Chapter

Combustion and Emissions of Gasoline Compression Ignition Engine Fuelled with Gasoline-Biodiesel Blends

Yanuandri Putrasari and Ocktaeck Lim

Abstract

A gasoline compression ignition (GCI) engine was proposed to be the next generation internal combustion engine for gasoline. The effect of exhaust gas recirculation (EGR) and intake boosting on combustion and emissions of GCI engine fueled with gasoline-biodiesel blends by partially premixed compression ignition (PPCI) combustions are investigated in this study. Tests were conducted on a single-cylinder direct-injection CI engine, with 5% by volume proportion of biodiesel in gasoline fuel blends. Engine control parameters (EGR rate, intake boosting rate, and various injection strategies) were adjusted to investigate their influences on combustion and emissions of this GCI engine. It is found that changes in EGR rate, intake boosting pressure and injection strategies affect on ignition delay, maximum pressure rise rate and thermal efficiency which is closely tied to HC, CO, NO_x and smoke emissions, respectively.

Keywords: gasoline, fuel, GCI, thermal efficiency, emission

1. Introduction

The stricter limitations of vehicle emission regulation especially for compression ignition (CI) engines with petroleum diesel fuel motivates many researchers to explore the utilization of low volatile and alternative fuels for CI engines to obtain high efficiency, but produce lower emissions or so-called low temperature combustion (LTC). Due to the low volatility and short ignition delay of diesel fuel, CI engines produce high nitrogen oxides (NO_x) and soot emissions. To obtain LTC combustion, which is increase engine efficiency, and improve exhaust emissions, a variety of combustion methods in CI engines have been investigated such as homogeneous charge compression ignition (HCCI), PPCI, multiple premixed compression ignition (MPCI), etc. HCCI engine is one method that potentially to achieve an advanced LTC, which possible to produce low particulate matter (PM) and NOx emissions to replace conventional diesel engine combustion [1]. However, many technical problems of HCCI engine strategy must be solved before released to the market. The maximum load limitation due to the surplus of pressure rise rate (PRR) and engine knocking phenomenon [2, 3] have been major obstacle as long as HCCI combustion influenced by fuel type and air fuel mixture quality.

Recently, several studies have shown that gasoline and some other fuels with low cetane number and higher volatility are potentially advantages for low temperature combustion in CI engine, which is popular as gasoline compression ignition engine (GCI) [4–19]. At the beginning, GCI was proposed to exploit the benefit of high volatility and long ignition delay of gasoline fuel and high ratio of CI engine to obtain both high engine efficiency and low exhaust emission [20]. Later on, the concept of GCI was improved with PPCI concept by injecting fuel in the compression stroke, then mixture stratification is formed before combustion event [21]. The PPCI concept may extend load range at the same time maintaining high thermal efficiency, low NOx and soot emissions. Even though, the pressure rise rate (PRR) is remaining too high at the high load operation. The most advanced injection strategy on GCI is MPCI which has purpose to control combustion noise by manage the injection and combustion process in an order as spray – combustion – spray – combustion [22, 23]. By using MPCI mode, the acceptable pressure rise rate can be achieved and extended load range can be obtained.

There are many challenges in the implementation of GCI mode in CI engines. Several techniques are used to realize the utilization of gasoline fuel in CI engines including PPCI, MPCI and other various parameters. The start of injection (SOI) timing has effects on the balance of GCI combustion based on the study by Kodavasal et al., [24]. The study indicated that, the perfect combustion of a GCI engine was earned with start of injection (SOI) timing at -30° after top dead center (ATDC) and misfired was happened at SOI advanced than -42° ATDC. More advance SOI has potentially to solve the exhaust gas deterioration by minimizing NO_x and keeps the high performance of LTC. However, more knocking, ringing, HC and CO will occur due to improper combustion. The multiple-injection strategy has been known as the solution for the NOx/soot and decrease of combustion noise with maintaining low fuel consumption in CI engine. The application of this method with higher volatility fuel which are suitable for well-premixed or properly stratified mixture prior to the ignition to reduce both NO_x and soot emissions while maintaining high efficiency compared to conventional diesel CI engines was conducted previously [21, 25]. Meanwhile, the double injection strategy for pure gasoline fuel in GCI engines successfully reduced MPRR and NOx levels in half of the single-injection [17]. However, the IMEP and fuel economy were decreased. Furthermore, the significant increasing of the CO and soot emissions were also happened, even though the levels less than conventional diesel CI engine.

Blending gasoline fuel with certain percent of biodiesel in CI engines is the one way to obtain the good combustion and emissions results. Biodiesel is proven to be appropriated as a substitution fuel for CI engines [11, 26]. Furthermore, biodiesel has evidently decreasing engine exhaust pollutant [27, 28], due to the high content of oxygen which important to minimize the soot development in combustion process [29]. The implementation of gasoline-biodiesel blends in GCI engine with single and double injection modes has been studied previously [30–32]. The effect of biodiesel-gasoline blends on GCI combustion using 5% and 10% biodiesel was studied by Adams [30], which focused on reducing required intake temperature and utilization of split injection. To overcome the auto ignition difficulty of gasoline fuel without modification on intake temperature in GCI engine, the authors using a high compression ratio around 19.5 and various SOI of single injection mode in the previous study [31].

Previous studies have presented detailed analysis and discussion of the combustion and emission characteristics of GCI for PPCI or MPCI modes, by fueled with gasoline-diesel blends or gasoline biodiesel blends using direct injection GCI concept [4, 7–10, 12, 30–34]. However, the combustion and emission characteristics of CI engines are also influenced by various other factors, such as fuel injection strategy, initial conditions and its combustion modes. Furthermore, based on previous study [30–32], the maintaining of high efficiency and emission reduction of GCI engine fueled with gasoline-biodiesel blends still challenging and need more to be optimized especially for its NOx and soot emissions. Since the auto-ignition sensitivity of gasoline fuel is influenced by several factors such as in-cylinder equivalence ratio, intake dilution, intake temperature and pressure, it is potentially to utilize EGR and boosting in GCI combustion. To overcome the load operation limit and adjusting heat release subject to engine speed by delaying the combustion phasing, EGR was used [35]. The high-load operation can be achieved without knocking by using EGR, in which increases of specific heats capacity and minimize of oxygen (O_2) concentration in the chamber promotes longer ignition delay and shifted combustion phasing far away from after top dead centre [36–39]. But, surplus of EGR supply leads to the decreasing of power and resulted more CO and HC emissions. Thus, boosting was utilized to increase the operating load simultaneously encourage fuel ignition reactivity. Furthermore, adjusting the CA50 is also necessary when boosting is operated. Because, the more intake charges mass leads to the high intensity of knocking due to the higher of pressure rise [40–42]. Suitable method of EGR and boosting was proven potentially to extend CI engine 12]. Thus, it is also potential to be used in GCI engines. The basic and control mechanism of EGR with boosting on GCI gasoline-biodiesel auto-ignition should be able to explain the relatively wide ranges of operating parameters. Thus, complementary experimental works are conducted to achieve a better understanding on the combustion process and emission characteristics of GCI engine fueled with gasoline-biodiesel blends. Information about the effects of EGR and boosting on GCI engines using gasoline-biodiesel blends are essential for advancing the theory and contributions to successfully implement gasoline in CI engines and biofuel into the transportation sector.

The objective of this study was to determine the effects of EGR and boosting on the combustion and emissions of a GCI engine fueled with gasoline-biodiesel blends. To obtain a clear and comprehensive analysis of the effect of various EGR and boosting rates on combustion and emissions of GCI engine the same basic energy input of injected fuels was used for comparing the various parameters. The PPCI combustion modes for the gasoline-biodiesel blend were utilized. Modification of several initial conditions, such as intake, oil, and coolant temperatures are also conducted. The combustion characteristics of cylinder pressure, heat release rate, ignition delay, and emission characteristics are analyzed accordingly as the focus of this study.

2. Method

2.1 Test system

The experimental study was conducted using a single-cylinder, four-stroke, direct injection, water-cooled, naturally aspirated diesel engine with 498 cm^3 of displacement and four-valve SOHC. **Figure 1** shows the schematic diagram of the test engine and measurement setup. The engine specifications are listed in **Table 1**. A standalone supercharger made by Engine Tech, a Korean local company, was used to supply the intake boosting. A conventional EGR system was used, in which the line is routed directly from exhaust manifold to the intake manifold. The engine was connected to the test system which is a 57 kW Dynamometer (Elin AVL Puma MCA325MO2). A data acquisition system (Dewetron DEWE-800-CA) in combination with an encoder (Autonics E40S8–1800-3-T-24), a pressure transducer (Kistler

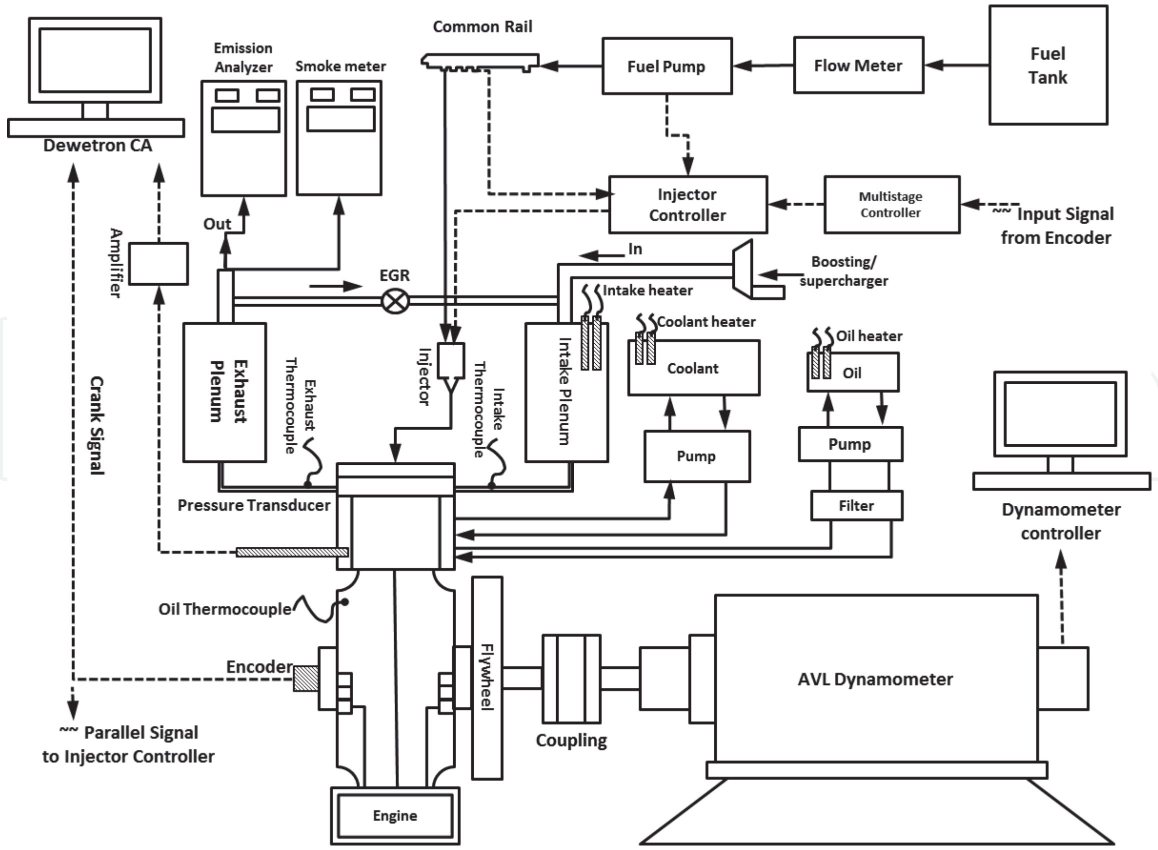


Figure 1.
Schematic diagram of the engine and measurement system setup.

