Doctor of Philosophy

Improvement of the efficiency and emission characteristics for compression ignition engine operated in gasoline compression ignition mode fueled with gasoline-biodiesel blends

The Graduate School

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Department of Mechanical Engineering

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ABSTRACT

Improvement of the efficiency and emission characteristics for compression ignition engine operated in gasoline compression ignition mode fueled with gasoline-biodiesel blends

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Among the internal combustion engines, compression ignition (CI) engine shows the most efficient as fuel to energy conversion. The potential improvement in thermal efficiency and emission characteristics of CI engines fueled with gasoline-biodiesel blends running on GCI mode was investigated in an experimental series in this study. Fuels were injected directly into the cylinder by using common rail injection system. Commercial gasoline (GB00), diesel (D100), pure soyabean biodiesel (B100) and four gasoline-biodiesel blends (GB05, GB10, GB15, and GB20) were used in this study. The first step in conducting this research was the experimental works on an injection flow rate measurement of gasoline-biodiesel blend fuel to obtain its characteristics during injected in the cylinder. The second step was the experimental works for GCI engine testing fueled with gasoline-biodiesel blends with single injection strategy. The third step was experimental works on GCI engine fueled with gasoline biodiesel blends using main and pilot injection strategies. The last but not the least, fourth step was the experimental work on the effect of EGR and intake boosting on GCI engine combustion and emissions when fueled with gasoline-biodiesel blends. In single injection strategy, the results showed that the earlier the SOI of GB blends, the shorter the ignition delay compared to diesel fuel. Furthermore, the thermal efficiency for GB blends was found to be almost equivalent to diesel fuel for all conditions. In the case of emission, GB blends produce lower HC compared to diesel, as expected, because of their homogeneous mixing capabilities. However, a higher NOx emission from GB blends was observed, which might be a result of excess oxygen in the fuel. Using multiple injections and increasing the temperatures of the intake, oil, and engine coolant could result in improved combustion and engine efficiency. Multiple injections of GB05 showed decreased CO emissions, which could be due to the pilot injection of GB05. The biodiesel content and using gasoline as a highly volatile fuel in GB05 showed the significant effect of lowering total hydrocarbon and CO emissions. 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, NOx, and smoke emissions, respectively.

Keywords: Gasoline compression ignition (GCI), multiple injections, emission, combustion, biodiesel, EGR, boosting.

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ABBREVIATION AND NOMENCLATURES

B	Biodiesel
B100	100% biodiesel
BTDC	Before top dead center
CA	Crank angle
CA50	Combustion phasing (50% the heat release)
CI	Compression ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
COV	Coefficient of variability
CR	Compression ratio, dimensionless
D100	100% diesel
DME	Di-methyl ether
EGR	Exhaust gas recirculation
EVO	Exhaust valve open
G	Gasoline
GB	Gasoline-biodiesel
GB00	100% gasoline
GB05	Blend of 95% gasoline and 5% biodiesel

GCI	Gasoline compression ignition
GHG	Greenhouse gas
HC	Hydrocarbons
HCCI	Homogeneous charge compression ignition
HR	Heat release
HRR	Heat release rate
HTHR	High temperature heat release
IMEP	Indicated mean effective pressure
IVC	Intake valve closure
IVO	Intake valve opening
kW	Kilowatt
LPG	liquefied petroleum gas
LTHR	low temperature heat release
MPa	Mega pascal
N_2	Nitrogen gas
NOx	Nitrogen oxides
NTC	Negative temperature coefficient
PCCI	Premixed charge compression ignition

PM	Particulate matter
PPRR	Peak pressure rise rate
PRR	Pressure rise rate
RTD	Resistance temperature detector
SI	Spark ignition
SOHC	Single overhead cam
SOI	Start of injection
TDC	Top dead center
THC	Total hydrocarbons
TI	Thermal ignition
TIP	Thermal ignition preparation

a, b, c	coefficients for convective heat transfer correlation
aTDC	after top dead center
bTDC	before top dead center

C_1, C_2, C_{11}, C_{12} constants for Woschni heat transfer correlation of average cylinder gas velocity

$CHR_{j,Tt}$ contribution ratio of jth elementary reaction to heat release at T_t, dimensionless

dP/dt	pressure rise rate, MPa/ms
$HR_{j,Tt}$	heat release rate by the jth elementary reaction at T_t, J/mol·cm^3
n_{DME}	mole number of DME during one cycle, mol/cycle
N_e	engine speed, rpm
$n_{n\text{-}Butane}$	mole number of n-Butane during one cycle, mol/cycle
Nu_h	Nusselt number
P	pressure, MPa
P_c	gas pressure [Pa]
P_{in}	intake pressure, MPa
P_m	motored cylinder pressure, MPa
PM	particulate matter
P_{max}	maximum pressure, MPa
Pr	Prandtl number, dimensionless
Qin	input heat calorie, J/cycle
R	ideal gas constant, kJ/mol·K
Re	Reynolds number, dimensionless
RI	ringing intensity, kW/m^2
S_p	mean piston speed, cm/s

T_{in}	intake temperature, K
T_{max}	maximum temperature, K
V_c	gas volume [m^3]
V_d	displacement volume, cm^3
V_{in}	volume, cm^3
w	average cylinder gas velocity, m/s
X_{DME}	mixing ratio of DME, dimensionless
γ	ratio of specific heats, dimensionless
φ	equivalence ratio, dimensionless
τ	ignition delay time, s

1. INTRODUCTION

1.1 Background

Energy saving, emission reduction, environmentally friendly and sustainable energy supply are becoming issues that attract much attention nowadays. Investment in the research and development of alternative fuels and energy resources is growing rapidly in both national and international level. Many attentions on future energy supply are stimulated by the anxiety of fossil fuel stock. Global energy consumption forecasts continue to predict increasing demand for liquid hydrocarbon fuels, for the near future by internal combustion (IC) engine as it is abundant, cheap and convenient. However, large-scale and extensive use of fossil fuels consequents in two bad situations of the oil reservoir and environmental deterioration in the form of air pollution, climate changes and the gradual increase in the average global temperature[1]. Therefore, it is important to promote and develop a novel engine technology and combustion strategy that pursuit of high efficiency and clean internal combustion engines.

Efforts have been made to develop alternative and efficient powertrains for IC engines including low-temperature combustion (LTC) concept. LTC has been extensively studied as a novel combustion mode which offers the potential to reduce both NOx and particulate matter (PM) via enhanced air-fuel mixing and intake charge dilution resulting in lower peak combustion temperatures[2][3]. Gasoline compression ignition (GCI) is a new combustion mode that pertained to the extensive category of LTC strategies. In this combustion mode, a fuel with high volatility property and low reactivity characteristic, such as gasoline, burns solely by compression process[4–9]. GCI was first proposed by Kalghatgi to take advantage of good volatility and long ignition delay of gasoline fuel and the high compression ratio of a diesel engine to achieve high efficiency and low emissions simultaneously[4]. However, the lubricity of market gasoline is not adequate to protect today's fuel injection components, so either the engine components must become more robust, or the routine use of fuel lubricity additives will be needed. Furthermore, the major challenge for GCI is gasoline's very low Cetane Number (CN), which is usually estimated to be no higher than about CN15 leads to the long ignition delay and misfire. Therefore, the utilization of gasoline-diesel blends was proposed to the engine combustion strategies, except for their extended ignition delay, their changed physical properties may potentially affect the injection and spray characteristics, which are equally important for the mixture formation and engine combustion[10][11]. Later on, to achieve fuel properties that appropriate for GCI engine, to overcome the autoignition of gasoline, a certain portion of biodiesel was added to the gasoline fuel[12].

Even though some researcher did many efforts, there are still many challenges associated with GCI operation for CI engine. Due to gasoline low reactivity characteristic, lead to unstable combustion for idle- to the low-load operation, which will affect the efficiency and emissions of the engine. The efficiency of a GCI vehicle is expected to be about the same as for a diesel engine and the emission as better as SI engine. This research project deals with the improving the high efficiency and emission characteristics of CI engine using GCI mode fueled with gasoline-biodiesel blends. Understanding the important fuel properties and quantify the effects of several parameters 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.

1.2 Gasoline-biodiesel blend as an alternative fuels for CI engine

Gasoline with high volatility has low ignitability and high octane number then so-called low reactivity fuel[13]. This fuel is appropriate for spark ignition (SI) or gasoline engine. However, the main problem of this application is knocking combustion as a limitation of the SI engine, which only operates in low compression ratio around 8 to 11. The disadvantage of low compression ratio is that produces low efficiency. Then, spark plug usually required in the gasoline engine application to provide spark ignition at the end of the compression stroke. Gasoline engine usually operated in the near stoichiometric mixtures, which will develop high-temperature combustion and finally produce NOx emission[14].

Biodiesel is a chemically modified alternative fuel for use in diesel engines, derived from vegetable oils and animal fats. The definition of biodiesel, according to ASTM biodiesel standard D6751, is defined as a ‘‘fuel comprised of mono-alkyl esters of long-chain fatty acids derived from vegetable oils or animal fats, designated B100’’. There are many types of crop plant that potentially can be produced to be biodiesel such as castor, grape seed, maize, camelina, pumpkinseed, beechnut, rapeseed, lupin, pea, poppy seed, peanut, hemp, linseed, chestnut, sunflower seed, palm, olive, soybean, cottonseed, and Shea butter[15]. Biodiesel should have the same properties with diesel fuel for its chemical and physical properties. The brief explanation of physicochemical properties of biodiesel and biodiesel standards around the world is presented in Table 1.1[16]. Meanwhile, the advantages or disadvantages of biodiesel compared to diesel or petroleum fuels are as follows [16]:

Advantages of biodiesel:

- Portability, availability, and renewability of biodiesel.
- Biodiesel emits fewer emissions such as CO_2, CO, SO_2, PM and HC compared to diesel.
- Producing biodiesel is easier than diesel and is less time consuming
- Biodiesel can make the vehicle perform better as it has a Cetane number of over 100. Moreover, it prolongs engine life and reduces the need for maintenance (biodiesel has better lubricating qualities than fossil diesel).
- Owing to the clarity and the purity of biodiesel, it can be used without adding additional lubricant unlike diesel engine.
- Biodiesel hold a great potential for stimulating sustainable rural development and a solution for energy security issue.
- Biodiesel does not need to be drilled, transported, or refined like diesel.
- Biodiesel is more cost efficient then diesel because it is produced locally.
- Biodiesel is better than diesel fuel in terms of sulfur content, flash point, aromatic content and biodegradability.
- It is safer to handle, being less toxic, more biodegradable, and having a higher flash point.
- Non-flammable and non-toxic, reduces tailpipe emissions, visible smoke and noxious fumes and odors
- No required engine modification up to B20.
- Higher combustion efficiency

Disadvantages of biodiesel:

- It emits Higher NOx emission than diesel.
- Higher pour and cloud point fuel freezing in cold weather causing a cold weather starting.
- Biodiesel has a corrosive nature against copper and brass.

- The high viscosity (about 11–17 times greater than diesel fuel) due to the large molecular mass and chemical structure of vegetable oils leads to problems in pumping, combustion and atomization in the injector systems of a diesel engine.
- Biodiesel lower engine speed and power. The biodiesels on the average decrease power by 5% compared to that of diesel at rated load degradation of biodiesel under storage for prolonged periods
- Coking of injectors on piston and head of engine.
- The high viscosity, in long-term operation introduces the development of gumming, formation of injector deposits, plugging of filters, lines and injectors, ring sticking as well as incompatibility with conventional lubricating oils.
- Carbon deposits on piston and head of engine.
- Biodiesel causes excessive engine wear.
- Biodiesel is not cost-competitive with gasoline or diesel.

Table 1.1 Physicochemical properties of biodiesel and biodiesel standards around the world[16].

Properties (units)	**Malaysia**	**Indonesia** **Indonesian National Standardization Agency**	**Thailand** **E 14214**	**USA** **ASTM D6751**	**EU** **E 14214**	**Brazil** **ANP 42**
Flash point (℃)	182 min.	100 min.	120 min.	130 min.	120 min.	100 min.
Viscosity at 40℃ (cSt)	4.415	2.3- 6.0	3.5- 5	1.9- 6	3.5- 5	-
Sulphated Ash (% mass)	0.01 max.	max. 0.02	max. 0.02	0.02 max.	0.02 max.	0.02 max.
Sulphur (% mass)	0.001 min.	0.001 min.	0.001 min.	0.001 min.	0.001 min.	-
Cloud point (℃)	15.2	Max. 18	-	-	-	-
Copper corrosion (3hr, 50℃)	Class 1	Class 3	Class 1	Class 3	Class 1	Class 1
Cetane number	-	51 min.	51 min.	47 min.	51 min.	-
Water content and sediment	0.05 max.	0.05 max.	-	0.05 max.	-	0.05 max.

(vol.%)						
CCR 100% (% mass)	-	-	0.3 max.	0.05 max.	-	0.1 max.
Neutralization value (mg, KOH/gm)	-	-	-	0.05	0.05	0.08
Free glycerin (% mass)	max. 0.01	max. 0.02	max. 0.02	max. 0.02	max. 0.02	max. 0.02
Total glycerin (% mass)	max.0.01	max. 0.24	max. 0.25	max. 0.24	max. 0.25	0.38
Phosphorus (% mass)	-	max.10 ppm(mg/kg)	max. 0.001	max. 0.001	max. 0.001	-
Distillation temperature	-	<360℃	-	<360℃	-	<360℃
Oxidation stability, hrs	-	-	6	3	6	6

The LTC is a famous combustion concept in CI engines, which offers the potential to reduce NOx and particulate matter (PM). It is possible to obtain LTC in CI engine by using an appropriate modification of fuel properties, for example, lower cetane number and higher volatility fuel[17,18] such as gasoline [19], then so-called GCI engines. The longer ignition delay of gasoline with high octane number allows sufficient mixing of fuel and air, and then the thermal efficiency improved by increasing pre-mixing combustion that resulted in low NOx and soot levels[17]. The advantages of GCI are the high thermal efficiency compared to a gasoline engine, low NOx and soot emission and low combustion temperature[20]. However, the disadvantages are the required high intake temperature, low lubricity and need for high compression ratio as like diesel engine. Biodiesel has high potential to solve several problems in GCI engine application when blended with gasoline, such as the low lubricity. Furthermore, due to the high oxygen content in biodiesel, the promising complete combustion is also possible to be achieved[21].

1.3 Problem statement

Particulates matter (PM) and NOx are the problems of CI engines. These emissions can be contributors to air pollutant, which have serious environmental and health implications. The emission control technology such as after treatment systems are being developed, however, expensive, complicated and reduce the main advantage of CI engines. Moreover, limitations of vehicle emission regulations especially CI engine are getting to be more stringent in all over the world. Therefore, to develop engine combustion systems that offer high efficiency but low emissions motivate engine researchers to study the utilization of high volatile fuels, such as gasoline and alternative fuels, for CI engines. Biodiesel, which is made from various renewable resources, is known to be very suitable as a sustainable alternative fuel for CI engines[22][23]. Furthermore, biodiesel has proven to have prominent advantages in reducing engine soot emissions[24][25], because the presence of oxygen in the biodiesel plays a significant role in lowering soot formation during combustion[26].

CI engines are much more efficient than SI engines for several reasons. First, as the main reason, CI engines do not suffer from knocking at high loads and, hence, can have higher compression ratios compared to SI engines. Second, CI engines can run at part load by reducing the amount of fuel injected, rather than controlling the mass of air trapped in the cylinder. Third, in a CI engine, during the compression stroke, only air is compressed, rather than a mixture of fuel and air, which brings the performance closer to the ideal cycle efficiency. However, CI engines running on diesel fuel always produce high emissions, especially NOx and soot/smoke/particulate matter, which are difficult to control through subsequent treatments. In contrast with CI engines, SI engines running on gasoline fuel have lower engine efficiency, but also produce lower emissions, especially NOx and particulate matter. Based on these realities, a breakthrough combustion method is required to obtain high efficiency like a diesel engine, but produce lower emissions like a gasoline engine[4,27]. Recently, GCI is being considered as the most promising concept for low-temperature combustion (LTC) due to its high thermal efficiency and low emission characteristics[20,28–33]. Although homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI) also offer attractive combustion phenomena under uniformly lean conditions, the complexity of auto-ignition controllability in HCCI and the fuel injection system in PCCI makes GCI combustion more practical to overcome the aforementioned issues in the others LTC's concept of CI engine[11,12,34–37]

Engine experiments are very important to analyze and explain the real phenomena of combustion and emission characteristics of an IC engine. However, it is usually costly, need big effort and complicated, therefore only a few of researchers interested to do so in their study. Traced from the extensive research literature, only a few references on the experimental study of GCI engines and more rarely for fueled with gasoline-biodiesel blends. Most of the studies are using simulation and numerical method. Thus, to fill these lack information, by not ruling out the important role of simulation method, sequential and 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.

1.4 Objectives of the study

The objectives of this study are as follows:

(i) The primary objective of this study is to improve the efficiency and emission characteristics of GCI engine fueled with gasoline-biodiesel blends.

(ii) To investigate experimentally the effects of biodiesel content in gasoline biodiesel-blends, injection pressure, and duration on the amount of injection quantity.

(iii) To investigate experimentally the effects of a various start of injection (SOI), high compression ratio and ambient temperature of initial conditions on combustion and emission of GCI engine fueled with gasoline-biodiesel blends.

(iv) To investigate experimentally the effects of small biodiesel content of the gasoline-biodiesel blends on performance and emission of GCI engine.

(v) To investigate experimentally the effects of various EGR ratios, intake boosting, and injection strategies, i.e., single or multiple-injection mode, which consists of a pilot and main injections, in combination with increasing intake temperature of initial conditions on the combustion and emissions of a GCI engine fueled with gasoline-biodiesel blends.

1.5 Scope of the study

This thesis makes efforts to improve the efficiency and emission characteristics of CI engine fueled with gasoline-biodiesel blends using GCI mode. The scopes of the study include:

(i) Modify a stationary, single cylinder diesel engine to run under GCI mode.

(ii) Explain the gasoline-biodiesel blending process, with the content of biodiesel from 5% to 20% by volume.

(iii) Injection quantity test comparison between gasoline-biodiesel blends and diesel fuel by using different injection timing, duration, and pressure.

(iv) Engine experimental under the different operating condition such us SOI, single and multiple injection modes, increasing temperature (oil, cooling and air intake) and application EGR and boosting, fueled with gasoline-biodiesel blends and diesel fuel.

(v) EGR and intake boosting application are realized by connecting exhaust port and intake port by using adjustable gate valve and installing a supercharger in the intake manifold.

(vi) Combustion analysis for experimental works is based on cylinder pressure trace and its derivative parameters.

(vii) Thermal efficiency was calculated based on fuel consumed during the combustion process.

(viii) Measurement and analysis of regular exhaust emissions such as CO, HC, NOx, and including smoke/soot/PM.

(ix) Combustion efficiency calculated based on the CO and HC emissions.

(x) The obtained results may only valid for the engine used in this study.

1.6 Organization of the thesis

In Chapter 2, a review of the most important findings of the previous work related to the objectives of the present study is given. This review summarized the application of gasoline in CI engine by blending it with diesel fuel or others reactivity fuel, the experimental works, and simulation studies. Based on this review, some issues can be inferred as a starting point for the present study. Chapter 3 explains the detail about basic of engine testing and some of the main parameters, which should be calculated for analyzing engine performance. Other engine parameters that will be discussed as part of the interest in this study are in-cylinder pressure and its derivatives such as mean effective pressure, heat release rate, cylinder temperatures, ignition delay, combustion phasing, ringing intensity, etc. were also discussed. Chapter 4 describes the experimental setup, fuel preparation, and engine operating conditions. Engine modification, coupling the engine to the dynamometer, instrumentation, injection strategies, initial conditions modification and experimental procedure are explained briefly. Method for calculating and analyzing data is also described. Chapter 5 discusses and analyses the experimental results. The effects of biodiesel content in gasoline biodiesel-blends, injection pressure, and duration on the amount of injection quantity are discussed. The effects of the various start of injection (SOI), high compression ratio and ambient temperature of initial conditions on combustion and emission of GCI engine fueled with gasoline-biodiesel blends are analyzed accordingly. The effects of small biodiesel content of the gasoline-biodiesel blends on performance and emission of GCI engine is presented. The effects of various injection strategies, i.e., single or multiple-injection mode, which consists of a pilot and main injections, in combination with increasing intake temperature of initial conditions on the combustion and emissions of a GCI engine fueled with gasoline-biodiesel blends are investigated. Last but not the least, the effect of EGR and intake boosting are also discussed. Chapter 6 concludes with a summary, discussion, and suggestion for further research.

2. Literature review

2.1 Introduction

The purpose of this chapter is to provide a review of past research efforts related to GCI engine, utilization of gasoline-biodiesel blends for internal combustion engine fuel, and experimental study of GCI combustion. A review of other relevant research studies is also provided. The review is organized chronologically to offer insight into how past research efforts have laid the groundwork for subsequent studies, including the present research effort. The review is detailed so that the present research effort can be properly tailored to add to the present body of literature as well as to justly the scope and direction of the present research effort.

2.2 Engine classifications for automotive purpose

The common and majority of currents automotive engines use a four-stroke cycle: (1) intake stroke or induction stroke, (2) compression stroke, (3) expansion or power stroke, (4) exhaust stroke. Combustion process happened when the piston is near the end of compression stroke, or top dead center (TDC) of piston position, between stroke 2 and 3. Most automotive engines manufacturing can be categorized into either gasoline (SI) engine or diesel (CI) engines. The advanced engine combustion strategies, which is different with conventional SI, or CI engine forms are now being developed to fulfill the target of fuel economy shortly. The brief theory and explanation of SI and CI engines are reviewed in this chapter followed by the discussion of the advanced engine operating strategy as the focus of this study.

2.2.1 SI Engine

SI engines are mainly defined by a well-mixed air-fuel charge, which is created by port fuel injection at the stoichiometric air-fuel ratio. Combustion is started by using a spark[38], which proposed flame propagates through the mixture burning fuel and releasing heat. Stoichiometric air-fuel ratios allow improved engine’s operating stability and reduced cycle-to-cycle variations as the combustion is more easily started and the flame propagates faster compared to more dilute air-fuel mixtures.

The thermal efficiency of SI engine is fully relying on the compression ratio (CR) of the engine. CR is the ratio of the maximum in-cylinder volume before compression stroke to the minimum in-cylinder volume after compression stroke. SI engine normally is an Otto cycle, with thermal efficiency $\eta_{th} = 1 - (1/CR^{\gamma-1})$, where $\gamma = c_p/c_v$, which is the ratio of specific heats. The increasing of CR in an SI engine is desirable to improve the efficiency. However, these engines are limited by a phenomenon that is known as engine knock. Engine knock is indicated by auto-ignition of the unburned gases earlier of the flame front as a result of the compression heating from the expanding burned gas. Engine knock generates an increase in pressure rise rate and pressure waves that can make deterioration of the engine. One way to avoid knocking is by increasing octane number fuels, which are more resistant to engine knock and are used in high-CR engines.

Due to the air-fuel ratio is fixed at stoichiometric, the power output of SI engines is determined by the amount of intake air supplied to the cylinder. The intake air mass can be reduced by throttle settings of the airflow into the cylinder, at the expense of increased pumping work, or raised by increasing the intake pressure using a turbocharger or supercharger.

Exhaust gas emissions from SI engine contain NOx, CO, and UHC. SI engine produces high emission of NOx due to the high peak temperature from the combustion of a stoichiometric air-fuel ratio (NOx is formed through the thermal route when the temperature of combustion exceeds more than 1800K). CO and UHC emissions are produced from SI engines because the unburnt air-fuel mixture is trapped in the cold crevice regions (between the piston and piston rings) during the compression stroke and exhaust stroke when the temperature is too low for complete oxidation to occur. However, SI engines have a low emission of particulate matter (PM). One of recommended strategy to reduce NOx, CO and UHC in SI engines is by using after treatment method. Exhaust after treatment is usually completed using a three-way catalyst that: (1) reduces NOx to nitrogen and oxygen, (2) oxidizes CO to CO_2 and H_2O, and (3) oxidizes unburnt hydrocarbons to CO_2 and H_2O. The three-way catalyst works well for air-fuel ratios near stoichiometric, with the oxidation of UHC and CO preferred when there is excess O_2, and the reduction of NOx preferred when there is negligible O_2.

A developed strategy to improve SI engine efficiency is direct-injection (DI) of gasoline fuel as contrary to port fuel injection. In direct-injection SI engines, air is sucked into the cylinder, and the fuel is injected directly into the cylinder during the intake or compression strokes (early enough for evaporation and mixing to occur before the time of spark). By using direct-injection, the fuel can be maintained away from the crevices (reducing the emission of CO and UHC) and cylinder walls (reducing heat losses). Furthermore, the cooling effect of evaporation reduces the in-cylinder temperature. Thus a higher CR can be used without suffering engine knock[39].

2.2.2 CI Engine

The combustion process in a CI engine starts when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression[38]. In conventional CI engines, fresh air enters the combustion chamber during the intake stroke then compressed at compression stroke, after that fuel is injected when piston position is near TDC. The fuel injected/spray atomizes, evaporates, mixes, and auto-ignites promptly. The heat release is dominated by the speed of evaporation and mixing rather than that of flame propagation as in SI engines. The power output in diesel engines is determined by the amount of fuel injected as opposed to the amount of air in SI engines. Due to the direct injection of fuel and the quick combustion of fuel upon mixing with air, diesel engines are operated without throttle and at greatly higher CR than SI engines. By using un-throttled operation, higher CR, and globally lean air-fuel mixtures lead to improved efficiency of CI relative to SI engines. The thermal efficiency of the ideal Diesel cycle is given in Equation 1, where r_c is called the cutoff ratio and defined as the ratio of the in-cylinder volumes before and after the combustion process. It should be noted that the Otto cycle (SI) has a higher efficiency than the Diesel cycle (CI) at the same CR ($r_c = 1$ for Otto, $r_c > 1$ for Diesel). However, CR of SI engines is limited by engine knock.

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}}\left(\frac{r_c^{\gamma}-1}{\gamma(r_c-1)}\right) \tag{1}$$

CI engines are commonly operated with excess air (globally) compared to stoichiometric. Thus UHC and CO emissions are lower than SI engines. Furthermore, only air enters into the cylinder during the intake stroke, and early compression strokes, so only air (without fuel) come to the crevices. However, high in-cylinder temperatures occur due to combustion happen at the stoichiometric air-fuel ratio, generate too high NOx emissions. When running on high loads, PM and soot are also produced. Because of the excess global air, the three-way catalyst cannot be used for exhaust after treatment in CI engines. Selective catalytic reduction (SCR) can be used to reduce NOx through reaction with ammonia or urea to produce nitrogen and water, although this has the extra difficulty of carrying SCR fluid on the vehicle. PM and soot are trapped using a particulate filter, which has to be periodically re-charged by oxidizing the accumulated soot by introducing excess fuel in the exhaust. UHC and CO can be oxidized in the exhaust using an oxidation catalyst.

As long as conventional diesel engines offer attractive high efficiencies, refineries can only shift their gasoline-diesel balance a little. For this reason, gasoline-fueled engines will stay important in the global transportation sector, and research is now being conducted to develop advanced gasoline-fueled engine operating strategies.

2.3 LTC Concept

The advanced engine operating strategies target is to increase efficiency and decrease exhaust emission in IC engines. The primary research focus is the development of LTC concepts. LTC's offer the potential to reduce both NOx and particulate matter (PM) via enhanced air-fuel mixing and intake charge dilution resulting in lower peak combustion temperatures[40]. Depending on the air-fuel distribution and combustion temperatures in the cylinder at various operating points, there exist regions of excessive soot or NOx formation (also referred to as soot or NOx 'islands')[41]. The soot or NOx islands and LTC concept derivatives (HCCI) has been shown in Fig. 1.1[2].

According Loeper et al.[41] the principal and explanation of LTC based on soot or NOx islands, as shown in Fig.1, reducing soot and NOx emissions simultaneously in conventional diesel combustion has long proved challenging due to operating points that exist within these "islands." This duality of achieving either low NOx or soot formation, but not both are referred to as the soot- NOx trade-off. Low NOx formation rates can be achieved by reducing combustion temperatures, but this is traditionally accompanied by either high soot formation (due to poor air utilization and incomplete combustion) or significant restrictions in engine load (or power). Of course, soot formation can be reduced with superior air utilization but results in higher combustion temperatures and NOx formation rates. These compromises in conventional diesel combustion have spurred engine researchers to seek combustion strategies (e.g., LTC, HCCI, etc.) that avoid the NOx/soot islands altogether.

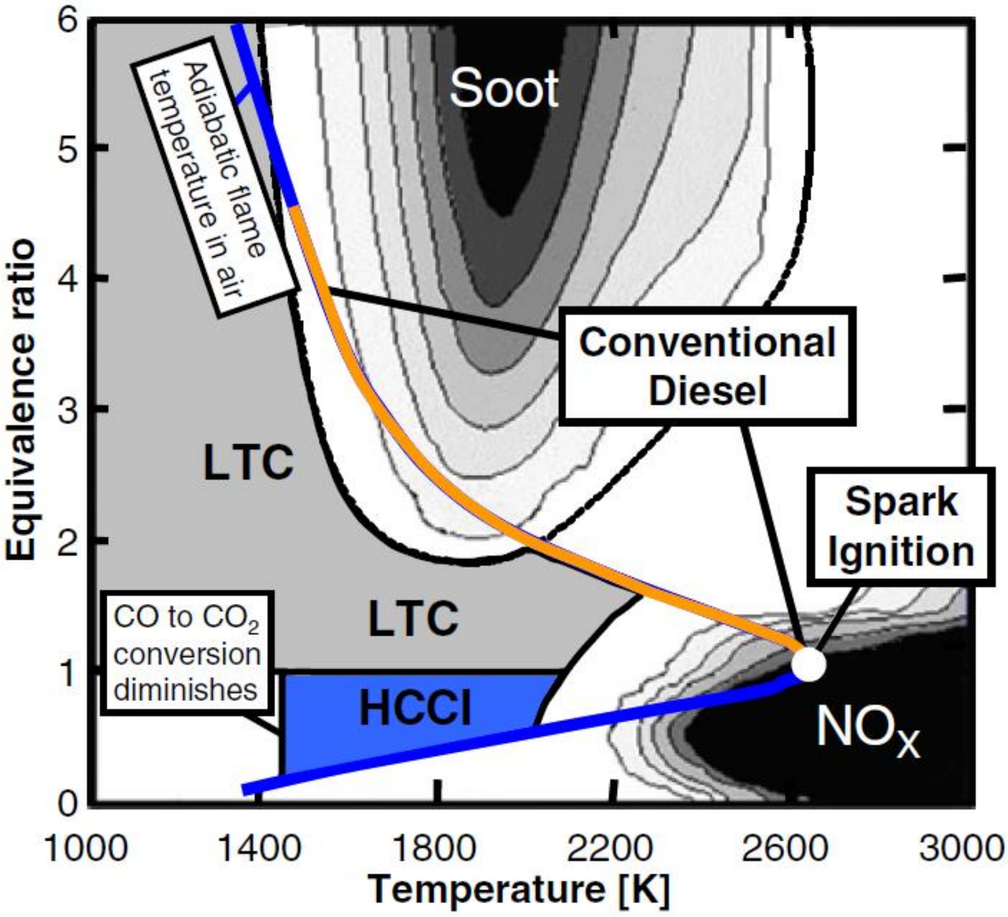


Fig. 2.1 Soot or NOx islands, LTC and conventional CI combustion regimes in ϕ -T space[2]

The most common form of LTC is HCCI combustion, which works on same fundamental principle as a 4-stroke engine and uses basic elements of CI and SI engines[42]. Homogeneous Charge Compression Ignition (HCCI) is one of the earliest forms of LTC and possibly the most widely researched[40]. In HCCI, a homogeneous of the air-fuel mixture is compressed until it auto ignite. HCCI, therefore, combines the homogenous, ultra-low soot characteristics of conventional well-mixed SI combustion with the high efficiencies (achieved through lean, unthrottled operation) usually typical of CI combustion. NOx emissions are kept low through high levels of dilution with air and residuals. Due to these factors, HCCI has demonstrated near-zero NOx and soot emissions at efficiencies similar to, or greater than, conventional diesel combustion[2]. However, HCCI is only achievable over a narrow, part-load operating range due to the lack of direct control over the start and rate of heat release. In order to provide control over heat release in HCCI, a better understanding of the fundamental combustion processes is required. A more practical LTC strategy that can solve the problems from HCCI engines is GCI, in which a low ignition quality fuel (gasoline) is directly injected into the combustion chamber using a common rail fueling system in a diesel engine[4].

2.4 GCI Engine

GCI is a new combustion mode that belongs to the wide category of LTC strategies. In this combustion strategy, a fuel with low reactivity and high volatility, such as gasoline, burns perfectly by compression[4,20,28,32,43,44]. GCI achieves same or better efficiencies than diesel engines because of throttle-less, lean, and cooler combustion operation. This combustion strategy also emits very low soot (due to better mixing of gasoline) and very low NOx emissions (due to low-temperature combustion and absence of flames) compared to traditional diesel combustion. The combustion in GCI is a result of sequential autoignition, based on reactivity stratification of fuel[45], without significant flame propagation. GCI aims to control ignition delay timing by managing air-fuel equivalence ratio stratification concentration inside the combustion chamber. The high volatile and low reactivity fuel such as gasoline fuel is ignited purely by compression, without the assistance of a spark plug, in order to prevent propagating flames, and control combustion temperatures and NOx emissions low. It has been referred in the literature by various names and terms such as gasoline direct compression ignition and gasoline direct injection compression ignition (GDICI); however, throughout this study, this mode will refer as GCI.

GCI was first studied by Kalghatgi[4] to take advantage of high volatility and long auto-ignition resistance of gasoline fuel and a high compression ratio of CI engine to achieve high efficiency and low emissions simultaneously. The GCI fuels can be categorized into two categories, high-octane fuels and low octane fuels[46]. A high-octane fuel usually refers to gasoline with octane number greater than 90. Low octane fuels have better ignitability and can be used in extended operating range to lower load if compared with high-octane fuels. The common method to obtain low octane fuels is by blending gasoline with diesel, and there is also blending with biodiesel, bioethanol and other cetane improvers[11,12,47,48].

Based on Cracknell et al.[28], the potential advantages of fuelling CI engines with gasoline as GCI combustion mode are due to the following reasons. First, CI engines have higher efficiency over SI engines and can use a broader range of fuels. Second, the ability of GCI engine concepts to use available market gasoline would allow these concepts to enter the engine application quickly without fuel constraints. Third, more gasoline consumption in passenger cars would help to rebalance several countries gasoline/diesel fuel demand on refineries and reduce GHG emissions from the fuel supply. Fourth, a successful GCI vehicle could potentially compete in predominantly gasoline markets in other parts of the world.

2.5 Review of previous studies on CI engines with gasoline fuel

The low load operation is a challenge for GCI combustion due to the high octane number of pure gasoline and its high resistance to auto-ignition at the low mixture temperatures typical of low-load operation, thus requiring several methods to obtain optimal combustion and good exhaust emission. The focus of this study is to improve the efficiency and emission characteristics of CI engine fueled with gasoline-biodiesel blends using GCI mode. Before a first attempt was made to improve the high efficiency and emission characteristics of the concept of CI engines fueled with gasoline-biodiesel blends, a literature study was performed. This study should give an overview of currently available research regarding the use of gasoline in CI engines using several combustion strategies.

At the beginning Kalghatgi et al.[4], investigated the effect of fuel auto-ignition quality, using four fuels ranging from diesel to gasoline, on such combustion at two inlet pressures and different EGR levels. The experiments were done in a single cylinder engine with a compression ratio of 14 with injection near TDC and with different EGR levels using three diesel-like fuels (CN > 30) and gasoline (RON ~ 95) at an engine speed of 1200 RPM. The engine can be easily run on gasoline with a single injection near TDC, even though it cannot be run with very early injection, in the HCCI mode, because of failure of auto-ignition or because of excessive heat release. From this study, Kalghatgi et al.[4], concluded that these results highlight the possibility of using the autoignition resistance of fuels-higher octane/lower cetane to attain high IMEP in low NOx, low smoke combustion system. However, Kalghatgi et al.[4], mentioned that further improvements in all parameters using gasoline should be possible by optimizing injector and engine design and operating conditions and bringing in other strategies such as multiple injections.

