\theta$ is the rate of change of the pressure, and $\mathrm{d}V/\mathrm{d}\theta$ is the rate of change of the cylinder volume. BSFC is defined as the ratio of fuel consumption rate to the brake power of the engine. The value was calculated based on the fuel consumption, engine torque, and speed data using the following formula:

$$b_\mathrm{f} = \frac{\dot{m}_\mathrm{f}}{2\pi N T_\mathrm{e}} \tag{3}$$

Here, b_f is the brake specific fuel consumption rate, $\dot{m}_\mathrm{f}$ is the fuel consumption flow rate into the cylinder, N is the engine speed, and T_e is the brake torque, which was directly measured using an engine dynamometer. The brake specific energy consumption (BSEC) is also known as the ratio of the energy consumption rate to the brake power of the engine, which is calculated from the fuel consumption and low heating calorific value using the following formula:

$$b_\mathrm{e} = \frac{Q_\mathrm{LHV} B_\mathrm{f}}{2\pi N T_\mathrm{e}} \tag{4}$$

Here, b_e is the brake specific energy consumption rate, B_f is the fuel consumption mass per hour, and Q_{LHV} is the low heating calorific value, which is directly measured by the engine dynamometer.

Table 4. Specifications of the exhaust gas analyzer. PM: particulate matter.

Method of detection	Species	Unit	Range	Resolution	Accuracy
Electrochemical	O_2	%	0%–30%	0.1%	±0.57%
Electrochemical	CO	ppm	0–4000 ppm	1 ppm	±0.62%
Pellistor	HC	%	0%–5%	0.01%	±0.8%
Electrochemical	NO	ppm	0–5000 ppm	1 ppm	±0.25%
Electrochemical	NO_2	ppm	0–1000 ppm	1 ppm	±0.25%
Smoke opacity	PM	%	0%–100%	0.1%	±1%

3. Results and Discussion

3.1. Effects of Biodiesel on Combustion and Emissions

3.1.1. Combustion and Engine Performance

Figure 2 shows the combustion pressure and HRR as functions of the engine speed at the various BD blend ratios in the diesel engine without EGR. As shown in Figure 2a, at the low engine speed of 1500 rpm, the combustion pressure decreased slightly by 5.9% with BD 10, 4.6% with BD 20, and 6.9% with BD 30 compared to that of BD 0. On the other hand, at the middle engine speed of 2500 rpm, the combustion pressure increased to 3.3% with BD 10, 2.8% with BD 20, and 6.6% with BD 30. These results demonstrate that the combustion pressure with BD 0 is slightly higher at the low engine speed of 1500 rpm. As Table 1 shows, the kinetic viscosity of BD is 50.4% higher than that of ULSD. For these reasons, in the case of BD 0, it is estimated that at low engine speeds in the range of 1500 rpm, this higher viscosity disturbs the atomization of fuel during injection. As shown in Figure 2b, at the middle engine speed range, as the BD blend ratio increases, the combustion pressure increases. This is due to the improvement of conditions caused by the higher fuel injection pressure at 2500 rpm including the spray droplet size, fuel injection speed, and catalyzed combustion activation. In addition, Figure 2 shows that the HRR is greater than that of ULSD. At an engine speed of 1500 rpm, as the BD blend ratio increases, the HRR increases by 9.5% with BD 10, 11.4% with BD 20, and 10.8% with BD 30 compared to the value obtained with BD 0. At an engine speed of 2500 rpm, the HRR increased considerably up to 22.5% with BD 10, 21.5% with BD 20, and 24.8% with BD 30 compared to the value obtained for BD 0. For this reason, it is seen that the HRR of BD is higher than that of ULSD because BD holds sufficient oxygen which catalyzes combustion activation.

Figure 3 shows the peak combustion pressure and indicated mean effective pressure (IMEP) as functions of the engine speed without EGR when increasing the BD blend ratio. As shown in Figure 3, the peak combustion pressure shows slight tendencies to decrease until the engine speed reached 2000 rpm with increasing BD blend ratio, however, IMEP presents an increasing trend as biodiesel percentage in the blend is increased under all different engine speed values. On the other hand, at 2500 rpm, the peak combustion pressure increased by 3.3% with BD 10, 2.8% with BD 20, and 6.6% with BD 30, compared to BD 0. For this reason, the combustion pressure increased at 2500 rpm with increasing BD blend ratio, which is caused by the accelerated combustion of fuel which is atomized by a high injection pressure and the oxygen contained in the BD.

Figure 2. Effects of the biodiesel blend ratio on the combustion characteristics at (**a**) 1500 and (**b**) 2500 rpm.