Engine Parameters	Value
Displacement	498 cm^3
Bore	83 mm
Stroke	92 mm
Compression Ratio	19.5
Con. Rod Length	145.8 mm
Crank Radius	43.74 mm
Valve System	4-valve SOHC
Fuel System	Electronic Common Rail

Table 1.
Engine specifications.

6056A), and an amplifier (Kistler 5018) were used to obtain the combustion data. The fuel was injected to the combustion chamber using a Bosch seven-hole injector. A multi-stage injection engine controller (*Zenobalti:* ZB-8035, ZB-5100, ZB-100) was used to adjust the combustion strategy. Meanwhile, the temperatures of air intake, engine coolant, and lubricant oil were controlled by using separate temperature controller. The exhaust emissions, including unburned total hydrocarbon, carbon monoxide, and NOx were measured using a Horiba MEXA-7100DEGR. AVL 415 smoke meter was used for soot emission measurement. Some thermocouple K and RTD types were installed on the certain part of the engine to measure including the intake, oil, coolant and exhaust temperature.

2.2 Test fuels preparation

Two fuels, which are diesel and gasoline-biodiesel blend were used in this study. The baseline fuels utilized in this study were commercial gasoline (GB00), neat diesel (D100) and pure soya bean biodiesel (B100). The chemical composition information of soya bean vegetable oil is given in **Table 2**. Biodiesel (5% by volume) and gasoline were blended and labeled as GB05. To maintain the homogeneity, the mixing process was conducted about 10 minutes then immediately used for experiment. The physical properties of baseline fuels and GB05 are given in **Table 3**.

2.3 Operating conditions

The engine was operated at stable condition with fixed 1200 rpm. An injection pressure of 70 MPa was used for PPCI. Single injection timing at 40 °CA BTDC was adopted and set for PPCI combustion mode. The total energy input of injected fuel was set at around 26 mg/cycle. The initial parameters of intake temperature, oil temperature, and coolant temperature were maintained at 85°C, 75°C, and 65°C, respectively. The reason why the intake temperature was maintained at 85°C is to promote the autoignition of the fuel easier. As already known that GB05 more likely as a pure gasoline which low autoignition characteristics. The homogeneous hot

Fatty Acid	System Name	Structure	Formula[a]	Composition (wt %)
Myristic	Tetradecanoic	14:0	$C_{14}H_{2}8O_{2}$	0
Palmitic	Hexadecanoic	16:0	$C_{16}H_{32}O_{2}$	12
Stearic	Octadecanoic	18:0	$C_{18}H_{36}O_{2}$	3
Arachidic	Eicosanoic	20:0	$C_{20}H_{40}O_{2}$	0
Behenic	Docosanoic	22:0	$C_{22}H_{44}O_{2}$	0
Lignoceric	Tetracosanoic	24:0	$C_{24}H_{48}O_{2}$	0
Oleic	cis-9-Octadecenoic	18:1	$C_{18}H_{34}O_{2}$	23
Linoleic	cis-9,cis-12-Octadecadienoic	18:2	$C_{18}H_{32}O_{2}$	55
Linolenic	cis-9,cis-12, cis-15-Octadecatrienoic	18:3	$C_{18}H_{30}O_{2}$	6
Erucic	cis-13-Docosenoic	22:1	$C_{22}H_{42}O_{2}$	0

xx:y indicates xx carbons in the fatty acid chain with y double bonds.

Table 2.
Chemical composition of soya bean vegetable oil.

Test Item	Unit	Test Method	Gasoline	GB05	B100	D100
Heating Value	MJ/kg	ASTM D240:2009	45.86	45.32	39.79	45.93
Kinematic Viscosity (40°C)	mm^2/s	ISO 3104:2008	0.735	—	4.229	2.798
Lubricity	mm	ISO 12156-1:2012	548	290	189	238
Cloud Point	°C	ISO 3015:2008	−57	−37	3	−5
Pour Point	°C	ASTM D6749:2002	−57	−57	1	−9
Density (15 °C)	kg/m^3	ISO 12185:2003	712.7	722.3	882.3	826.3

Table 3.
Physical properties of the fuels.

EGR and air mixture were applied in this study with 0%, 20% and 50% of flow rates by using a pair of gate valve. The EGR ratio was calculated using Eq. 1 as follows.

$$EGR\% = \frac{m_E}{m_E + m_i} \times 100\% \tag{1}$$

where m_E and m_i are the mass of EGR and intake fresh air, respectively.

The air boosting were set at 0.1 and 0.12 MPa in the intake manifold. The more detail engine operating parameters and injection strategies are presented in **Tables 4** and **5**, respectively. The data of 100 consecutive cycles such as in-cylinder pressure was recorded for combustion analysis.

The analysis and discussion was performed on several engine parameters such as in cylinder heat release rate, temperature, peak of pressure rise rate, IMEP, COV of IMEP, knocking/ringing intensity, thermal efficiency and combustion efficiency. Rate of heat release was calculated using Eq. 2.

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} \tag{2}$$

Where, γ is the specific heat ratio, V is the instantaneous cylinder volume, and p is the cylinder pressure. The normal and suitable value of γ for a CI engine is 1.3. The in-cylinder pressure and volume data were used to calculate the in-cylinder temperature using ideal gas law, as shown in Eq. 3.

$$T = \frac{p.V}{n.R} \tag{3}$$

where p is for pressure, V for volume, n is the amount of substance, and R is the gas constant.

Parameter	Diesel/GCI
Speed (rpm)	1200
Inj. Pressure (MPa)	70
Injection strategy	PPCI
Inj. Quantity (mg)	26
T intake (°C)	85
T oil (°C)	75
T coolant (°C)	65
EGR (%)	0, 20 and 50
Intake boosting (MPa)	0.1 and 0.12

Table 4.
Operating parameters.

Combustion modes	Injection strategies	Injection timing and duration	Injected fuel	
			D100	**GB05**
PPCI	Single	40 °CA BTDC (1000 μs)	(26 mg)	

Table 5.
Injection strategies.

Furthermore, the emission of CO, HC, NO_x and particulate matter (smoke) were also discussed and analyzed in detail.

3. Experimental results and discussion

The main purpose of this experiment is to improve the efficiency and emission characteristics of CI engine fueled with gasoline-biodiesel blends using GCI mode. To replace the utilization of diesel fuel with gasoline fuel in CI engine due to high demand of diesel fuel in the market, thus a small amount of biodiesel (5%) was added as the lubricity improver to overcome the wear problems in the fuel system. The performance results were compared with only pure diesel fuel, because the basic of the engine is diesel engine. The engine was run on single injections mode (PPCI) at 1200 rpm with various EGR rate (0%, 20% and 50%) and intake boosting 0.1 to 0.12 MPa to investigate the effect of EGR and boosting on performance, combustion and emissions. The performance, combustion and emissions characteristics data was analyzed and presented graphically for in-cylinder pressure, temperature, HRR, ignition delay, MPRR, PPRR, IMEP, thermal efficiency and its emissions including HC, CO, NO_x and smoke opacity.

3.1 Effect of EGR and PPCI injection strategy

The total fuel consumption per cycle in PPCI mode is maintained at 26 mg per cycle and single injection timing at 40 °CA BTDC. The others engine operating conditions i.e. air intake, engine coolant and engine oil temperatures were set at 358 K, 338 K and 348 K, respectively. Meanwhile, the fixed intake pressure 0.1 MPa and various EGR rates for 0%, 20% and 50% were used to characterized the effect of EGR and PPCI injection strategy on combustion and emissions of GCI engine fueled with gasoline-biodiesel blends.

Figure 2 shows the in-cylinder pressure, temperature and HRR of PPCI mode at various EGR rate for 0%, 20% and 50%, and fixed intake boosting 0.1 MPa. It can be seen from the figure that CI engine fueled with diesel fuel reveal the decreasing in-cylinder pressure when the EGR rate is increase. Similar with the diesel fuel, gasoline-biodiesel blend also indicates the same trend when EGR rate increase the in-cylinder pressure decrease. The in-cylinder temperature for diesel fuel decreasing as the trend of in-cylinder pressure when EGR rate increase. However, the in-cylinder temperature trends of gasoline-biodiesel blend show that EGR 50% lead to the highest value among the others EGR rates. Observing at heat release rates curves, it is seen that the heat release process of both diesel and gasoline-biodiesel blends fuels show a marked two-stage ignition. The first stage ignition of diesel fuel consistently higher than 20 J/deg., even though all of the curves reveal decreasing trends for various increasing EGR rate. Meanwhile, the first stage ignitions from gasoline-biodiesel blends are very low for all various EGR rates, and it is almost very difficult to be recognized. The highest peak of heat release rate can be obtained from gasoline-biodiesel blends with 50% EGR rate. The highest peak of heat release rate can be used to determine that the excessive pressure rise rate is happened. The excessive of PRR means that the combustion is not stable or some time when in the high load condition, the rapid pressure rise rate can result in heavy knocking operation.