In the other study, Kalghatgi et al.[43], improved the GCI concept by proposed 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 diesel fuel. The results show that, with gasoline, pilot injection helps reduce the maximum heat release rate for a given IMEP and enables heat release to occur later with low cyclic variation compared to single injection. This enables higher mean IMEP to be reached with lower smoke, NOx and maximum heat release rate compared to single injection. One of the operating points reached with gasoline with double injection had mean IMEP of 15.95 bar (stdev. 0.112 bar), AVL smoke opacity of 0.33% (FSN < 0.07), ISNOx of 0.58 g/kWh, ISFC of 179 g/kWh, ISHC of 2.9 g/kWh, ISCO of 6.8 g/kWh and peak pressure of ~ 120 bar. However, there is still challenging for further improvements by increasing intake pressure and the EGR level and through optimization of the injection and mixture preparation strategy, e.g., more injection pulses and injector design, e.g., more holes[43].

An investigation into partially premixed combustion in a light-duty multi-cylinder diesel engine fuelled with a mixture of gasoline and diesel was conducted by Weall and Collings[49]. From their study, the results show that an increased proportion of gasoline fuel reduced smoke emissions at higher operating loads through an increase in charge premixing resulting from an increase in ignition delay and higher fuel volatility. The results of this investigation confirm that a combination of fuel properties, exhibiting higher volatility and increased ignition delay, would enable a widening of the low emission-operating regime, but that consideration must be given to combustion stability at low operating loads[49]. Furthermore, engine idling was possible using a 50% gasoline proportion test fuel, but cold start issues would emerge as a significant problem with such combustion regimes when using alternative fuels with present technology diesel cold start systems.

The effects of gasoline fumigation have been investigated experimentally in a single cylinder direct injection (DI) diesel engine by Sahin et al.[50]. In the presented study, diesel fuel has been supplied to the combustion chamber by using the original fuel injection pump–injector system. No modification in this fuel system is done, and fuel injection timing has not been changed during the tests. The gasoline has been introduced through the inlet airflow using an elementary carburetor, and no other modification on the engine has been done. The engine speeds were set 900–1600 rpm and at the selected compression ratios of 18–23. From the experimental results, it is determined that by application of gasoline fumigation effective power output increases at the levels of 4–9%, effective efficiency increases by approximately 1.5–4% and specific fuel consumption decreases by approximately 1.5–4%. However, the exhaust emissions of the engine were not analyzed.

Numerical parametric study of diesel engine operation with gasoline has been conducted by Ra et al.[6]. The results show that high-pressure direct injection gasoline engine combustion and emissions are successfully predicted and are in good agreement with available experimental measurements under various operating conditions. Gasoline has a much longer ignition delay than diesel fuel for the same combustion phasing; thus, oxides of nitrogen (NOx) and particulate emissions are significantly reduced compared to corresponding diesel-fueled cases. The results also suggest that by advancing the main injection timing in gasoline combustion, an optimal injection timing that gives minimum CO and UHC emissions was found. However, the main injection timing should be selected as a compromise between the reduced emissions and increasing pressure rise-rate levels.

The other in-cylinder fuel blending of gasoline with diesel fuel has been investigated by Curran et al.[37], by which introduced gasoline at the port-fuel-injection system. This study applied multi-cylinder diesel engine, with the parameter sweeps included gasoline-to-diesel fuel ratio, intake charge mixture temperature, in-cylinder swirl level, and diesel start of injection timing. The results of this study show that there is a strong influence of intake charge temperature on cylinder pressure rise rate, while the thermal efficiency increased along with greater than 90% decreases in NO_x and PM. However, indicated thermal efficiency for the multi-cylinder experiments were less than expected based on modeling and single-cylinder results. The lower indicated thermal efficiency as compared to the model predictions suggests a need for improved cylinder-to-cylinder control and further optimization within the dual-fuel operation.

The emission characteristics of a diesel engine operating with in-cylinder gasoline and diesel fuel blending have been studied by Prikhodko et al.[36]. In this study, the engine-out emissions of a compression ignition engine operating in RCCI regime using an in-cylinder blending of gasoline and diesel fuel have been characterized, and the results were compared to conventional diesel and diesel PCCI combustion. The results from this study show that in dual-fuel RCCI combustion, 87% and 99% reductions in NOx and PM emissions were achieved, and thermal efficiency was increased by 1.7% as compared to conventional diesel combustion. However, HC, CO, aldehyde and ketone emissions increased significantly for dual-fuel RCCI combustion compared to conventional diesel and diesel PCCI combustion.

Shi and Reitz conducted a comprehensive optimization study of a heavy-duty CI engine operated under mid- and high-load conditions and fueled with diesel, gasoline, and E10[18]. The focus of this study was optimization of injection system parameters, including the pilot and main injection timings, pressures, and amounts. CFD tool with detailed fuel chemistry was used to evaluate the engine performance and pollutant emissions. The study indicates that with an optimized injection system gasoline-like fuels are promising for heavy-duty CI engines due to their lower NOx and soot emissions and higher fuel economy compared to traditional diesel fuels. However, the high PRR associated with partially premixed combustion of gasoline-like fuels may become problematic at high-load, and the low-load operating limit is a challenge.

Experiments were carried out in two single-cylinder engines, Scania D12 and Scania D13, using a total of three different engine setups to show the advantages of using gasoline-type fuels (research octane number (RON) from 80 to 69) in a heavy-duty compression ignition engine[51]. The three engines showed similar behavior regarding efficiency, NOx, and maximum pressure rise rate, while soot was quite different. All three setups showed gross indicated efficiency between 53 and 55 percent, with a peak of 57 percent for the high-CR D12. In all the three cases NOx was below 0.30 g/ kWh, while the relative maximum pressure rise rate was below 8 bar/CAD. From this study, it is concluded that if the engine layout is kept constant the most suitable gasoline for this combustion concept should have ON in the range of 70, if higher-ON fuels are used in gasoline PPC, either a variable-CR mechanism or a glow plug must be used to run lower-load operations.

A commercial diesel fuel and a lighter fuel blend of diesel (80%) and gasoline (20%), named G20, were tested for two injection pressures (70 and 140 MPa) and injection timings in the range 11 CAD BTDC to 5 CAD ATDC in a single cylinder optically accessed high swirl multi-jets compression ignition engine[52]. Optical imaging and UV-visible detection of in-cylinder combustion phenomena were made engine operating with two different fuels and two EGR levels. The results demonstrate as the G20 fuel blend, at late injection timing and high EGR (50%), increases the ignition delay, allowing operating at late injection timing, in a partially premixed low temperature combustion (PPLTC) regime in which the fuel is completely injected before the start of combustion. In this regime, strong reduction of engine out emissions of smoke and NOx were obtained with a penalty on engine efficiency.

Ra et al[7] conducted an investigation of high-speed direct injection (DI) compression ignition (CI) engine combustion fueled with gasoline (termed GDICI for Gasoline Direct-Injection Compression Ignition) in the low temperature combustion (LTC) regime. The results showed good agreement between the experiments and the model predictions. Furthermore, due to the high volatility and low cetane index of gasoline combined with reduction of combustion temperature through utilization of EGR, both PM and NOx emissions could be reduced to levels of about 0.1 g/kg-f while maintaining experimental gross ISFC at about 180 g/kw-hr.

Another PPCI stdudy of Dieseline (blending of diesel and gasoline) on CI engine was conducted by Zhang et al[53]. In this study, a series of tests were carried out on a Euro IV light-duty common-rail diesel engine, and different engine modes, from low speed/load to middle speed/load were all tested, under which fuel blend ratios, EGR rates, injection timings and quantities were varied. From this study, the results showed that dieseline had great advantages as a PPCI combustion fuel in terms of emission reduction. Then, it was also found that with a blend of 50% gasoline in diesel, the particle numbers total concentration could be reduced by 90% while low NOx level and high brake fuel conversion efficiency (around 30%) were maintained at all the loads tested.

There is also another study on gasoline DICI engine operation in the LTC regime using triple-pulse injection by Ra et al[8]. This work aims to extend the operation ranges for a light-duty diesel engine, operating on gasoline, that have been identified in previous work via extended controllability of the injection process[7]. From this study, it was shown that combustion stability and maximum pressure rise rate can be controlled by the 2nd pulse while the 3rd pulse can be used to control engine operation load. Furthermore, using the triple-injection strategy along with EGR, both PM and NOx emissions could be reduced to levels of about 0.1 g/kg-f while achieving ISFC at about 173 g/kW-hr.

Experimental study has been carried out to compare the combustion and emission characteristics of the early and late injection highly premixed charge combustion (E-HPCC and L-HPCC respectively) modes and the blended fuel LTC mode[17]. In this study, the fuels used for the HPCC modes are 80% gasoline (by mass) which is port-fuel injected and 20% diesel, which is direct injected. From the results it can be drawn a conclusion that the fuel stratification in LTC mode leads to a fast heat release rate and high MPRR because combustion takes place in regions of higher fuel concentration so that the fuel consumption rate is higher. Because of fuel stratification, combustion with the L-HPCC and LTC may occur with local regions that are close to stoichiometric. Hence, they have relatively high NOx emissions, which could be lowered with increasing EGR rates. The soot emissions for the three modes of combustion are lower and are below the Euro 6 regulation. The HC emissions of the E-HPCC and LHPCC modes are higher than those of the LTC because the trapping of the premixed gasoline mixture in the crevice region.

Experimental investigation of light-medium load operating sensitivity in a GCI light-duty diesel engine have been comducted by Loeper et al[44]. The objective of this work is to identify andquantify the effects of variation in input parameters on overall engine operation. Input parameters including inlet temperature, inlet pressure, injection timing/duration, injection pressure, and engine speed were varied in a ~0.5L single-cylinder engine based on a production GM 1.9L 4-cylinder high-speed diesel engine. Based on the results, it can be conclude that input parameters can be effectively manipulated to maintain low NOx emissions<0.6 g/kg-fuel with good combustion stability (COV of IMEP<3%) over a wide inlet temperature range. Furthermore, optimization (with respect to combustion efficiency and CO/UHC emissions) was realized with additional adjustment of these input parameters.

Adams et al[12] studied the effects of biodiesel–gasoline blends on gasoline direct-injection compression ignition (GCI) combustion. In this study, combustion effects of two levels of biodiesel addition (5% and 10%) to gasoline are compared to neat gasoline while using a partially premixed, split-injection combustion strategy. The results show that stable combustion was achieved for all three fuels at 3 bar IMEP and 5.5 bar IMEP conditions. Then, the biodiesel content at the 5% and 10% levels significantly reduced ignition delay and therefore advanced the phasing of combustion compared with operation on neat gasoline. The most important finding in this study is that intake temperature requirements were reduced by

15 ^{o}C and 30 ^{o}C for the 5% and 10% blends respectively compared with neat gasoline at a matched phasing condition.

An experimental study has been carried out focusing on evaluating emissions, performances and combustion noise in PCCI conditions when using diesel-gasoline blends[11]. In this study, a parametrical study has been performed varying injection timing and fuel type in a High Speed Direct Injection (HSDI) Diesel engine. The results show that increasing the gasoline percentage in the fuel blend lead in an enlarged ignition delay, therefore achieving an extended mixing time between the End of Injection (EoI) and the Start of Combustion (SoC). Consequently, lower local equivalence ratios are achieved and therefore lower soot emissions are reached. Additionally, performances and the sound quality of combustion noise are improved. By contrast, NOx levels are slightly increased.

A novel combustion concept namely ‘‘multiple premixed compression ignition’’ (MPCI) in gasoline direct injection compression ignition (GDICI) regime has been proposed by Yang[54]. The mos important feature in this concept is the first premixed and followed quasi-premixed combustion processes in a sequence of ‘‘spray-combustion- spray-combustion’’ around the compression top dead center. In the experiment gasoline fuels with the research octane number (RON) of 66, 76 and 86 were tested under 1400 rpm, 0.8 MPa IMEP conditions as injection timing sweeping, without EGR and intake conditioning. The results show that compared to the single-stage diffusion combustion mode of conventional diesel engines, the RON66 MPCI mode achieves lower emissions of soot, NO, CO, as well as higher thermal efficiency, with a penalty of higher THC emissions. However, for RON76 and RON86, it is harder to realize the same performance due to a poorer auto-ignition quality than RON66.

The spray and combustion characteristics of gasoline and diesel were investigated in a direct injection compression ignition engine equipped with a common rail injection system[55]. Related to this study a series of combustion experiments was performed in order to investigate the performance and emissions in a metal engine and the flame characteristics in an optical engine. From the results, it can be inferred that the gasoline combustion created a larger amount of hydrocarbon, carbon monoxide, and comparable NOx but had a lower soot emission compared with diesel combustion. However, the NOx emission of the gasoline combustion was significantly reduced with the premixed charge compression ignition (PCCI) combustion via early injection.

Experimental and computational assessment of inlet swirl effects on GCI light-duty diesel engine have been conducted by Loeper[41]. The objective of this work was to investigate, both experimentally and computationally, GCI combustion operating sensitivity to inlet swirl ratio (Rs) variations when using a single fuel (87-octane gasoline) in a 0.475-liter single-cylinder engine based on a production GM 1.9-liter high speed diesel engine. Experimental results showed significant changes in CA50 due to changes in inlet swirl ratio. A reduction in swirl ratio (from 2.2 to 1.5) caused a 6 CAD advancement of CA50, while increasing swirl ratio (from 2.2 to 3.5) resulted in a 2 CAD retard of CA50. This advancement in CA50 at the 1.5 swirl ratio operating point was accompanied with significant increases in NOx emissions (from 0.2 to 1.6 g/kg-fuel).

Another experimental investigation of different blends of diesel and gasoline (Dieseline) in a CI Engine has been conducted by Zhang et al[14]. In this study, combustion behaviour and emissions characteristics of different blending ratios of diesel and gasoline fuels (Dieseline) were investigated in a light-duty 4-cylinder compression-ignition (CI) engine operating on partially premixed compression ignition (PPCI) mode. Three different blends, 0% (G0), 20% (G20) and 50% (G50) of gasoline mixed with diesel by volume, were studied and results were compared to the diesel-baseline with the same combustion phasing for all experiments. Results show that, compared to the diesel baseline, the total particle number concentration of G50 was reduced by up to 50% and 90% and count median diameter (CMD) was reduced by 25% and 75% at medium and low loads respectively. Then, the G50 blend had the lowest smoke emission level (0.5 FSN) for all tested load conditions. While, NOx emissions decreased by 50% at low load and increased by 20% at medium load for G50 compared with the diesel. Furthermore, dieseline led to lower in-cylinder pressure rise and heat release rate peaks at low loads and it prolonged the combustion delay by up to 7 angles compared to the diesel-baseline condition.

Characterisation of ignition delay period for a compression ignition engine operating on blended mixtures of diesel and gasoline has been conducted by Thoo et al[56]. In this work, experimental studies have been carried out on a light-duty turbocharged direct injection diesel engine with pump injection system to test diesel gasoline fuel blends comprising up to

80%vol. gasoline. The results show that ignition delay increases with the gasoline content of the blend, and generally gives rise to an increase in the premixed heat release. Higher gasoline blends retarded the start of fuel injection by up to 3 crank angle (CA) due to changes in physical properties. The change in injection timing affected combustion phasing but not ignition delay directly.

An effort which provides an analysis of how stable gasoline compression ignition (GCI) engine operation was achieved down to idle speed and load on a multi-cylinder compression ignition engine using only 87 anti-knock index (AKI) gasoline was done by Kolodziej et al[57]. The variables explored in this study to extend stable engine operation to idle included: uncooled exhaust gas recirculation (EGR), injection timing, injection pressure, and injector nozzle geometry. Although uncooled EGR was successfully able to significantly increase intake temperature, the simultaneous reduction to in-cylinder trapped mass and slight reduction of charge oxygen concentration had a more significant effect on reducing the mixture reactivity.

The combustion and emission characteristics of three low octane fuels: naphtha, the blend of gasoline and diesel (G70D30), and the blend of gasoline and n-heptane (G70H30) in Multiple Premixed Compression Ignition (MPCI) mode have been studied by Wang et al[58]. The results show that the combustion delay of the gasoline-type fuels is extended with the increase of injection pressure. Then, the soot emission decreases at high injection pressure with a penalty of higher CO and HC emissions. Furthermore, increasing the injection pressure also reduces the particle number in accumulation mode, but produces more in nucleation mode. It is also can be seen from among the test fuels, naphtha has the lowest NOx emission due to low combustion temperature but the highest CO and HC emissions. For the three fuels, there is no significant difference in particle size distribution. In term of indicated thermal efficiency, gasoline-type fuels increases with the rise of injection pressure and is higher than that of diesel at high injection pressure. The last but not the least from the results, naphtha has the highest efficiency as a result of its low heat transfer and exhaust loss.

Another experimental study on fuel economies and emissions of direct-injection premixed combustion engine fueled with gasoline/diesel blends have been conducted by Du et al[59]. In this study, the effects of gasoline/diesel blended fuel composed of diesel fuel with gasoline as additives in volume basis, on combustion, fuel economies and exhaust emissions were experimentally investigated. From this study, the results indicated that with the fraction of gasoline increasing in blends, the ignition delay was prolonged and the combustion phasing was retarded with the common injection timing. This condition led to a significant increase of premixed burning phase, which was in favor of smoke reduction; although, too much gasoline might be adverse to fuel consumption. It is also proved from this study that a combined application of EGR and blended fuel with a high gasoline fraction was confirmed effective in reducing the oxides of nitrogen and smoke emissions simultaneously at the optimum combustion phasing without giving significant penalty of fuel consumption.

The performance and emission formation of a direct injection engine fueled with gasoline/diesel blend fuel have been investigated numerically by Yang and Chou[60]. In their study, the simulations were conducted on pure diesel and its blend fuels with 10%, 20%, 30% and 40% gasoline at an engine speed of 2800 rpm under 10%, 50% and 100% loads. From the reults, it is found that the ignition delay time is extended by increasing the ratio of gasoline in blend fuels. However, this extended ignition delay has diverse effects on engine performance for different engine loads. At low load, pure diesel condition achieves a better performance; in contrast, a better performance could be realized by blend fuels at medium and high loads, though a slightly higher NOx emission level.

The other study focuses on the experimental investigation on the effect of fuel injection strategies on LTC with gasoline on a single-cylinder CI engine have been conducted by Yang et al[61]. The engine performance and emissions have been explored by single and double injections. The results indicated that the double-injection strategy enables successful expansion of high-efficiency and clean combustion region, with increasing soot, CO, and THC emissions at high loads and slightly declining combustion efficiency and indicated thermal efficiency. However, MPRR and soot emission are considered to be the predominant constraints to the load expansion of gasoline LTC, and they are related to their trade-off relationship.

The effects of injection parameters, boost, and swirl ratio on gasoline compression ignition operation at idle and low-load conditions have been studied by Kodavasal[32]. Closed-cycle computational fluid dynamics simulations using a 1/7th sector mesh representing a single cylinder of a four-cylinder 1.9 L diesel engine, operated in gasoline compression ignition mode with 87 anti-knock index (AKI) gasoline, were performed in this study. Then, two different operating conditions were studied the first is representative of idle operation (4 mg fuel/cylinder/cycle, 850 r/min), and the second is representative of a low-load condition (10 mg fuel/cylinder/cycle, 1500 r/min). Form the results, it was found that narrower nozzle inclusion angles allow for more reactivity or propensity to ignition (determined qualitatively by computing constant volume ignition delays) and are suitable over a wider range of injection timings. Meanwhile, under idle conditions, it was found that lower injection pressures helped to reduce overmixing of the fuel, resulting in greater reactivity and ignitability (ease with which ignition can be achieved) of the gasoline. However, under the low-load condition, lower injection pressures did not increase ignitability, and it is hypothesized that this is because of reduced chemical residence time resulting from longer injection durations.

The combustion process and the emissions properties of a multi-cylinder CI engine fueled with pure diesel, diesel/gasoline, diesel/n-butanol, and diesel/gasoline/n-butanol blends have been experimentally investigated by Huang et al[62]. In their study, the fuel characteristics of diesel, n-butanol and gasoline were compared and the effects of EGR were studied in CI engine. The results show that the addition of gasoline or n-butanol increase BSFC and NOx but decrease soot. Then, as EGR ratio increased, the total PM number concentrations of four fuels increased. Furthermore, gasoline or n-butanol decreases the total PM number concentrations. The most interesting finding in this research is that at 25%EGR approximately, the emissions of D70B30 reached the optimum values.

Experiments and numerical simulations to improve the fuel efficiency of compression ignition engine using a gasoline-diesel blended fuel and an optimization technology have been conducted by Lee et al[63]. Combustion and emission characteristics were investigated to explore the effects of gasoline ratio on fuel blend. The ignition delay and maximum pressure rise rate increased with the proportion of gasoline. The results show that as the gasoline fraction increased, the combustion duration and the indicated mean effective pressure decreased. The homogeneity of the fuel-air mixture was improved due to longer ignition delay. Soot emission was significantly reduced up to 90% compared to that of conventional diesel. The nitrogen oxides emissions of the blended fuel increased slightly when the start of injection was retarded toward top dead center. The last but not the least, the optimized chamber geometry enhanced the fuel efficiency, for a level of nitrogen oxides similar to that of conventional diesel over a variety of operating ranges.

The experiments in a diesel engine fueled with pure diesel, gasoline-diesel (GD) blends, and gasoline-diesel- polyoxymethylene dimethyl ethers (GDP) blends have been carried out by Liu et al[64]. The results show that both GD blends and GDP blends have single-stage premixed heat release.Then, GDP blends have shorter ignition delay, lower max pressure rise rate and COVIMEP (coefficient of variation of indicated mean effective pressure) than GD blends. Furthermore, GD blends have lower soot emissions than diesel fuel, while GDP blends have lowest soot emissions and exhibit best NOx-soot trade-off. GDP blends also have higher combustion efficiency and thermal efficiency than GD blends, even slightly higher than diesel fuel.

Another study of MPCI with mixtures of gasoline and diesel which was performed on a light-duty single cylinder diesel engine have been conducted by Wang et al[65]. The results show that compared with diesel combustion, the gasoline and diesel mixtures can reduce NOx and soot emissions simultaneously while maintaining or achieving even higher indicated thermal efficiency, but the HC and CO emissions are high for the mixtures. Meanwhile, the NOx emission of the gasoline and diesel mixture is kept less than 0.4 g/kWh with about 25% EGR and at some points the soot emission is less than 0.06 1/m, which is equivalent to 0.5FSN.

Yu et al [46] conducted an experimental investigation on the efficiency of PPCI and MPCI combustion using five gasoline-like fuels: gasoline, gasoline/ ethylhexyl-nitrate (EHN) blends, gasoline/diesel blends, gasoline/polyoxymethylene dimethyl ethers (PODEn) blends, gasoline/diesel/PODEn blends, denoted as G, GE, GD, GP, GDP, respectively. The results showed that in PPCI mode, the order of thermal efficiency from the highest to the lowest was GP, G, GDP, GE, and GD, while the order in MPCI mode was GP, GDP, GE, G, and GD. In PPCI mode, NOx and soot emissions were below 0.4 g/kWh and 0.01 g/kWh, respectively, satisfying the Euro VI emission standard. Meanwhile, NOx emission in MPCI mode can meet the requirement of Euro VI. However, the soot emission exceeded the limit because the second stage combustion was partially diffusion combustion, implying an extra Gasoline Particle Filter (GPF) was needed to meet the Euro VI standard.

Closed-cycle computational fluid dynamics (CFD) simulations have been performed by Kodavasal[45] on GCI combustion mode using a sector mesh in an effort to understand effects of model settings on simulation results. The goal of this work is to provide recommendations for grid resolution, combustion model, chemical kinetic mechanism, and turbulence model to accurately capture experimental combustion characteristics. The CFD model with the following settings: (a) 0.175 mm minimum cell size, (b) multizone chemistry (5 K/0.05 / bins), (c) RANS turbulence model and, (d) 48-species RF mechanism, was found to perform satisfactorily with respect to experiments, requiring more or less fixed calibration at all points within experimental uncertainties. Pressure rise rates and HRRs are over predicted by the model, as are NOx, CO, and HC emissions. The over prediction of HRR and NOx might be consequence of the "wellmixed" assumption for the combustion model.

2.6 Summary

This literature review has revealed the study on CI engine fueled with gasoline and/or blended with the others fuels especially running on GCI mode or its similar method. The apparent significant progress in the GCI research, both experimental works and simulation studies, has been achieved. There are still many areas that need further study due to the complexity of CI engine, which is operated in GCI combustion mode. The information on the combustion characteristics of CI engine when gasoline and GCI acts as the main fuel and the combustion mode are still limited. An extensive review showed that most research was dealt with gasoline and biodiesel as a fuel supplement/additive in CI engine.

In general, there is an apparent lack of the study on the experimental study of CI engines fueled with gasoline-biodiesel blends. The characteristics of emissions, ignition, and combustion process of the gasoline-diesel blends that greatly influences the efficiency of CI engine are well understood. However, in the CI engine in which the gasoline-biodiesel fuel is used, the ignition, combustion, efficiency and emission characteristics are not well understood.

Fuel properties and its formation play an important role in the combustion process. The experimental and simulation study on the gasoline-diesel blends mixing strategy inside the cylinder would be a good contribution to reveal the combustion of GCI engine. Some simulation results on combustion process have a good agreement with the experimental ones. However, there exist many areas, which are unaddressed by the previous simulation and experimental studies. The study on gasoline-biodiesel blends as an alternative fuel for CI engines especially for experimental work is very rare if cannot be mentioned as "never conducted". This eventually shows the limitation of the study on GCI engines fueled with gasoline-biodiesel blend especially in experimental work and opens the doors for further investigation. Combustion and emissions analysis on GCI combustion fueled with gasoline-biodiesel blend on experimental work would give a better understanding on the effort to improve the efficiency of CI engines.

To fill these gaps, some sequential experimental works will be conducted to identify the important parameters and phenomena related to combustion and emission characteristics of CI engine fueled with gasoline-biodiesel blends in order to improve its efficiency. From the literatures studied its can be arranged, planed and proposed an experiment procedures and hypotheses to obtain the high efficiency and low emissions targets which is applied in the CI engines fueled with gasoline-biodiesel blends. The brief explanation on how to obtain the targets can be summarized in the effect flowchart as seen in the Fig 2.2.The information on basic of engine testing and performance parameters that will be used to investigate the important parameters of GCI engine will be described and discussed in Chapter 3.

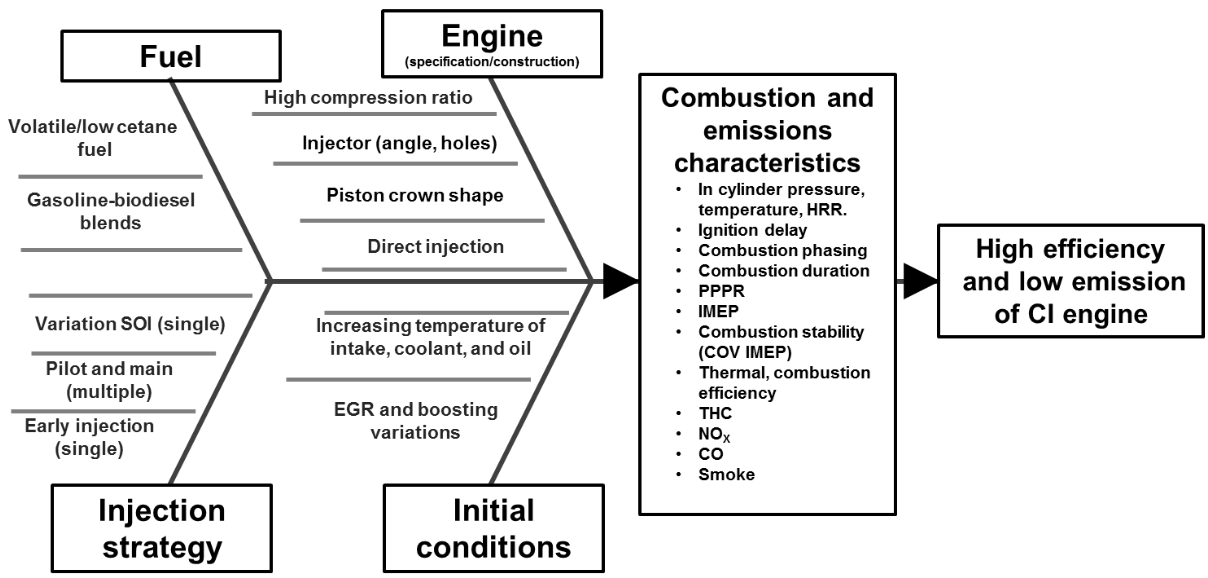


Fig. 2.2 Effect flowchart of the potential strategies to obtain high efficiency and low emission of CI engines fueled with gasoline-biodiesel blends

3. Basic of engine testing and performance parameters

In the previous chapter, a literature review on the background and history of GCI engine, the previous experiment and simulation/modelling has been discussed. The research gaps and their specific problems that will be solved have been outlined. This chapter provides the information on basic of engine testing and performance parameters that will be used to investigate the important parameters of GCI engine. This study contains sequential of experimental works as an effort to improve high efficiency of CI engines using GCI mode fueled with gasoline-biodiesel blends. The main parts of the experiments are engine performance tests, which were carried out in accordance with the SAE standard for measuring CI engines' performance and emissions. The following are explanation about basic of engine testing and some of main parameters which should be calculated for analysing engine performance. Other engine parameters that will be discussed as part of interest in this study are in-cylinder pressure and its derivatives such as mean effective pressure, heat release rate, cylinder temperatures, ignition delay, combustion phasing, ringing intensity etc.

3.1 Basic of four stroke engine

Four-stroke engine is widely used in various purposes for both transportation and industrial sectors. In every single cycle, it requires four piston motion (stroke) and two crankshaft revolutions. Here is a four-stroke's cycles based on the very common literature for CI engine [38,66]:

Intake stroke. When piston moves form TDC to BDC with the intake valve open and exhaust vale close. No fuel is added to the incoming air.

Compression stroke. Intake and exhaust valves closed and piston moves from TMB to BDC. Therefore, air inside the cylinder was compressed. Air's pressure and temperature inside cylinder increase. Late of the compression stroke, fuel is injected directly into the combustion chamber, where it mixes with the very hot air. This causes the fuel to evaporate and self-ignite, causing combustion start.

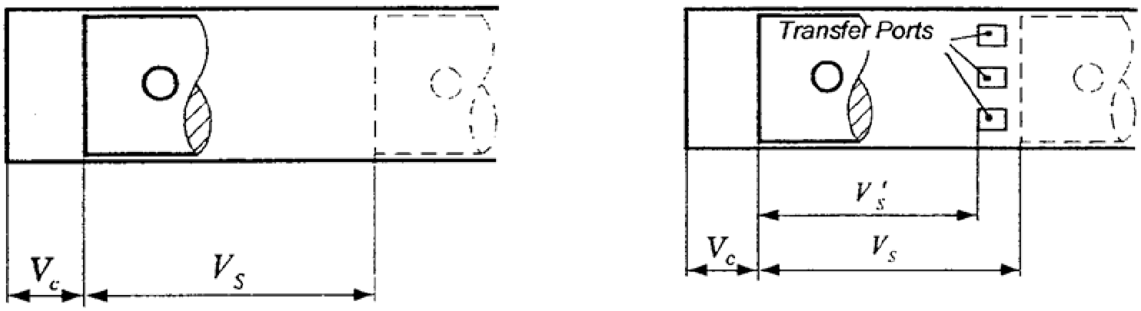
Combustion stroke. Combustion is fully developed by TDC and continues at about constant pressure until fuel injection is complete and the piston started towards BDC.

Power stroke. The power stroke continues as combustion ends and piston travels towards BDC.

Exhaust stroke. Exhaust valve is opened during piston moves towards TDC and exhaust blowdown occurs. The exhaust gas goes out from the cylinder due to higher pressure inside cylinder than outside, then pushes by piston motion.

3.2 Compression ratio

One of the important parameters in the ICE is compression ratio (CR). Higher CR means higher engine efficiency. In CI engine, CR is commonly from 12 to 22. Compression ratio is ratio between volume cylinder maximum and volume cylinder minimum, as illustrated in the Fig. 3.1.



$$r_c = \frac{V_s + V_c}{V_c} \qquad (1)$$

$$r_c = \frac{V'_s + V_c}{V_c} \qquad (2)$$

Fig. 3.1 CR left side is for four stroke, right side is for two stroke.

3.3 Torque, power and work

The engine characteristics such as speed, torque, fuel consumption, air inlet, pressure and temperature are measured on the engine test bed using tachometer, dynamometer, flowmeter, thermometer and manometer, as depicted in Fig. 3.2. From the figure its can be explained that the engine is put on the engine test bed which is the engine's shaft is connected to a dynamometer. The shaft connection mechanism can be electromagnetically, hydrolically and by friction-mechanically to the stator of dynamometer which is stabilized with a load and measured by using a load cell to obtain brake force value (F), as illustrated in Fig. 3.3.

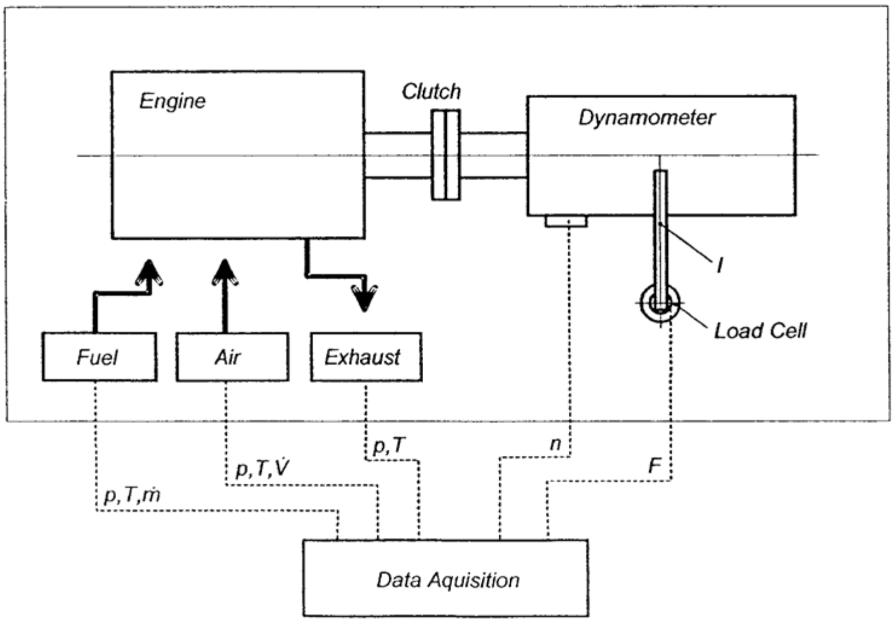


Fig. 3.2 Engine test bed to measure engine's characteristics

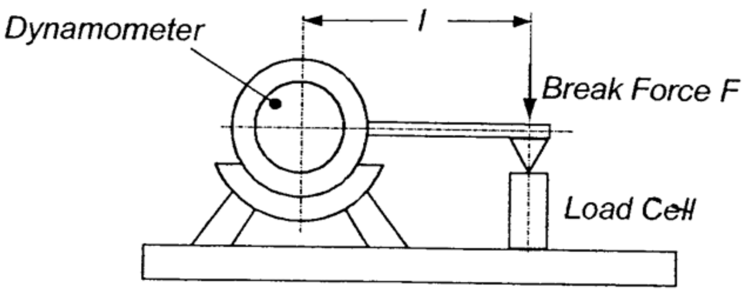


Fig. 3.3 Basic principal of dynamometer

From the figure it can be calculated an effective torque (T_e)

$$\boldsymbol{T_e = F.l} \tag{3}$$

Effective power is calculated from the torque and engine angular speed. This power is that can be transferred to be brake load.

$$P_e = T_e . \omega \tag{4}$$

Where the ω is angular speed that determined from the revolution speed of the engine. If the revolution speed in rpm, therefore:

$$P_e = T_e . 2\pi . n \tag{5}$$

Effective work on the crankshaft can be calculated from brake power divided by time.

$$W_e = \frac{P_e}{n/2} \tag{6}$$

3.4 Indicatd work

The pressure data inside the cylinder during engine operation can be used to calculate the work on the piston. Pressure versus cylinder volume can be ploted on P-V diagram, and then the work can be called as indicated work is as follows:

$$W_i = \int PdV \tag{7}$$

3.5 Mean effective pressure (MEP)

The pressure in the cylinder of and engine is continuously changing during the cycle. An average or mean effective pressure (MEP) is defined by

$$W = MEP . \Delta V \tag{8}$$

or

$$MEP = \frac{W}{\Delta V} = \frac{W}{V_d} \tag{9}$$

Where W is work of one cycle and V_d is engine displacement volume.