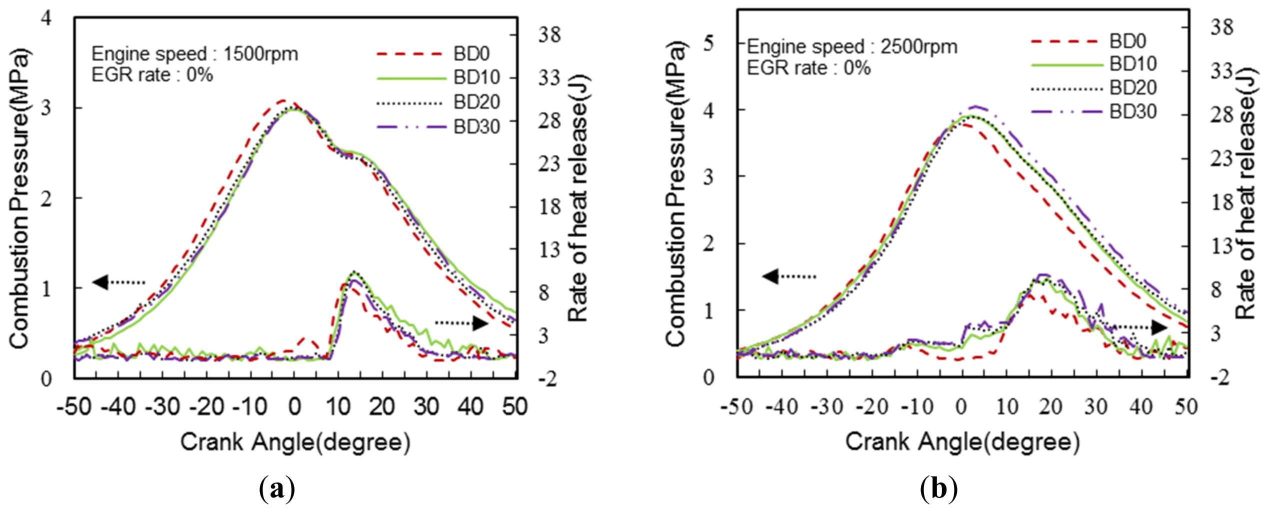


Figure 3. Effects of the engine speed and biodiesel blend ratio on the (**a**) peak combustion pressure and (**b**) indicated mean effective pressure (IMEP).

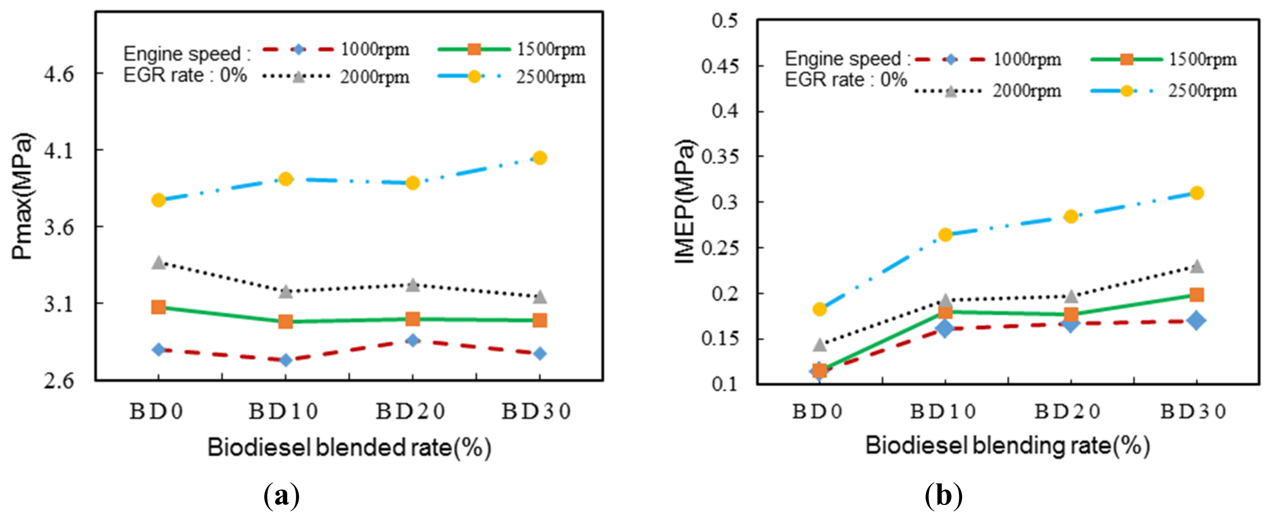
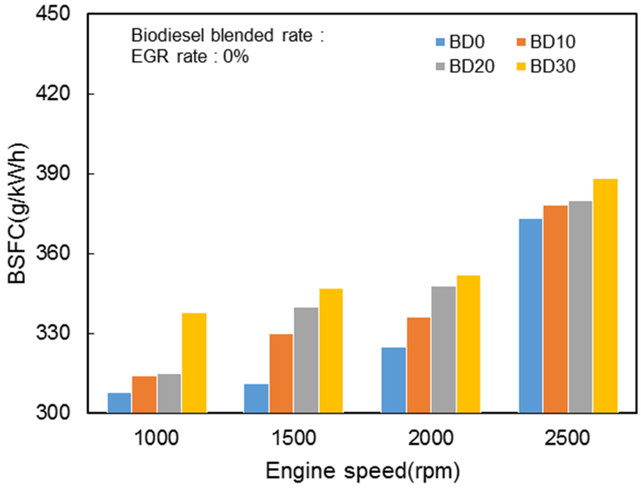