Figure 3 shows the effect of EGR on ignition delay when engine operated using PPCI mode. The higher EGR rate results the longer ignition delay for both of diesel and gasoline-biodiesel blends. However, it can be observed that gasoline-biodiesel blends lead to the much longer ignition delay compared to diesel fuel in every EGR

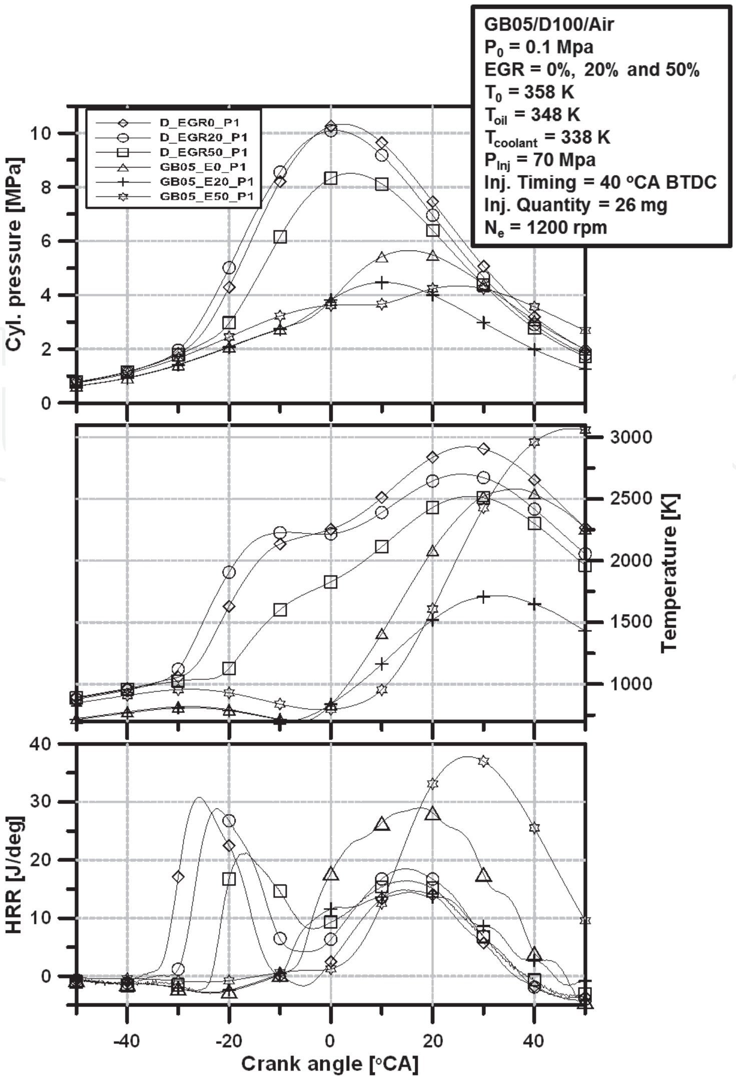


Figure 2.
Effect of EGR on cylinder pressure, temperature and HRR of PPCI mode.

rate variations. This condition is the advantage of gasoline fuel, which is longer ignition delay due to high volatile and low cetane number, thus there is a possibility of complete mixing period before combustion occurred. However, the longer ignition delay caused the shifted of maximum in-cylinder pressure far away form TDC which can reduce the performance of the engine. To overcome this condition the earlier injection timing can be applied among many other solutions.

Figure 4(a) shows the effect of EGR on maximum of in-cylinder pressure and (b) peak of pressure rise rate of PPCI strategy. The higher of EGR rate generate the lower in-cylinder pressure maximum and lower the maximum pressure rise rate for both diesel fuel and gasoline-biodiesel blends. Similarly, the increasing of EGR rates also reducing the maximum of pressure rise rate for both diesel fuel and gasoline-biodiesel blends. This condition happened due to the slowdown of combustion process. One of the reasons when utilizing EGR to slowing down of combustion process is the concentration of O_2 is lowered and the concentrations of CO_2 and H_2O unintentionally increased. Therefore, this slows down the reactions in the oxidizing direction and speeds up the reactions of reduction process direction.

The effect of EGR on IMEP of GCI engine using PPCI strategy is presented in **Figure 5**. The increasing of EGR rates does not give any effect on IMEP of GCI engine fueled with diesel fuel. However, the 50% EGR rate results the highest IMEP value for gasoline-biodiesel blends, even much higher if compared with diesel fuel that is almost 1.0 MPa. Related to the IMEP value, the engine efficiencies especially

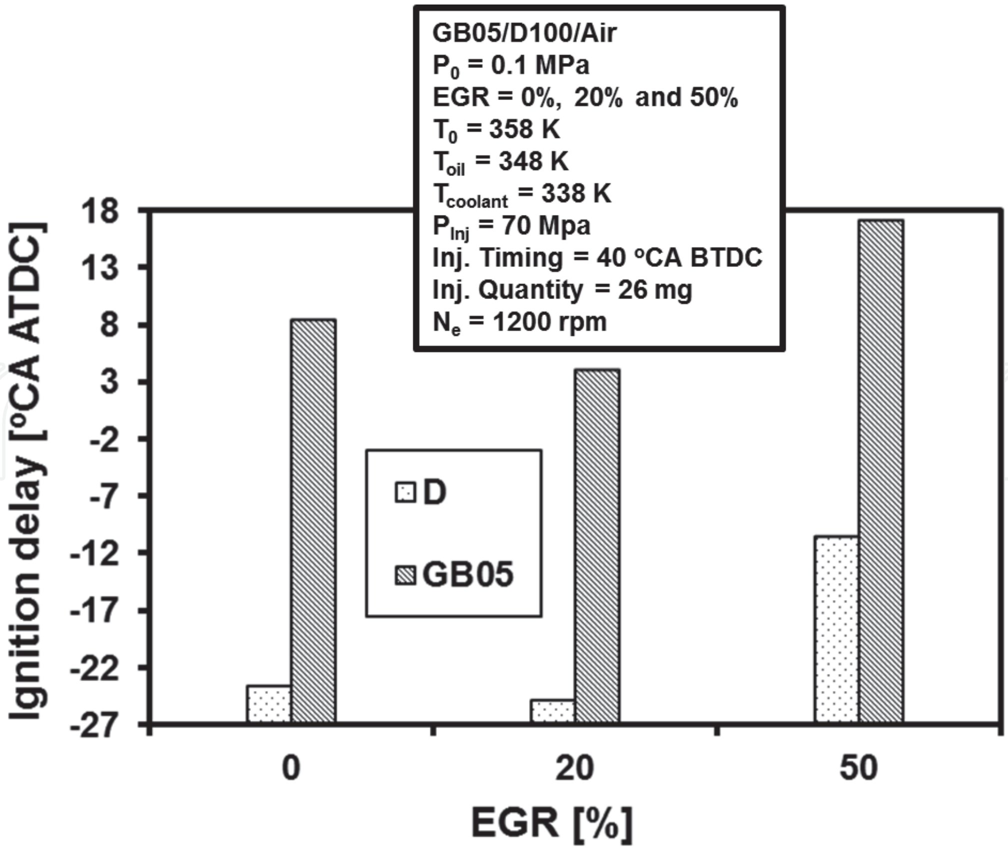


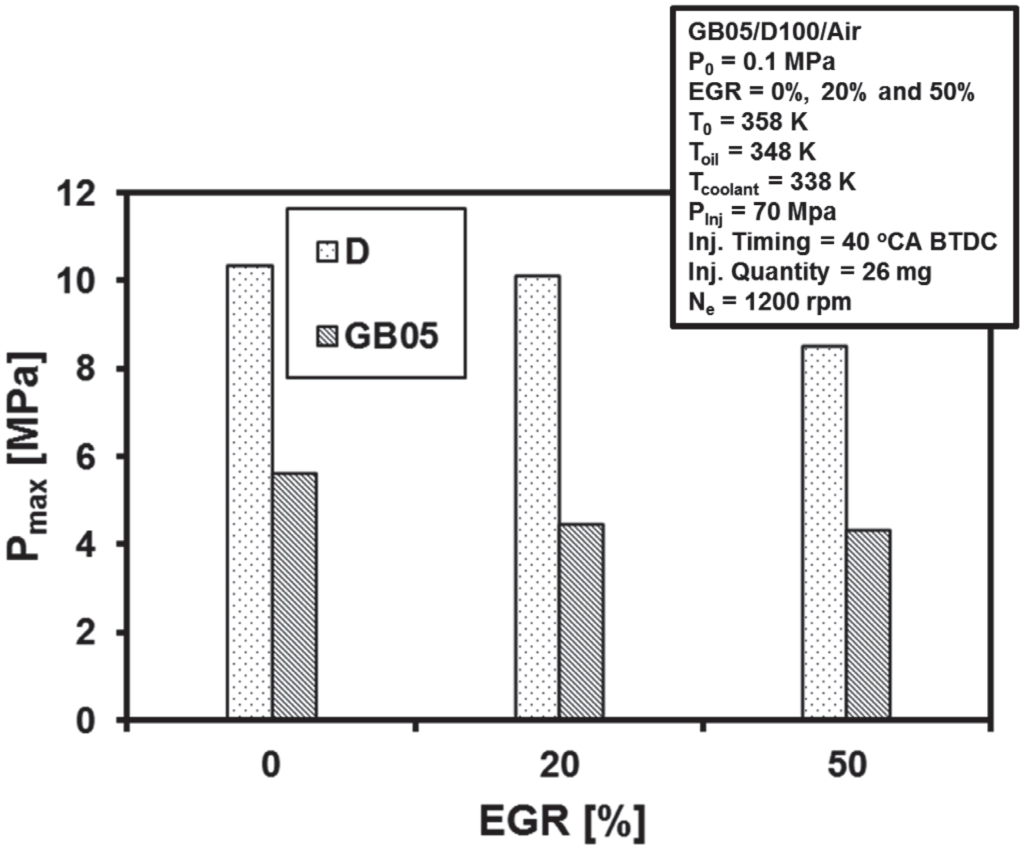
Figure 3.
Effect of EGR on ignition delay of PPCI mode.

indicated thermal efficiency also can be calculated by using it derivative that is indicated power/work. The effect of various EGR rates on indicated thermal efficiency of GCI engine using PPCI strategy can be seen in **Figure 6**. It can be seen that by increasing EGR rate the value of indicated thermal efficiencies are decreased for both of diesel and gasoline-biodiesel blends. The 50% EGR rate for diesel fuel lead to a little increasing value of indicated thermal efficiency is compared with 20% of EGR rate. However, it caused the significant drop value of indicated thermal efficiency in case of gasoline-biodiesel blends fuel.