Mean effective pressure is a good parameter for comparing engines with regard to design or output because it is independent of both engine size and speed. If torque is used for engine comparison, a larger engine will always look better. If power is used as the comparison, speed becomes very important.

Various mean effective pressures can be defined by using different work terms. If brake work is used, brake mean effective pressure is obtained:

$$\boldsymbol{BMEP = \frac{W_b}{V_d}} \tag{10}$$

Indicated work gives indicated mean effective pressure:

$$\boldsymbol{IMEP = \frac{W_i}{V_d}} \tag{11}$$

Friction work gives friction mean effective pressure:

$$\boldsymbol{FMEP = \frac{W_f}{V_d}} \tag{12}$$

$$\boldsymbol{FMEP = IMEP - BMEP} \tag{13}$$

3.6 Indicated thermal efficiency

The indicated thermal efficiency is defined in the classical manner following the reference by Pulkrabek Willard W[38]. It can be calculated as indicated power/work divided by energy input per cycle. In this case, the power indicated can be obtained by multiplying the IMEP with displacement volume. However, other researchersalso defined the indicated thermal efficiency as the indicated fuelconversion efficiency based on the equation by Kalghatgi[67]. In this case, if all the components of the formula are similar to the Ref. [38], the indicated fuel conversion efficiency is obtained as indicated power/ work divided by energy input per cycle. The indicated thermal efficiency formula is as follows:

$$\boldsymbol{\eta_t = \frac{P_i}{Q_{in}} \times 100} \tag{14}$$

where

- $\boldsymbol{\eta_t}$ = indicated thermal efficiency [%]

- $\boldsymbol{P_i}$ = indicated work/power [J/cycle]

- $\boldsymbol{Q_{in}}$= input energy [J/cycle]

3.7 Heat release rate

The pressure inside the cylinder provides information about the thermodynamic state of charge. Using the first law of thermodynamics and several simplifying assumptions, it is possible to obtain information about the rate at which the combustion is taking place. The rate of heat release can be calculated using the following equation;

$$\frac{dQ}{d\theta} = \frac{1}{\gamma-1} V \frac{dp}{d\theta} + \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} \quad (15)$$

where γ is the specific heat ratio, p is the pressure inside the combustion chamber, and V is the instantaneous combustion chamber volume. The normal and suitable value of γ for a CI engine is 1.3.

In the experiments, the ignition timing can be defined based on the heat release data. Therefore, the combustion duration was defined based on the burning point versus crank angle. CA10 is the 10% burning point of heat release. CA50 and 90 are the 50 and 90% heat release points, respectively. It has been widely acknowledged that CA10 is the start of combustion and CA90 is the termination point of combustion. Therefore, combustion duration was considered to be the time between CA10 and CA90. Ignition delay was defined as the region between the start of injection and CA10.

3.8 Cylinder temperature

The The in-cylinder temperature can be calculated from the in-cylinder pressure and volume data based on the ideal gas law using the following equation:

$$T = \frac{p.V}{n.R} \quad (16)$$

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

3.9 Ringing intensity

The knocking level of a CI engine for low-temperature combustion can be quantified using ringing correlation based on a previous study by Eng[68]. The knocking limit is calculated using the ringing intensity formula as follow.

$$\boldsymbol{Ringing\ Intensity} \approx \frac{1}{2\gamma} \cdot \frac{\left(0.05\left(\frac{dp}{dt}\right)_{max}\right)^2}{P_{max}} \cdot \sqrt{\gamma R T_{max}} \qquad (17)$$

where

– $(dp/dt)_{max}$ = maximum value of the pressure rise rate (in real time, MPa/ms)

– P_{max} = maximum pressure (MPa)

– T_{max} = maximum temperature (K)

– γ = ratio of specific heats (C_p/C_v)

– R = gas constant in air

The ringing intensity as given by this equation is in kW/m2, which is proportional to the absolute maximum PRR (in MPa/ms) squared, and Pmax is also in MPa. The speed of sound is expressed in m/s. The use of a time variable in this equation, which is based on the pressure-rise for computing ringing/knocking, is justified by the fact that the acoustic time scales are independent of engine speed. In this study, five (5) MW/m2 was chosen as the limit value for allowable ringing.

3.10 COV of IMEP

In the further analysis, the IMEP values can be used to investigate the stability of combustion in the engine. The coefficient of a variant of IMEP (COV of IMEP), which is used generally as an index of combustion stability, is analyzed from the recorded in-cylinder pressure data.

The data is usually obtained from the calculation of 100 consecutive engines cycles. The COV of IMEP can be calculated using the formula as follows[17]:

$$\boldsymbol{COV_{IMEP}} = \frac{\mathbf{1}}{\boldsymbol{IMEP_{mean}}}\sqrt{\frac{\sum_{i=1}^{N}(\boldsymbol{IMEP_i}-\boldsymbol{IMEP_{mean}})^2}{\boldsymbol{N}-\mathbf{1}}} \tag{18}$$

Recently, the cyclic variability analysis was considered necessary to determine the stability of engine combustion by a number of researchers [69–71] A coefficient of variation (COV) was also used to evaluate the cyclic variations of CI engine combustion. COV was calculated by the following equations:

$$\boldsymbol{COV} = \frac{\boldsymbol{\sigma}}{\bar{\boldsymbol{x}}} \times \mathbf{100}\% \tag{19}$$

where σ is calculated from Equation 6. x¯is an average of x.

$$\boldsymbol{\sigma} = \sqrt{\frac{\sum_{i=1}^{n}(\boldsymbol{x_i}-\bar{\boldsymbol{x}})^2}{\boldsymbol{n}-\mathbf{1}}} \tag{20}$$

where n is the number of cycles in a sample. The combustion parameters such as cylinder peak pressure (P_{max}) and CA10 were used to quantify cycle-by-cycle variations in this study.

3.11 Combustion efficiency

The combustion efficiency commonly evaluated based on exhaust gas composition. Normally, there are still combustible species left in the exhaust gas of the engines, such as CO, HC, and unburned hydrocarbons. A high content of these species in the exhaust gas of the engine indicates combustion inefficiency. The combustion inefficiency mainly depends on the concentrations of CO and THC emissions in the exhaust gases. Therefore, the in-cylinder combustion efficiency of the engine can be calculated with strong accuracy using formula, which is related to the exhaust gas analysis based on references [66,72], as follows:

$$\boldsymbol{\eta_c} = \mathbf{1} - \frac{\sum \boldsymbol{X_i Q_{LHV_i}}}{\left(\frac{\dot{\boldsymbol{m}}_{\boldsymbol{f}}}{\dot{\boldsymbol{m}}_{\boldsymbol{a}}+\dot{\boldsymbol{m}}_{\boldsymbol{f}}}\right)\boldsymbol{Q_{LHV_i}}} \tag{21}$$

where X_i is the mass fraction of THC and CO in the exhaust gas of the engine, Q_{LHVi} is the lower heating values of these species, and Q_{LHVf} is the lower heating value of the fuel.

3.12 Summary

This chapter has explained the detail about basic of engine testing and some of main parameters, which should be calculated for analysing engine performance. Other engine parameters that will be discussed as part of interest in this study are in-cylinder pressure and its derivatives such as mean effective pressure, heat release rate, cylinder temperatures, ignition delay, combustion phasing, ringing intensity etc were also discussed.

Based on the explanation in this chapter, the experimental details and procedures will be presented and discussed in Chapter 4. Experimental setups, detail of instrumentation, and measurement method and procedures will be outlined.

4. Experimental details and procedures

4.1 Introduction

In the previous chapter, the detail about basic of engine testing and some of main parameters, which should be calculated for analysing engine performance were explained. Other engine parameters as part of interest in this study which is in-cylinder pressure and its derivatives such as mean effective pressure, heat release rate, cylinder temperatures, ignition delay, combustion phasing, ringing intensity etc were also discussed. This chapter provides the information on the engine experimental details and procedures to investigate the effort for improving the high efficiency and emission characteristics of CI engine using GCI mode fueled with gasoline-biodiesel blends. Detail of engine and test bed instrumentation, combustion analysis, and emission measurement are explained.

4.2 Experimental setup

The experimental study was conducted using an engine test bed, which is consists of several main components such as engine, dynamometer and measurement equipments to obtain engine experimental data. The detail schematic diagram and components of the test engine and measurement setup is showed in Fig. 4.1.

4.2.1 Engine specifications

A single-cylinder, four-stroke, direct injection, water-cooled, naturally aspirated diesel engine with 498 cm^3 of displacement and four-valve SOHC was used in this experiment. This engine was equipped with a cylinder pressure transducer and crank angle sensor to investigate the combustion process. There are minor modifications applied to the engine in order to run in GCI mode fueled with gasoline-biodiesel blends. A gasoline-biodiesel blend was injected directly to the engine cylinder using commonrail injection system. The commonrail injection system, which was used, is originaly diesel fuel injection system to control the flow rate of the fuel. The main specifications of the engine are summarized in Table 4.1.

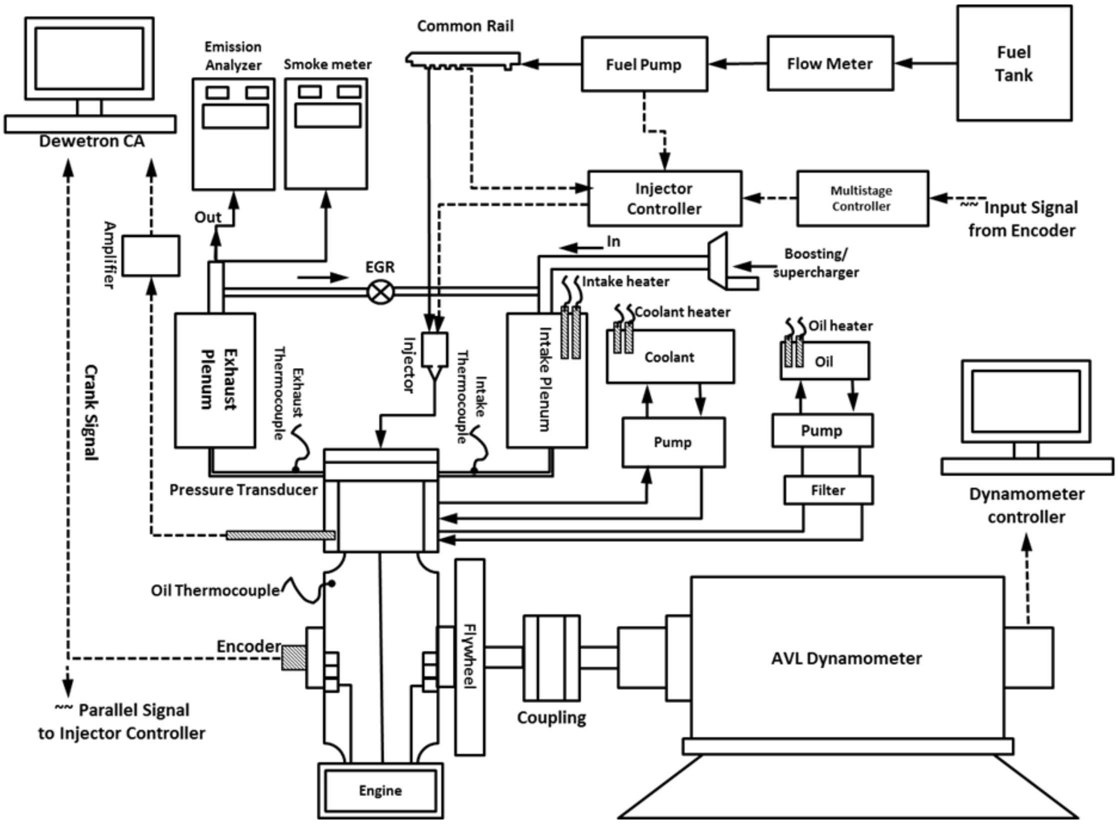


Fig. 4.1 Schematic diagram for engine and measurements system setup

Table 4.1. Engine specifications

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
IVO	7^o BTDC
IVC	43^o ATDC
EVO	52^o BBDC
EVC	6^o ATDC

4.2.2 Dynamometer

In this experiment, the engine was coupled with a 57 kW Dynamometer series 3 phase asynchrongenerator, Elin AVL Puma engine test system (MCA325MO2) as showed in Fig. 4.2. The dynamometer was used to control the engine speed, while measuring the torque (Torque) and the output (Power) over the whole ranges of speed and load. The dynamometer specifications are provided in Table 4.2.

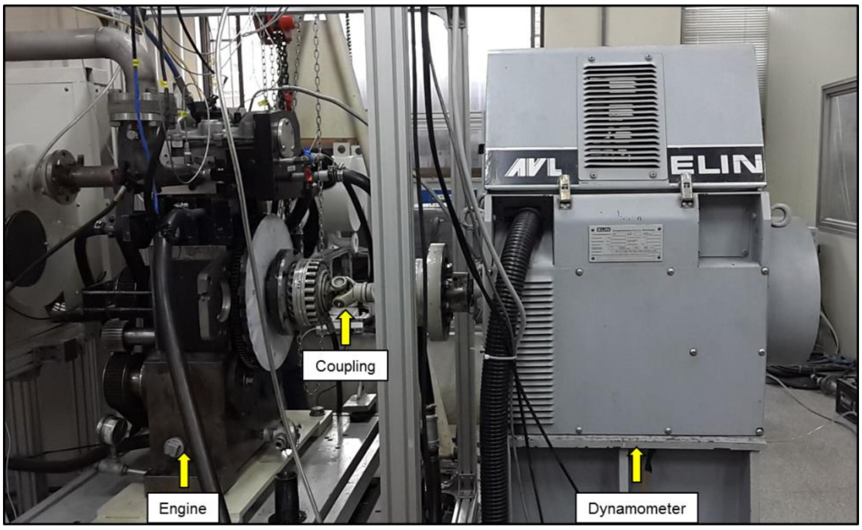


Fig. 4.2 Arrangement of engine coupled with dynamometer

Table 4.2. Dynamometer specifications

Manufacturer	AVL ELIN
Model No.	3˜ ASYNCHRONGENERATOR MCA-325M02
Weight (approximately)	600 kg
Power [kW]	0-63/57
Voltage [V3˜]	380
Current [A3˜]	112
Frequency [Hz]	0.5-67/150
Power factor	0.90
Speed [min^{-1}]	0-4000/9000
Weight [kg]	600

4.2.3 Arrangement of engine, fuel injection, and measurements systems

An Autonics E40S8-1800-3-T-24 encoder and Kistler 6056A pressure transducer combined with a Kistler 5018 amplifier were installed on the engine and connected to a Dewetron DEWE-800-CA combustion analyzer (Fig. 4.3). A Bosch seven-hole injector was used to deliver the fuel directly into the engine combustion chamber. The injector was controlled using a Zenobalti multi-stage injection engine controller (ZB-8035) combined with a common rail solenoid injector peak and hold driver (ZB-5100) and encoder interfacing box (ZB-100) to adjust the injection timing and duration. The common rail system and injection controller showed in Fig. 4. 4. 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. A standalone supercharger made by Engine Tech, a Korean local company, was used to supply the intake boosting as shown in Fig. 4.5. A conventional EGR system was used, in which the line is routed directly from exhaust manifold to the intake manifold. The EGR system was prepared by using 2 pieces of valve gate type to adjust the flow direction and ratio as shown on Fig. 4.6.

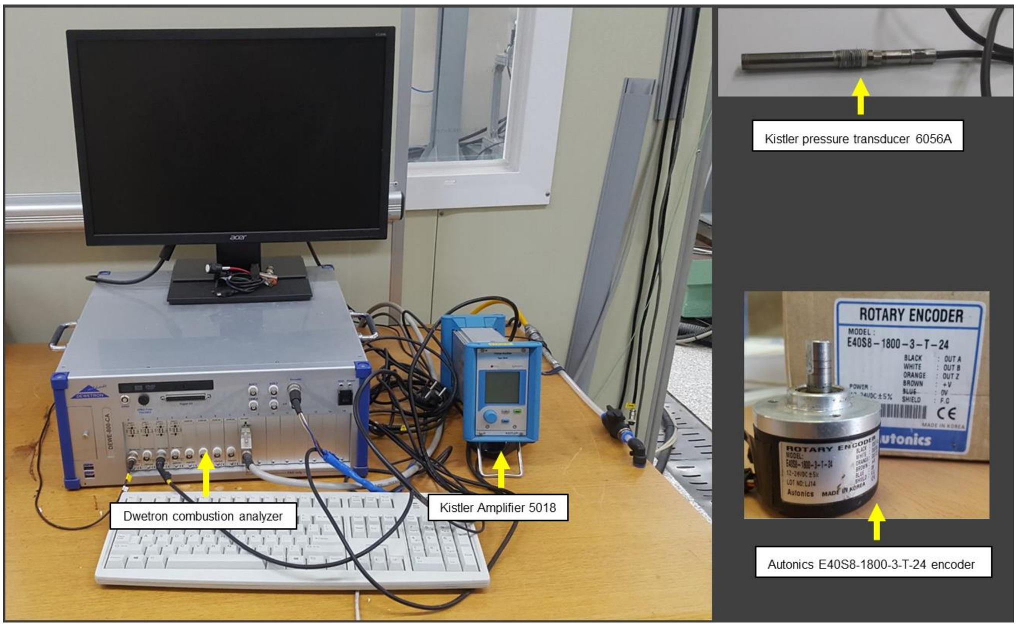


Fig. 4.3 Combustion analyzer system including pressure transducer and encoder

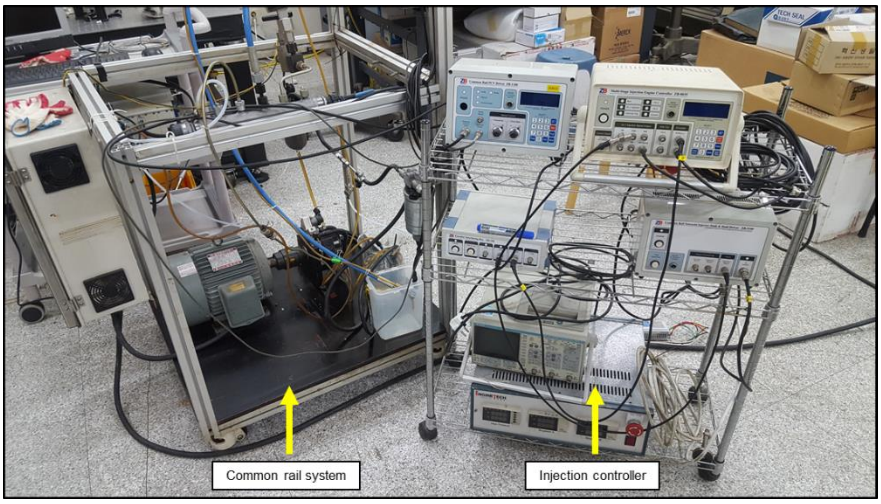


Fig. 4.4 Arrangement of fuel injection system

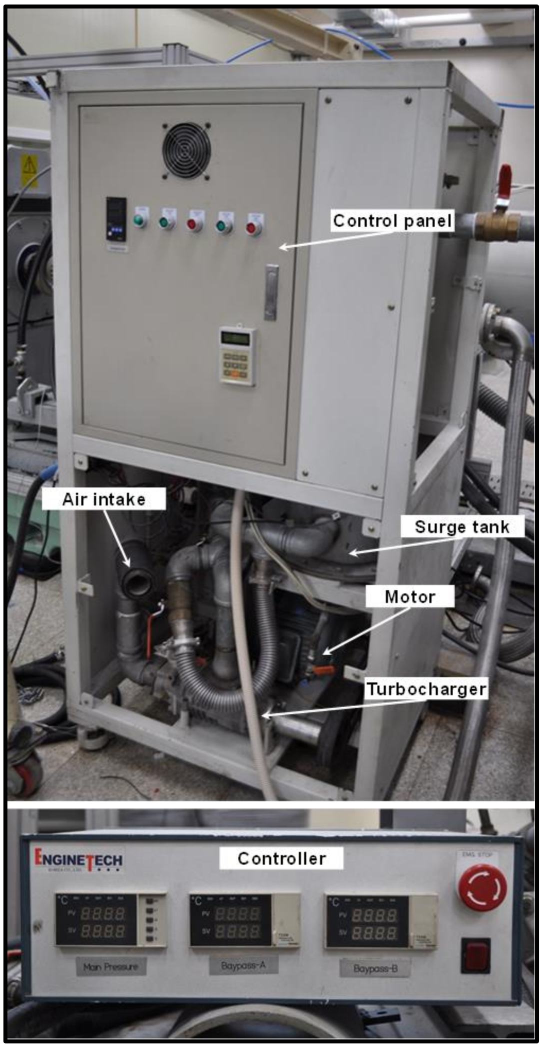


Fig. 4.5 Boosting system

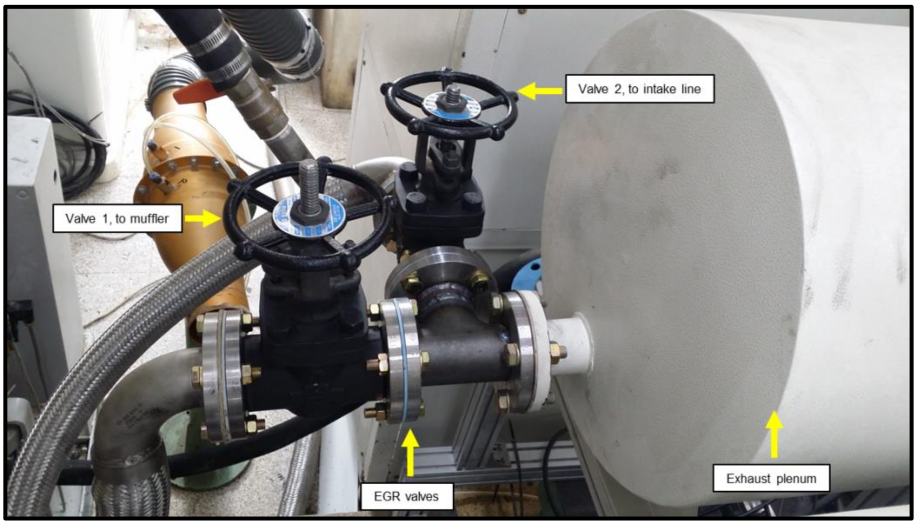


Fig. 4.6 EGR system

4.3 Experimental procedures

There are at least four factors that can be identified which are influences to the combustion and emission characteristics of CI engines fueled with gasoline-biodiesel blends, based on an effect flowchart in Fig. 2.2. These following discussions explain the experimental procedures that were conducted in this study related to the four factors to obtain high efficiency and low emission of CI engines fueled with gasoline-biodiesel blends.

4.3.1 Fuel preparation

Commercial gasoline (GB00) and diesel (D100) from a common fuel station, pure Soya bean biodiesel (B100) from an industrial source in Korea, and four gasoline-biodiesel blends (GB05, GB10, GB15, and GB20) were used in this study. The chemical composition of Soya bean vegetable oil is presented in Table 4.3[73]. The concentrations of biodiesel in the blends were set at 5, 10, 15 and 20 % by volume and called GB05, GB10, GB15 and GB20, respectively. The term "G" stands for gasoline and "B" for biodiesel, and the numeric value refers to the percentage content of biodiesel in the blends. The gasoline-biodiesel blends were prepared by a mixing/shaking process for about 2-10 minutes to produce homogeneous blends. However, the blending fuel was prepared immediately before the experiment was conducted to minimize fuel stratification. Several beaker glasses and measuring cylinders were used in the blending process. The properties of gasoline and biodiesel from laboratory tests based on an international standard are presented in Table 4.4[74].

Table 4.3 Chemical composition of soya bean vegetable oil[73]

Fatty Acid	**System Name**	**Structure**	**Formula**[a]	**Composition (wt%)**
Myristic	Tetradecanoic	14:0	C14H28O2	0
Palmitic	Hexadecanoic	16:0	C16H32O2	12
Stearic	Octadecanoic	18:0	C18H36O2	3
Arachidic	Eicosanoic	20:0	C20H40O2	0
Behenic	Docosanoic	22:0	C22H44O2	0
Lignoceric	Tetracosanoic	24:0	C24H48O2	0
Oleic	cis-9-Octadecenoic	18:1	C18H34O2	23
Linoleic	cis-9,cis-12-Octadecadienoic	18:2	C18H32O2	55
Linolenic	cis-9,cis-12, cis-15-Octadecatrienoic	18:3	C18H30O2	6
Erucic	cis-13-Docosenoic	22:1	C22H42O2	0

[a]*xx:y indicates xx carbons in the fatty acid chain with y double bonds.*

Table 4.4 Physical properties of the fuels[74]

Test Item	**Unit**	**Test method**	**Gasoline**	**GB05**	**GB10**	**GB15**	**GB20**	**B100**	**Diesel**
Heating value	MJ/kg	ASTM D240:2009	45.86	45.32	44.92	44.57	43.6	39.79	45.93
Kinematic Viscosity (40 ^{o}C)	mm^{2}/s	ISO 3104:2008	0.735	-	-	-	-	4.229	2.798
Lubricity	µm	ISO 12156-1:2012	548	290	282	252	236	189	238
Cloud Point	^{o}C	ISO 3015:2008	-57	-37	-32	-20	-16	3	-5
Pour Point	^{o}C	ASTM D6749:2002	-57	-57	-57	-57	-57	1	-9
Density (15 ^{o}C)	kg/m^{3}	ISO 12185:2003	712.7	722.3	732.2	742.6	757.1	882.3	826.3

The lubricity test of gasoline-biodiesel blends and the limitation of acceptable standar diesel based European Committee for Standardization. EN-590: automotive fuels diesel requirements and test methods can be seen in Fig. 4. 7.

4.3.2 Injection flow rate measurement

The injection flow rate was measured using a Bosch 7 hole injector attached to an injection rate-measuring vessel (Fig. 4.8). The injector was triggered internally using a Zenobalti multistage injection engine controlled ZB-8035 combined with a common rail solenoid injector peak and hold driver ZB-5100 to adjust the 18 to 75 ^{o}C A BTDC degree SOI timing, 800 µs injection duration, and 500 to 1000 engine cycles. The injection pressures were 30, 50, and 70 MPa produced from common rail and a high-pressure pump controlled by a common rail PCV driver ZB-1100 and three-phase electric motor controller, respectively. Even though the injection pressure controller’s working range is from zero up to 200 MPa, the minimum injection pressure was set at 30 MPa, corresponding to the low-pressure limit with reliable accuracy of the solenoid Bosch injector. The backpressures of 0.1 to 3 MPa were provided by connecting the vessel to a high-pressure nitrogen tank with several valves. A measuring cylinder with 0.05 ml of accuracy was used to measure the injection rate. A schematic diagram of the injection rate measurement system is shown in Fig. 4.9.

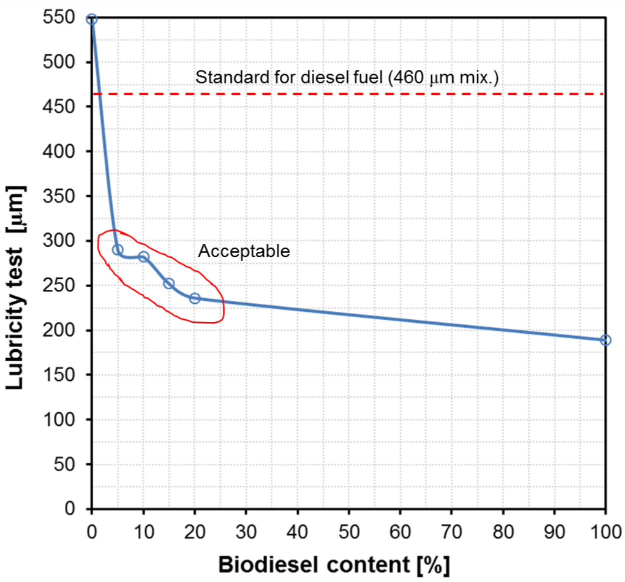


Fig. 4.7 Lubricity test of gasoline biodiesel blends

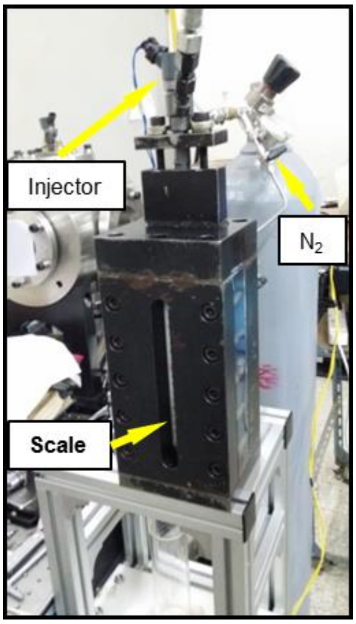


Fig. 4.8 Measuring vessel

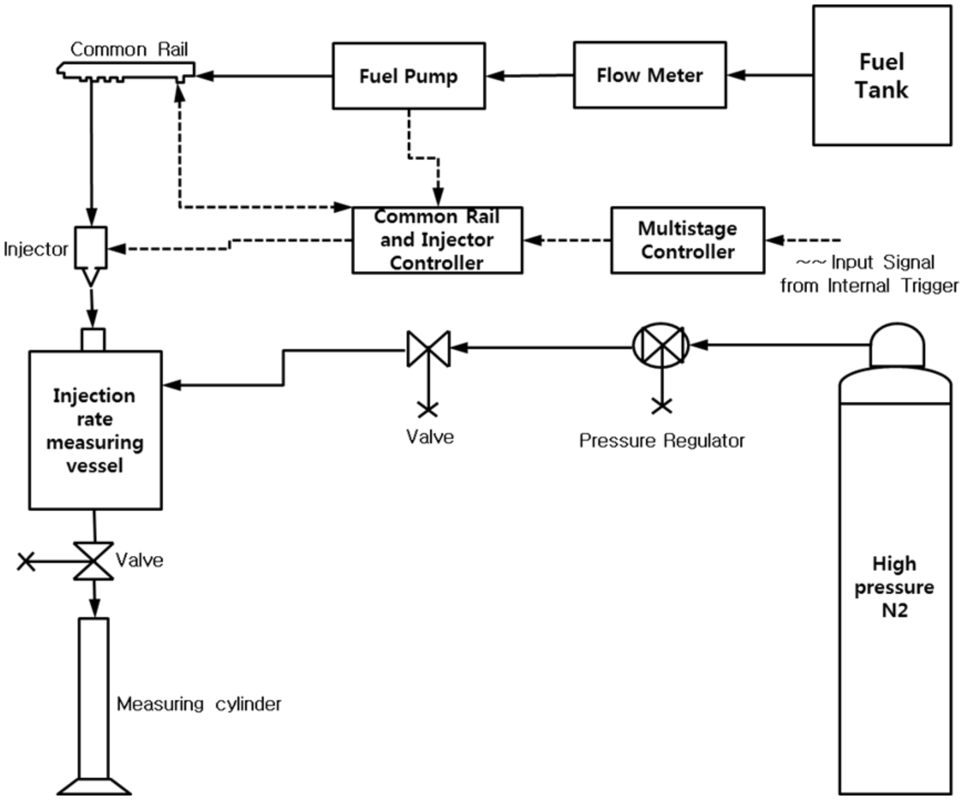


Fig. 4.9 Schematic diagram of the injection rate measuring system

4.3.3 Single injection strategy

The engine was run at 1200 rpm with an injection pressure of 70 MPa and a single injection mode for a duration of 800 μs and an SOI timing of 18 to 75 ^{o}C A BTDC. In order to avoid manipulation and maintain several initial parameters such as high intake air temperature, high engine oil temperature and high engine coolant temperature, and to make engine ignition occur easily, a high compression ratio of the engine (19.5) was used in this experiment. Meanwhile, to avoid fuel blend freezing and maintain fuel temperatures while making ignition easier, only B20 and diesel fuels were used in the engine experiment. The engine operating conditions are presented in Table 4.5. The in-cylinder pressure data was recorded for 100 engine cycles. In this step only the regular emissions of unburned total hydrocarbon and NOx were measured using a Horiba MEXA-7100DEGR.

Table 4.5. Engine operating conditions for single injection strategy

Parameters	Value
Engine speed	1200 rpm
IVO	7^{o} BTDC
IVC	43^{o} ATDC
EVO	52^{o} BBDC
EVC	6^{o} ATDC
Rail pressure	700 bar
SOI	18 to 75 BTDC
Duration	800 μs
Intake pressure	0.1 MPa
Intake temperature	298 K
EGR	0 %

4.3.4 Pilot and main injection strategies

In this sequence, the engine was operated at 1200 rpm with a common rail injection pressure of 70 MPa, single and multiple injection strategies and lean of equivalence ratio around 0.8. The injection quantity test was conducted 1000 cycles prior to the engine testing to obtain same energy input for each fuel. Based on the low heating value (LHV) of GB05 and D100 at Table 4.4, to get equal energy input for both fuels the average of total injected fuel per cycle must be 41 mg, for either single or multiple injections. Therefore, to obtain average 41 mg per cycle of injected fuel in the single injection mode at 40 ^{o}C A BTDC the injection duration for around 1200 μs was applied. Multiple injections were utilized which comprised of the pilot injection at 350 ^{o}C A BTDC for approximately 1140 μs (31 mg), followed by the main injection at 40 ^{o}C A BTDC for around 350 μs (10 mg). For diesel fuel, the initial parameters of intake temperature, oil temperature, and coolant temperature were set at 25 ^{o}C. Meanwhile, to overcome the auto-ignition problem for GB05, the initial temperatures of the intake air, oil, and coolant were maintained at 85 ^{o}C, 75 ^{o}C, and 65 ^{o}C, respectively. The intake air temperature was increased based on previous results from experts to control the auto-ignition combustion of gasoline fuel in a CI engine [20,28,44,75–78] Gasoline is more challenging to ignite at light-medium load by compression ignition [44]. However, if the inlet temperature is increased sufficiently through the use of internal EGR, this can be achieved with gasoline even at low loads [4]. The impact of inlet temperature on gasoline compression engine has been studied by several research groups. Rose et al. [20] and Cracknell et al. [28] considered intake air up to 75 ^{o}C to simulate an EGR-cooler bypass and to improved combustion stability. Loeper et al. [44] studied engine load sensitivity due to changes in inlet temperature and maintained the starting inlet temperature between 80-90 ^{o}C until PRR of 8 bar/deg was achieved. Meanwhile, Dong Han et al. [30] studied the CI engine fueled by blends of diesel and gasoline, to obtain stable combustion for all operating conditions the inlet temperatures are increased by maintained EGR temperatures at 85 ^{o}C. In this study, the intake temperature was maintained at 85 ^{o}C to obtain stable combustion, which expected to result in small effects on indicated mean effective pressure, combustion phasing, and maximum pressure rise rate. The engine operating parameters and injection strategies are presented in Table 4.6 and Table 4.7, respectively. An illustration of the injection scheme is depicted in Fig. 4.10. The in-cylinder pressure data were recorded for 100 engine cycles. The emissions of total unburned hydrocarbon, carbon monoxide, and NOx were measured using a Horiba MEXA-7100DEGR.