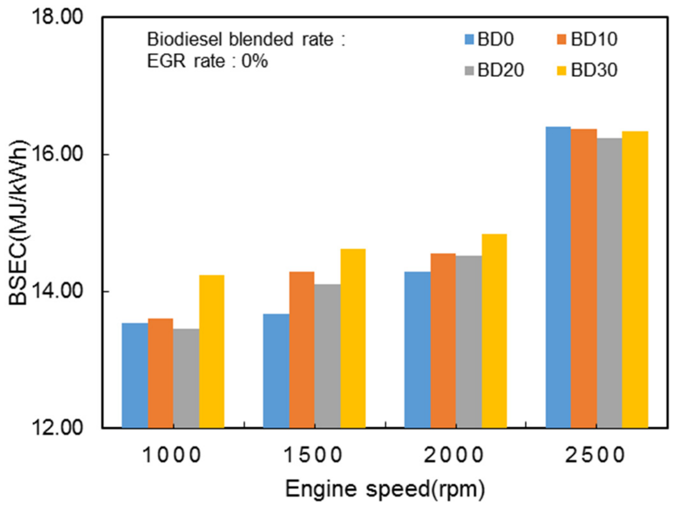
Figure 4 shows the variation of the BSFC of the engine with ULSD and BD blended fuels at engine speeds of 1000, 1500, 2000, and 2500 rpm. With increasing BD blend ratio, the BSFCs of the BD blended fuels compared to ULSD increased at all speeds with the maximum value obtained with BD 30 at 2500 rpm. The higher BSFC of BD 30 indicates that more fuel was consumed to produce the same amount of power. This is expected because of the relatively low calorific value of BD 30 compared to ULSD. However, the BD 10 and BD 20 blends showed less of a BSFC increase (1.9% and 2.2% BSFC increases, respectively) at 1000 rpm. This indicates that they have higher fuel consumption efficiencies than those of the different BD blended fuels at all speeds.

Figure 4. Effects of the biodiesel blend ratio and engine speed on the brake specific fuel consumption (BSFC).



Another important performance parameter is the BSEC. This parameter is often used to compare the performance of fuels with different calorific values. BSEC is defined as the product of the BSFC and heating calorific value of a fuel and corresponds to the amount of energy consumed to develop a unit of output power in one hour. Generally, the value of BSEC decreases with increasing energy consumption. Figure 5 shows the BSEC values of the engine fueled with the various canola oil biodiesel blends compared to BD 0 (pure diesel) at various engine speeds. From the results, it can be seen that engine fueled with BD 0 consistently resulted in the lowest BSEC. The biodiesel fuel blend demonstrates a higher BSEC, which may be related to the lower brake thermal efficiency and some dissimilarities in the combustion processes. However, it did not increase further as the degree of the COME blending increased. This is due to the fact that at a higher degree of COME, the initiation of combustion at an earlier period and efficient combustion due to the increased oxygen content in fuel spray yields a lower BSEC. The highest increase of the BSEC was approximately 6.9% for BD 30 compared to BD 0 at an engine speed of 1500 rpm.

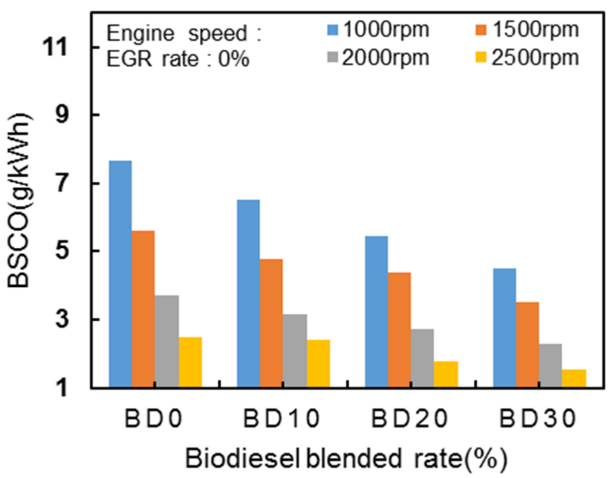
Figure 5. Effects of the biodiesel blend ratio and engine speed on the brake specific energy consumption (BSEC).