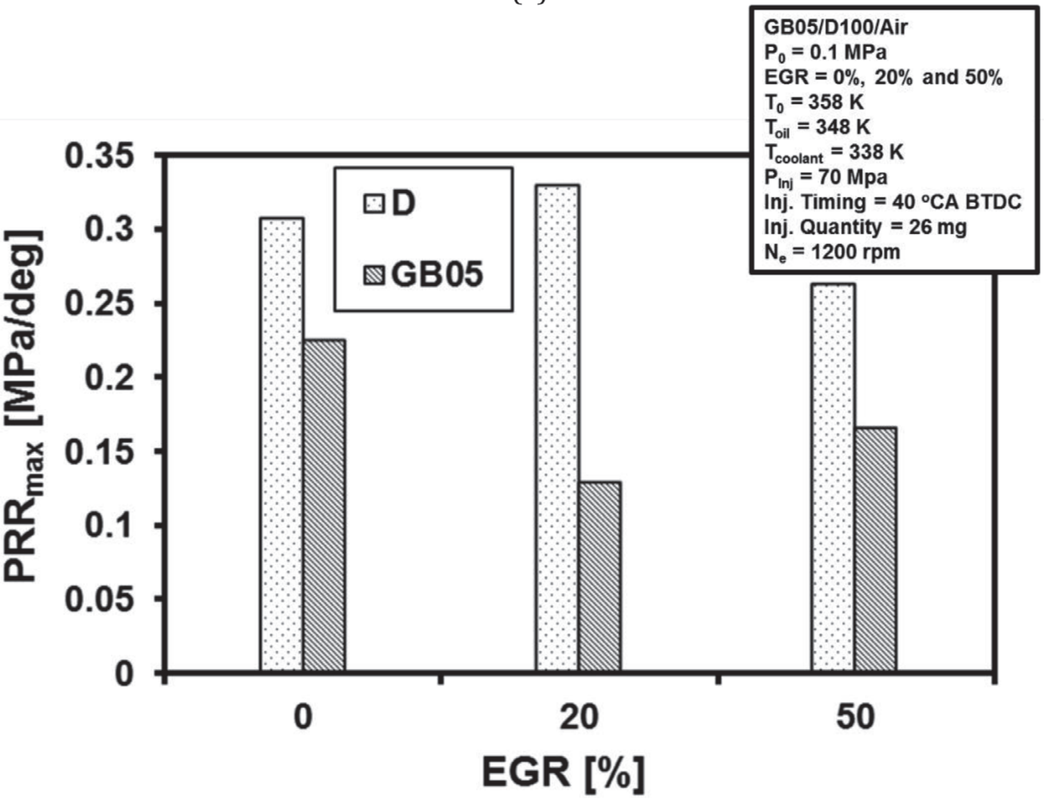
Effect of EGR rates on CO emission of GCI engine using PPCI mode can be observed on **Figure 7**. All variation of EGR rates showed that CO emission of gasoline-biodiesel blends are lower than diesel fuel due to the volatile properties of gasoline and higher oxygen content of biodiesel, which make more complete mixing and produce more perfect combustion. However, in general, the increasing of EGR rates caused no different of CO emission for both diesel and gasoline-biodiesel blends fuels. A little decreasing value of CO emission was only happened on GCI engine fueled with gasoline-biodiesel blends when running on 50% EGR rate.

Figure 8 shows the effect of various EGR rate on HC emission of GCI engine running on PPCI strategy. As like the trend of CO emission, HC emission of GCI engine fueled with gasoline-biodiesel blends was also showed a lower value compared to diesel fuel. This condition can be explained also due to the properties of gasoline fuel and the oxygen content of biodiesel. The 20% of EGR rate value gives the lowest effect of HC emission both for diesel and for gasoline-biodiesel blends. Therefore, it is assumed in the PPCI mode the 20% of EGR rate as an optimum value to obtain lowest HC emission.

The NO_x emission and its effect by using various EGR rate on GCI engine using PPCI mode can be seen in **Figure 9**. Normally, the increasing of EGR rates will lead to the lower NO_x emission. However, in this case, for diesel fuel, the 20% of EGR



(a)



(b)

Figure 4.
Effect of EGR on (a) max pressure and (b) peak pressure rise rate of PPCI mode.

rate gives highest NO_x emission. Even though, when 50% EGR was applied the NO_x emission will also decreasing. However, there are no effects of EGR rate variations on NO_x emission of GCI engine fueled with gasoline-biodiesel blends. This

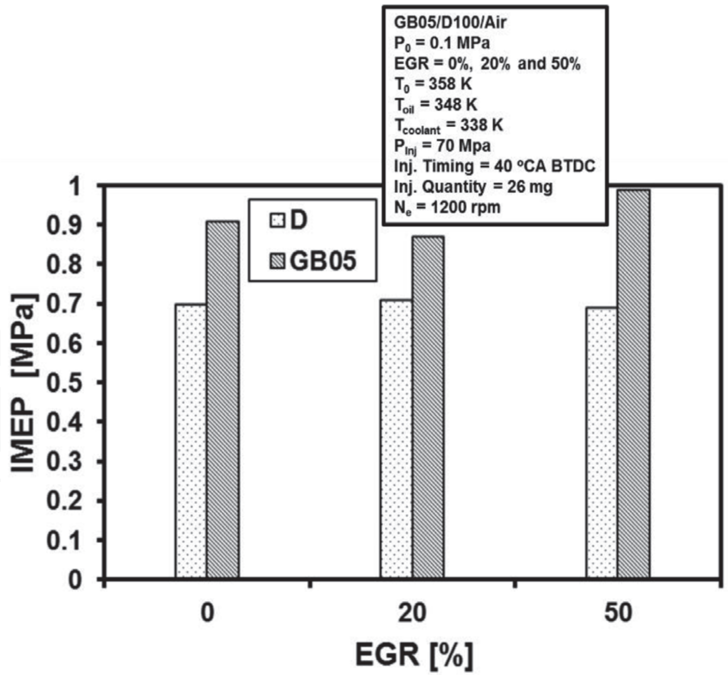


Figure 5.
Effect of EGR on IMEP of PPCI mode.

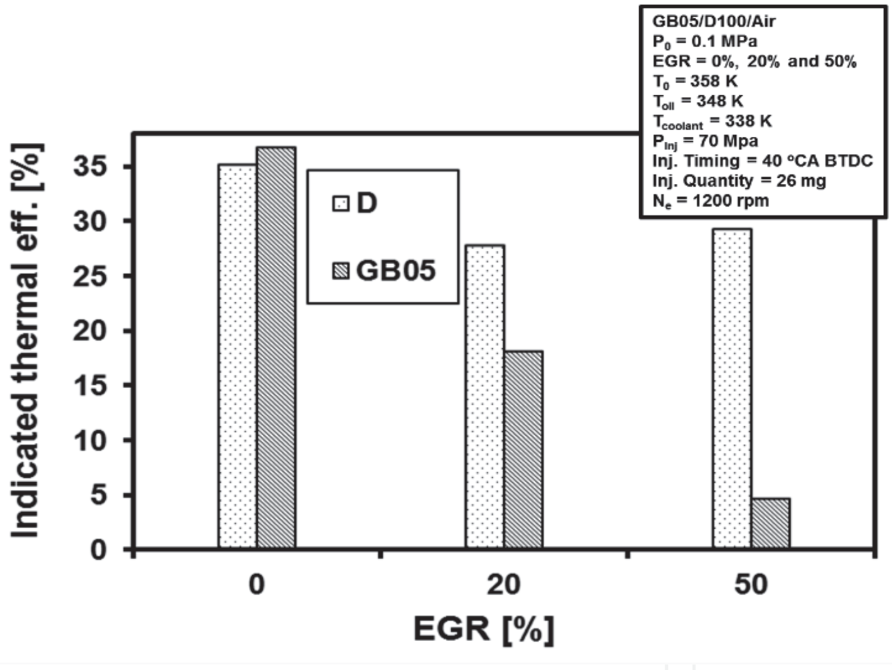


Figure 6.
Effect of EGR on indicated thermal efficiency of PPCI mode.

condition can be seen in the trend of graph that from the three EGR rate variation resulted almost same NO_x emission value.

The smoke emission of CI engine usually contrasts with NO_x emission. When the NO_x higher, the smoke will be a lower and vice versa. The effect of EGR rate variation on the smoke emission of GCI engine can be seen in **Figure 10**. Smoke emission of GCI engine fueled with diesel in the high level for all variation of EGR rate, even when the rate increased. However, the smoke emission of GCI engine using gasoline-biodiesel blends obtain its lowest value when EGR rates at 20%. It can be said that the optimums of EGR rate that can maintain lowest smoke emission while lowest NO_x emission is 20%.

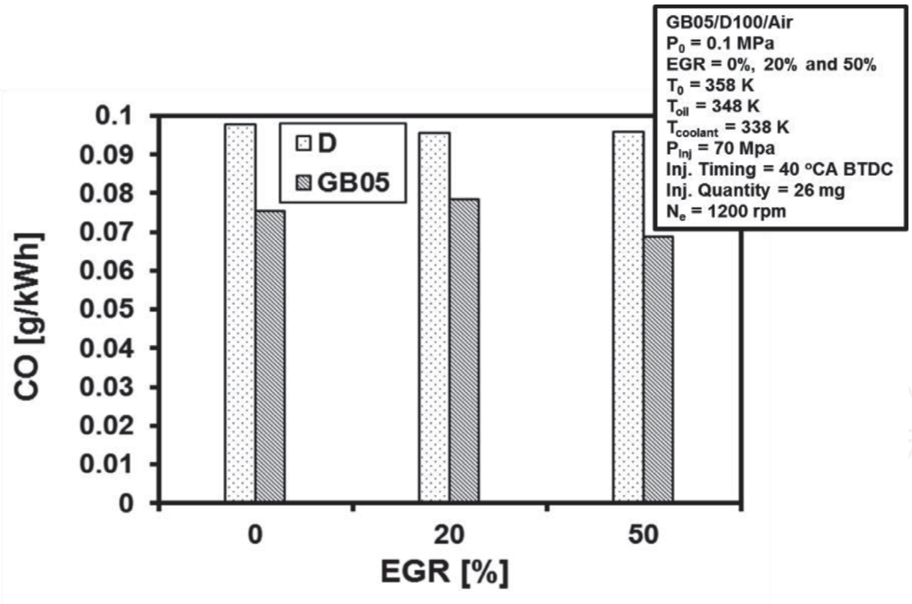


Figure 7.
Effect of EGR on CO emission of PPCI mode.