Table 4.6 Operating conditions

Parameters	**Diesel**	**GCI**
Speed (rpm)	1200	
Inj. Pressure (MPa)	70	
Inj. Timing	Single, Pilot and Main	
Inj. Quantity (mg)	41, 31, 10	
T intake (^{o}C)	25	85
T oil (^{o}C)	25	75
T coolant (^{o}C)	25	65

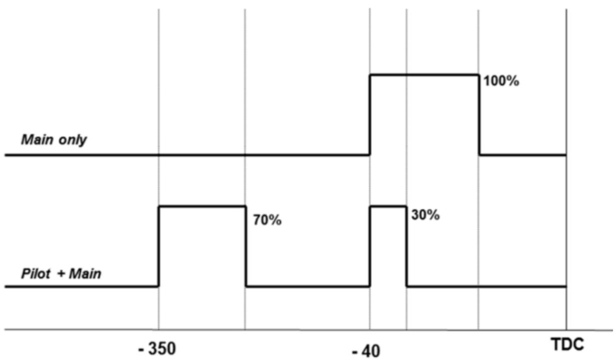


Fig. 4.10 Schematic diagram of the injection modes

Table 4.7 Injection strategies

Injection mode	Timing	D100	GB05
Pilot	350 ^{o}C A BTDC	1135 μs (31.34 mg)	1140 μs (31.36 mg)
Main	40 ^{o}C A BTDC	350 μs (9.66 mg)	350 μs (9.64 mg)
Single	40 ^{o}C A BTDC	1165 μs (41 mg)	1200 μs (41 mg)

4.3.5 Effect of EGR and intake boosting

The engine was operated at stable condition with fixed 1200 rpm. An injection pressure of 70 MPa was used both for single and for multiple injection strategy. Injection timing at 40 ^{o}CA BTDC was adopted and set for single injection combustion mode. The multiple injection was set with pilot injections timing at 350 ^{o}CA BTDC, and a main injection at 40 ^{o}CA BTDC. The total energy input of injected fuel was set at the same value around 26 mg/cycle and 16 mg/cycle for single injection and multiple injection, respectively. The initial parameters of intake temperature, oil temperature, and coolant temperature were maintained at 85 ^{o}C, 75 ^{o}C, and 65 ^{o}C, respectively. The homogeneous hot 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 Equation 1 as follows.

$$\boldsymbol{EGR}\% = \frac{\boldsymbol{m_E}}{\boldsymbol{m_E + m_i}} \times \mathbf{100}\% \tag{22}$$

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 Table 4.10 and Table 4.11, respectively. The data of 100 consecutive cycles such as in-cylinder pressure was recorded for combustion analysis.

Table 4.8 Operating parameters

Parameter	Diesel	GCI
Speed (rpm)	1200	
Inj. Pressure (MPa)	70	
Injection strategy	single and multiple injection	
Inj. Quantity (mg)	26 and 16	
T intake (^{o}C)	85	
T oil (^{o}C)	75	
T coolant (^{o}C)	65	
EGR (%)	0, 20 and 50	
Intake boosting (MPa)	0.1 and 0.12	

Table 4.9 Injection strategies

Combustion modes	Injection strategies	Injection timing and duration	Injected fuel	
			D100	**GB05**
Single injection	Single	40 ^{o}CA BTDC (1000 μs)	(26 mg)	
Multiple injection	Pilot	350 ^{o}CA BTDC (700 μs)	(16 mg)	
	Main	20 ^{o}CA BTDC (300 μs)		

4.4 Summary

This chapter has explained the detail step of the research work. Experimental setups, detail of instrumentation, and measurement method have been outlined. The first step in conducting this research was the experimental works on an injection flow rate measurementa of gasoline-bidiesel blend fuel to obtain its characteristics during injected in the cylinder. The second step was the experimental works for GCI engine testing fueled with gasoline-biodiesel blends with single injection strategy. The experiment for single injection mode swept the engine using SOI timing of 18 to 75 ^{o}C A BTDC. The third step was experimental works on GCI engine fueled with gasoline biodiesel blends using main and pilot injection strategies. In this method, a multiple injections were utilized which comprised of the pilot injection at 350 ^{o}C A BTDC for approximately 1140 μs (31 mg), followed by the main injection at 40 ^{o}C A BTDC for around 350 μs (10 mg). The last but not the least, fourth step was the experimental work on the effect of EGR and intake boosting on GCI engine combustion and emissions when fueled with gasoline-biodiesel blends. The sequential of experimental work, to improve the efficiency and emission of CI engine in GCI engine combustion, were briefly outlined.

Based on the experimental works explained in this chapter, the results will be presented and discussed in Chapter 5.

5. Results and discussion

Based on the experimental works explained in chapter 4, this chapter presents and disscuss in detail the results of the experimental series. The first part will show and discuss the effects of back pressure and injection pressure on fuel injection flow rate, and also combustion and emission characteristics when using single injection strategy. Second part will mainly focus on the analysis of pilot and main injection effect on GCI combustion and emission. While, third part will concern on the analysis of the effect of EGR and boosting on GCI engine fueled with gasoline-biodiesel blends. In the part of combustion process, in-cylinder pressure and heat release will be discussed thoroughly. Regular emission such as CO, HC and NOx will be discussed on all injection strategies. However, smoke or shoot emission phenomena will be analyzed and presented only in the last part,-the effect of EGR and boosting on GCI engine fueled with gasoline-biodiesel blends.

5.1 Single injection strategy

This experimental part discusses a GCI engine fueled with biodiesel blended into gasoline in some percentage by volume with single injection mode and variable start of injection. It is possible to obtain low temperature combustion (LTC) by using an appropriate modification of fuel properties, for example, a lower cetane number and higher volatility, such as in the GCI engine concept. The LTC concept offers the potential to reduce NOx and particulate matter (PM). The longer ignition delay of gasoline with high octane number allows sufficient mixing of fuel and air, and then the thermal efficiency is improved by increasing pre-mixing combustion that results in low NOx and soot levels.

Biodiesel has a strong potential to solve several problems in GCI engine applications, such as low lubricity when blended with gasoline. Due to the high oxygen content in biodiesel, complete combustion is also possible. Increasing the compression ratio may result in further improvements in the ignition delay, which may overcome obstacles to auto ignition. By adding biodiesel at an appropriate level to gasoline, auto ignition and combustion performance of a GCI engine can be improved. Further study on the properties of the blends is necessary, especially for stability due to the large density difference between gasoline and biodiesel. A key property of biodiesel that currently limits its application to blends of 20 % or less is its relatively poor low-temperature properties. A conventional fuel containing 25 % biodiesel blend was observed to be the best-suited blend for an engine without heating and without any engine modification. A fuel blends with less than 5 % biodiesel do not affect cold flow properties. However, it is important to understand the behavior of the fuel injector under these conditions for the different fuels. Manipulation at the start of injection is also likely to improve combustion when the CI engine operates in single injection mode. The combustion process and emission characteristics of the compression ignition engine are mainly affected by the fuel injection process and fuel atomization.

This study discusses a GCI engine fueled with biodiesel blended into gasoline in some percentage by volume with single injection mode and variable start of injection. The objectives of blending a small amount of biodiesel into gasoline are to increase cetane number, against auto ignition resistance and increase oxygen content to reduce emissions. This study uses a common rail injection system to obtain stable high-pressure fuel with a homogeneous injection process. The injection flow rate of several fuels with various injection pressures is presented and discussed in this section. Combustion characteristics such as cylinder pressure, heat release rate, combustion stability, ignition delay and emission characteristics were also discussed.

5.1.1 Effects of back pressure on fuel injection flow rate

The combustion process and emission characteristics of the compression ignition engine are mainly affected by the fuel injection process and fuel atomization. The quality of fuel spray, atomization, and flow rate are influenced by several factors, one of which is backpressure [79–81] Backpressure is a pressure which results from piston motion inside the cylinder when the engine running. In this experiment, the backpressure is represented by pressure from the N_2 gas inside of a high-pressure tank connected to an injection rate measuring vessel. The ranges of backpressures in this experiment were 0.1 to 3 MPa and were assumed to be the characteristics of in-cylinder pressure for motoring conditions. This parameter represents the estimation of in-cylinder pressure during the period from where the intake valve opens to just before the fuel injection is applied and combustion occurs. Based on the study by Desantes[80], the injection fuel flow rate into the cylinder for higher density and higher kinematic viscosity fuel resulted in a higher mass flow rate when the injector fully opens. Table 5.1 shows the injection flow rate of D100 fuel with injection pressure and different backpressures of the test rig vessel. The flow rates for backpressures of 0.1 to 3 MPa are 0.005 ml/cycle, 0.016 ml/cycle and 0.025 ml/cycle for injection pressure values of 30, 50 and 70 MPa, respectively. It can be seen that there is no difference in the amount of injection flow rate from 0.1 to 3 MPa of cylinder backpressure with injection pressures of 30, 50 and 70 MPa. This condition is suspected to be the result of the backpressure value applied in this experiment being too small. Normally, the backpressure in the cylinder affects the injection fuel flow rate, based on the previous study when the backpressure was more than 5 MPa[79–81]. Fundamentally, it is very complex to determine the process of injector flow rate; however, the main factor that influences the injection flow rate is the cavitation phenomena inside the injector nozzle. The variation in back pressure has influenced cavitation that may reduce the injection flow rate of the injector nozzle[79]. The backpressure from 0.1 to 3 MPa is indicated to be very small, therefore, the effect of backpressure is negligible since it is low compared with the injection pressure.

Table 5.1 Injection flow rate of D100 with various injection pressures and backpressures

Back Pressure (MPa)	**100% Diesel, Injection pressure**		
	30 MPa (ml/cycle)	**50 MPa (ml/cycle)**	**70 MPa (ml/cycle)**
0.1	0.005	0.016	0.025
1	0.005	0.016	0.025
2	0.005	0.016	0.025
3	0.005	0.016	0.025

5.1.2 Effects of injection pressure on fuel injection flow rate

Fig 5.1 hows the injection flow rate with injection pressures of 30, 50 and 70 MPa. Since there is no influence of backpressure below 5 MPa on injection flow rate as discussed previously, only 0.1 MPa of backpressure (ambient pressure) was applied in the next experiment. For an injection pressure of 30 MPa, the smallest average injection flow rate is for B100, which is 0.0045 ml/cycle followed by D100, which is 0.005 ml/cycle, and GB00, GB05, GB10, GB15 and GB20 with the same value are 0.006 ml/cycle. For an injection pressure of 50 MPa, the lowest was also B100, which is 0.014 ml/cycle followed by D100 which is 0.016 ml/cycle and the highest is GB00 at 0.019 ml/cycle. The same condition for injection pressure is 70 MPa, while the highest injection rate is for GB00 with a value of 0.028 ml/cycle followed by D100, which is 0.025 ml/cycle and the smallest is B100, which is 0.022 ml/cycle. From the graph in Fig. 5.1, it can be seen that the higher the injection pressure, the higher the injection flow rate. Then, in every change in injection pressure, the higher gasoline level or lower biodiesel content generates a higher injection flow rate. Consistent with the previous study about biodiesel[82,83], which was blended into petroleum-based fuel, more biodiesel in the gasoline results in a lower injection rate. This occurs due to the higher viscosity of biodiesel compared with diesel and gasoline. In this study the viscosity change of the blended fuels cannot be assumed to be linear with respect to the biodiesel concentration, since no noticeable effect on injection rate up to 20 % biodiesel occurs. This can indicates that the mixing of biodiesel and gasoline cannot be modeled as ideal mixtures.

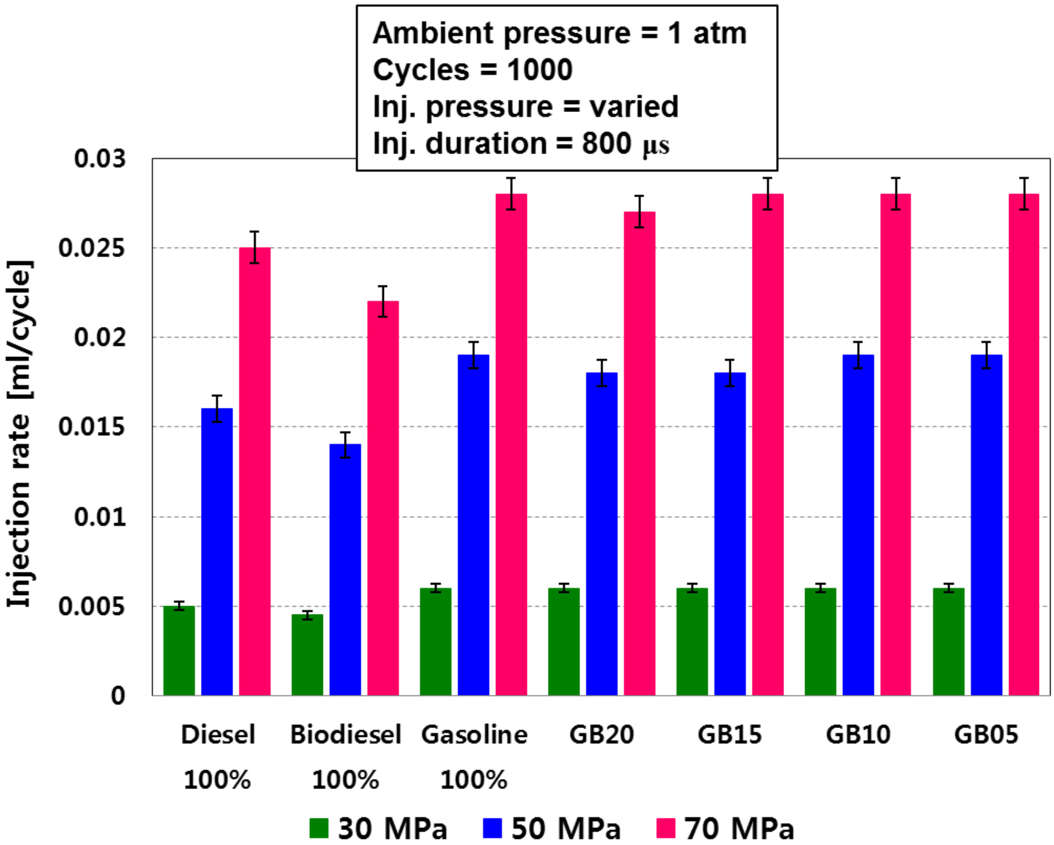


Fig. 5.1 Injection rate with various injection pressures

5.1.3 Cylinder pressure, temperature and heat release rate

Fig. 5.2 shows the in-cylinder pressure, temperature and heat release rate of D100 when the engine is operated with different SOI. The SOI range was adjusted from 18 to 65 ^{o}C A BTDC. The reason is that the combustion occurred when the engine was fueled with D100 and operated with SOI only in this range. The highest maximum in-cylinder pressure occurred when the engine operated with SOI at 40 ^{o}C A BTDC at about 7.8 MPa. Meanwhile, the lowest maximum in-cylinder pressure occurred when SOI was at 20 ^{o}C A BTDC, with the value of approximately 5.5 MPa.

The highest maximum in-cylinder temperature occurred when the SOI at 20 ^{o}C A BTDC is 2388 K and gradually decreases when SOI becomes more advanced until the lowest maximum in-cylinder temperature for SOI at 60 ^{o}C A BTDC is 1843 K. This trend indicates that the more advanced SOI will decrease the in-cylinder temperature and will potentially reduce NOx emissions. The highest HRR occurred for SOI at 20 ^{o}C A BTDC, about 35 J/deg, then the lowest maximum HRR occurred for SOI at 50 ^{o}C A BTDC up to 19 J/deg. As seen in the Fig. 5.2, a low maximum in-cylinder temperature with a high maximum in-cylinder pressure happened for SOI at 40, 50 and 60 ^{o}C A BTDC. Based on this condition, it is possible to obtain lower NOx formation but produce high engine power due to lower in-cylinder temperature and high in-cylinder pressure when combustion occurred at SOI 40, 50, and 60 ^{o}C A BTDC, as seen in the graph of Fig. 5.2 and the further discussion of NOx emissions. From the HRR graph, several engine modes with different SOI's showed a low-temperature heat release (LTHR) peak found in the combustion process then followed by the main heat release process. When the SOI retarded, for example at 20 ^{o}C A BTDC, no LTHR occurred. However, when SOI was more advanced, LTHR obviously occurred. The clearest LTHR could be observed on SOI advanced at 40, 50 and 60 ^{o}C A BTDC.

Fig. 5.3 shows the in-cylinder pressure, temperature and heat release rate of GB20 when the engine operated with different SOI. Combustion occurred when the engine fueled with GB20 and operated over an SOI range from 18 to 75 ^{o}C A BTDC. These are slight differences for D100. The combustion occurred with a wider SOI range for GB20 than D100. In this experiment, the results show that SOI was more advanced at 70 and 75 ^{o}C A BTDC for GB20 when combustion occurred. This is because adding biodiesel to gasoline fuel assumed increases the cetane number of the fuel, allowing the blending fuel (GB20) to ignite more readily even though in the ambient intake air condition compared to the 100 % gasoline. This result is consistent with the study by Adams[12]. The ignition possibilities for a gasoline biodiesel blend without increasing the intake air temperature can be improved by using a compression ratio over 17, as mentioned by Rose[20]. In this experiment, it was proved that combustion happened in a compression ratio of 19.5 was associated with an SOI range from 18 to 75 ^{o}C A BTDC. The maximum in-cylinder pressure occurred when an engine was operated with SOI at 40 ^{o}C A BTDC and about 7.5 MPa, which is slightly lower than D100. The lowest maximum in-cylinder pressure is for SOI at 20 ^{o}C A BTDC at about 5 MPa.

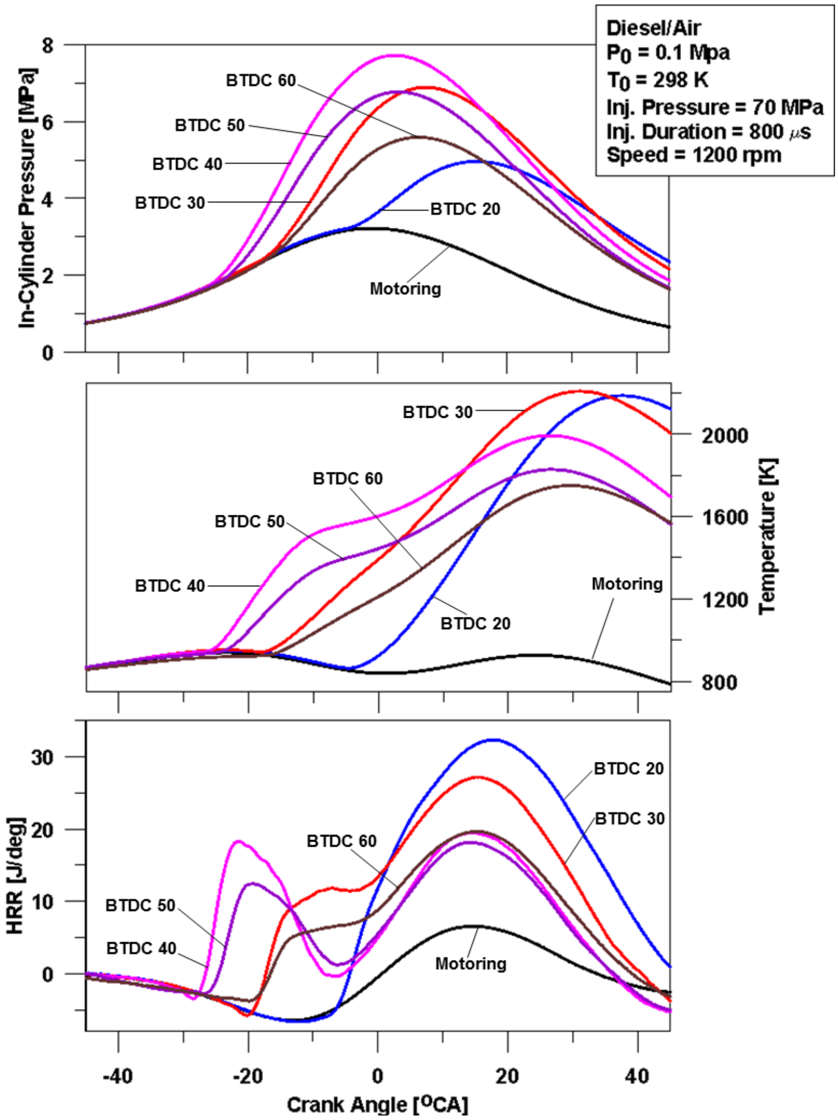


Fig. 5.2 In-cylinder pressure, temperature and heat release rate of D100

The highest in-cylinder temperature occurred when the SOI for 20 ^{o}C A BTDC was 2095 K. Meanwhile, the lowest maximum in-cylinder temperature is for SOI 70 ^{o}C A BTDC, around

1608 K. The highest maximum HRR occurred for SOI 20 ^{o}C A BTDC with the value around 35 J/deg, and the lowest is for SOI at 50 ^{o}C A BTDC at around 17 J/deg. As seen on HRR traces, LTHR also clearly occurred not only for SOI at 30, 40 and 50 ^{o}C A BTDC but also for SOI when it was more advanced at 60 and 70 ^{o}C A BTDC. Then, when the combustion phenomena of D100 and GB20 are compared to each other in detail for the in-cylinder pressure, in-cylinder temperature and HRR are all similar. In GB20, adding biodiesel into gasoline fuel may increase the cetane number of the blended state and also improve the chemical or physical properties of the blend, although more studies are needed to confirm this hypothesis.

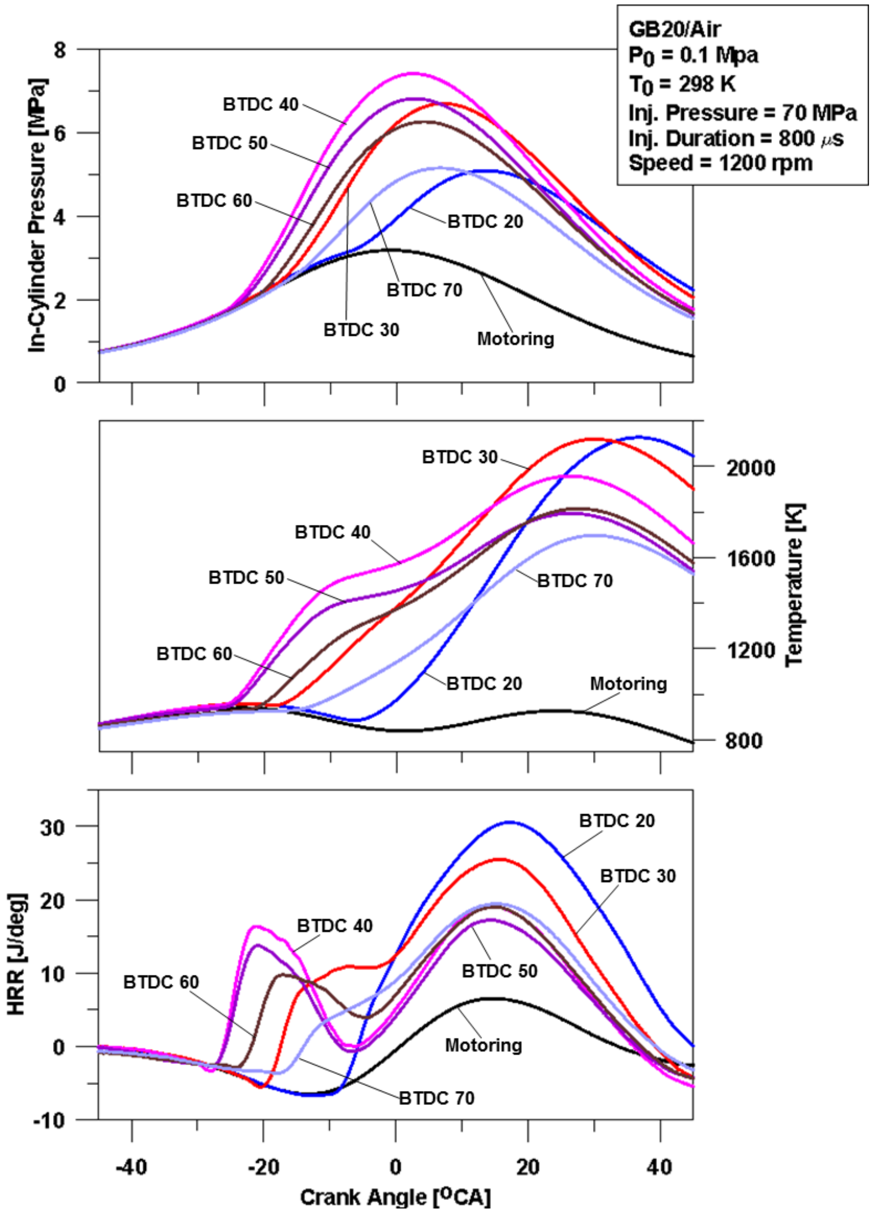


Fig. 5.3 In-cylinder pressure, temperature and heat release rate of GB20

5.1.4 Peak of pressure rise rate and combustion phasing

Fig. 5.4 shows the peak of pressure rise rate (PPRR), meanwhile, Fig. 5.5 shows the combustion phasing as measured by CA50. The PPRR of GB20 is almost the same with D100 for SOI retarded from 20 to 30 ^{o}C A BTDC. Then, the PPRR of GB20 is lower than D100 when SOI is slightly more advanced at 35 and 40 ^{o}C A BTDC. The PPRR GB20 becomes higher than D100 after SOI becomes more advanced from 50 to 75 ^{o}C A BTDC. This condition can be explained because in the gasoline biodiesel blend, there are more fuel stratifications so that when the higher gasoline mixture ignites faster than the other part of the fuel mixture, the heat release rate will be higher. The other reason is injection timing more advanced than 40 ^{o}C A BTDC will promote excessive liner spray impingement (resulting in oil dilution). Related to this, more advanced injection timing will not result in sufficient fuel for homogeneous mixture combustion, but incomplete combustion due to being undermixed (rich fuel zones resulting in incomplete combustion) or overmixed (air-fuel mixture too lean to properly ignite). Therefore, the quality of combustion become worse (possible to produce higher hydrocarbon emissions) and the thermal efficiency will be lower. This can be confirmed using combustion phasing (CA50) that also shows that the combustion phasing value would be lower. Even though both D100 and GB20 show unsatisfactory combustion results for earlier injection that is more advanced than 40 ^{o}C A BTDC, the early injection strategy will not automatically produce a sufficient fuel/air mixture before combustion. Furthermore, if it is assumed that the CN of GB20 is around 26 and D100 is around 46, the results of combustion should result in different characteristics, especially for engine noise with PPRR as one of the indicators. Normally, the lower CN of fuel will result in a longer ignition delay and cause a higher peak of pressure rise rate. However, overall, based on PPRR traces, the combustion results of GB20 and D100 are the same, even though no increase in intake temperature or EGR was applied. The effects of CN difference will be reduced if the compression temperature (usually associated with a high compression ratio) are very high. However, based on Rose[20], ignition could not be achieved at CR 17 using standard intake air pressure and temperature. Therefore, CR values higher than 17 (around 19.5) were used in this study. As seen in Fig. 5.2 and 5.3, even though the motoring trace indicated doubtful in-cylinder pressure results, the combustion of GB20 resulted from the experiment with the engine operating condition with an SOI range from 75 to 18 ^{o}C A BTDC.

The combustion phasing of GB20 seems to be more advanced than D100 for SOI at 18 to 40 ^{o}C A BTDC. However, the combustion phasing of GB20 more retards than D100 for SOI at 45 and 50 ^{o}C A BTDC. Then, the combustion phasing of GB20 becomes slightly more advanced when SOI advances from 55 to 70 ^{o}C A BTDC. The advanced combustion phasing of GB20 occurs due to the addition of biodiesel to the gasoline fuel. This result almost matches the study conducted by Adams[12].

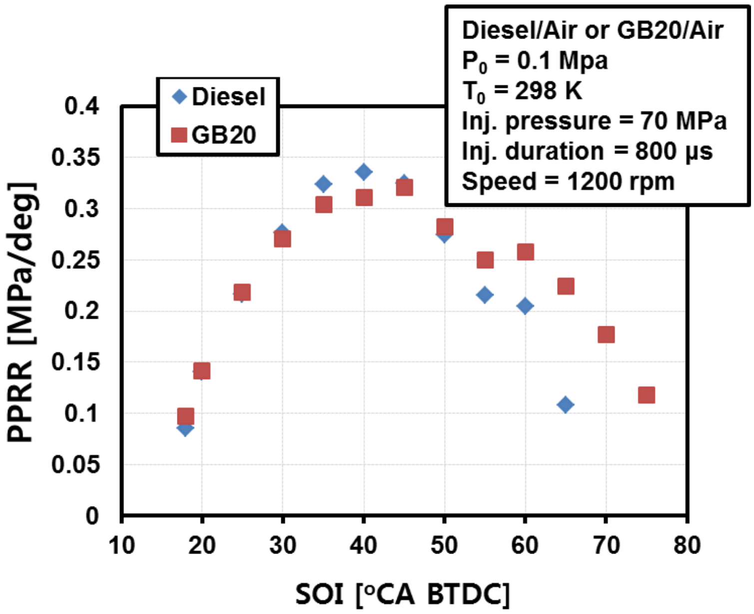


Fig. 5.4 Peak of pressure rise rate

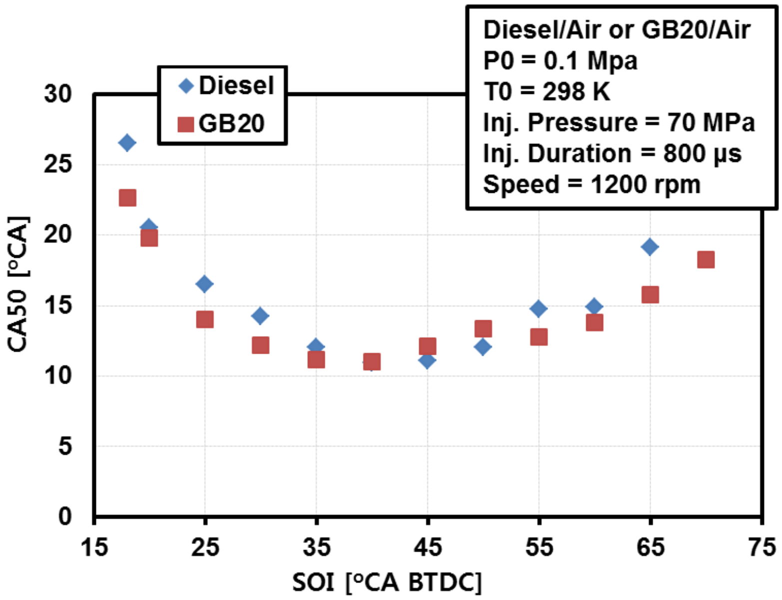


Fig. 5.5 Combustion phasing

5.1.5 Combustion duration and ignition delay

Fig. 5.6 shows the CA 10, 50 and 90 of the engine when operated using D100 and GB20 fuels. Fig. 5.7 shows the combustion duration from 10 % to 90 %, while Fig. 5.8 shows the ignition delay of D100 and GB20 with different SOI. From the figure, it can be concluded that when SOI was retarded, the combustion phasing advanced. The combustion duration for every SOI of D100 fuel and GB20 is almost the same except for the SOI range from 55 to 65 ^{o}C A BTDC. In this range, the combustion duration of GB20 is longer than for D100 fuel. The ignition delays associated with D100 fuel and GB20 are almost the same in every SOI except for an SOI range from 55 to 65 ^{o}C A BTDC. The ignition delay of GB20 is shorter than that of diesel fuel at SOI ranges from 55 to 65 ^{o}C A BTDC. The ignition delay values are consistent with the PPRR, which is the GB20 with SOI in the range from 55 to 65 ^{o}C A BTDC. The ignition delay is shorter than D100. However, the combustion duration shows a different trend in that the combustion duration of GB20 is longer than that of D100 for the advanced SOI range from 55 to 65 ^{o}C A BTDC. The shorter ignition delay and the advanced combustion phasing result in higher bulk in-cylinder temperatures that may increase NOx emissions following combustion inside the cylinder.

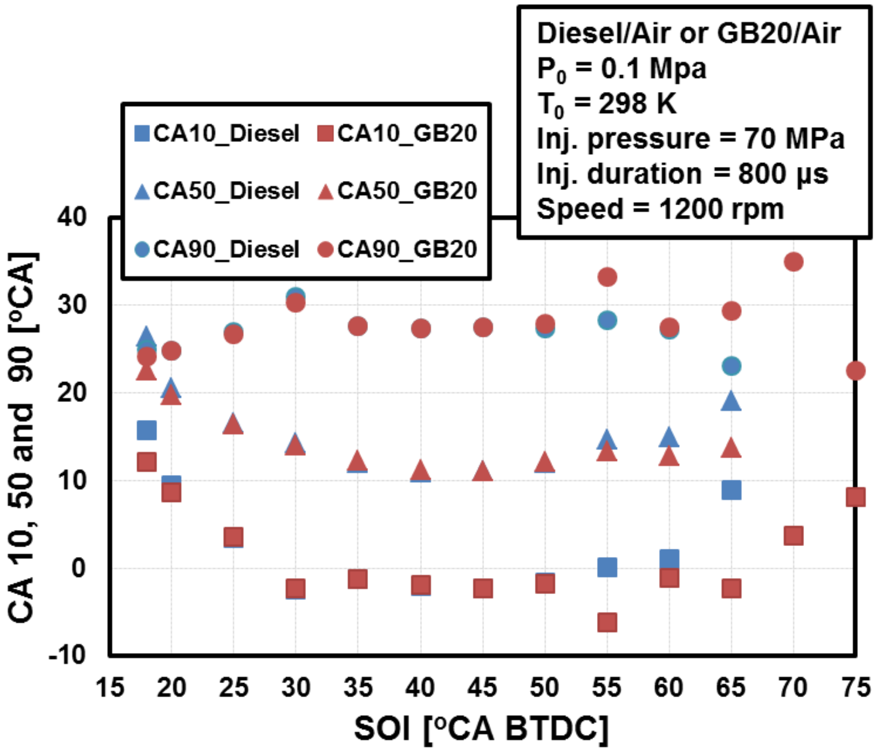


Fig. 5.6 CA10, CA50, and CA90

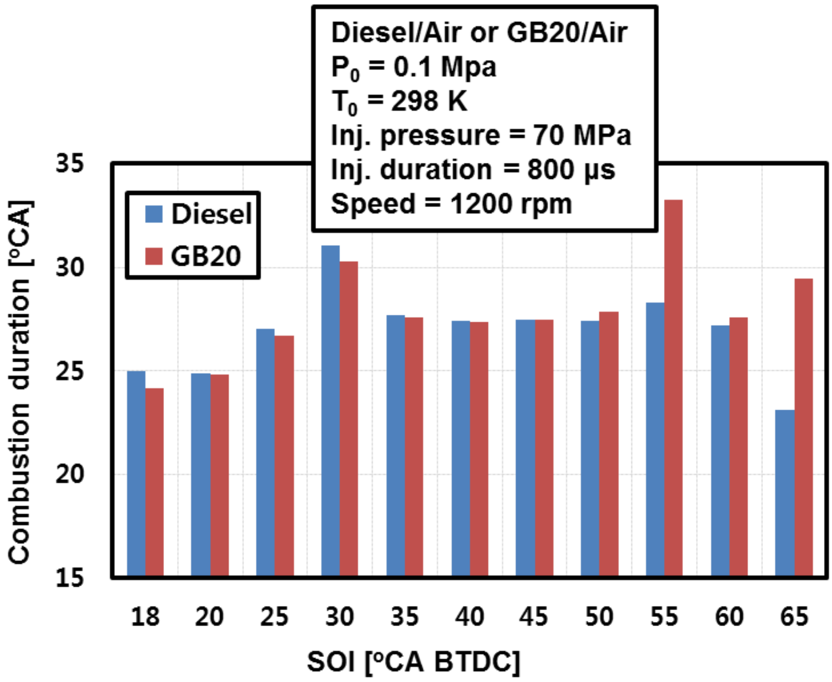


Fig. 5.7 10-90 % burn duration

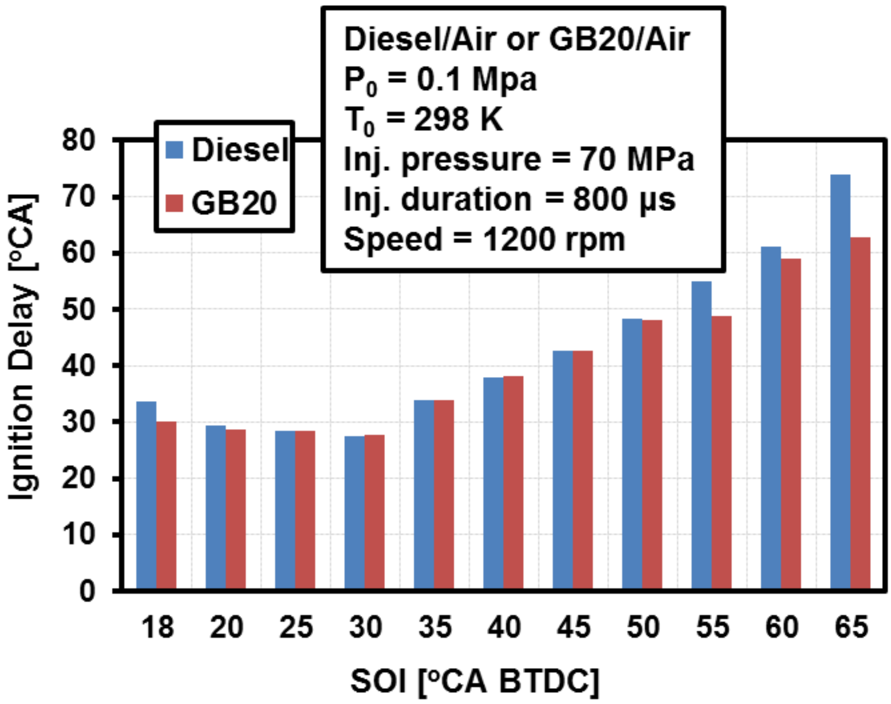


Fig. 5.8 Ignition delay

The burning duration at 55 ^{o}C A of GB20 is long, which means that the mixing-controlled combustion process is longer and may increase HC or soot. The same condition shows a shorter ignition delay, meaning that the GB20 containing biodiesel has a lower calorimetry value. The fatty acid composition of biodiesel has been identified as the main element for shorter ignition delay [84]. The lower compressibility and viscosity of GB20 leads to the advanced start of injection (shorter ignition delay). This relation shows that by adding some biodiesel into a gasoline reduced ignition delay, due to the increase of the cetane number, the combustion duration increase can be used to predict whether higher HC or soot will result.