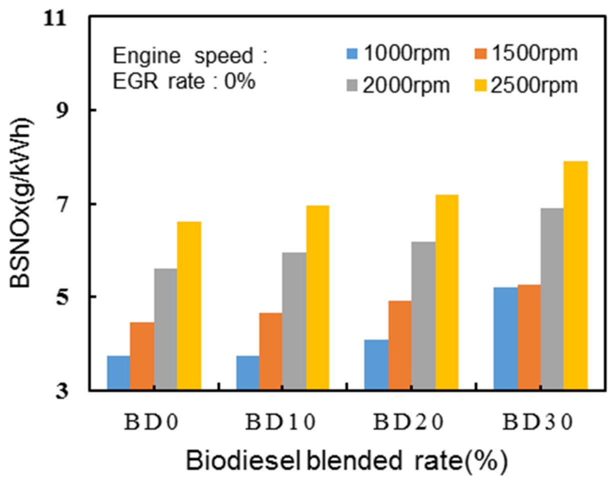
3.1.2. Exhaust Emissions Characteristics

Figure 6 shows the CO, NO_x, and PM emissions depending on the engine speed and the blend ratio of BD without EGR. As shown in Figure 6a, the CO emissions decreased considerably by 34.8% with BD 10, 68.5% with BD 20, and 82.4% with BD 30 at an engine speed of 1000 rpm compared to the CO emissions obtained for BD 0. At 1500 rpm, the CO emissions were reduced by 26.5% with BD 10, 58.1% with BD 20, and 66.4% with BD 30. In the case of an engine speed of 2000 rpm, the CO emissions were reduced by 18.5% with BD 10, 50% with BD 20, and 83.2% with BD 30 compared to the CO emissions obtained for BD 0. At an engine speed of 2500 rpm, the CO emissions decreased by 23.4% with BD 10, 52.6% with BD 20, and 83.2% with BD 30. As the comparative analysis of the experimental results indicates, increases of the BD blend ratio and engine speed resulted in decreased emissions. CO is a colorless toxic gas that is exhausted when the fuel burns without a sufficient oxygen supply. The increase of the BD blend ratio results in an increase of the oxygen content in the fuel itself and the increasing engine speed induces an increase of the fuel injection pressure.

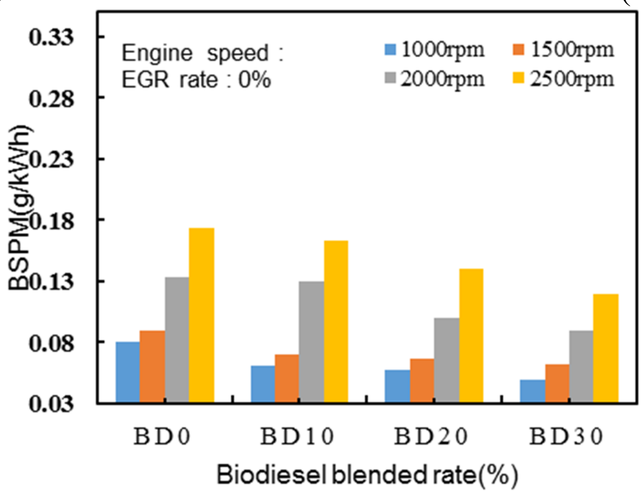
Figure 6. Effects of the engine speed and biodiesel blend ratio on the (**a**) brake specific carbon monoxide (BSCO); (**b**) brake specific nitrogen oxide ($BSNO_x$); and (**c**) brake specific particulate matter (BSPM).



(**a**)



(**b**)



(**c**)

For these reasons, fuel is atomized, leading to strong combustion activation. Figure 6b shows the NO_x emission results. The NO_x emissions increased under high temperature and high pressure conditions in the combustion chamber. As seen in the Figure 6, the NO_x emissions increased as the BD blend ratio and engine speed increased as described similarly in the article of Ozsezen *et al*. [15]. The increase of the NO_x emission rate is similar at engine speeds of 1000 and 1500 rpm, namely the low speed range, while the NO_x emissions significantly increased up to 14.8% at 2000 rpm and 22.8% at 2500 rpm compared to 1000 rpm with BD 20. In addition, as the BD blend ratio was increased, the NO_x emissions increased more than that of ULSD. It is expected that the high oxygen content in BD results in the increased combustion temperature due to improvement of the burning efficiency. The PM emission results are summarized in Figure 6c.