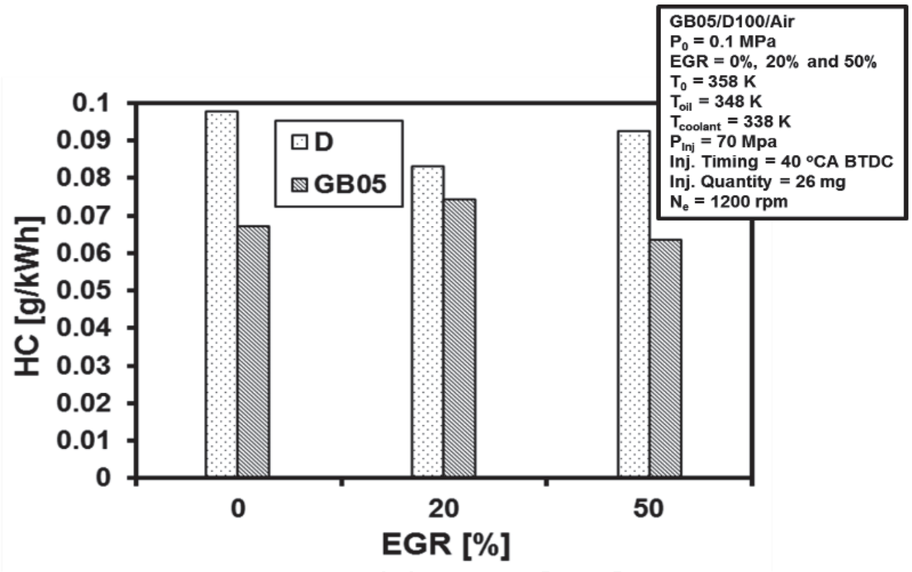


Figure 8.
Effect of EGR on HC emission of PPCI mode.

3.2 Effect of intake boosting and PPCI injection strategy

To understand the effects of intake boosting on GCI engine fueled with gasoline-biodiesel blends on PPCI mode in a simple and easy way, only the 20% of EGR rate was chosen an explained in this study. The intake boosting was set at 0.1 MPa and 0.12 MPa. **Figure 11** shows the effect of boosting on in-cylinder pressure, temperature, and heat release rate of GCI engine fueled with gasoline-biodiesel blends when running on PPCI strategy. Normally found that the increasing of intake boosting rate, increasing the in-cylinder pressure for both diesel fuel and gasoline-biodiesel blends fuel. An ambient pressure of intake boosting gives a higher in-cylinder pressure of GCI engine fueled with diesel compared to gasoline-biodiesel blends. Even, the in-cylinder of gasoline biodiesel-blends with intake boosting 0.12 MPa is lower than diesel fuel with ambient intake boosting. It was also same, that the implementation of 0.12 MPa intake-boosting leads to a higher in-cylinder pressure

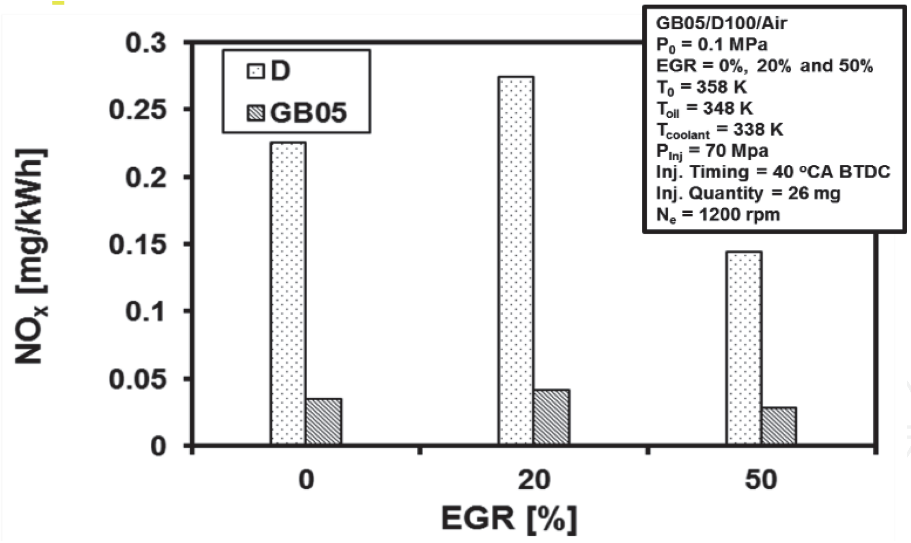


Figure 9.
Effect of EGR on NO_x emission of PPCI mode.

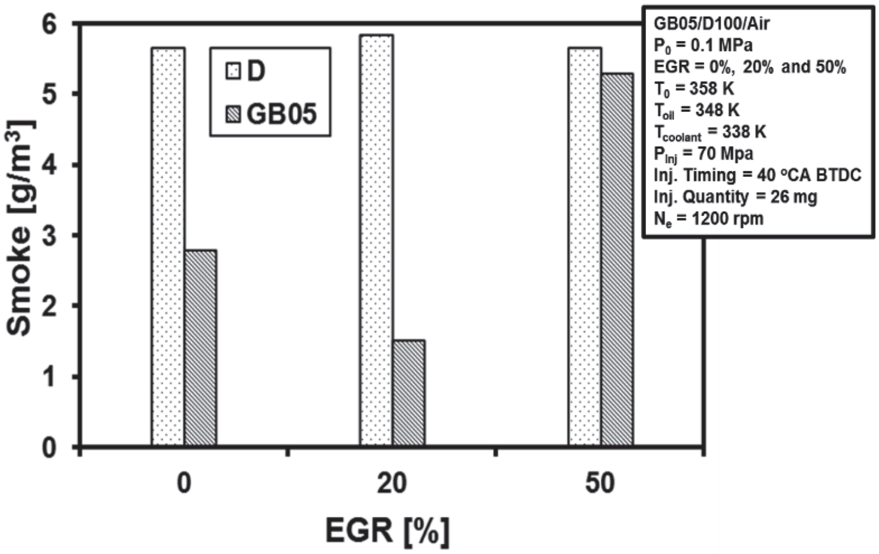


Figure 10.
Effect of EGR on smoke emission of PPCI mode.

of GCI engine fueled with gasoline-biodiesel blends than gasoline-biodiesel with 0.1 MPa intake boosting. To obtain in cylinder pressure of gasoline biodiesel blend at least equal to pure diesel fuel, the higher intake boosting can be applied as long as the engine material supports for high pressure condition and the real engine booster in this case turbocharger can achieves maximum desired pressure. Similar with in-cylinder pressure, the in-cylinder temperature curves show that the highest value is for GCI engine fueled with diesel fuel when intake boosting 0.12 MPa was applied. The lowest in-cylinder temperature, which is below 2000 K, was happened for GCI engine fueled with gasoline-biodiesel blends fuel when using ambient pressure 0.12 MPa. The HRR curves show that the highest value is for GCI engine fueled with gasoline-biodiesel fuel using 0.1 MPa intake boosting. The higher HRR value, the higher-pressure rise rate that can be determines the more unstable engine combustion. The lowest HRR value was obtained from GCI engine fueled with diesel fuel in the ambient pressure condition, which is the most stable combustion.

The effect of intake boosting on ignition delay of GCI engine using PPCI strategy is presented in **Figure 12**. The intake boosting gives effect on the lower ignition

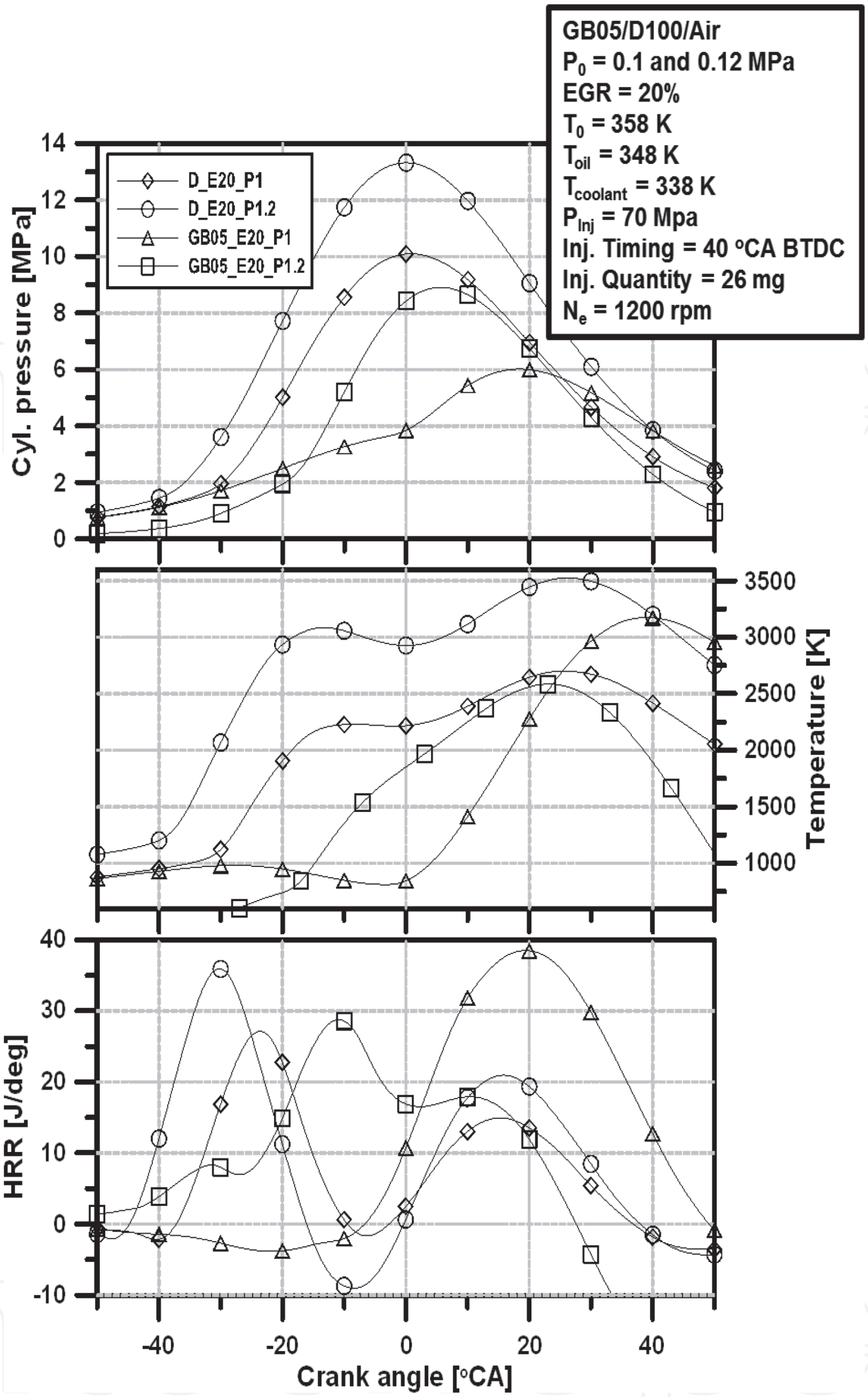


Figure 11.
Effect of boosting on cylinder pressure, temperature, and heat release rate of PPCI mode.