The low CN of gasoline causes longer ignition delays or more resistance to autoignition. To a certain extent, this behavior is profitable because it provides more time for fuel mixing after injection and before combustion starts, therefore allowing combustion to take place near TDC. However, too long ignition delay could result in delayed combustion phasing, leading to an inefficient engine work or low thermal efficiency. Therefore, to overcome the too-long ignition delays, more reactive fuel (more reactive stratification fuel) is necessary in the gasoline. In the gasoline biodiesel blend fuel, there are more fuel stratifications, so that when the higher biodiesel content of the gasoline mixture ignites faster than the other part of the fuel mixture, it will also influence the ignition delay, combustion phasing and combustion duration. The shorter ignition delay of GB20 compared to D100 in this experiment is associated with the higher reactivity of biodiesel in the blend.

5.1.6 IMEP, COV-IMEP and thermal efficiency

Fig. 5.9 shows the indicated mean effective pressure (IMEP) of the engine when operated using D100 and GB20 fuels. The IMEP value obtained from the calculation involves dividing the indicated work per cycle by the displacement volume according to Pulkrabek[38] and Heywood[66]. Fig 5.10 shows the coefficient of variability (COV) of IMEP, and the thermal efficiency is presented in Fig. 5.11. From the figures, it can be concluded that when the SOI range is retarded from 18 to 40 ^{o}C A BTDC, the IMEP of D100 is higher than GB20. Meanwhile, the advanced SOI for 65 ^{o}C A BTDC with an IMEP of D100 is lower than GB20. The combustion phenomena of the engine fueled with D100 and GB20 is in the stable condition for every SOI range except D100 at 65 ^{o}C A BTDC based on the COV of IMEP calculation. It was known that the combustion stability was represented by the coefficient of variability (COV) of IMEP.

Some references use a combustion stability limit for 5 % of COV[12][17]; however, to improve the stability limit analysis used in this study, 3 % of COV of IMEP was used[44]. The thermal efficiency of the engine shows that in the retarded SOI range from 18 to 35 ^{o}C A BTDC, the D100 fuel is higher than GB20. Meanwhile, when the SOI ranges were advanced from 55 to 65 ^{o}C A BTDC, the thermal efficiency of D100 is lower than GB20. The higher thermal efficiency of D100 in the retarded SOI is due to the higher LHV of the diesel fuel compared to GB20. However, in some cases of advanced SOI, the thermal efficiency of GB20 is higher than D100. This condition can explain the sensitivity of the engine to volatile fuel, which in this case, is the gasoline content in the blend that caused the chemical and physical properties of the fuel to have a significant effect on the combustion process. Because of the high gasoline volatility in the GB20 and the earlier or retarded injection timing, the fuel air mixture has enough time to become a more homogeneous fuel-air mixture. Therefore, a complete combustion process will be achieved.

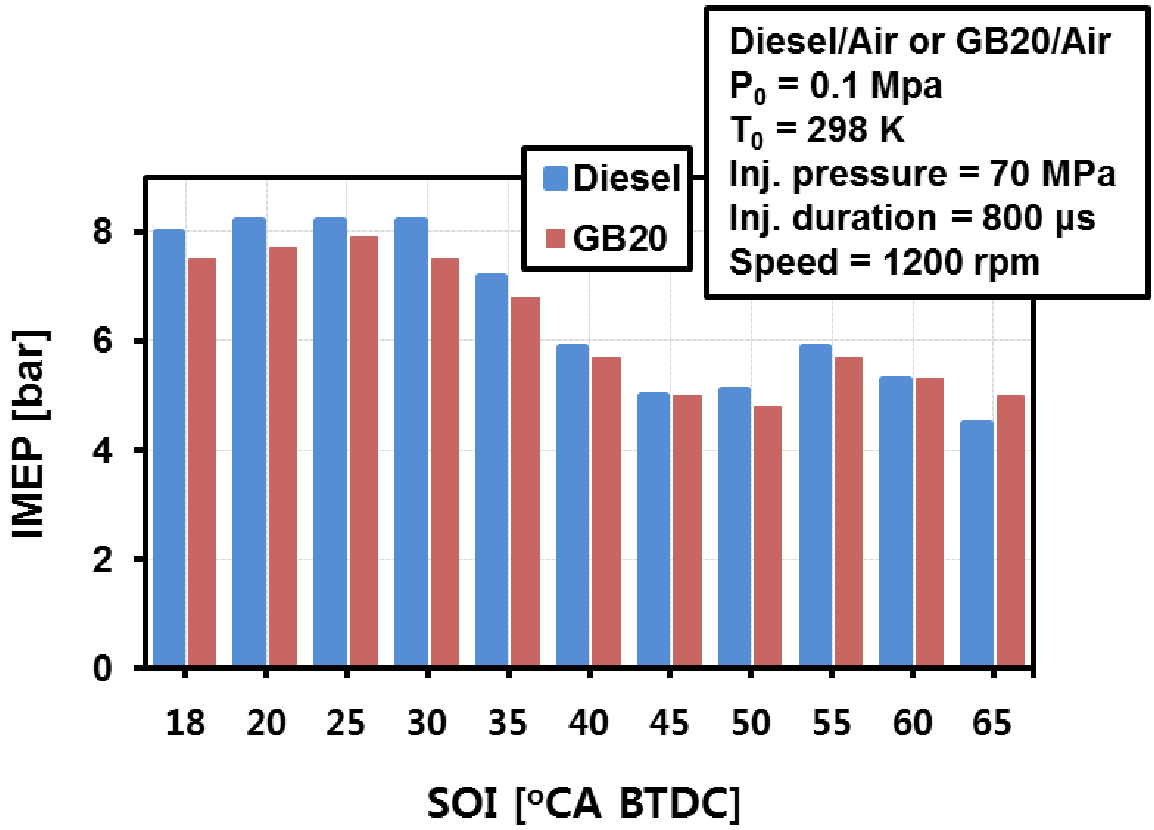


Fig. 5.9 IMEP

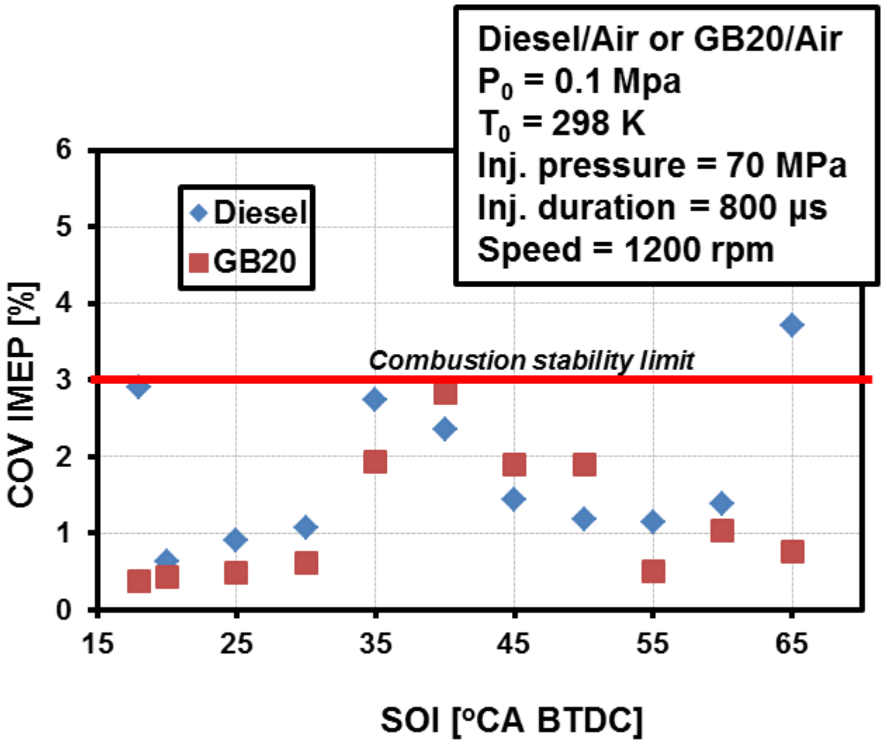


Fig. 5.10 COV of IMEP

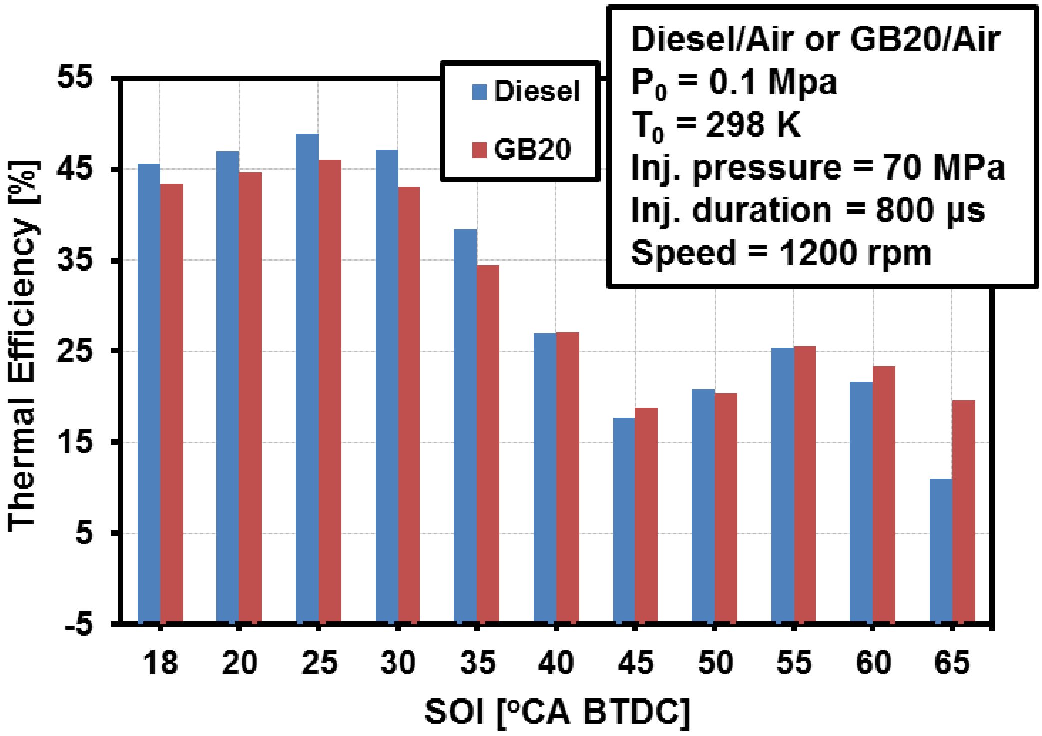


Fig. 5.11 Thermal efficiency

5.1.7 Combustion efficiency

Fig. 5.12 shows the combustion efficiencies of the engine for various SOI and different fuels, which are Diesel and GB20. The combustion efficiencies were evaluated based on the exhaust gas composition. Commonly, there are still combustible species left in exhaust gas, i.e., CO, HC and unburned hydrocarbon. The high content of this species in the exhaust gas indicates combustion inefficiency. The combustion inefficiency mainly depends on the concentration of HC and THC in exhaust gases. Little combustion temperature occurs in the exhaust pipe of GCI engine.

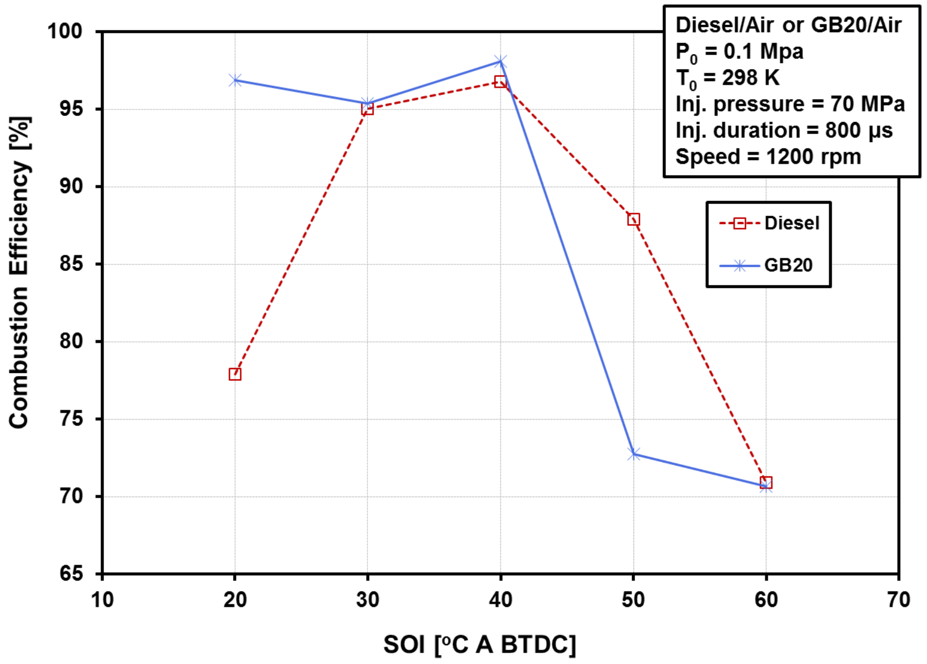


Fig. 5.12 Combustion efficiency

5.1.8 THC emission

The total hydrocarbon emission in a CI engine can be used to predict the inefficiency of the engine. The higher total hydrocarbon emission is a consequence of incomplete combustion of the fuel. Fig. 5.13 shows the THC emission of the engine when operated using D100 and GB20 fuels. From the figure, it can be seen that the THC emission of GB20 is lower than D100 fuel in every SOI variation. The retarded SOI range from 20 to 40 °C A BTDC for both fuels usually resulted in higher in-cylinder bulk gas temperature and greater HRR that increase NOx and reduce THC. The low thermal efficiencies for the more advanced SOI at 50 and 60 °C A BTDC may result in substantial low in-cylinder bulk gas temperature. The substantial low in-cylinder bulk gas temperature and lower HRR may increase THC emission. The lower THC emission resulting from GB20 than D100 in every variation of SOI, especially for retarded SOI, is suspected to be due to the effect of higher gasoline content causing the homogeneous fuel-air mixture to combust completely. The 80 % gasoline and 20 % biodiesel in the blend may have high fuel volatility (temperature distillation decrease) that will promote evaporation and mixing, and reduce the liquid fuel film (caused by impingement) on the cylinder surface. Additionally, with the 20 % of gasoline in the blend, the mixing process of the air fuel is improved and leaner mixtures are achieved. Therefore, higher homogeneous levels are attained. This condition may result in better combustion in the cylinder during the cycle. The THC emission of D100 is higher than GB20, which is also caused by the characteristics of diesel fuel, which is very non-homogeneous during the simultaneous injection and combustion processes. The local spots range from the very rich to the very lean for non-homogeneous air-fuel mixing with very limited combustion duration, resulting in some amount of fuel not burning properly. Therefore, higher THC emissions will result.

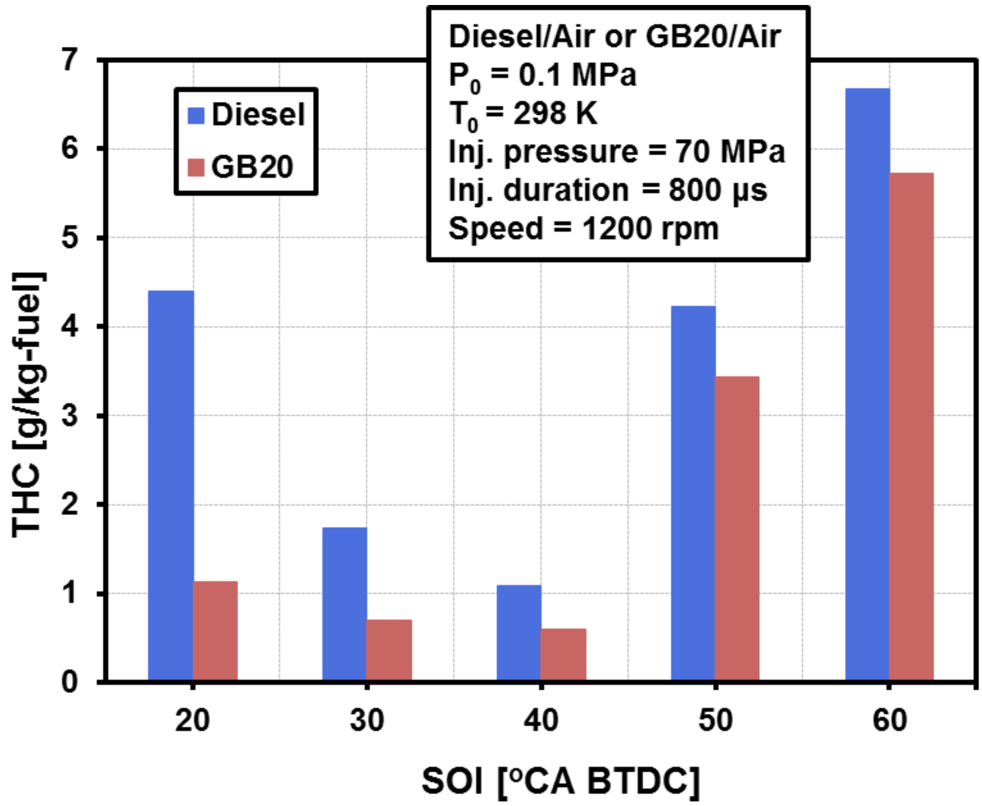


Fig. 5.13 THC emission

5.1.9 NOx emission

Fig. 5.14 shows the NOx emission of the engine when operated using D100 and GB20 fuels. From the figure, it can be seen that the NOx emission of D100 is lower than GB20 fuel in every SOI variation. This occurs due to the emission characteristics of biofuel or high oxygen content fuel in the gasoline fuel blends. The SOI ranges from 40 to 50 °C A BTDC of GB20, resulting in a higher in-cylinder pressure and higher in-cylinder temperature of more than 1800 K, and also low combustion duration. This approach is likely to produce high NOx emissions, since NOx formation is believed to result from combustion at temperatures over 1800 K[44]. However, the other reference[17] mentioned that since the flame temperature is below approximately 1650 K, the low NOx emission will be achieved. Therefore, the reason why NOx emission resulting from an engine fueled with GB20 is higher than D100 can be explained by the effect of high-temperature combustion and high oxygen content in the fuel.

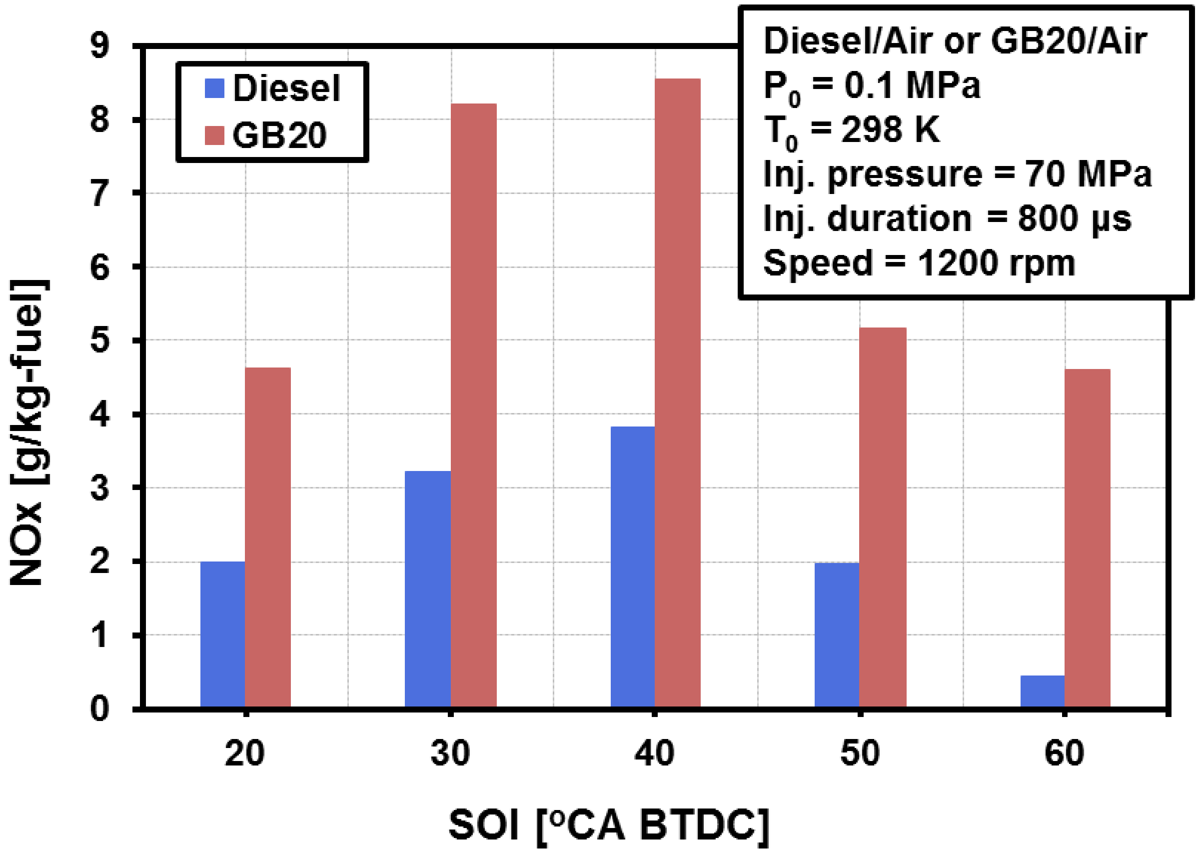


Fig. 5.14 NOx emission

5.1.10 Summary

Engine performance, combustion, and emission have been revealed and discussed in this experimental step. In single injection strategy, the results showed that the earlier the SOI of GB blends, the shorter the ignition delay compared to diesel fuel. The low cetane number of gasoline causes longer ignition delays or more resistance to autoignition. To a certain extent, this behavior is profitable because it provides more time for fuel mixing after injection and before combustion starts, therefore allowing combustion to take place near top dead centre. However, too long ignition delay could result in delayed combustion phasing, leading to an inefficient engine work or low thermal efficiency. Therefore, to overcome the too-long ignition delays, more reactive fuel (more reactive stratification fuel such as biodiesel) is necessary in the gasoline. In the gasoline biodiesel blend fuel, there are more fuel stratifications, so that when the higher biodiesel content of the gasoline mixture ignites faster than the other part of the fuel mixture, it will also influence the ignition delay, combustion phasing and combustion duration.

The thermal efficiency for GB blends was found to be almost equivalent with diesel fuel for all conditions. In the retarded SOI range from 18 to 35 ^{o}C A BTDC, the D100 fuel is higher than GB20. Meanwhile, when the SOI ranges were advanced from 55 to 65 ^{o}C A BTDC, the thermal efficiency of D100 is lower than GB20. The higher thermal efficiency of D100 in the retarded SOI is due to the higher LHV of the diesel fuel compared to GB20. However, in some cases of advanced SOI, the thermal efficiency of GB20 is higher than D100. This condition can explain the sensitivity of the engine to volatile fuel, which in this case, is the gasoline content in the blend that caused the chemical and physical properties of the fuel to have a significant effect on the combustion process. Because of the high gasoline volatility in the GB20 and the earlier or retarded injection timing, the fuel air mixture has enough time to become a more homogeneous fuel-air mixture. Therefore, a complete combustion process will be achieved.

The combustion efficiencies were evaluated based on the exhaust gas composition. Commonly, there are still combustible species left in exhaust gas, i.e., CO, HC and unburned hydrocarbon. The high content of this species in the exhaust gas indicates combustion inefficiency. The combustion inefficiency mainly depends on the concentration of HC and THC in exhaust gases. Little combustion temperature occurs in the exhaust pipe of GCI engine. In the case of emission, GB blends produce lower HC compared to diesel, as expected, because of their homogeneous mixing capabilities. However, a higher NOx emission from GB blends was observed, which might be a result of excess oxygen in the fuel. To overcome the higher emission of NOx, the next experimental step was conducted using pilot and main injection strategies.

5.2 Pilot and main injection strategies

The combustion and emission characteristics of CI engines are influenced by various factors, such as fuel injection strategy. Various injection strategies can lead to discrepancies in characterizing engine combustion and emissions. Therefore, it is very important to describe the effect of injection fuel strategy, e.g., multiple injections, on the combustion and emissions of GCI engines fueled with gasoline-biodiesel blends. The multiple-injections strategy, which is comprised of a pilot and main injections applied directly to a cylinder engine, and the earlier split-port injection method, which is used to create fuel stratification combustion, was applied to successfully reduce NOx. The pilot and main injection strategies created a decreasing NOx profile that was attributed to the decreases in combustion pressure and temperature peaks along the combustion phasing generated by the progressively later combustion. At medium engine loads and speed, the pilot-pilot-main injection strategy, which is optimized using the design of experiments, allows the NOx emissions to be decreased significantly and has led to higher mean combustion pressure, lower heat release rates, shorter ignition delays, and lower brake specific fuel consumption.

There is less experimental data characterizing the combustion and emissions of the GCI engine using the multiple-injection strategy with fossil gasoline and biodiesel blends, and the theory that discusses GCI with biodiesel utilization is less well-developed for the multiple-fuel injection strategy. Thus, experimental studies to identify the important fuel properties and quantify the effects of the pilot and main injection strategies 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 experimental squence was to determine the effects of various injection strategies, i.e., multiple-injection mode, which consists of a pilot and main injections, and single-injection mode, 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 injection strategy on combustion and emissions of GCI engine the same basic energy input of injected fuels was used for comparing the various parameters. The pilot and main injection modes for the gasoline-biodiesel blend were also combined with modification of several initial conditions, such as intake, oil, and coolant temperatures. The analysis of the combustion characteristics of cylinder pressure, heat release rate, combustion stability, ignition delay, and emission characteristics are discussed as the focus of this study.

5.2.1 Cylinder pressure, temperature, and heat release rate

In this study, a lean equivalence ratio of fuels for both GB05 and D100, for both multiple and single injections modes, were injected into the cylinder with the same energy input. Recall that the LHV of GB05 was around 45.32 MJ/kg and that of D100 was around 45.93 MJ/kg, with an equivalence ratio of 0.8; based on these results we determined that a similar amount of energy input resulted in an average total injected fuel of around 41 mg per cycle. For the multiple-injection mode, around 70% (31 mg) of the total average fuel was injected into the pilot injection part and 30% (10 mg) into the main injection part. Fig. 5.15 shows the cylinder pressure, temperature, and heat release rate (HRR) of GB05 and diesel fuel in the single and multiple-injection modes. The pressure data were taken from 100 consecutives engine cycles, and the mean values are plotted in Fig. 5.15. The highest peak for cylinder pressure was obtained from D100 by using single-injection mode, followed by GB05 in multiple-injection mode and then GB05 in single-injection mode. Finally, the lowest peak for cylinder pressure was for D100 in multiple-injection mode. It was normal for D100 to produce high cylinder pressure due to the high density and energy content of the fuel. Furthermore, the multiple injection strategies showed significant reductions of the in-cylinder peak pressure and heat release rate, as this is the main feature of multiple injection strategies with early pilot injection compared to single injection based on the previous research study[84]. The other reason is that by using early pilot injection strategy, the premixed combustion of the pilot injection become weaker, which decreased temperature and pressure before the main injection caused both peak values of the in-cylinder pressure and the heat release rate was reduced. Then, the combustion of the main injection was retained during the shorter period. This condition agreed with the previous study by Kim and Bae [85]. The impact of the higher intake temperature for GB05 with single or multiple injections resulted in lower in-cylinder pressure trace compared to a single injection of diesel fuel, even though the injection fuel had the same energy content. The higher intake temperature leads to a reduction in the volumetric efficiency of the engine; thus, a very small amount of fresh air enters the combustion chamber and less fuel is burned. Therefore, the in-cylinder pressure trend of GB05 is lower than that of diesel fuel. However, from Fig. 5.15, it can be inferred that, by increasing the intake air temperature, the auto-ignition combustion issues of gasoline fuel in the CI engine were successfully resolved. As can be seen from the in-cylinder pressure trace, the gasoline combustion achieved a stable state. Unstable combustion or misfiring usually occurs when gasoline fuel is utilized in the CI engine because gasoline is a single-stage ignition fuel. Therefore, increasing the air intake temperature is required to obtain stable auto-ignition combustion[75].

Similar to the cylinder pressure, the highest in-cylinder temperature was for D100. Logically, since the in-cylinder temperature is calculated using pressure data based on the ideal gas law, at a fixed in-cylinder pressure, a low intake temperature results in a high inlet mass and high inlet air density. Therefore, diesel fuel should have a lower in-cylinder temperature. However, the lowest in-cylinder temperature in the present study was observed for multiple injections of GB05, with a value of approximately 1800 K. The reason for this is that, as the intake air temperature increases, the air density will decrease, resulting in lower inducted air mass. Since the injected fuel in the combustion chamber was constant, the injected fuel for both diesel and GB05 was adjusted to the same energy content; in addition, the fuels also unintentionally had the same weight. Therefore, less fuel was burned, which led to a lower indicated mean effective pressure (IMEP). The combustion phasing was also changed, but it had a small effect on the IMEP. Thus, reduction of the amount of fuel burned will contribute to lower combustion temperatures and lower residual temperatures. This has the potential to be thought of as low-temperature combustion, which produces low NOx and soot emissions and usually occurs below 1800 K[44].

The pressure inside the cylinder provides information about the thermodynamic state of charge. Using the first law of thermodynamics and several simplifying assumptions, it is possible to obtain information about the rate at which the combustion is taking place. In Fig. 5.14, the heat release rate for multiple injections of GB05 is lower than for a single injection of D100, but higher than for multiple injections of D100 or a single injection of GB05. These results demonstrate that the GCI engine fueled with a low-cetane number fuel (GB05) in multiple injections mode has the ability to control its combustion. However, the sensitivity of the intake air temperature should be considered.

5.2.2 Ignition delay, combustion phasing, and combustion duration

In a CI engine, ignition delay (CA10), or auto-ignition timing, is commonly defined as the crank angle at which the integrated value of heat release reaches 10% of the total heat value of the introduced fuel. Ignition delay can be affected by many variables, such as equivalence ratio, intake pressure, and intake temperature. This is particularly true for intake temperature, as it increases as ignition delay timing decreases. A higher intake temperature leads to an earlier start for the first stage of ignition (cool flame) and, therefore, reduces the primary ignition delay. Increasing the intake temperature shifts compression ignition combustion, but the controllable area is relatively narrow. In a CI engine fueled with diesel, increasing the intake temperature causes an earlier ignition delay, which leads to an increase of smoke due to a shorter time allowed for the fuel-mixing process. This is where gasoline has an advantage for a CI engine, because it has a very high resistance to auto-ignition, so that the ignition delay is increased, and more mixing takes place before combustion. Increasing the intake charge air temperature overcomes the auto-ignition issues of the gasoline fuel. This condition can help reduce the amount of smoke and contributes to the reduction of NOx. The larger is the ignition delay value, the lower are the NOx emissions. This can be explained by the findings that, with a higher ignition delay, the content of unmixed fuel and air decreases and the difference between the local mixture strength and the whole mixture strength decreases. If the whole mixture is lean in the CI engine, NOx emission production will be low. This condition will be discussed in more detail in the emission section.

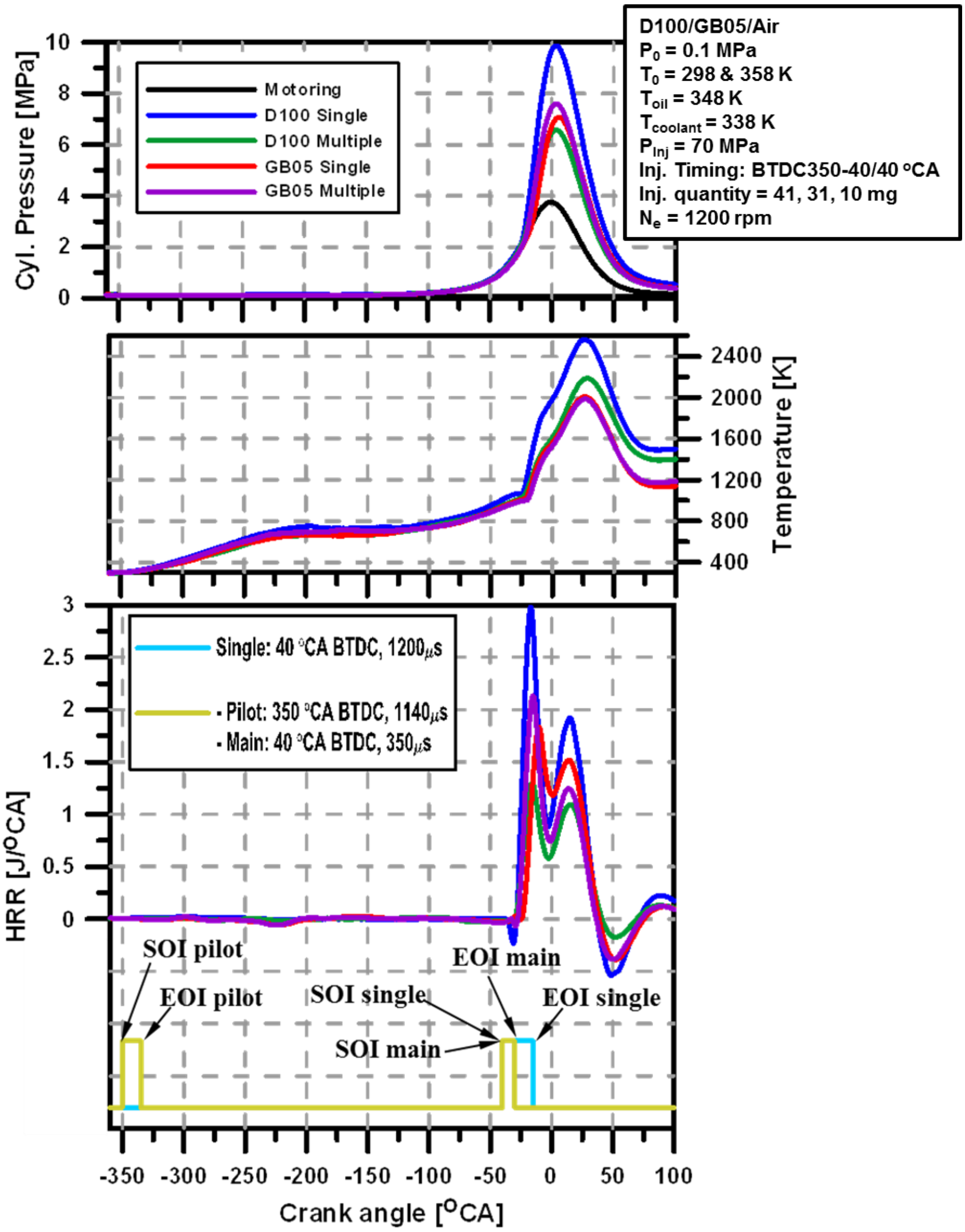


Fig. 5.15 Cylinder pressure, temperature, and heat release rate

Fig. 5.16 shows the ignition delay of the GCI engine fueled with diesel and GB05 using single- and multiple-injection modes. The ignition delay of multiple injections and single injection of GB05 is more advanced than multiple injections of D100. Gasoline fuel is usually more resistant to auto-ignition; therefore, the ignition delay will be increased. However, GB05 contains 5% biodiesel, which has a high cetane number, and the initial temperature in this experiment was maintained at approximately 85 °C in order to overcome the autoignition resistance. The blend of gasoline with 5% biodiesel had an estimated cetane number of 17.96 using the linear blending formula developed by a previous researcher[12].