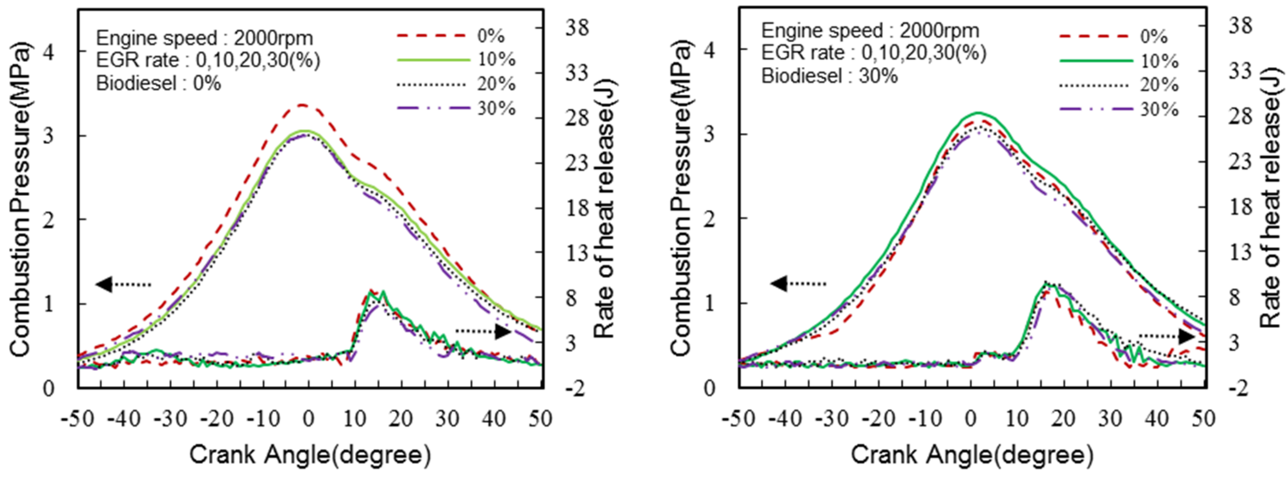
The PM emissions decreased with increasing BD blend ratio and increased with increasing engine speed. PM is produced by unburned carbon particles when using a densely blended fuel. It is observed that the PM emissions depend on the degree of combustion diffusion related to both the diffusion of fuel vapor separated from fuel drops and the flame diffusion in a dense fuel zone. Accordingly, it can be concluded that exhaust emissions using BD were evaluated to similar results as shown in other experiments [4,15,34].

3.2. Effect of Biodiesel on the Combustion and Emissions with Exhaust Gas Recirculation

3.2.1. Combustion and Engine Performance

Figure 7 shows the combustion pressure and HRR obtained at an engine speed of 2000 rpm with EGR at the various BD blend ratios. The combustion pressure decreased as the EGR rate and BD blend ratio increased at 2000 rpm. With BD 0, the combustion pressure decreased by 10.1% at a 10% EGR rate, 11.9% at a 20% EGR rate, and 12.3% at a 30% EGR rate. At BD 30, the combustion pressure increased by 3% with a 10% EGR rate, decreased 2.2% with a 20% EGR rate, and decreased 4.3% with a 30% EGR rate compared to the value obtained with BD 0. When comparing the HRR at BD 0 with that at BD 30, as the EGR rate increased, the HRR at BD 0 increased more than that at BD 30. In the case of BD 30, the dissolved oxygen of the BD leads to a small decrease of the ratio. At a 10% EGR rate with BD 30, the combustion pressure increased 3% more than that at the 0% EGR rate because the combustion activation is immediately catalyzed by the dissolved oxygen in the BD itself. As shown in Figure 7, with BD 0, the HRR decreased by 0.9% with a 10% EGR rate, 13.6% with a 20% EGR rate, and 18.6% with a 30% EGR rate. This behavior appears to be influenced by the poor combustion conditions resulting from the intake of inert gases into the cylinders. With BD 30, even though the EGR rate increases, the HRR increases more than that of ULSD because of the high cetane number and high oxygen content.