delay for both diesel and gasoline-biodiesel blend fuel. The ambient pressure of intake boosting resulted ignition delay timing for diesel fuel at around 25 °CA BTDC, then the 0.12 MPa intake boosting lead to the slightly earlier of ignition delay timing at around 27 °CA BTDC. Similar trend happened on gasoline-biodiesel fuel, that ambient pressure of intake boosting resulted ignition delay timing at around 11 °CA BTDC, then when 0.12 MPa intake boosting was applied the ignition delay timing also more advanced at around 2 °CA BTDC. The higher volatile and lower cetane number properties of gasoline fuel caused the longer ignition delay timing if compared with diesel fuel. However, the application of intake boosting resulted a shifting of ignition delay timing earlier. The longer ignition delay timing is possible to produce more complete mixing period of air and fuel prior to combustion,

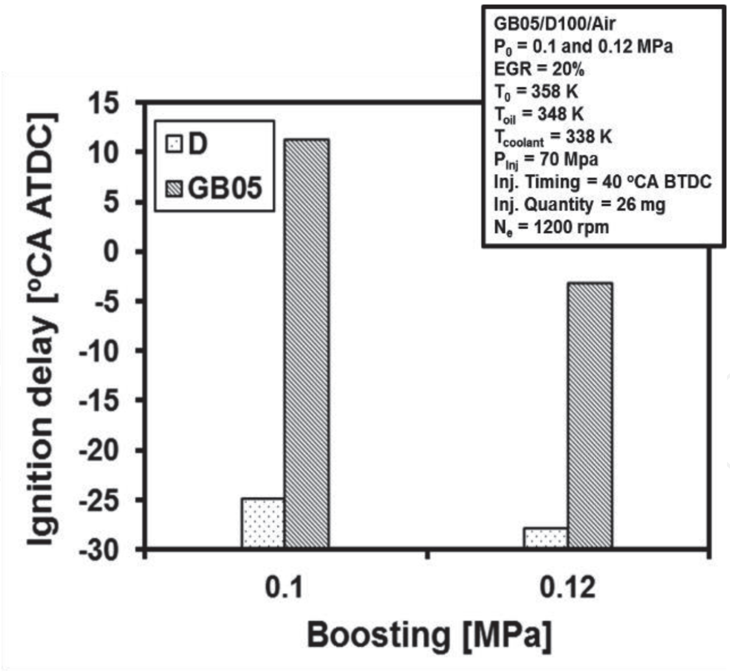


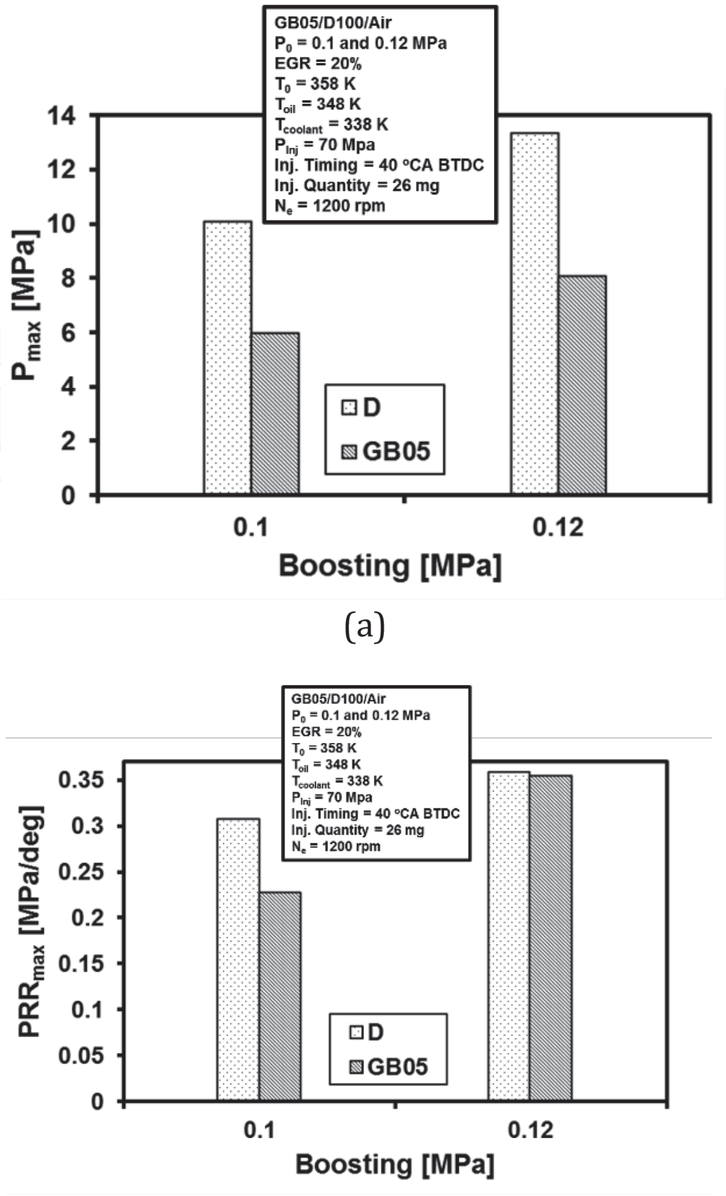
Figure 12.
Effect of boosting on ignition delay of PPCI mode.

however, too long ignition delay timing sometimes caused problem in the engine emission and efficiency.

Figure 13 shows the effect of various intakes boosting on maximum of in-cylinder pressure and its maximum pressure rise rate. A normal condition happened that the increasing intake boosting, the increasing maximum in-cylinder pressure for both diesel and gasoline-biodiesel blends fuels. However, the increasing level of maximum in-cylinder pressure of diesel fuel is much higher than gasoline-biodiesel fuel. It is suspected that intake boosting caused the mixing of air fuel in diesel fuel more optimum than gasoline-biodiesel blend. The increasing of intake boosting leads to the increasing maximum pressure rise rate of GCI engine fueled with gasoline-biodiesel blend in almost same value with diesel fuel. It is mean that the GCI engine running with intake boosting for gasoline-biodiesel blend has an almost similar stability compared with diesel fuel. However, very high-pressure rise rate indicated that the engine in unstable condition.

Effect of various intakes boosting on IMEP of GCI engine fueled with gasoline-biodiesel blends in PPCI strategy can be seen in **Figure 14**. The IMEP of GCI engine fueled with gasoline-biodiesel blend in ambient pressure of intake boosting is higher than when intake boosting is 0.12 MPa. The opposite condition was happened for diesel fuel, which is the IMEP value of GCI engine is higher when 0.12 MPa intake boosting was applied compared with ambient pressure. The condition for IMEP of diesel fuel as the effect of increasing the intake boosting is the normal phenomenon; however, for gasoline-biodiesel blend it is quiet special. This condition suspected by the effect of high volatile and low cetane number of gasolines, which resulted higher-pressure rise rate as shown in **Figure 13**. Fluctuate of in-cylinder pressure may lead to the unstable combustion and resulted the lower IMEP value.

The indicated thermal efficiency of GCI engine using PPCI strategy affected by various intake boosting is presented in **Figure 15**. The indicated thermal efficiency of GCI engine fueled with diesel fuel increased due to the increasing of intake boosting. Similarly, for GCI engine fueled with gasoline-biodiesel blend, even though the IMEP reduced when the intake boosting increased to be 0.12 MPa. This condition, in any case, is expected in the GCI engine fueled with gasoline-biodiesel blend. Furthermore, both for ambient and 0.12 MPa intake boosting showed that



(a)

(b)

Figure 13.
Effect of boosting on (a) max pressure and (b) peak pressure rise rate of PPCI mode.

the indicated thermal efficiency of GCI engine with diesel fuel is higher than gasoline-biodiesel blend.

Figure 16 shows the effect of intake boosting on CO emission of GCI engine using PPCI strategy. It is already known that the utilization of gasoline-biodiesel blend in GCI engine resulted lower CO emission compared to diesel fuel. Similarly, in the single injection method of PPCI strategy also obtained the lower CO emission of GCI engine fueled with gasoline-biodiesel blend compared to diesel fuel. The increasing of intake boosting from 0.1 to 0.12 MPa in GCI engines gives effect on the decreasing of CO emission for both gasoline-biodiesel blend and diesel fuels. It is suspected due to the combination of 20% EGR and 0.12 MPa of intake boosting, which may lead to the complete combustion.

The effect of intake boosting on HC emission of GCI engine can be observed in **Figure 17**. Similar with the trend on CO emission, the HC emission of GCI engine fueled with of GCI engine with gasoline-biodiesel blend originally is lower than diesel fuel as it can be seen in the ambient intake boosting condition. When the intake boosting increased to be 0.12 MPa HC emission of GCI engine decreased around a half value than 0.1 MPa of intake boosting. For GCI engine fueled with

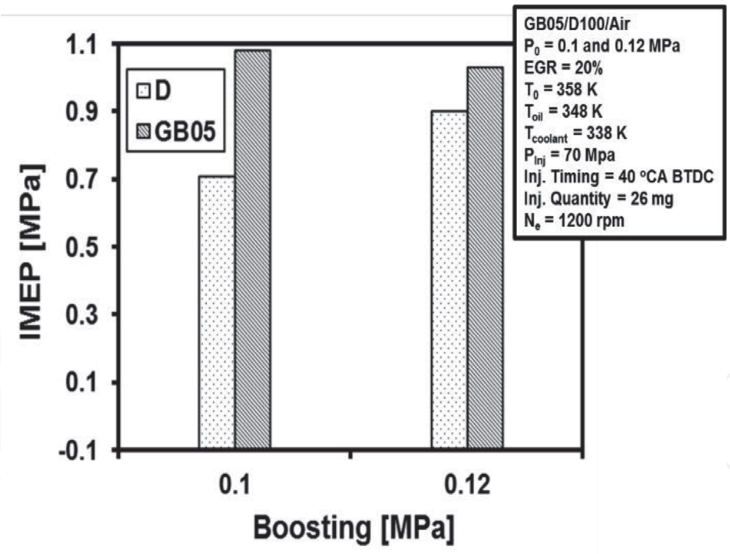


Figure 14.
Effect of boosting on IMEP of PPCI mode.