Therefore, using this strategy combined with increasing intake temperature, the high auto-ignition resistance of gasoline fuel can be resolved in a CI engine.

Fig. 5.17 shows the combustion phasing of the GCI engine fueled with diesel and GB05 using single- and multiple-injection modes. The combustion phasing of multiple injections of GB05 is similar to multiple and single injections of D100. Multiple-injection mode consists of pilot and main injections with intake air, coolant, and oil heating to successfully maintain the combustion phasing (CA50) within a very narrow range, even with major changes in the fuel properties. When GB05 is used in single-injection mode, the combustion phasing is more advanced, even though the intake air, oil, and coolant temperatures are maintained at values higher than 65 ^{o}C. Multiple-injection mode with GB05 successfully preserved the combustion phasing near top dead center (TDC) and was similar to single-injection mode with D100.

Fig. 5.18 shows the combustion duration of the GCI engine fueled with diesel and GB05 using single- and multiple-injection modes. The combustion duration was defined as the start of combustion at around 10% of the total heat value of the introduced fuel (CA10) until complete combustion at 90% of the total heat value of the introduced fuel (CA90). The longest combustion duration was for GB05 in single-injection mode. The longer combustion duration indicates that the mixing-controlled combustion process is longer and might increases HC or soot. The lowest combustion duration was for D100 with multiple injections. The multiple injections had a decreased the combustion duration for both D100 and GB05. In the homogeneous combustion strategy, the combustion duration is a function of many parameters. The parameters with the strongest influence on the combustion duration are usually λ, exhaust gas recirculation (EGR) rate, combustion timing, compression ratio, and engine speed. In this study, the increasing intake temperature can affect the intake air charge. As the intake air temperature increases, the air density will decrease, resulting in less inducted air mass. GB05, which is more volatile, will lead to a longer ignition delay and then affect the combustion duration. However, increasing the intake temperature and using multiple-injection mode, shorter combustion duration is generated. The combustion duration of multiple injections with GB05 was slightly longer than multiple injections with D100. This condition could be a reason for complete combustion and resulted in a lower amount of unburned fuel. The gasoline fuel with low cetane number can cause longer ignition delays or more resistance to auto-ignition. To a certain extent, this behavior is an advantage of gasoline fuel, because it provides more time for the mixing process after injection and before combustion starts, thereby allowing combustion phasing to take place near TDC. However, an ignition delay that is too long could result in delayed combustion phasing, leading to an inefficient engine or low thermal efficiency. Therefore, to solve this problem, instead of blending with biodiesel, increasing the intake air temperature is necessary as a solution in the GCI engine fueled with gasoline.

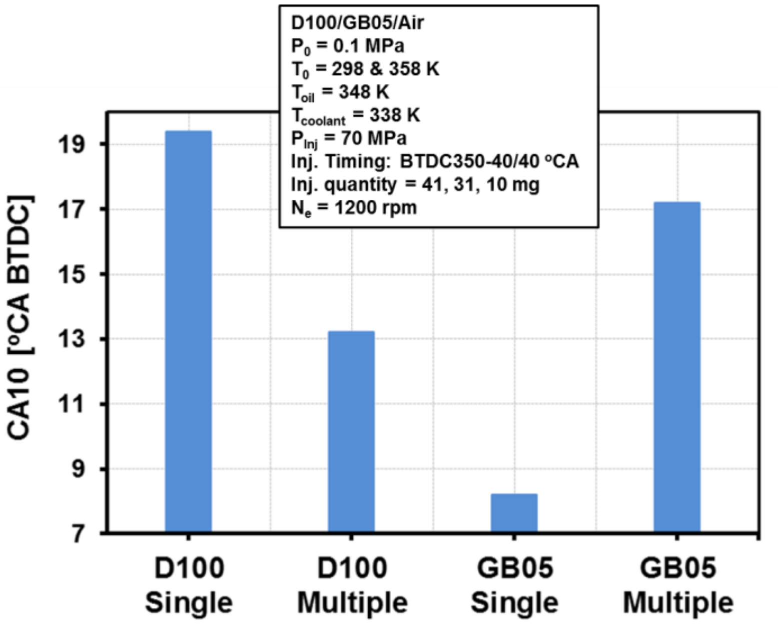


Fig. 5.16 Ignition delay

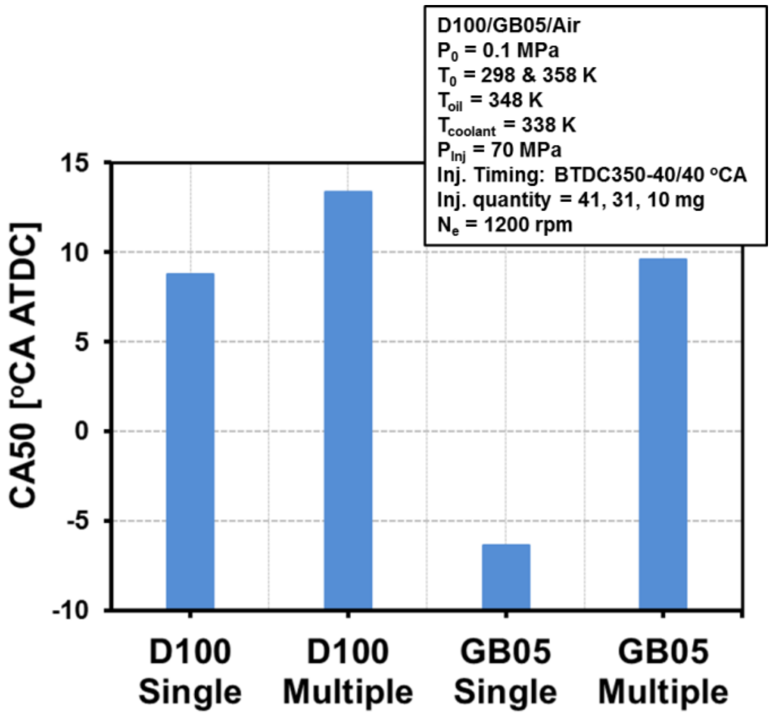


Fig. 5.17 Combustion phasing

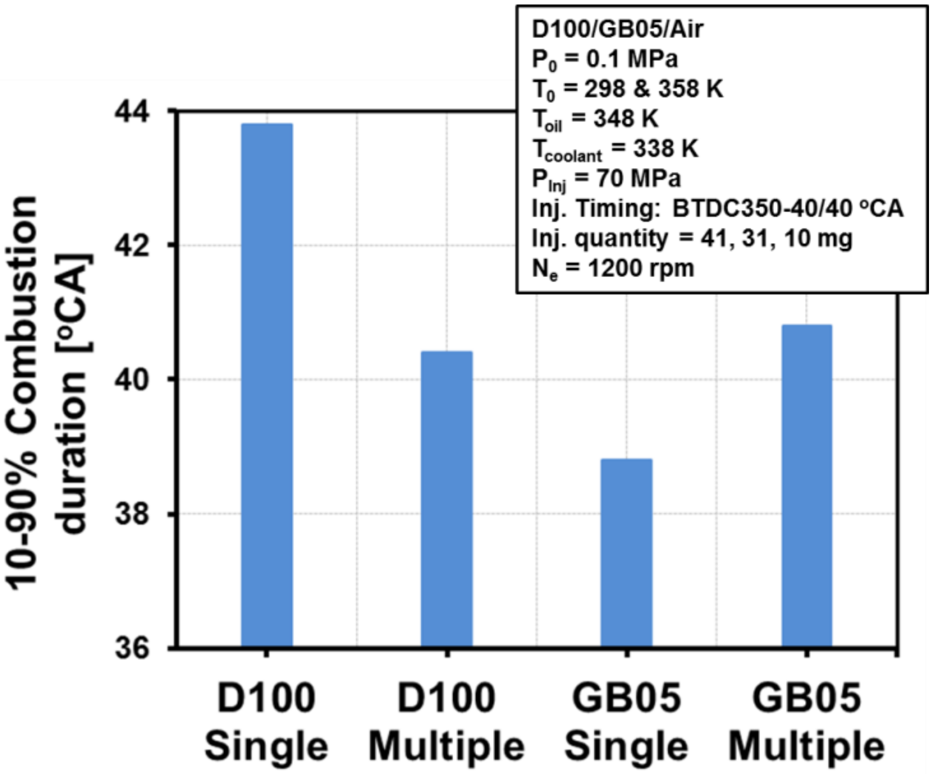


Fig. 5.18 Combustion duration

5.2.3 PPRR, knocking, and misfire phenomena

The utilization of gasoline fuel in the CI engine is challenging due to the autoignition difficulties of gasoline fuel when applied in CI engine. The implementation of biodiesel in the gasoline blend and increasing inlet temperature lead to the improvement of autoignition of GB05. Thus, the firing and misfiring phenomena should be determined when the engine operated by using gasoline-biodiesel blends, to understand wheatear the combustion occurred properly or not. The PPRR can be used as an indicator of engine noise or knocking, while a misfire can be identified using in-cylinder gas temperature. If the in-cylinder gas temperature is lower than 1100 K and high-temperature heat release does not occur, a misfire is noted. However, the results of this study and the data in Fig. 5.15 showed that combustion occurred for GB05.

Similar to the HCCI engine, the knocking phenomenon is one of the operating limits that are relevant to an engine with low-temperature combustion including GCI engine. Fig. 5.19 shows the peak pressure rise rate (PPRR) for GB05 and D100 in multiple- and single-injection modes. The PPRR was calculated based on the mean pressure from 100 consecutives individual cycle data set. It can be seen that the highest PPRR occurred with the engine fueled with D100 in single-injection mode. Using multiple injections, PPRR can be reduced, meaning that the engine noise is also reduced. As reported in the previous research results that the multiple injection strategy is and effective tool for the reduction of the combustion noise, but it causes some loss of the fuel economy due to the increased heat loss that occurs during the longer combustion durations by multiple injections [85]. Usually, high volatility fuel like gasoline used in CI engines has higher in-cylinder gas pressure rise rate (thus engine noise) and unburned hydrocarbon (UHC) emissions than conventional diesel engines[12]. For GB05, there is more fuel stratification, so that, when the portion of the fuel with higher gasoline content ignites faster than the rest of the fuel mixture, the heat release rate will be higher. The other reason is that the single-injection timing in this study is more advanced than in a common CI engine, which is at 40 ^{o}C A BTDC, and will promote excessive liner spray impingement (resulting in oil dilution). Related to this, more advanced injection timing will not result in sufficient fuel for homogeneous mixture combustion, but incomplete combustion due to under mixing (rich fuel zones resulting in incomplete combustion) or overmixing (air-fuel mixture being too lean to properly ignite). However, multiple injections can be used to solve the higher-pressure rise rate problem. Then, by combining gasoline blended with a small amount of biodiesel and maintaining the initial conditions, complete combustion and low emissions can be obtained. Early pilot injection (350 ^{o}C A BTDC) is one method to obtain HCCI mode in the CI engine. However, early injection with low volatile fuel caused higher PPRR. In previous discussion already mentioned that higher PPRR lead to produces noise and knocking as the operating limits of HCCI mode. However, a high-sensitivity fuel, diesel fuel in this case, usually produces lower PPRR than a low-sensitivity fuel (gasoline fuel). Therefore, multiple injections of diesel fuel resulted in a lower PPRR than a single injection of diesel fuel, while multiple injections of GB05 resulted in a higher PPRR than a single injection of GB05.

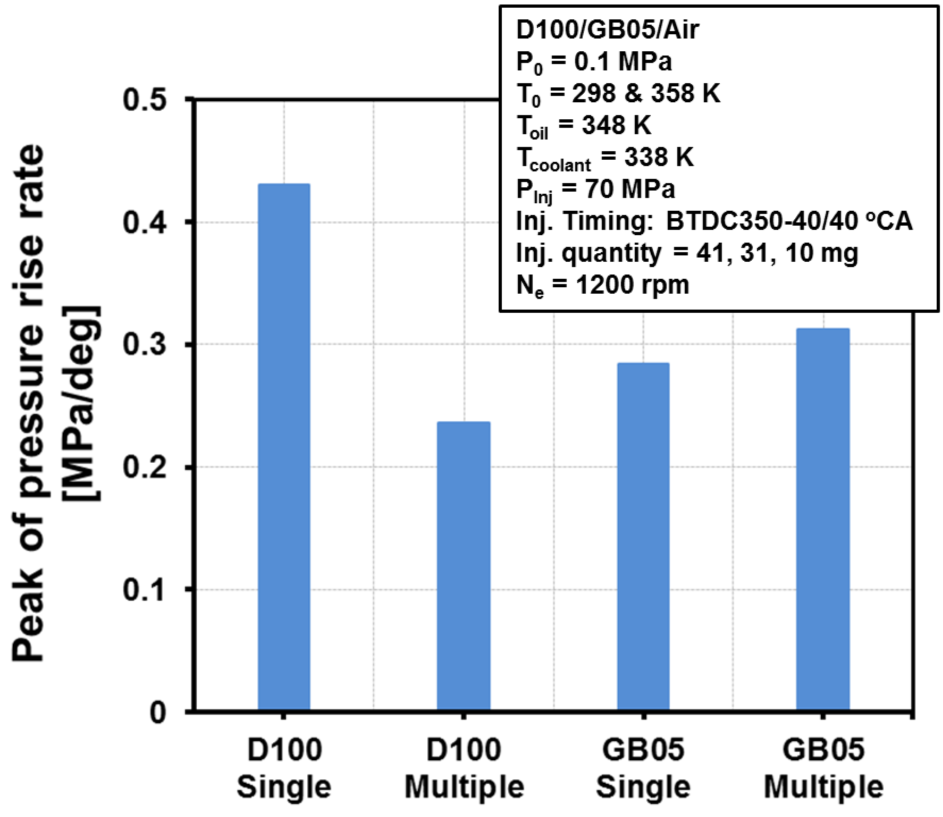


Fig. 5.19 Peak pressure rise rate

The knocking level of a CI engine for low-temperature combustion can be quantified using ringing correlation based on a previous study by Eng[68]. In this study, five (5) MW/m2 was chosen as the limit value for allowable ringing. It is already known that knocking commonly occurs in spark-ignition engines. However, instead of being of marginal importance, the knocking phenomenon is one of the main problems for the HCCI engine. Since low-temperature combustion is also a goal for a GCI engine, discussion on this phenomenon is necessary for a GCI engine. Even though knocking characteristics in HCCI engines are slightly different for SI engines in terms of amplitudes of the pressure oscillations[38][67], many researchers agree that knocking is one limitation for operation of HCCI engines. HCCI combustion can simultaneously reduce the exhaust emissions and increase engine thermal efficiency; however, the narrow operating range (limited by misfire and knocking) and the control of ignition timing (dominated by fuel chemical kinetics) are the main obstacles to its commercial use[86]. Knocking from excessively advanced and rapid combustion limited the operating range of the HCCI combustion phase[87]. There are many methods to determine the knocking phenomenon; however, in this study, ringing intensity was used even though it cannot be considered as a standard approach. The main reason to use this method is due to easily obtained based on in-cylinder pressure and temperature. In this study, the limitation of ringing intensity is 5 MW/m2. The computation of the RI was performed based on the PPRR averaged over 100 cycles together with the peak mass-averaged temperature and pressure from the ensemble-averaged pressure trace. The use of this method and the reason of this choice are clearly discussed in the study by Dernotte et al. [88]. As an example, for D100 in single-injection mode, if it is determined that the highest maximum in-cylinder pressure is 9.8 MPa and the highest maximum in-cylinder temperature is 2560 K, as can be observed in Fig. 5.15, the PRR maximum calculation result will be 558.8 MPa/ms. In the same way, for multiple injections of D100, a single injection of GB05, and multiple injections of GB05, the PPRR calculation results will be 476 MPa/ms, 504.6 MPa/ms, and 524.4 MPa/ms, respectively. Therefore, as shown in Fig. 5.19, as long as the value of PPRR for all conditions is less than this limitation value, the engine will run under stable combustion conditions without knocking. The detailed ringing intensity calculation can be seen in Fig. 5.20. Based on Fig. 5.20, it can be concluded that the ringing intensity for all conditions is much lower than the limit value or knocking criterion.

5.2.4 IMEP, engine power, thermal efficiency, and combustion efficiency

Fig. 5.21 shows the IMEP of the GCI engine fueled with diesel and GB05 using single- and multiple-injection modes. The indicated mean effective pressure is the average value of pressure resulting from the combustion of fuel in the chamber. Using multiple injections and maintaining the initial conditions, the obtained IMEP value of the GCI engine fueled with GB05 was similar to that of D100 for both single and multiple injections. Even though GB05 has a lower energy content and lower ignition sensitivity compared to D100, the combustion quality of GB05 related to the engine output (IMEP) can be improved by using multiple-injection mode and maintaining the initial conditions (increasing intake temperature).

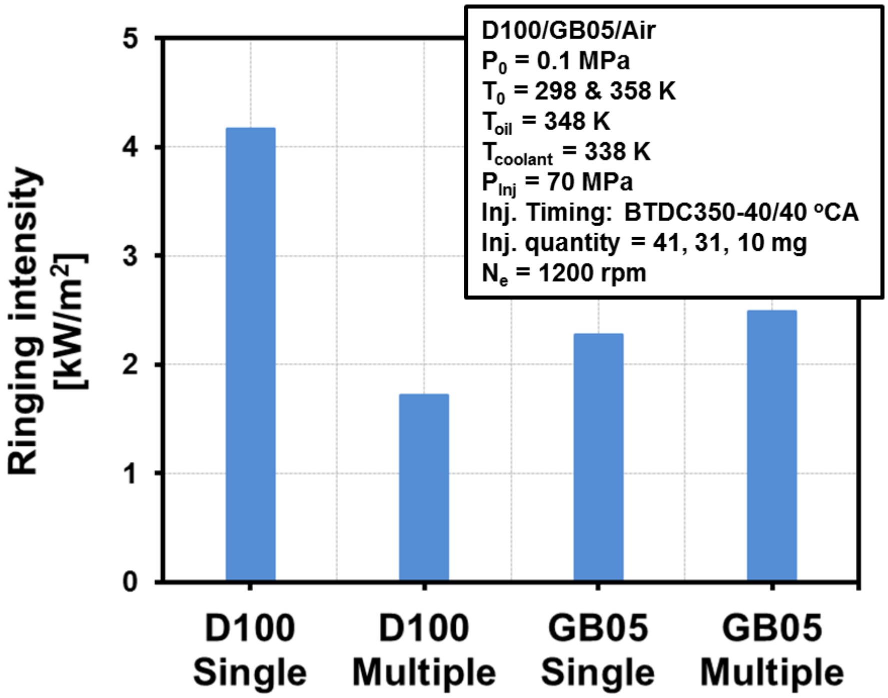


Fig. 5.20 Ringing intensity

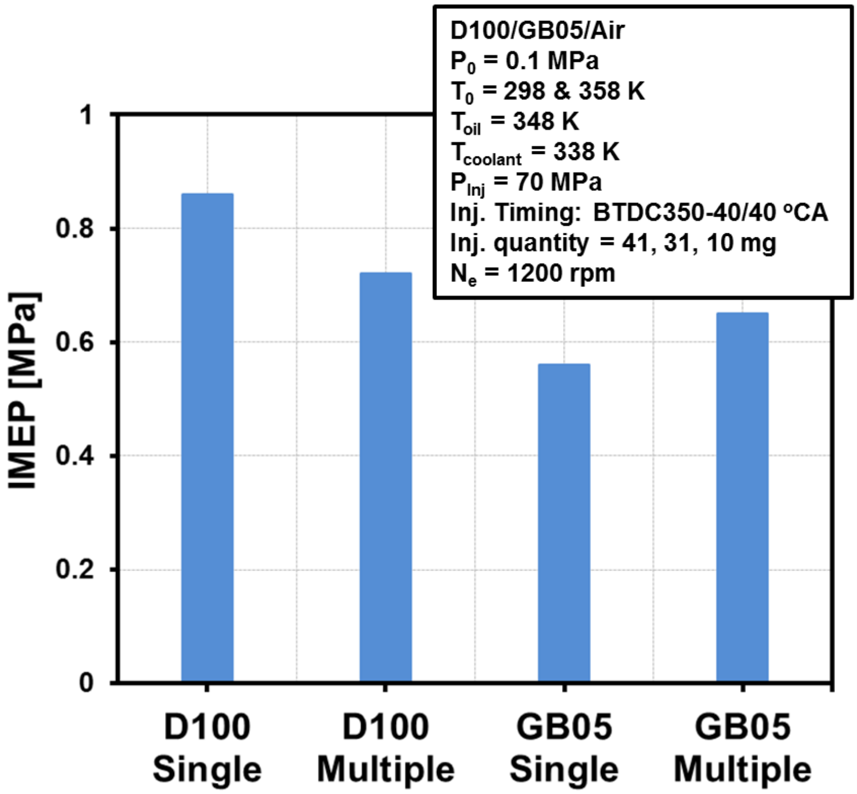


Fig. 5.21 IMEP

In the further analysis, the IMEP values can be used to investigate the stability of combustion in the engine. To investigate the cycle-to-cycle stability of the combustion performance of blended fuels, especially GB05, the coefficient of a variant of IMEP (COV of IMEP), which is used generally as an index of combustion stability, is analyzed from the recorded in-cylinder pressure data. The data is usually obtained from the calculation of 100 consecutive engines cycles.

Fig. 5.22 shows the COV of IMEP of the GCI engine fueled with D100 compared to GB05. Combustion with gasoline fuel was very sensitive to intake air temperature. Increasing the intake air temperature improved combustion, but it might be followed by combustion failure in some subsequent cycles. Previous research[44] showed that 3% of the COV of IMEP provides a consistent stability threshold over various GCI operating conditions. However, other researchers used a higher value for the combustion stability limit criteria (5% of the COV of IMEP)[12]. Based on the 5% combustion stability limit criteria from this experiment, the GCI engine fueled with GB05 for both single and multiple injections exhibited stable combustion.

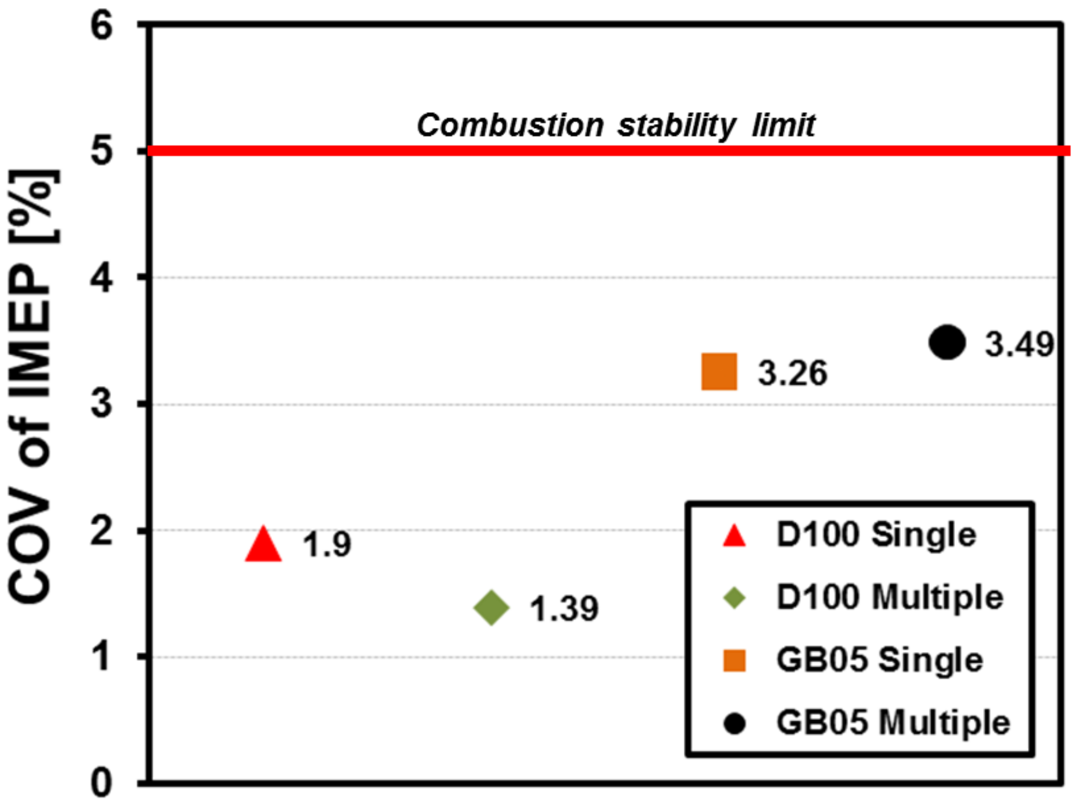


Fig. 5.22 Coefficient of variant of indicated mean effective pressure

As already known that CI engine running with highly volatile and low cetane fuel such as gasoline is very challenging. Large cyclic variations might be occurred due to instability of

GCI combustion. Therefore, cycle-by-cycle variations are very interesting and relevant in examining the combustion process of GCI engine, primarily when the analysis focuses on combustion stability. Recently, the cyclic variability analysis was considered necessary to determine the stability of engine combustion by a number of researchers[69–71] A coefficient of variation (COV) was also used to evaluate the cyclic variations of CI engine combustion. The visual variation of cylinder pressure for 100 consecutive test cycles under different condition can be observed in waterfall graph at Fig. 5.23. However, it is difficult to obtain detail analysis of cylinder pressure variation only by using waterfall plot graph. Fig. 5.24 illustrates the cycle-by-cycle variations in Pmax for 100 consecutive test cycles for D100 and GB05 in single and multiple injection modes. The highest average value of Pmax is for D100 single injection and the lowest if for GB05 single injection. It also can be observed that the more volatile and smaller cetane number of fuel GB05, with multiple injections, leads to a relatively higher COV of Pmax, and as more reactive fuel D100 with a single injection, the COV of Pmax decreases. GCI using GB05 as more volatile fuel and multiple injection modes indicated the tendency to knock by which can result in higher-pressure fluctuation. However, it can be obtained from all test points in Fig. 5.24, by using D100 and GB05 both for single and multiple injections, the COV of Pmax is smaller than 3% (COVPmax $<$ 3%). Therefore, the overall cyclic variations in Pmax by using D100 and GB05 both for single and multiple injections are smaller which mean the stability of combustion achieved. Fig. 5.25 shows the cycle-by-cycle variations in CA10. As like the trend of Pmax, the start of combustion is more extended by using more volatile and low cetane fuel, GB05, and multiple injection modes, while it was shorter when using more reactive fuel D100 and single injection mode. COV of CA10 by using GB05 fuel also significantly increases when running in multiple injection modes, even though the intake temperature was maintained in the fixed value. COV of CA10 for overall conditions obtained more than 3%, even for D100 single injection, and much higher when using GB05 in multiple injection modes. The COV of CA10 of GB05 in multiple injection modes is much higher which indicated ignition-timing fluctuation is in particularly remarkable which would more easily cause knock and misfire. From this condition, it shows that there are many factors influence the autoignition of highly volatile and low cetane number fuel, in this case, GB05 when applied in GCI engine. It is well known that the difficulty of autoignition control combustion as one of the constraints in low temperature and homogeneous charge ignition combustion strategy. The parameters that affected to control autoignition commonly are different intake temperature, intake pressure and the application of various EGR rate in order to modify the air intake dilution. The other possible explanation according to the previous study for this condition is that the relation of λ as significant effect on COV of IMEP at constant intake temperature [70]. As the mixture becomes leaner due to high volatile fuel in early pilot injection implementation, cyclic variations increase in this case COV of CA10 increases.