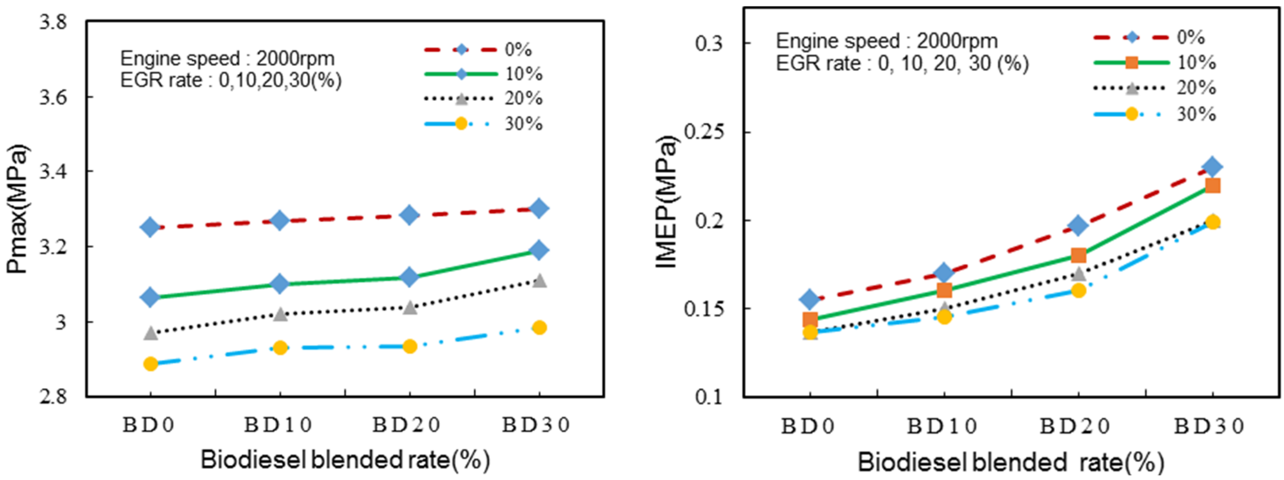
Figure 7. Effects of EGR at biodiesel blend ratios of (**a**) 0% and (**b**) 30%.



(**a**) (**b**)

Figure 8 shows graphs of the peak combustion pressure and IMEP at an engine speed of 2000 rpm with EGR for the various BD blend ratios. Figure 8a shows that the peak combustion pressure decreases with increasing EGR rate and increases slightly from BD 0 to BD 20 with increasing BD blend ratio. At BD 30, the peak combustion pressure increases 1.6% with a 0% EGR rate, 3.9% with a 10% EGR rate, 4.5% with a 20% EGR rate, and 3.3% with a 30% EGR rate compared to the value obtained for BD 0. In Figure 8b, it is seen that the IMEP decreases with increasing EGR rate and increases with increasing BD blend ratio. As shown in Figure 8b, when increasing EGR rate at 2000 rpm, the peak combustion pressure increased slowly with increasing BD blend ratio. At the same time, the IMEP increases remarkably. This occurs because the dissolved oxygen of BD has a strong influence on combustion.

Figure 8. Effects of EGR and the biodiesel blend ratio on the (**a**) peak combustion pressure and (**b**) IMEP.



(**a**) (**b**)

Figure 9 presents the BSFC of the BD blended fuels obtained at the various EGR rates at 2000 rpm. From the results, it is observed that the baseline results without EGR showed the lowest BSFC in the BD blended fuels with EGR. The BSFC increased in proportion to the EGR rate. BD 20 showed the lowest BSFCs, which means that BD 20 has the highest fuel conversion efficiency among the different BD blended fuels. These results are in agreement with the results obtained by Cenk *et al.* [30].

Figure 9. Effects of EGR and the biodiesel blend ratio on the BSFC.

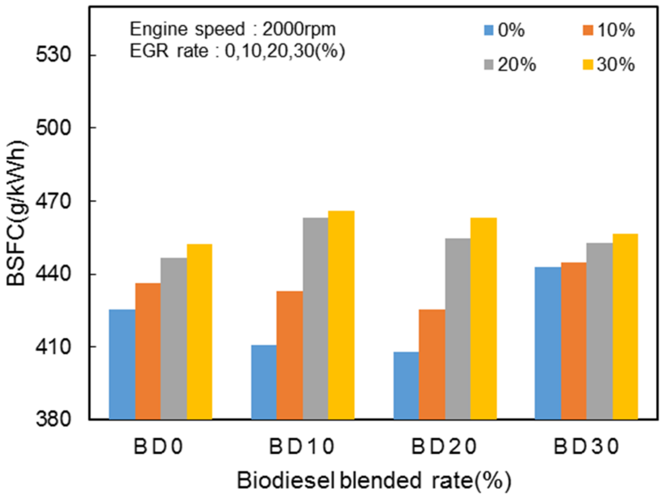
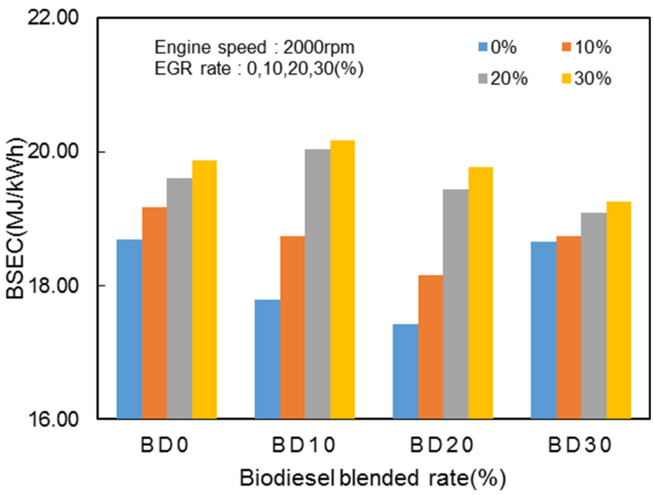