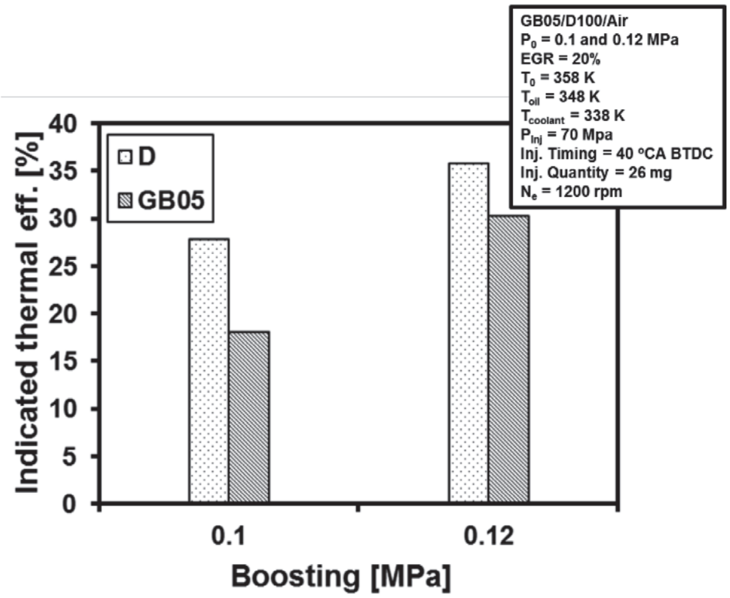


Figure 15.
Effect of boosting on indicated thermal efficiency of PPCI mode.

gasoline-biodiesel blend, it is obtained greatly decreasing of HC emission when the 0.12 MPa of intake boosting applied compared with 0.1 MPa. The decreasing value of HC emission in 0.12 MPa of intake boosting is almost 90% lower from the ambient pressure of intake boosting.

Figure 18 shows the effect of intake boosting on NO_x emission of GCI engine with PPCI strategy. Overall, the NO_x emission of GCI engine fueled with diesel is higher than GCI engine fueled with gasoline-biodiesel blend when using PPCI mode for either ambient intake pressure or increasing intake pressure at 0.12 MPa. The trend of graph shows that the increasing intake boosting also followed by increasing the NO_x emission for both diesel and gasoline-biodiesel blend. It is mean that the increasing of intake boosting has opposite function with 20% EGR. In this case, by using only 20% EGR rate, the NO_x emission of GCI engine fueled with gasoline-biodiesel blend is very low under 0.05 mg/kWh. However, increasing intake boosting 0.12 MPa, leads the deterioration on NO_x emission to be around 0.2 mg/kWh.

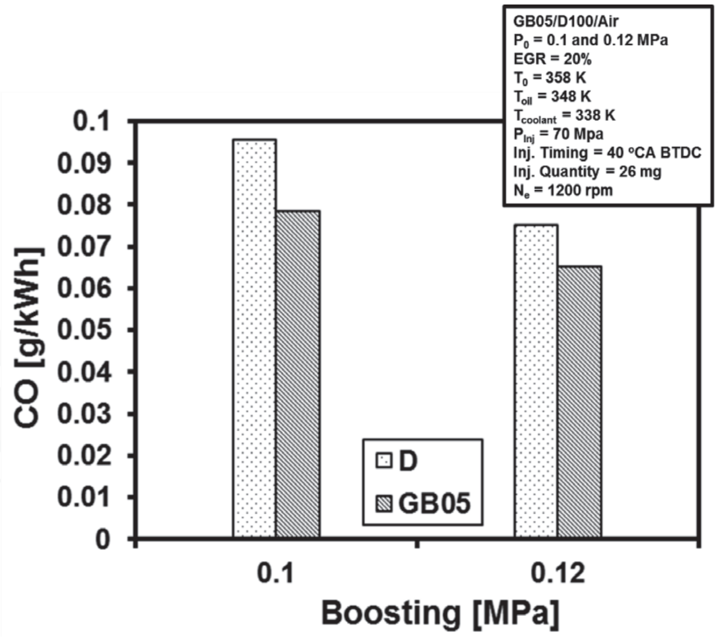


Figure 16.
Effect of boosting on CO emission of PPCI mode.

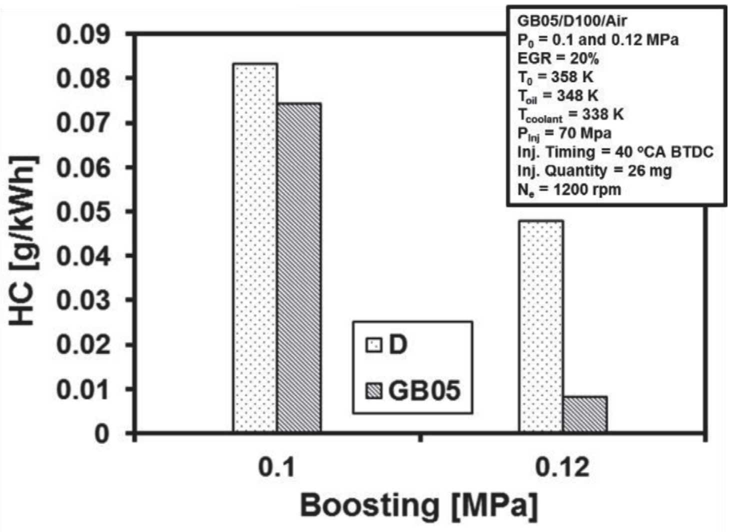


Figure 17.
Effect of boosting on HC emission of PPCI mode.

The effect of intake boosting on smoke emission of GCI engine with PPCI strategy can be seen in **Figure 19**. Smoke emission of GCI engine fueled with diesel fuel is very high almost 6 g/m^3 when running on PPCI mode by 20% of EGR rate and ambient pressure of intake boosting. While, in this condition smoke emission of GCI engine fueled with gasoline-biodiesel blend much lower than diesel fuel at around 1.5 g/m^3. Increasing intake boosting to be 0.12 MPa makes smoke emission of GCI engine fueled with diesel fuel decrease very significant around 3 g/m^3. However, the increasing of intake boosting to be 0.12 MPa for GCI engine fueled with gasoline-biodiesel caused the increasing of smoke emission, even though still lower than the emission of GCI engine fueled with diesel fuel which is to be around 2.5 g/m^3. If the point of view of GCI engine fueled with gasoline-biodiesel blend running on PPCI mode focused simultaneously on NOx emission and smoke emission, then it can be stated that the optimum effort to reduce both of emission parts is by using 20% EGR rate and 0.1 MPa intake boosting.

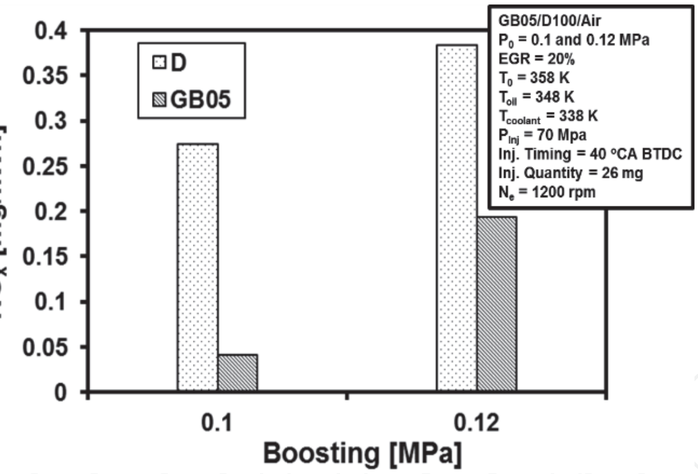


Figure 18.
Effect of boosting on NO_x emission of PPCI mode.

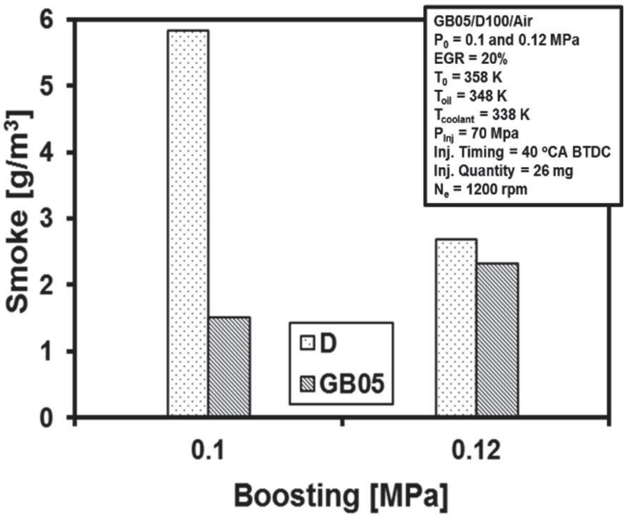


Figure 19.
Effect of boosting on smoke emission of PPCI mode.

4. Conclusions

The study on GCI engine was conducted in an experiment using biodiesel addition 5% into gasoline, compared to neat diesel with single injection (PPCI) strategy combined with the application of EGR and intake boosting in order to obtain high efficiency and low emission of GCI engine. The engine testing was set in the same of energy input that is injected fuel amount around 26 mg per cycle. Based on the results and comprehensive analysis, the following general conclusions may be drawn from this study:

Increasing EGR rate the value of indicated thermal efficiencies are decreased for both of diesel and gasoline-biodiesel blends. The highest 50% EGR rate for diesel fuel leads to a little increasing value of indicated thermal efficiency is compared with 20% of EGR rate. However, it caused the significant drop value of indicated thermal efficiency in case of gasoline-biodiesel blends fuel. By using diesel fuel, the 20% of EGR rate gives highest NO_x emission. Even though, when 50% EGR was applied the NO_x emission will also decreasing. The utilization of EGR gives effect on the drop of NO_x emission value for gasoline-biodiesel blend much lower than diesel fuel. However, there are no effects of EGR rate variations on NO_x emission of GCI engine fueled with gasoline-biodiesel blends. Smoke emission of GCI engine fueled with diesel in the high level for all variation of EGR rate, even when the rate increased. However, the smoke emission of GCI engine using gasoline-biodiesel blends obtains its lowest value when EGR rates at 20%.

The indicated thermal efficiency of GCI engine fueled with diesel fuel increased due to the increasing of intake boosting. Similarly, for GCI engine fueled with gasoline-biodiesel blend, the indicated thermal efficiency was also increased when the intake boosting increased to be 0.12 MPa. The NO_x emission of GCI engine fueled with diesel is higher than GCI engine fueled with gasoline-biodiesel blend when using PPCI mode for either ambient intake pressure or increasing intake pressure at 0.12 MPa. The increasing intake boosting also followed by increasing NO_x emission for both diesel and gasoline-biodiesel blend. Increasing intake boosting to be 0.12 MPa makes smoke emission of GCI engine fueled with diesel fuel decrease very significant around 3 g/m^3. However, the increasing of intake boosting to be 0.12 MPa for GCI engine fueled with gasoline-biodiesel caused the increasing of smoke emission, even though still lower than the emission of GCI engine fueled with diesel fuel.

Acknowledgements

This research was supported by the University of Ulsan, Korea. Yanuandri Putrasari acknowledges the support from the Indonesian Institute of Sciences (LIPI) and the Ministry of Research and Technology/National Research and Innovation Agency of Republic Indonesia (RISTEK-BRIN).

Author contribution

All authors contributed equally as the main contributor of this chapter. All authors read and approved the final version of this chapter.

Conflict of interest

The authors declare no conflict of interest.