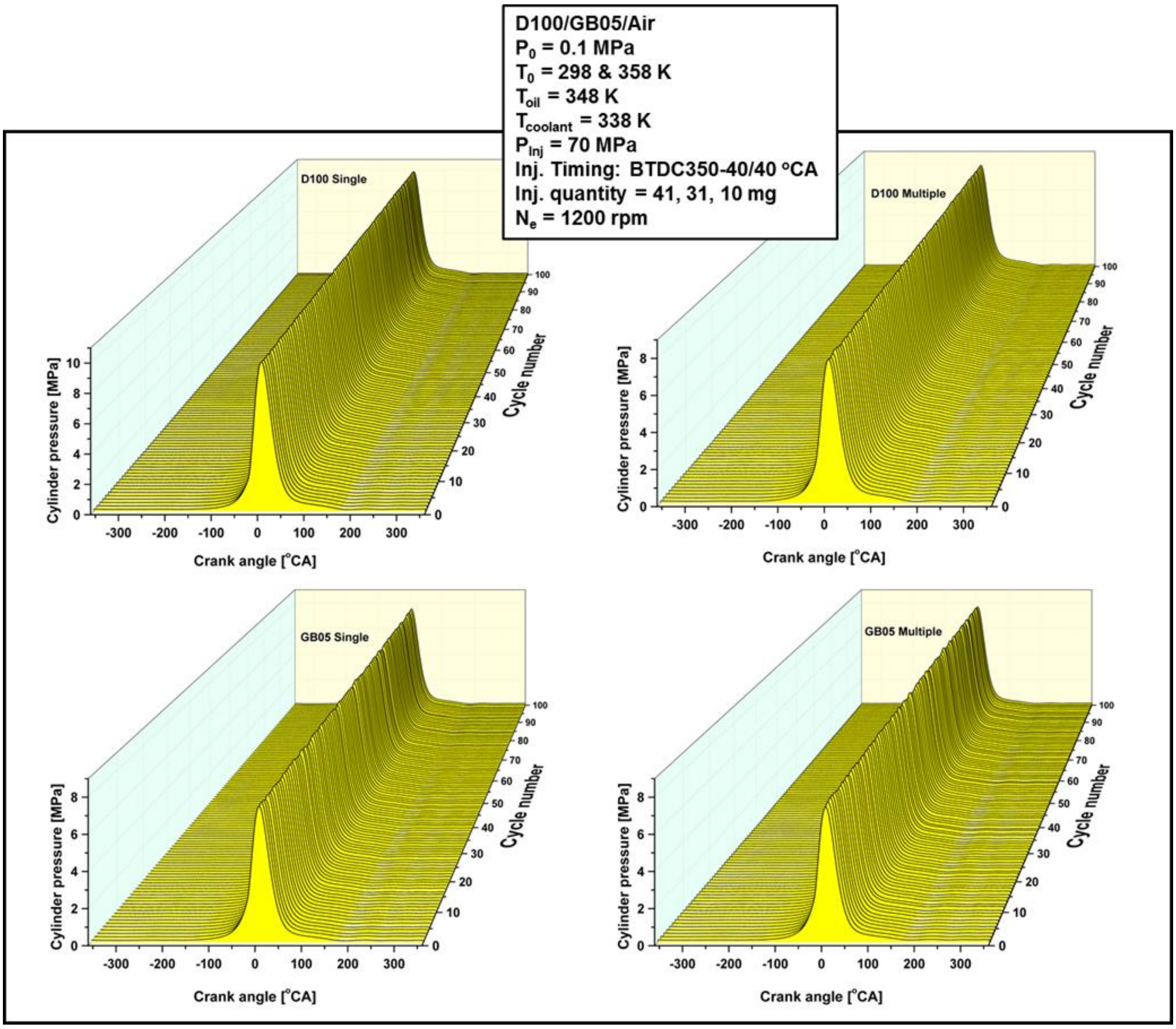


Fig. 5.23 Cylinder pressure variation under different conditions

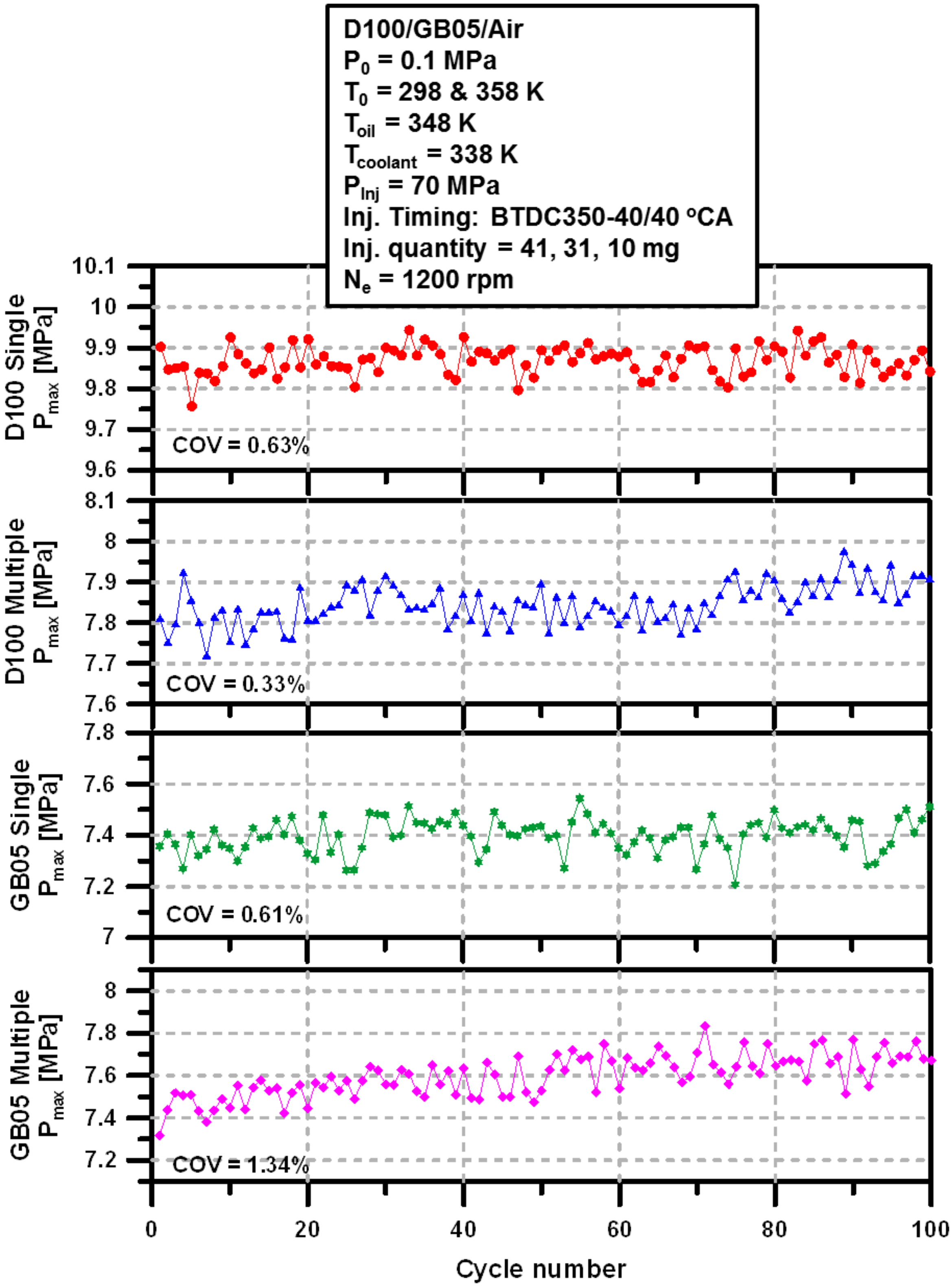


Fig. 5.24 Cycle-by-cycle variations in Pmax under different conditions

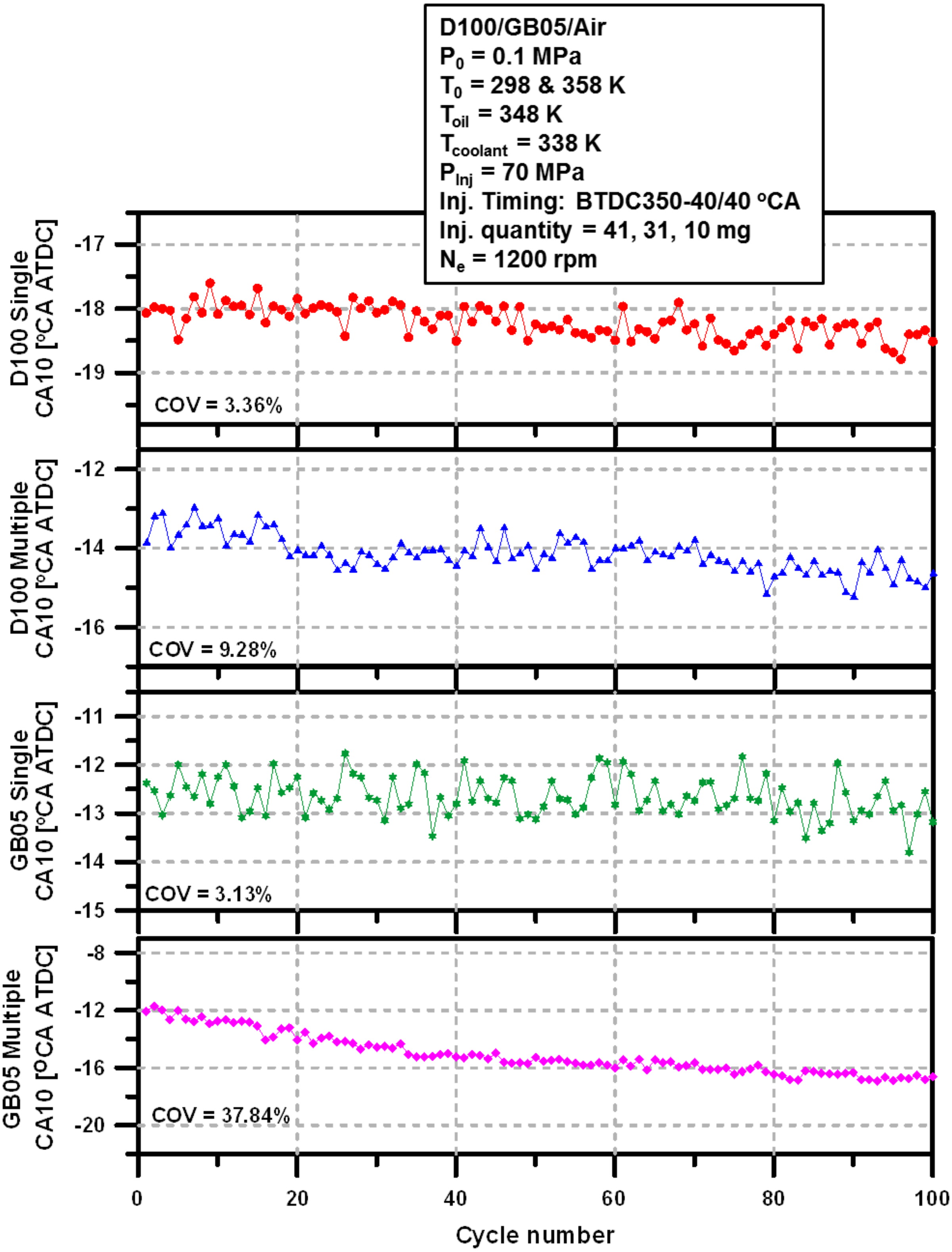


Fig. 5.25 Cycle-by-cycle variations in CA10 under different conditions

Fig. 5.26 shows the torque and power of the GCI engine fueled with GB05 and D100. The indicated power is obtained from the in-cylinder pressure data, while the brake power is obtained as torque multiplied by engine speed. The brake power is always lower than the indicated power due to friction loss in the engine system. The highest indicated power is for D100, which has a higher heating value than GB05, resulting in a higher in-cylinder pressure trace and, thereby, a higher indicated power. Using the indicated power, the indicated thermal efficiency can be calculated. Fig. 5.27 shows the indicated thermal efficiency of the GCI engine fueled with GB05 and D100 as a function of injection mode. The indicated thermal efficiency is defined in the classical manner following the reference by Pulkrabek Willard W[38]. It can be calculated as indicated power/work divided by energy input per cycle. In this case, the power indicated can be obtained by multiplying the IMEP with displacement volume. However, other researchers also defined the indicated thermal efficiency as the indicated fuel conversion efficiency based on the equation by Kalghatgi[67]. In this case, if all the components of the formula are similar to the reference[38], the indicated fuel conversion efficiency is obtained as indicated power/work divided by energy input per cycle.

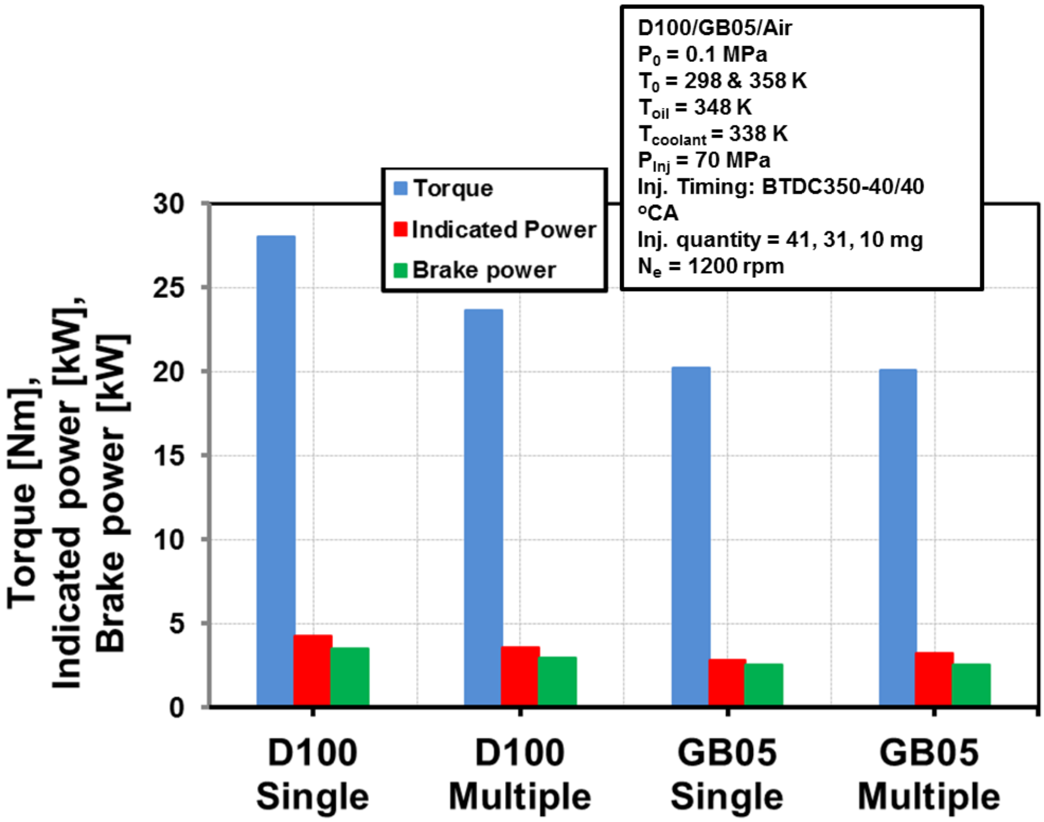


Fig. 5.26 Torque and engine power

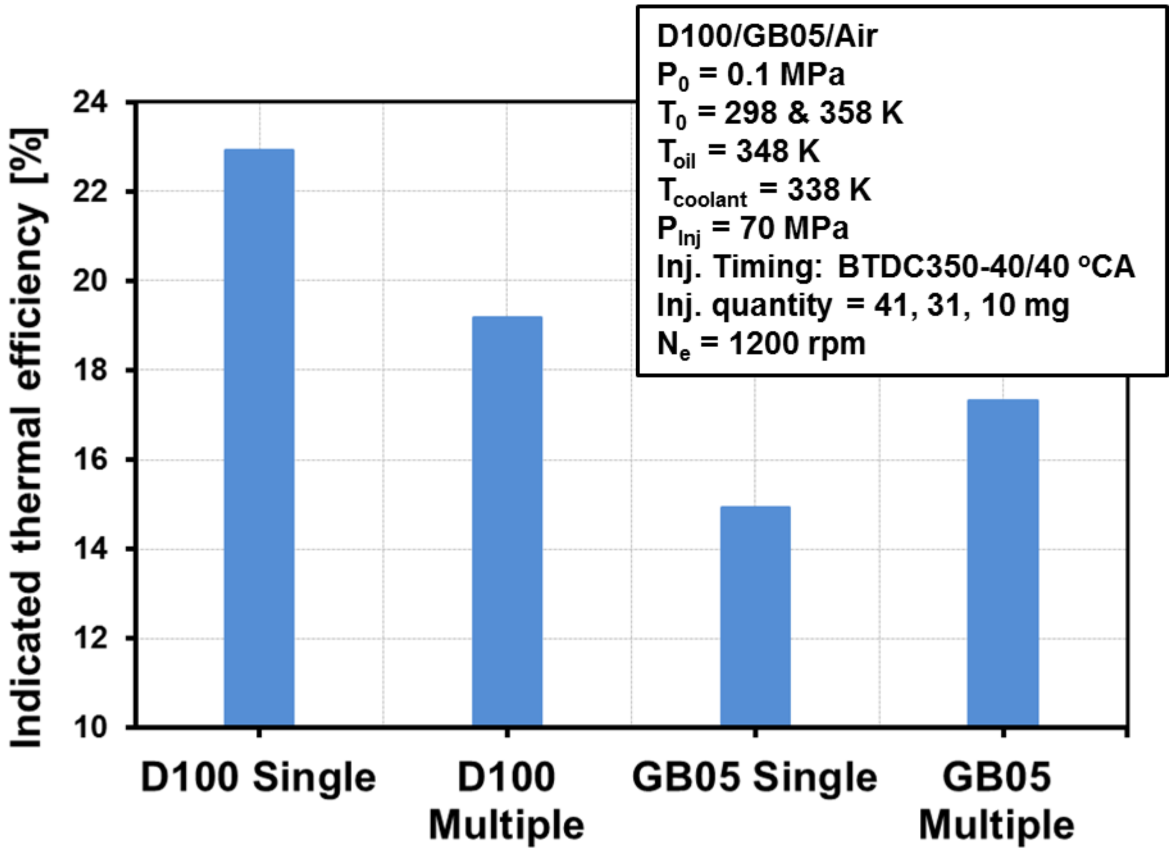


Fig. 5.27 Thermal efficiency

Fig. 5.27 reveals that the thermal efficiency of D100 decreased when multiple-injection mode was applied. In contrast, the thermal efficiency of GB05 increased when multiple-injection mode was applied. The higher thermal efficiency of D100 is due to the higher LHV of diesel compared to GB05. However, using multiple injections and increasing the intake air, oil, and coolant temperatures could indicate the ability to improve combustion and efficiency of the engine. Multiple injections, which consist of a pilot injection that supplies approximately 70% of the fuel amount at earlier timing, could provide sufficient mixing time for the gasoline-biodiesel blended fuel with a longer ignition delay. Then, supplying the main injection of approximately 30% of the fuel amount could induce combustion. Therefore, improvement in combustion will be obtained, resulting in a higher indicated thermal efficiency.

Fig. 5.28 shows the combustion efficiency of the GCI engine fueled with GB05 and D100 as a function of injection mode. In this study, the combustion efficiency was evaluated based on exhaust gas composition. Normally, there are still combustible species left in the exhaust gas of the engines, such as CO, HC, and unburned hydrocarbons. A high content of these species in the exhaust gas of the engine indicates combustion inefficiency. The combustion inefficiency mainly depends on the concentrations of CO and THC emissions in the exhaust gases. Fig. 5.28 shows that the combustion efficiency of a single injection of D100, a single injection of GB05, and multiple injections of GB05 are similar and higher than 93%. Meanwhile, the combustion efficiency of multiple injections of D100 is the lowest at approximately 86%. The high combustion efficiency of GB05 is mainly the result of complete combustion due to the longer mixing period of the injected GB05 and its volatile properties. This is supported by a relative high intake air temperature and high wall temperature because the engine oil and coolant temperature were also increased; therefore, more completing burning was achieved. The combustion efficiency increases for two reasons. First, the in-cylinder wall temperatures progressively increase, which helps maintain reactions close to the wall. Second, the more homogeneous mixing process of GB05 causes a proportional reduction of HC and CO emissions.

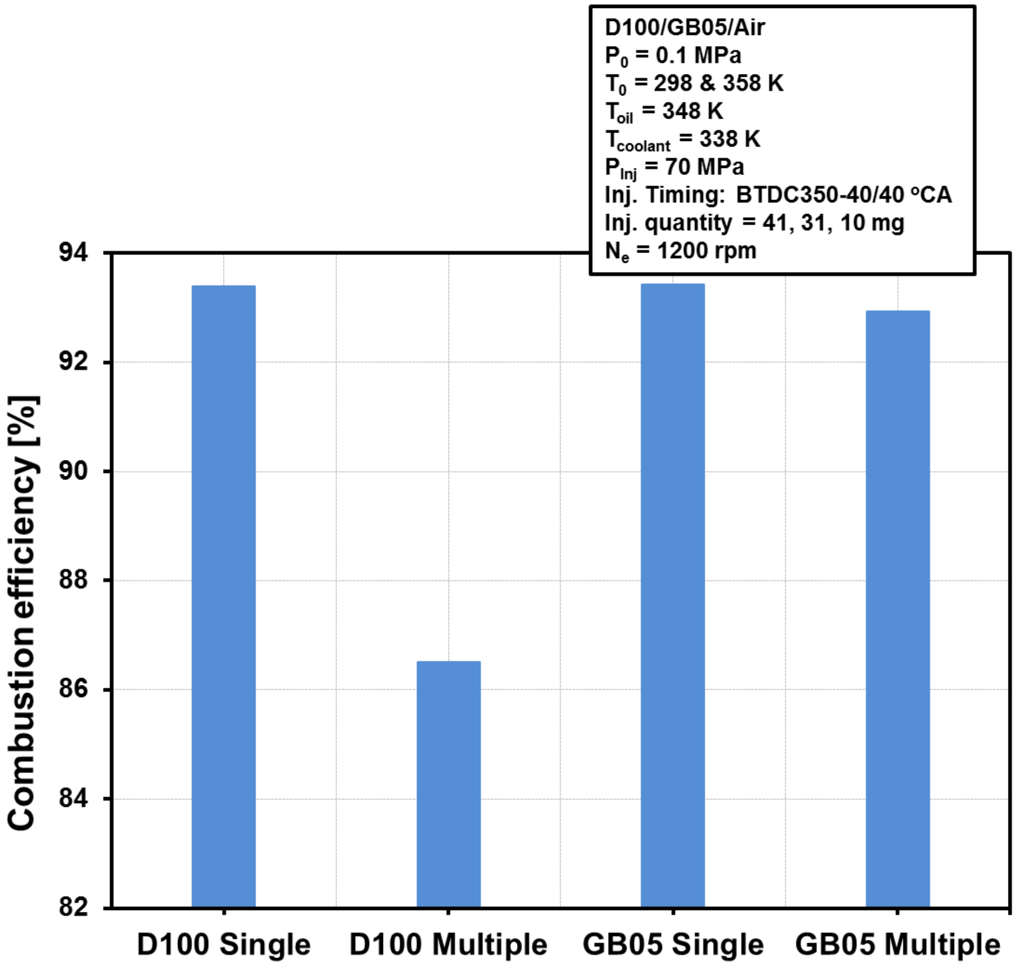


Fig. 5.28 Combustion efficiency

5.2.5 Exhaust emissions

Fig. 5.29 shows the CO emissions of the GCI engine fueled with D100 and GB05 for single and multiple injections. It can be seen that the CO emissions of diesel fuel for both multiple and single injections are much higher compared to those of GB05 for either single or multiple injections. The 5% biodiesel content and gasoline as the highly volatile fuel in GB05 equipped with a high intake air temperature showed the significant effect of lowering the CO emissions for the GCI engine. CO emissions are related to mixing and the temperature effect. CO emissions will increase if the local equivalence ratio becomes too lean (overmixing) to permit complete combustion; in other words, CO emissions are generated mainly by incomplete combustion of fuel. It is already known that diesel fuel has a high cetane number and is a so-called reactive fuel. Based on the ignition delay result shown in Fig. 5.16, it has a very short ignition delay. With a shorter ignition delay, more incomplete combustion occurs, resulting in increased CO emissions. In this study, GB05 is a more volatile fuel than diesel fuel, which is beneficial for promoting the premixing process. However, the volatility and low cetane number of GB05 will lead to high auto-ignition characteristics. High auto-ignition sometimes causes misfiring or a lack of combustion. In this condition, even though multiple injections promote sufficient premixing before combustion, misfiring occurs, and the CO emissions will also be higher. The higher intake temperature method helps to achieve complete combustion for GB05, which leads to low CO emissions. The CO emission content in the exhaust product can reflect the combustion efficiency. Therefore, decrease of CO emissions could be one indication of increased combustion efficiency.

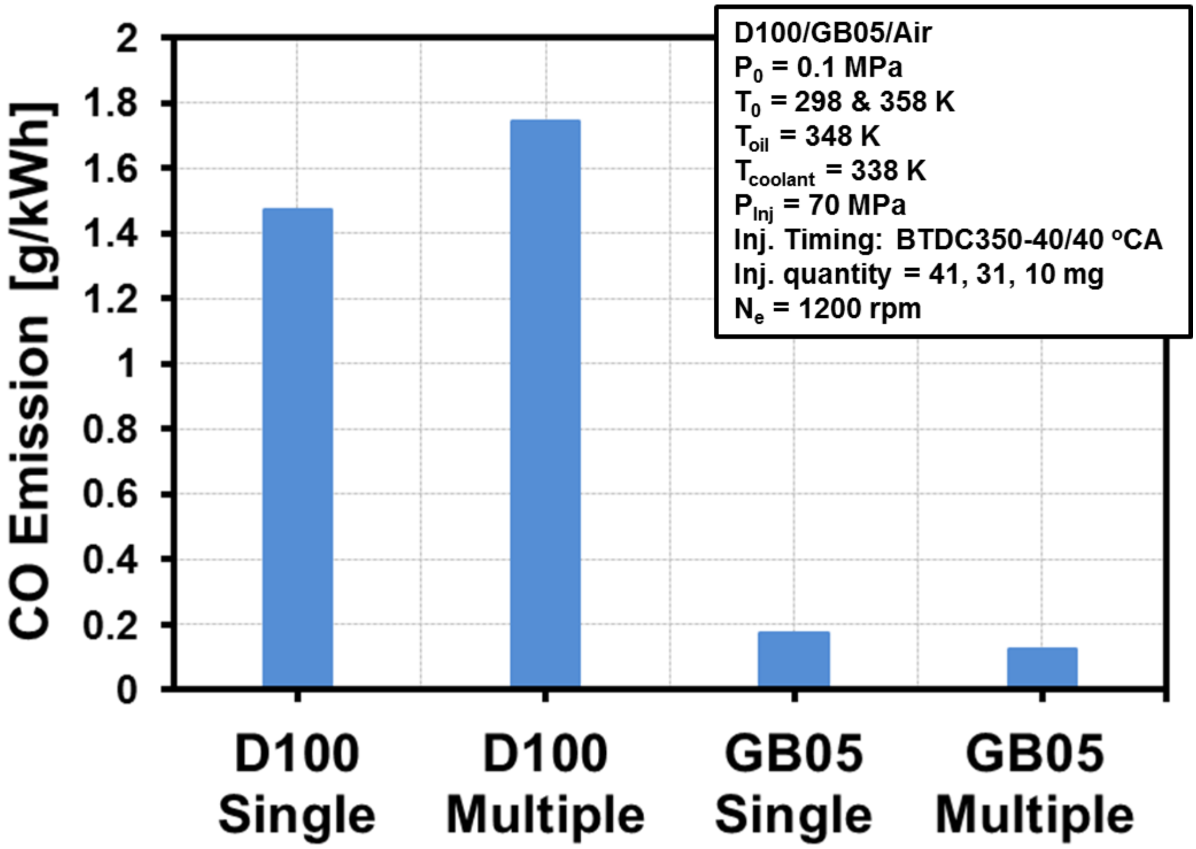


Fig. 5.29 CO emissions

Fig. 5.30 shows THC emissions of the GCI engine fueled with D100 and GB05 for single and multiple injections. The total hydrocarbon emissions in a CI engine can be used to predict the inefficiency of the engine. A higher amount of total hydrocarbon emissions is a consequence of incomplete combustion of the fuel. The higher THC emissions for the multiple injection strategies, in this case, D100 can be observed in the figure, which may indicate that the multiple injection timing and mass distribution may be far from its optimum point. Ra et al. [6] showed that a significant UHC originated from the pilot injection and entered the crevice region, and optimum injection timing should be determined by the trade-off between NOx and UHC/CO emissions. The pilot injection that applied in this study might be too far from the followed main injection prior to the start of combustion, causes a possibility to a wall-impingement of fuel resulted in a higher incomplete combustion of the fuel and THC emission. There is also a possible interaction between the injected fuel and the oil film on the cylinder liner walls during compression stroke caused the UHC emissions might be over-predicted. The other reason is that an early pilot injection may cause the over-lean mixture region, which is the main source of HC emissions, this condition agreed with the previous study by Kim and Bae [85]. In case of GB05, similar to CO emissions trend, the biodiesel content, and gasoline as a highly volatile fuel in the GB05 showed the significant effect of lowering THC emissions in the GCI engine. The primary sources of THC emissions are crevices, boundary layers, and bulk quenching. THC emissions become higher proportional to the fuel quantity supplied until auto-ignition occurs. The pilot injection of GB05 could be the reason for the decreasing THC emissions, where only 70% of the fuel amount was injected before auto-ignition. After that, the main injection was applied with 30% of the total fuel amount in every cycle. Furthermore, using a higher intake air temperature could also support the decrease of THC emissions. THC emissions are mainly generated by incomplete oxidation of the fuel. The content of oxygenated fuel, in this case, 5% biodiesel, could be the reason for decreasing THC emissions. With the presence of biodiesel in the blend, the THC emissions decrease. However, if the fuel and air are sufficiently premixed before combustion, fuel volatility plays a less important role. Since the high auto-ignition resistance of GB05 sometimes causes misfiring or a lack of combustion, THC emissions will also be higher.

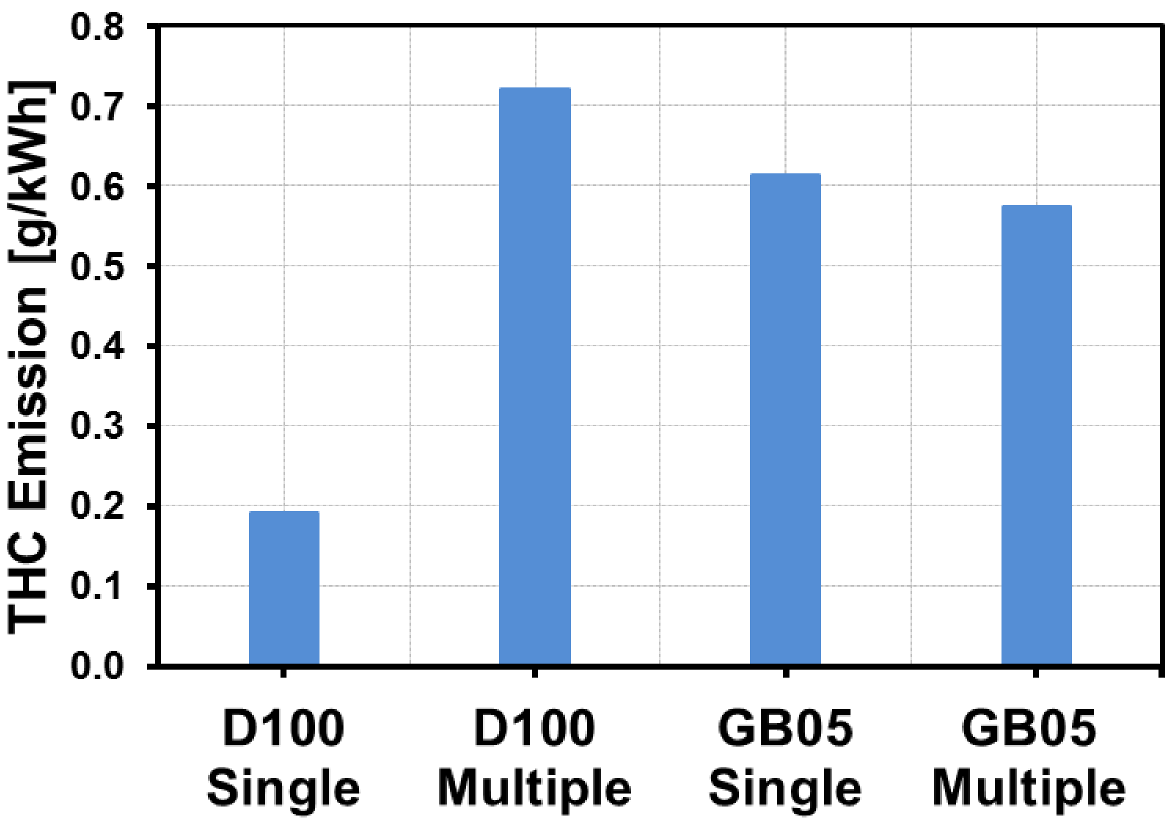


Fig. 5.30 THC emissions

Fig. 5.31 shows NOx emissions of the GCI engine fueled with D100 and GB05 for single and multiple injections. The in-cylinder maximum gas temperature, in-cylinder maximum gas pressure, and combustion efficiency increased proportional to the intake temperature. In this experiment, the intake temperature was increased to be above 80 ^{o}C. The NOx emissions from GB05 for both multiple and single injections seem to be higher than those of multiple injections of D100, and even higher than a single injection of D100. Normally, the oxygenated fuel has a unique characteristic that will produce a higher amount of NOx in the exhaust product. However, further reductions in NOx emissions could be realized by using a split (or two) injection strategy, as reported in a previous study[44]. In this experiment, the pilot injection (70% of the total fuel injected) and main injection (30% of the total fuel injected) for GB05 resulted in lower NOx emissions or at least maintained NOx emissions at a level similar to D100. The NOx amount in the fuel combustion product is always related to the temperature of the combustion process. NOx formation is very sensitive to the temperature history during the cycle. For combustion temperatures greater than 1800 K, the NOx formation rate increases rapidly with increased temperature [89]. Diesel fuels have a cetane number higher than 40 and are very easy to autoignition that they can autoignite before the fuel is thoroughly mixed with air. Contrarily, gasoline fuels have cetane number lower than 30. Due to a sufficiently-mixed charge status before ignition is necessary for CI engines to obtain low emission, therefore fuel characteristics such as those of gasoline that produce longer ignition delay to lead fuel thoroughly mixing with air are expected to be the advantageous. In this study, the blend of gasoline with 5% biodiesel had an estimated cetane number of 17.96 [12], applied on GCI engine with a pilot injection can establish homogeneous combustion of a premixed mixture. The temperature from this burning is expected to be the same in the entire combustion chamber, except near the walls. This condition, in combination with the high volatility characteristics of gasoline, will be as if a lean mixture of air and fuel experiences low maximum temperature combustion during the cycle, promotes lower NOx emission. Many researchers have investigated the effect of cetane number on NOx emissions, then, some of them confirmed that high cetane number increases NOx emission, some other proved that higher cetane number was leading to lower NOx emissions[90]. As reported that gasoline had a low cetane number and a much more extensive ignition delay so that the engine could be run at significantly higher loads with low smoke on gasoline compared to diesel fuel with no detriment to NOx, CO, UHC and fuel consumption [6]. However, the increasing intake temperature leads to more advanced combustion phasing, and the shorter combustion duration results in higher bulk in-cylinder temperatures, which can increase NOx emissions following combustion inside the cylinder. Moreover, the chemical properties of GB05 containing 5% biodiesel in gasoline mean that the oxygen content in the fuel will be higher. A higher oxygen content in the fuel can lead to improved combustion. However, the temperature in the combustion chamber is expected to be higher because a higher amount of oxygen is also present, leading to formation of a higher quantity of NOx in biodiesel-fueled engines[73].

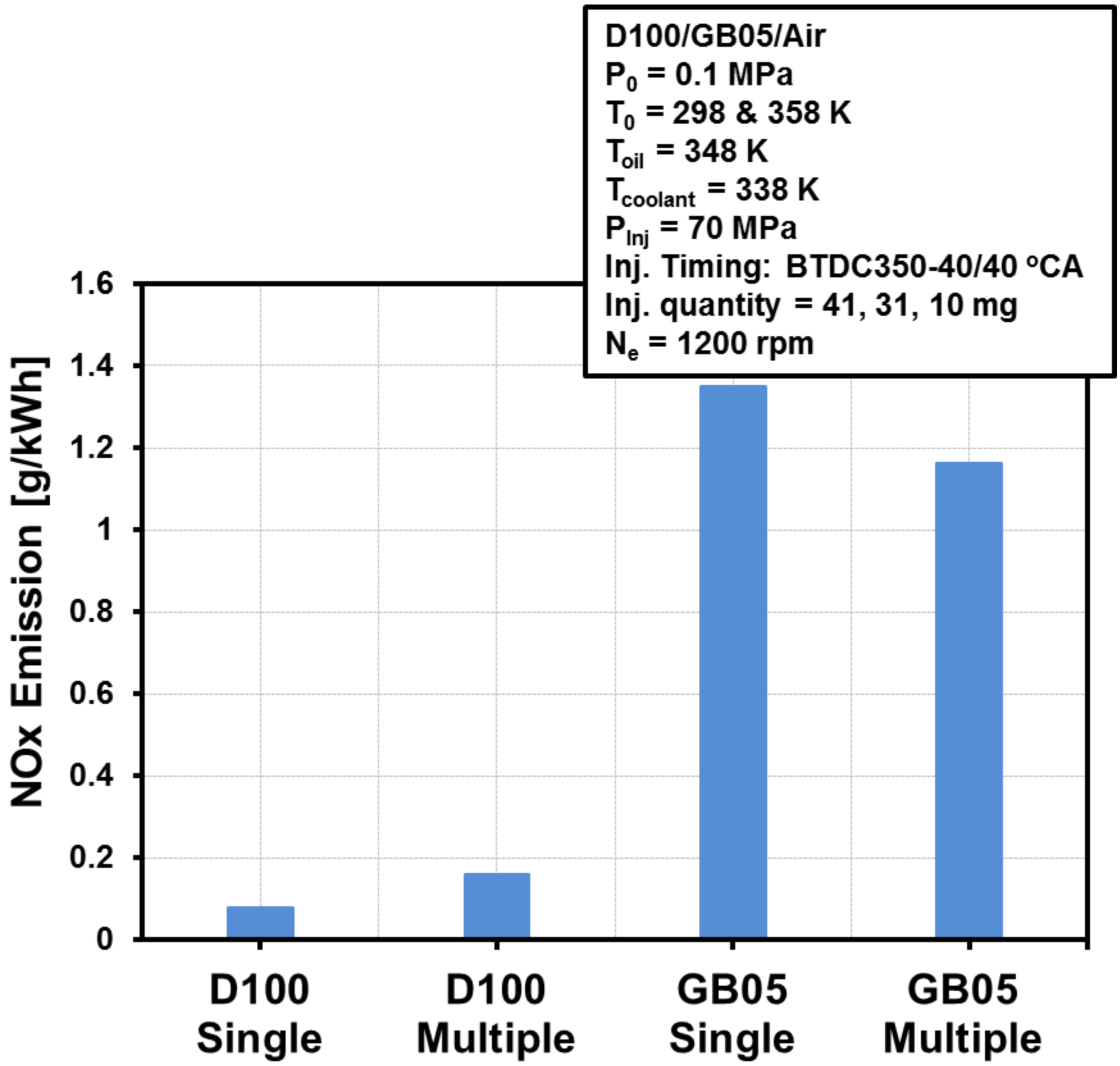


Fig. 5.31 NOx emissions

5.2.6 Summary

Using multiple injections and increasing the temperatures of the intake, oil and engine coolant could result in improved combustion and engine efficiency. The thermal efficiency of D100 decreased when multiple-injection mode was applied. In contrast, the thermal efficiency of GB05 increased when multiple-injection mode was applied. The higher thermal efficiency of D100 is due to the higher LHV of diesel compared to GB05. However, using multiple injections and increasing the intake air, oil, and coolant temperatures could indicate the ability to improve combustion and efficiency of the engine. Multiple injections, which consist of a pilot injection that supplies approximately 70% of the fuel amount at earlier timing, could provide sufficient mixing time for the gasoline-biodiesel blended fuel with a longer ignition delay. Then, supplying the main injection of approximately 30% of the fuel amount could induce combustion. Therefore, improvement in combustion will be obtained, resulting in a higher indicated thermal efficiency.

The combustion efficiency of a single injection of D100, a single injection of GB05, and multiple injections of GB05 are similar and higher than 93%. Meanwhile, the combustion efficiency of multiple injections of D100 is the lowest at approximately 86%. The high combustion efficiency of GB05 is mainly the result of complete combustion due to the longer mixing period of the injected GB05 and its volatile properties. This is supported by a relative high intake air temperature and high wall temperature because the engine oil and coolant temperature were also increased; therefore, more completing burning was achieved. The combustion efficiency increases for two reasons. First, the in-cylinder wall temperatures progressively increase, which helps maintain reactions close to the wall. Second, the more homogeneous mixing process of GB05 causes a proportional reduction of HC and CO emissions.

Multiple injections of GB05 showed decreased CO emissions, which could be due to the pilot injection of GB05. The biodiesel content and using gasoline as a highly volatile fuel in GB05 showed the significant effect of lowering total hydrocarbon and CO emissions. A significant UHC originated from the pilot injection and entered the crevice region, and optimum injection timing should be determined by the trade-off between NOx and UHC/CO emissions. The pilot injection that applied in this study might be too far from the followed main injection prior to the start of combustion, causes a possibility to a wall-impingement of fuel resulted in a higher incomplete combustion of the fuel and THC emission. There is also a possible interaction between the injected fuel and the oil film on the cylinder liner walls during compression stroke caused the UHC emissions might be over-predicted. The other reason is that an early pilot injection may cause the over-lean mixture region, which is the main source of HC emissions.