Figure 10 shows the variation of the BSEC of the engine at the different EGR rates and various BD fuel blends at an engine speed of 2000 rpm. From the results, it is clear that as the EGR rate increases, the BSEC gradually increases for all fuel blends. In addition, it can be seen that engine fueled without EGR consistently resulted in the lowest BSEC. The biodiesel fuel blend results in a higher BSEC, which is related to the lower brake thermal efficiency and dissimilarities in the combustion processes. The lowest BSEC was approximately 18.15 MJ/kW·h for BD 20 at a 10% EGR rate. However, for BD 30, the BSEC decreased significantly compared to the different blend fuels. This is attributed to the increase of the oxygen content in the BD blended fuels. It is estimated that the combustion of higher BD blend fuels is better than pure diesel fuel combustion.

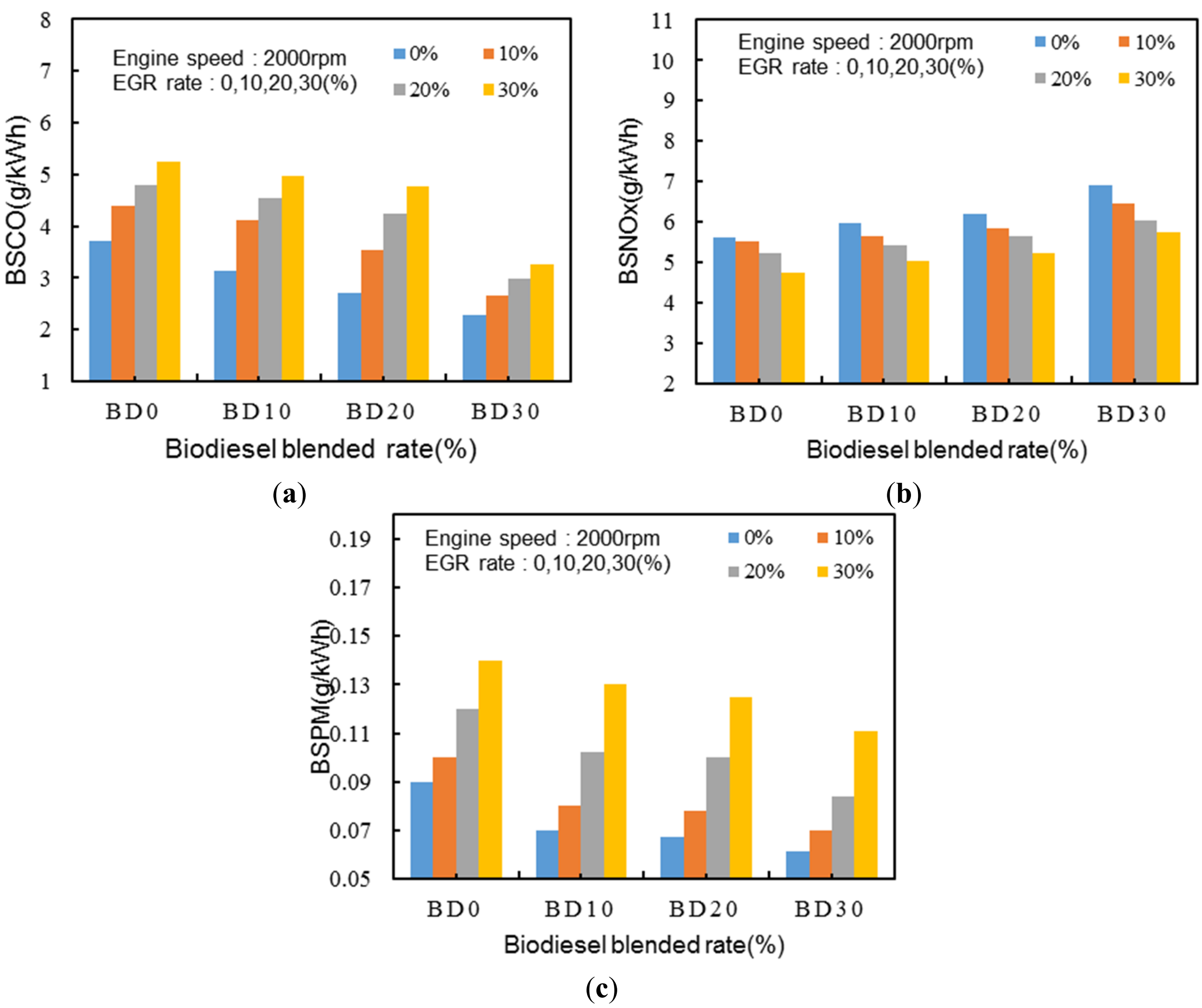
Figure 10. Effects of EGR and the biodiesel blend ratio on the BSEC.