Author details

Yanuandri Putrasari[1] and Ocktaeck Lim[2]*

1 Research Centre for Electrical Power and Mechatronics – Indonesian Institute of Sciences (LIPI), Bandung, Indonesia

2 School of Mechanical Engineering, University of Ulsan, Ulsan, South Korea

*Address all correspondence to: otlim@ulsan.ac.kr

References

[1] Dec JE. Advanced compression-ignition engines - Understanding the in-cylinder processes. Proc Combust Inst 2009;32 II:2727–42. doi:10.1016/j.proci.2008.08.008.

[2] Eng JA. Characterization of Pressure Waves in HCCI Combustion Reprinted From : Homogeneous Charge Compression Ignition Engines. SAE Tech Pap 2002;1:15. doi:10.4271/2002-01-2859.

[3] Das P, Subbarao PM V, Subrahmanyam JP. Control of combustion process in an HCCI-DI combustion engine using dual injection strategy with EGR. Fuel 2015;159:580–9. doi:10.1016/j.fuel.2015.07.009.

[4] Han D, Ickes AM, Bohac S V, Huang Z, Assanis DN. HC and CO emissions of premixed low-temperature combustion fueled by blends of diesel and gasoline. Fuel 2012;99:13–9. doi:10.1016/j.fuel.2012.04.010.

[5] Yang B, Li S, Zheng Z, Yao M, Cheng W. A comparative study on different dual-fuel combustion modes fuelled with gasoline and diesel. SAE Tech Pap 2012;2012–01–06. doi:10.4271/2012-01-0694.

[6] Kweon C, Foster D, Shibata G. The effects of oxygenate and gasoline-diesel fuel blends on diesel engine emissions. SAE Tech Pap 2000;2000–01–11. doi:https://doi.org/10.4271/2000-01-1173.

[7] Iannuzzi SE, Valentino G. Comparative behavior of gasoline-diesel / butanol-diesel blends and injection strategy management on performance and emissions of a light duty diesel engine. Energy 2014;71:321–31. doi:10.1016/j.energy.2014.04.065.

[8] Prikhodko VY, Curran SJ, Barone TL, Lewis S a, Storey JM, Cho K, et al. Emission Characteristics of a Diesel Engine Operating with In-Cylinder Gasoline and Diesel Fuel Blending. SAE Int 2010;2266:946–55.

[9] Leermakers CAJ, Van den Berge B, Luijten CCM, Somers LMT, de Goey LPH, Albrecht BA. Gasoline-Diesel Dual Fuel: Effect of Injection Timing and Fuel Balance. SAE Pap 2011; 2011–01–24. doi:10.4271/2011-01-2437.

[10] Liu H, Wang Z, Wang J, He X. Improvement of emission characteristics and thermal efficiency in diesel engines by fueling gasoline / diesel / PODEn blends. Energy 2016;97:105–12. doi:10.1016/j.energy.2015.12.110.

[11] Bae C, Kim J. Alternative fuels for internal combustion engines. Proc Combust Inst 2017;36:3389–413. doi:10.1016/j.proci.2016.09.009.

[12] Ra Y, Yun JE, Reitz RD. Numerical Parametric Study of Diesel Engine Operation with Gasoline. Combust Sci Technol 2009;181:2:350–78. doi:10.1080/00102200802504665.

[13] Rose KD, Ariztegui J, Cracknell RF, Dubois T, Hamje HDC, Pellegrini L, et al. Exploring a gasoline compression ignition (GCI) engine concept. SAE Int 2013;2013–01–09:1–54. doi:10.4271/2013-01-0911.

[14] Badra J, Viollet Y, Elwardany A, Im HG, Chang J. Physical and chemical effects of low octane gasoline fuels on compression ignition combustion. Appl Energy 2016;183:1197–208. doi:10.1016/j.apenergy.2016.09.060.

[15] Kim K, Kim D, Jung Y, Bae C. Spray and combustion characteristics of gasoline and diesel in a direct injection compression ignition engine. Fuel 2013; 109:616–26. doi:10.1016/j.fuel.2013.02.060.

[16] Yang B, Yao M, Zheng Z, Yue L. Experimental Investigation of Injection Strategies on Low Temperature Combustion Fuelled with Gasoline in a Compression Ignition Engine 2015;2015.

[17] Kim D, Bae C. Application of double-injection strategy on gasoline compression ignition engine under low load condition. Fuel 2017;203:792–801. doi:10.1016/j.fuel.2017.04.107.

[18] Kodavasal J, Kolodziej CP, Ciatti SA. Effects of injection parameters , boost , and swirl ratio on gasoline compression ignition operation at idle and low-load conditions. Int J Engine Res 2016:1–13. doi:10.1177/1468087416675709.

[19] Loeper P, Ra Y, Foster D, Ghandhi J. Experimental and computational assessment of inlet swirl effects on a gasoline compression ignition (GCI) light-duty diesel engine. SAE 2014 World Congr Exhib 2014;1. doi:10.4271/2014-01-1299.

[20] Kalghatgi GT, Risberg P, Ångström H. Advantages of Fuels with High Resistance to Auto-ignition in Late-injection, Low-temperature, Compression Ignition Combustion. SAE Int 2006:SAE 2006-01-3385. doi:10.4271/2006-01-3385.

[21] Kalghatgi GT, Risberg P, Angstrom H-E. Partially Pre-Mixed Auto-Ignition of Gasoline to Attain Low Smoke and Low NOx at High Load in a Compression Ignition Engine and Comparison with a Diesel Fuel. SAE Tech Pap 2007;2007–01–00. doi:10.4271/2007-01-0006.

[22] Yang H, Shuai S, Wang Z, Wang J. Fuel octane effects on gasoline multiple premixed compression ignition (MPCI) mode. Fuel 2013;103:373–9. doi:10.1016/j.fuel.2012.05.016.

[23] Wang B, Wang Z, Shuai S, Xu H. Combustion and emission characteristics of Multiple Premixed Compression Ignition (MPCI) mode fuelled with different low octane gasolines. Appl Energy 2015;160:769–76. doi:10.1016/j.apenergy.2015.01.115.

[24] Kodavasal J, Kolodziej CP, Ciatti SA, Sibendu S. Computational Fluid Dynamics Simulation of Gasoline Compression Ignition 2017;137:1–13. doi:10.1115/1.4029963.

[25] Won HW, Peters N, Pitsch H, Tait N, Kalghatgi G. Partially premixed combustion of gasoline type fuels using larger size nozzle and higher compression ratio in a diesel engine. SAE Tech Pap 2013;11. doi:10.4271/2013-01-2539.

[26] Tesfa B, Mishra R, Zhang C, Gu F, Ball AD. Combustion and performance characteristics of CI (compression ignition) engine running with biodiesel. Energy 2013;51:101–15. doi:10.1016/j.energy.2013.01.010.

[27] Cordiner S, Mulone V, Nobile M, Rocco V. Impact of biodiesel fuel on engine emissions and Aftertreatment System operation. Appl Energy 2016;164:972–83. doi:10.1016/j.apenergy.2015.07.001.

[28] Rakopoulos CD, Rakopoulos DC, Hountalas DT, Giakoumis EG, Andritsakis EC. Performance and emissions of bus engine using blends of diesel fuel with bio-diesel of sunflower or cottonseed oils derived from Greek feedstock. Fuel 2008;87:147–57. doi:10.1016/j.fuel.2007.04.011.

[29] Wang Z, Li L, Wang J, Reitz RD. Effect of biodiesel saturation on soot formation in diesel engines. Fuel 2016;175:240–8. doi:10.1016/j.fuel.2016.02.048.

[30] Adams CA, Loeper P, Krieger R, Andrie MJ, Foster DE. Effects of biodiesel-gasoline blends on gasoline direct-injection compression ignition

(GCI) combustion. Fuel 2013;111:784–90. doi:10.1016/j.fuel.2013.04.074.

[31] Putrasari Y, Lim O. A study on combustion and emission of GCI engines fueled with gasoline-biodiesel blends. Fuel 2017;189:141–54. doi: 10.1016/j.fuel.2016.10.076.

[32] Putrasari Y, LIM O. A study of a GCI engine fueled with gasoline-biodiesel blends under pilot and main injection strategies. Fuel 2018;221:269–82. doi: 10.1016/j.fuel.2018.01.063.

[33] Yang B, Li S, Zheng Z, Yao M, Cheng W. A Comparative Study on Different Dual-Fuel Combustion Modes Fuelled with Gasoline and Diesel. SAE Int 2012;694. doi:10.4271/2012-01-0694.

[34] Liu J, Shang H, Wang H, Zheng Z, Wang Q, Xue Z, et al. Investigation on partially premixed combustion fueled with gasoline and PODE blends in a multi-cylinder heavy-duty diesel engine. Fuel 2017;193:101–11. doi: 10.1016/j.fuel.2016.12.045.

[35] Cairns A, Blaxill H. The Effects of Combined Internal and External Exhaust Gas Recirculation on Gasoline Controlled Auto-Ignition 2005;2005. doi:10.4271/2005-01-0133.

[36] Zhao H, Peng Z, Williams J, Ladommatos N. Understanding the Effects of Recycled Burnt Gases on the Controlled Autoignition (CAI) Combustion in Four-Stroke Gasoline Engines 2001. doi:10.4271/2001-01-3607.

[37] Olsson J-O, Tunestål P, Ulfvik J, Johansson B. The effect of cooled EGR on emissions and performance of a turbocharged HCCI engine. Soc Automot Eng 2003;2003:21–38. doi: 10.4271/2003-01-0743.

[38] Yao M, Chen Z, Zheng Z, Zhang B, Xing Y. Effect of EGR on HCCI Combustion fuelled with Dimethyl Ether (DME) and Methanol Dual-Fuels. SAE Tech Pap 2005;2005–01–37. doi:10.4271/2005-01-3730.

[39] Sjöberg M, Dec JE, Hwang W. Thermodynamic and Chemical Effects of EGR and Its Constituents on HCCI Autoignition. SAE Tech Pap 2007;2007–01–02:776–90. doi:10.4271/2007-01-0207.

[40] Christensen M, Johansson B. Supercharged Homogeneous Charge Compression Ignition (HCCI) with Exhaust Gas Recirculation and Pilot Fuel. SAE Tech Pap 2000:SAE 2000-01-1835. doi:10.4271/2000-01-1835.

[41] Sjöberg M, Dec JE. EGR and Intake Boost for Managing HCCI Low-Temperature Heat Release over Wide Ranges of Engine Speed 2007:776–90. doi:10.4271/2007-01-0051.

[42] Saxena S, Bedoya ID. Fundamental phenomena affecting low temperature combustion and HCCI engines, high load limits and strategies for extending these limits. Prog Energy Combust Sci 2013;39:457–88. doi:10.1016/j.pecs.2013.05.002 Review.