3.2.2. Exhaust Emissions Characteristics

The emissions characteristics with EGR at an engine speed of 2000 rpm are shown in Figure 11 as a function of the BD blend ratio. As shown in Figure 11a, the CO emissions increase with increasing EGR rate and decrease with increasing BD blend ratio. The CO emissions at an EGR rate of 0% were reduced by 23.4% with BD 10, 40.3% with BD 20, and 64.2% with BD 30. At a 10% EGR rate, the CO emissions decreased by 15.9% with BD 10, 28.4% with BD 20, and 34.8% with BD 30. At a 20% EGR rate, the CO emissions were reduced by 11.3% with BD 10, 25.5% with BD 20, and 44.8% with BD 30 compared to the values obtained with BD 0. At an EGR rate of 30%, the CO emissions decreased by 13.1% with BD 10, 24.9% with BD 20, and 45.9% with BD 30. In these results, it is also obvious that the exhaust gas emissions increase because the exhaust gases recirculated into the cylinders are inert. However, at the same EGR rate, the exhaust gas emissions decreased further with increasing BD blend ratio, which holds sufficient oxygen. Figure 11b shows the NO_x emissions depending on the BD blend ratio at 2000 rpm as a function of the EGR rate. As shown in the figure, the NO_x emissions tended to increase slightly with increasing BD blend ratio and decreased considerably as the EGR rate was increased.

Figure 11. Effects of EGR and the biodiesel blend ratio on the (**a**) BSCO; (**b**) $BSNO_x$; and (**c**) BSPM.



Under ideal combustion conditions, namely under a high temperature and high pressure, the NO_x emissions increase. However, recirculated gases, which are burned gases and hold a remarkably low oxygen content in the cylinders, result in poor combustion, which results in reduced NO_x emissions with increasing EGR rate. Figure 11c presents the PM emissions depending on the BD blend ratio obtained at an engine speed of 2000 rpm as the EGR rate was varied. In the graph, the PM emissions decreased with increasing BD blend ratio and increased with increasing EGR rate. As the EGR rate increased, more carbon particles are formed due to incomplete combustion.

4. Conclusions

Experiments were conducted to investigate the effects of the BD blend ratio and EGR on the characteristics of combustion and exhaust emissions in a common rail diesel engine. The following conclusions can be drawn from this study:

- When increasing the BD blend ratio at engine speeds of 1000, 1500, 2000, and 2500 rpm without EGR, the combustion pressure and IMEP slightly decreased in the low engine speed range below 2000 rpm while it showed a tendency to increase in the middle engine speed range of 2500 rpm.
- As the BD blend ratio increased, compared to ULSD, the BSFC and BSEC increased at all speeds both with and without EGR.
- On the other hand, the CO and PM emissions decreased considerably as the BD blend ratio was increased at every engine speed. In particular, the CO emission decreased by up to 34.8% with BD 10 at 1000 rpm, up to 58.1% with BD 20 at 2000 rpm, up to 83.2% with BD 30 at 2500 rpm, as compared with BD 0. The PM emissions decreased by about 33% with BD 30 compared to BD 0 at all engine speeds.
- The PM emissions decreased by about 33% on average with BD 30 compared to BD 0 at all engine speeds of 1000, 1500, 2000, and 2500 rpm without EGR.
- However, the NO_x emissions increased by up to 14.8% at 2000 rpm and 22.8% at 2500 rpm compared to 1000 rpm for BD 20 without EGR.
- While increasing the EGR rate at 10% intervals from 0% to 30% at an engine speed of 2000 rpm, it was found that the combustion pressure decreased as the BD blend ratio increased. The NO_x emissions decreased by 4%–5% on average compared to those obtained at a 0% EGR rate. They also decreased considerably with increasing EGR rate.
- In the case of using the BD 20 blended fuel with EGR in the middle speed range of the engine, the exhaust emissions were considerably reduced without negatively impacting the engine performance.

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Author Contributions

All authors contributed equally to this work. All authors designed the experimental apparatus, discussed the results and implications, and commented on the manuscript at all stages. Sam Ki Yoon performed the engine performance experiments. Han Joo Kim measured the exhaust emissions and Min Soo Kim led the development of the paper. Nag Jung Choi performed the results analysis and discussion.

Conflicts of Interest

The authors declare no conflict of interest.

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