The increasing intake temperature leads to more advanced combustion phasing, and the shorter combustion duration results in higher bulk in-cylinder temperatures, which can increase NOx emissions following combustion inside the cylinder. Moreover, the chemical properties of GB05 containing 5% biodiesel in gasoline mean that the oxygen content in the fuel will be higher. A higher oxygen content in the fuel can lead to improved combustion. However, the temperature in the combustion chamber is expected to be higher because a higher amount of oxygen is also present, leading to formation of a higher quantity of NOx in biodiesel-fueled engines. Therefore, the next experimental step by using combination of single injection and multiple injection strategy and also the application of EGR and intake boosting to obtain the high efficiency and low emission of engine, simultaneously

5.3 EGR and boosting effects on GCI engine

Previous sections have presented detailed analysis and discussion of the combustion and emission characteristics fueled with gasoline biodiesel blends using direct injection gasoline compression ignition (GCI) for single and multiple injection concept. However, the combustion and emission characteristics of CI engines are also influenced by various other factors, such as fuel injection strategy and its combustion modes. 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. The purpose of using EGR is to retard combustion by the means of controlling the heat release over wide ranges of engine speed, lower the peak pressure, and further expanding the limits of high-load operation[91]. A combination of increase of specific heats capacity and reduction of oxygen (O_2) concentration associated with EGR addition results in suppression of auto-ignition and combustion-phasing retard, allowing high-load operation without knocking phenomena[75,92–94]. However, too much amount of EGR results the lower power output and increase the unburned CO and HC emissions since the constituents forming EGR such as CO_2 and H_2O suffer the rapid burning rate. Therefore, one way to increase the load range is to increase the intake-pressure boost, which intensifies the fuel auto-ignition reactivity. Controlling the combustion phasing is particularly important under boosted conditions since the greater charge mass with boost increases the amount of pressure rise, which is the main reason of knocking[72,95,96] In addition, combustion phasing, can influence on auto-ignition of fuel combustion process from any effect of engine operating conditions. Appropriate strategies of EGR and intake-pressure boost have the capability of extending the CI engine operation range to higher loads and further potential to be used in GCI engines[6]. The principal 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 section 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 single and multiple injection modes for the gasoline-biodiesel blend were utilized. Modification of several initial conditions, such as intake, oil, and coolant temperatures are also conducted. 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, NOx and smoke opacity.

5.3.1 Effect of EGR and single injection strategy

The total fuel consumption per cycle in single injection mode is maintained at 26 mg per cycle and single injection timing at 40 oCA 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.

Fig. 5.32 shows the in-cylinder pressure, temperature and HRR of single injection 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 indicate 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.

Fig. 5.33 shows the effect of EGR on ignition delay when engine operated using single injection 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 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.

Fig. 5.34 shows the effect of EGR on maximum of in-cylinder pressure and Fig. 5.35 peak of pressure rise rate of single injection 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 single injection strategy is presented in Fig. 5.36 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 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 single injection strategy can be seen in Fig.5.37. 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 single injection mode can be observed on Fig. 5.38. 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.

Fig. 5.38 shows the effect of various EGR rate on HC emission of GCI engine running on single injection 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 single injection mode the 20% of EGR rate as an optimum value to obtain lowest HC emission.

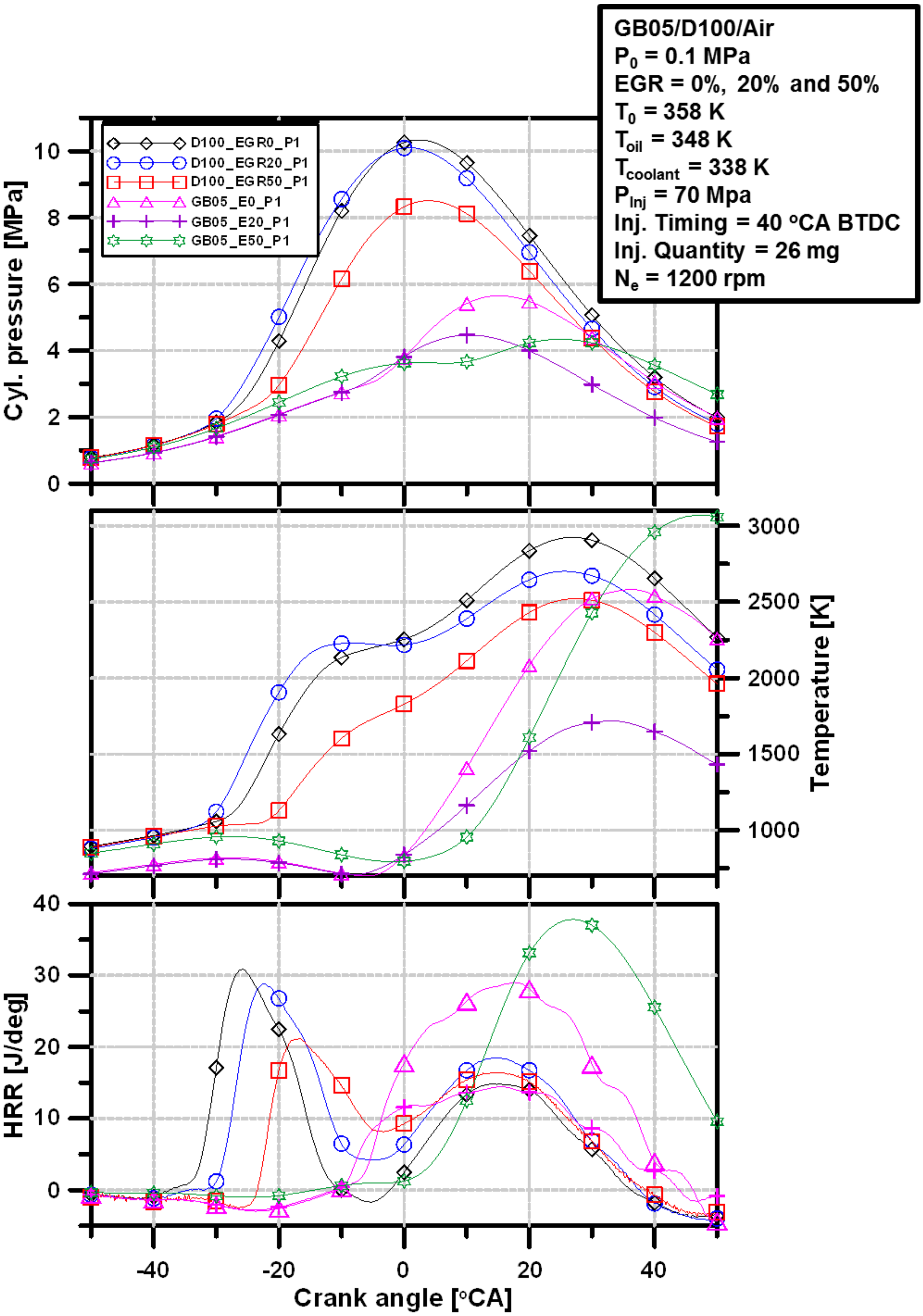


Fig. 5.32 Effect of EGR on cylinder pressure, temperature and HRR of single injection mode

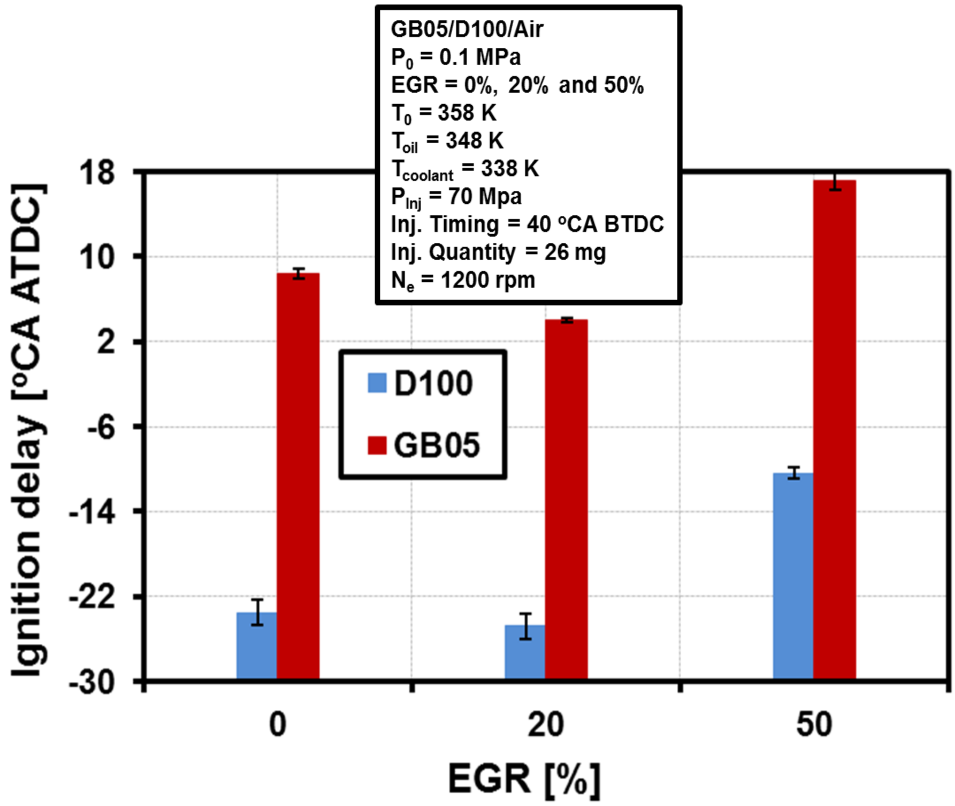


Fig. 5.33 Effect of EGR on ignition delay of single injection mode

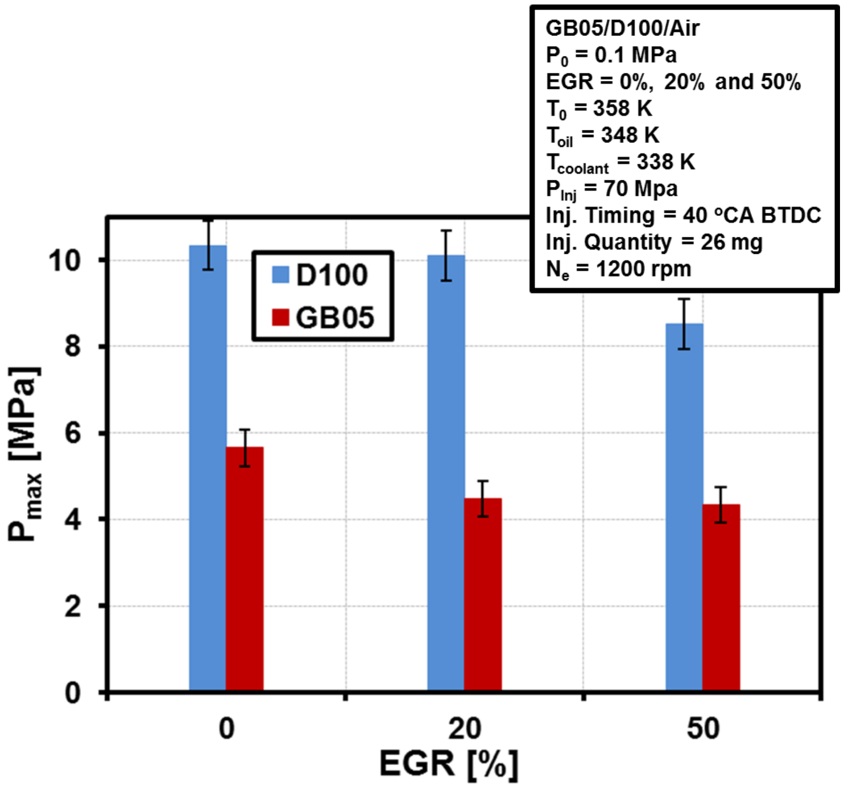


Fig. 5.34 Effect of EGR on in-cylinder max pressure of single injection mode

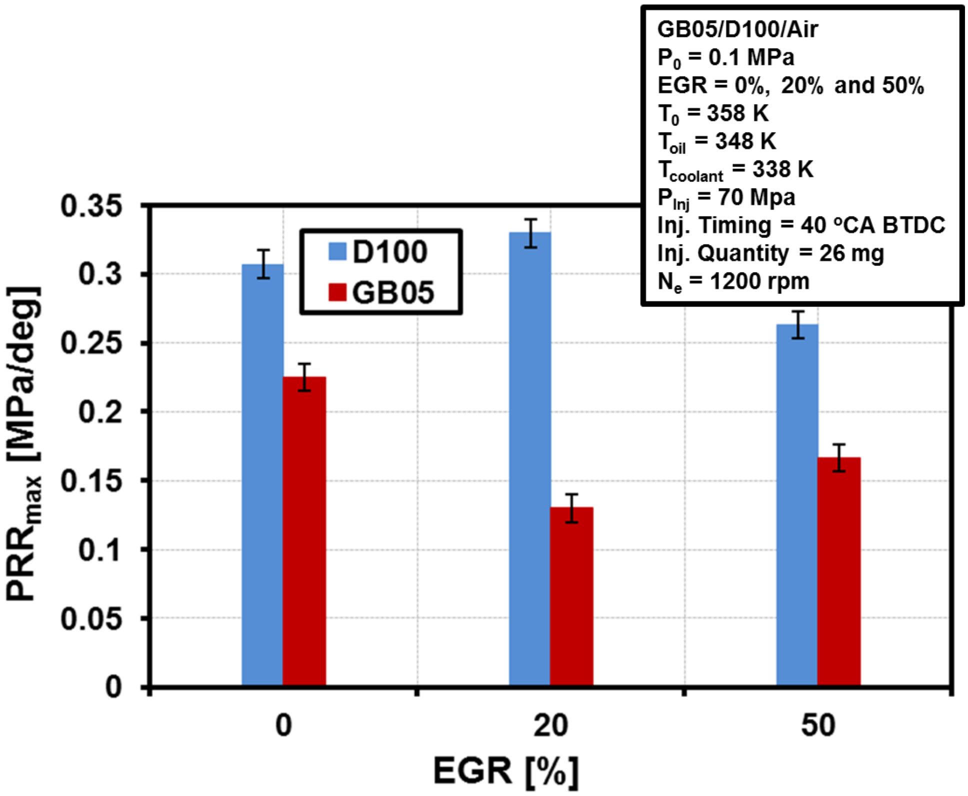


Fig. 5.35 Effect of EGR on peak pressure rise rate of single injection mode

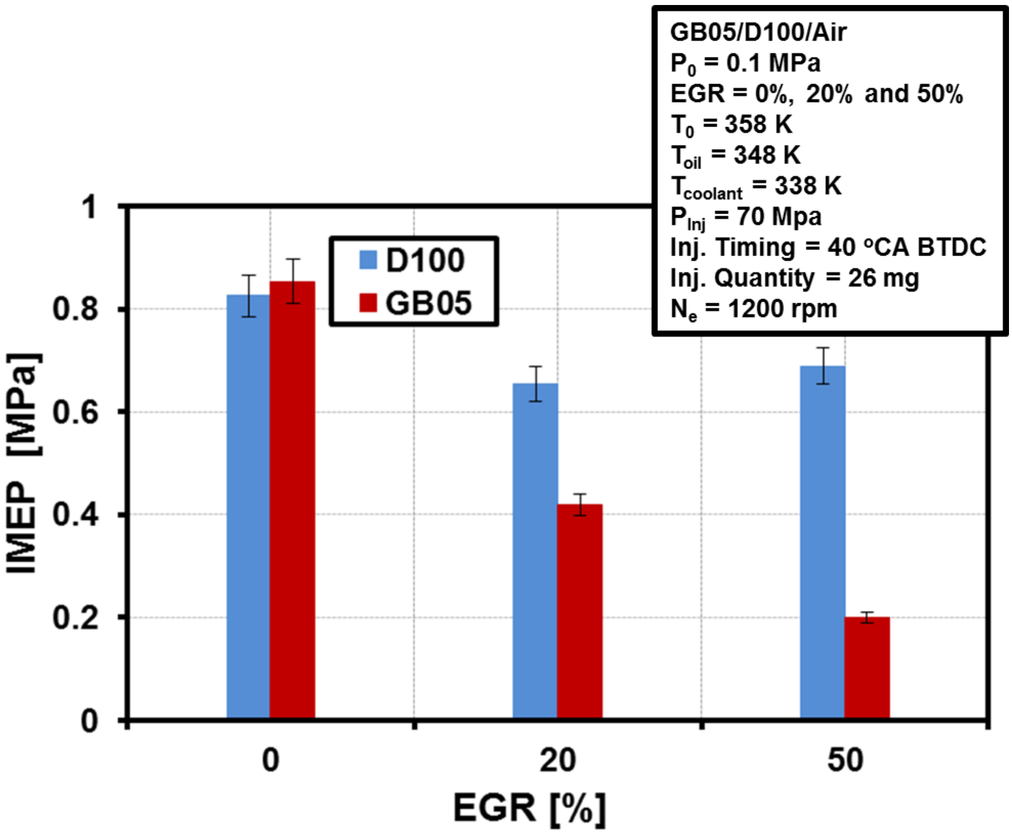


Fig. 5.36 Effect of EGR on IMEP of single injection mode

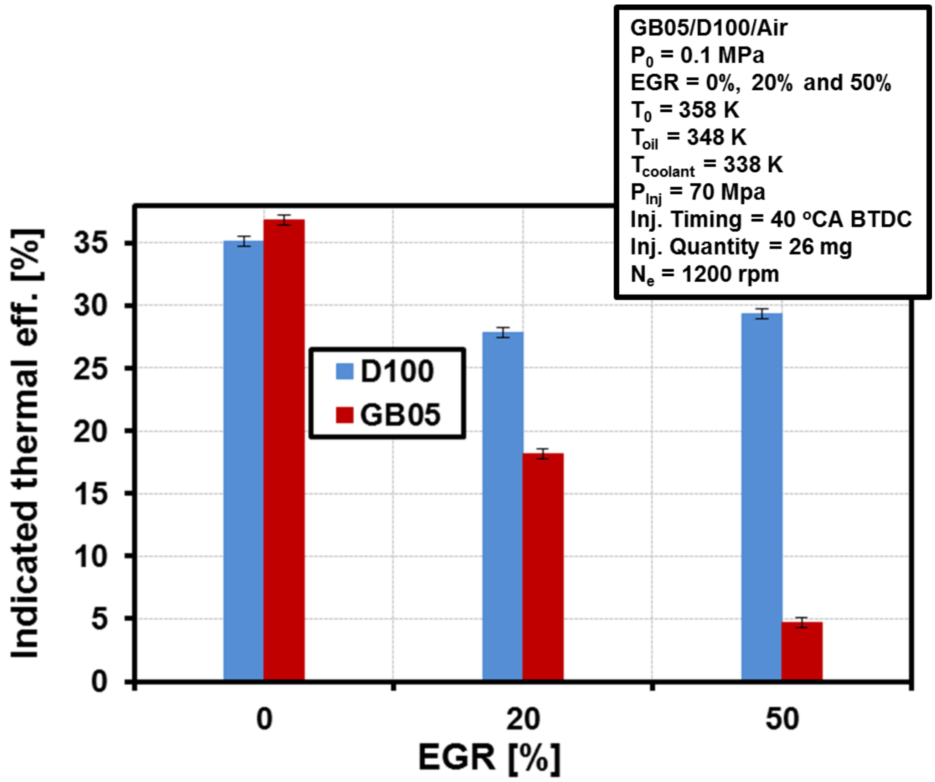


Fig. 5.37 Effect of EGR on Indicated thermal efficiency of single injection mode

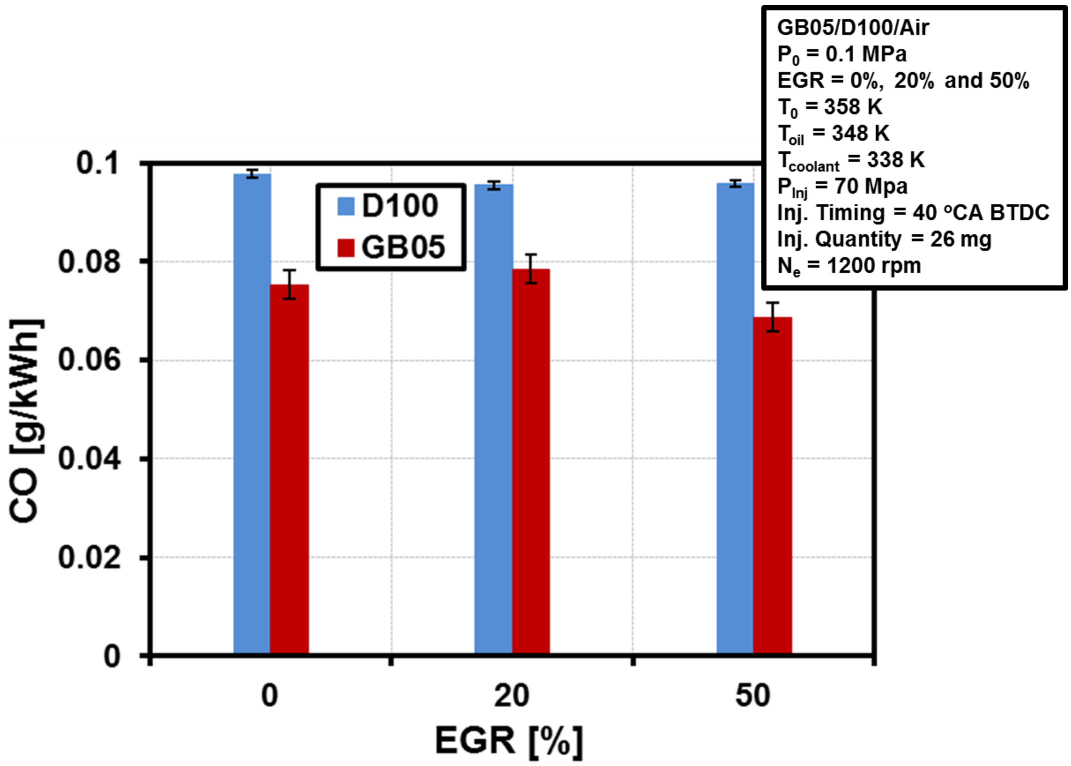


Fig. 5.38 Effect of EGR on CO emission of single injection mode

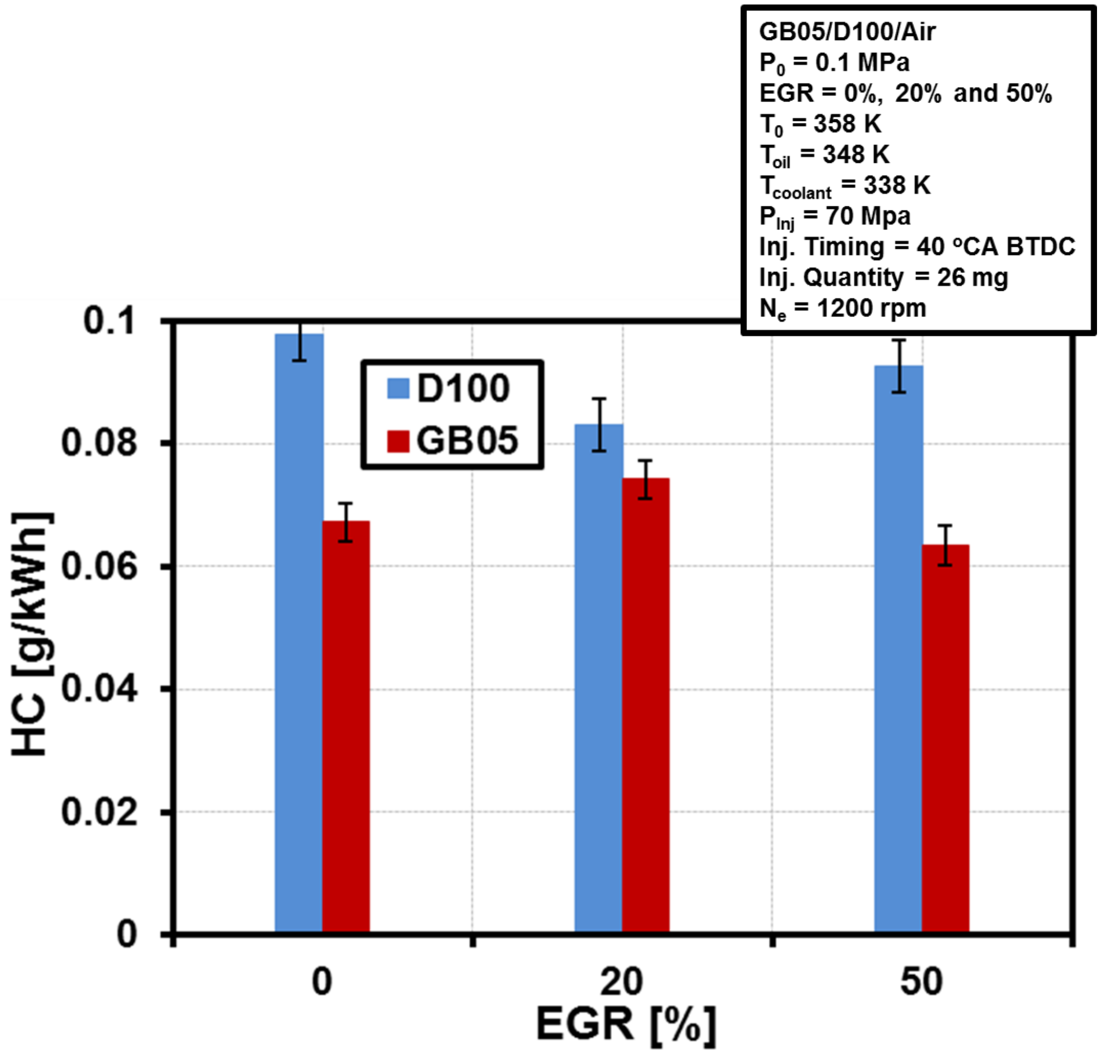


Fig. 5.39 Effect of EGR on HC emission of single injection mode

The NOx emission and its effect by using various EGR rate on GCI engine using single injection mode can be seen in Fig. 5.40. Normally, the increasing of EGR rates will lead to the lower NOx emission. However, in this case, for diesel fuel, the 20% of EGR rate gives highest NOx emission. Even though, when 50% EGR was applied the NOx emission will also decreasing. However, there are no effects of EGR rate variations on NOx emission of GCI engine fueled with gasoline-biodiesel blends. This condition can be seen in the trend of graph that from the three EGR rate variation resulted almost same NOx emission value.

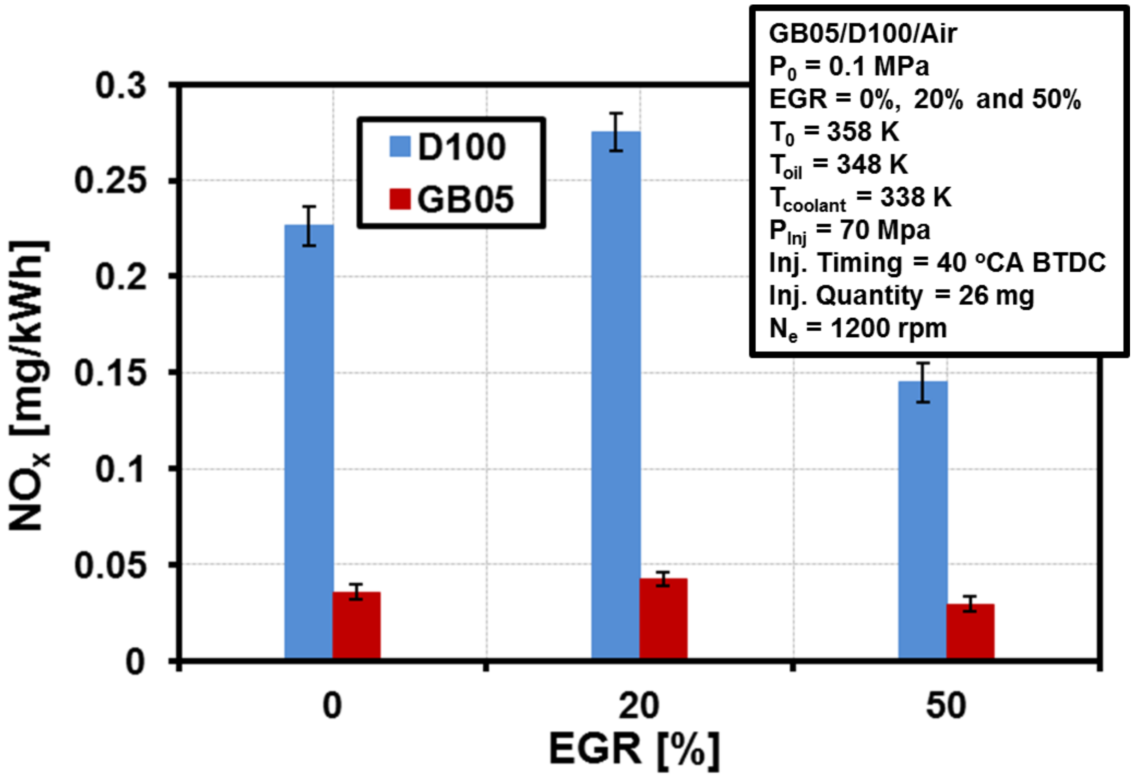


Fig. 5.40 Effect of EGR on NOx emission of single injection mode

The smoke emission of CI engine usually contrasts with NOx emission. When the NOx 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 Fig 5.41. 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 NOx emission is 20%.

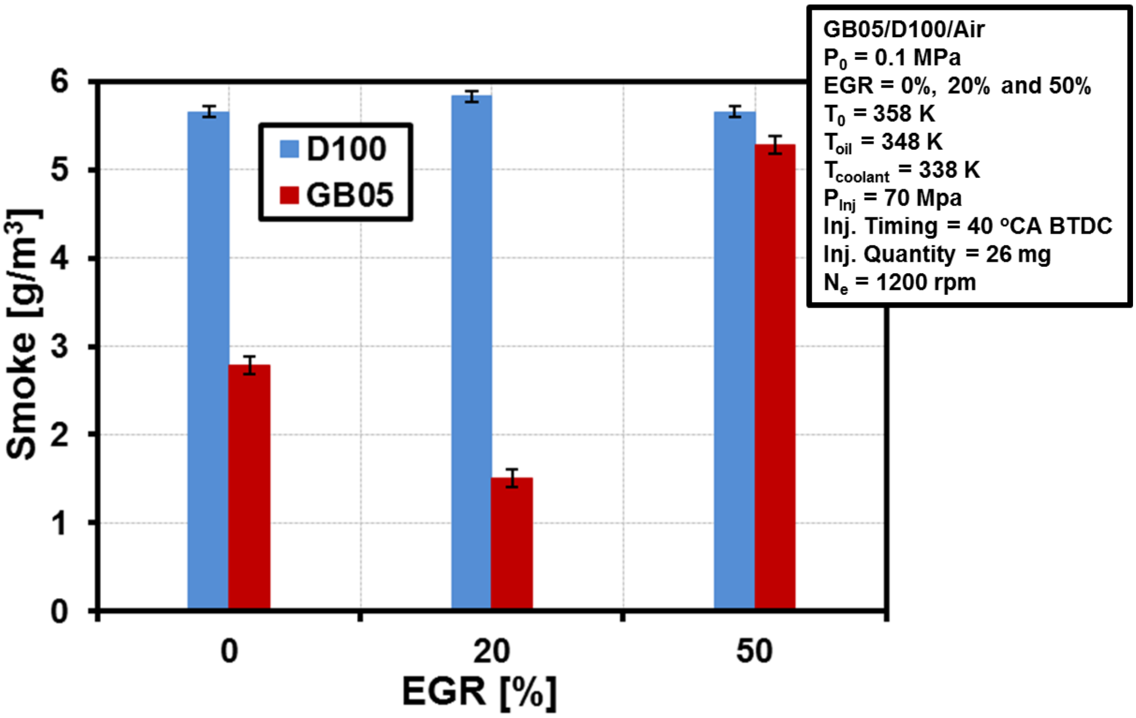


Fig. 5.41 Effect of EGR on smoke emission of single injection mode

5.3.2 Effect of boosting and single injection strategy

To understand the effects of intake boosting on GCI engine fueled with gasoline-biodiesel blends on single injection 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. Fig. 5.42 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 single injection 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 of GCI engine fueled with gasoline-biodiesel blends than gasoline-biodiesel with 0.1 MPa intake boosting. 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.

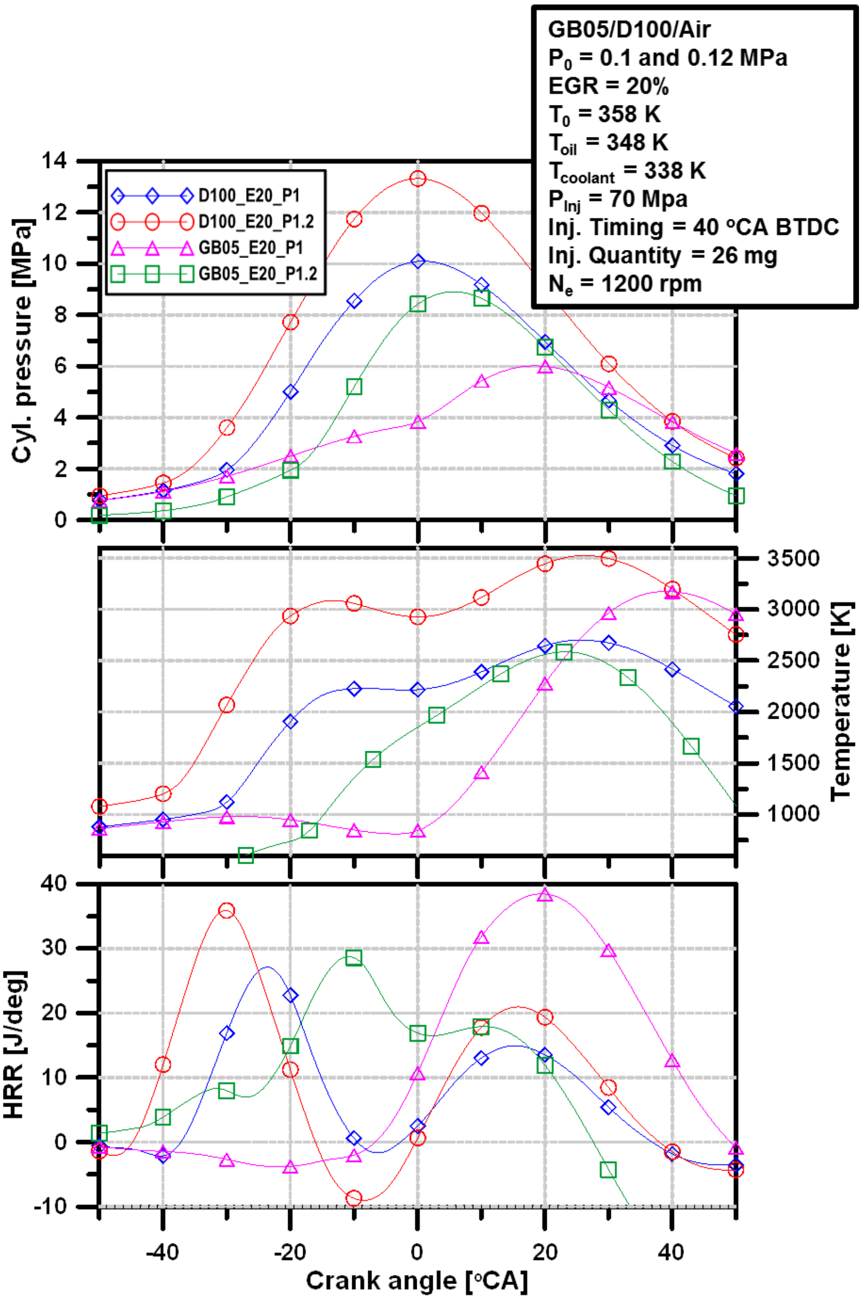


Fig. 5.42 Effect of boosting on cylinder pressure, temperature, and heat release rate single injection mode

The effect of intake boosting on ignition delay of GCI engine using single injection strategy is presented in Fig. 5.43. The intake boosting gives effect on the lower ignition 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 ^{o}CA BTDC, then the 0.12 MPa intake boosting lead to the slightly earlier of ignition delay timing at around 27 ^{o}CA BTDC. Similar trend happened on gasoline-biodiesel fuel, that ambient pressure of intake boosting resulted ignition delay timing at around 11 ^{o}CA BTDC, then when 0.12 MPa intake boosting was applied the ignition delay timing also more advanced at around 2 ^{o}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, however, too long ignition delay timing sometimes caused problem in the engine emission and efficiency.

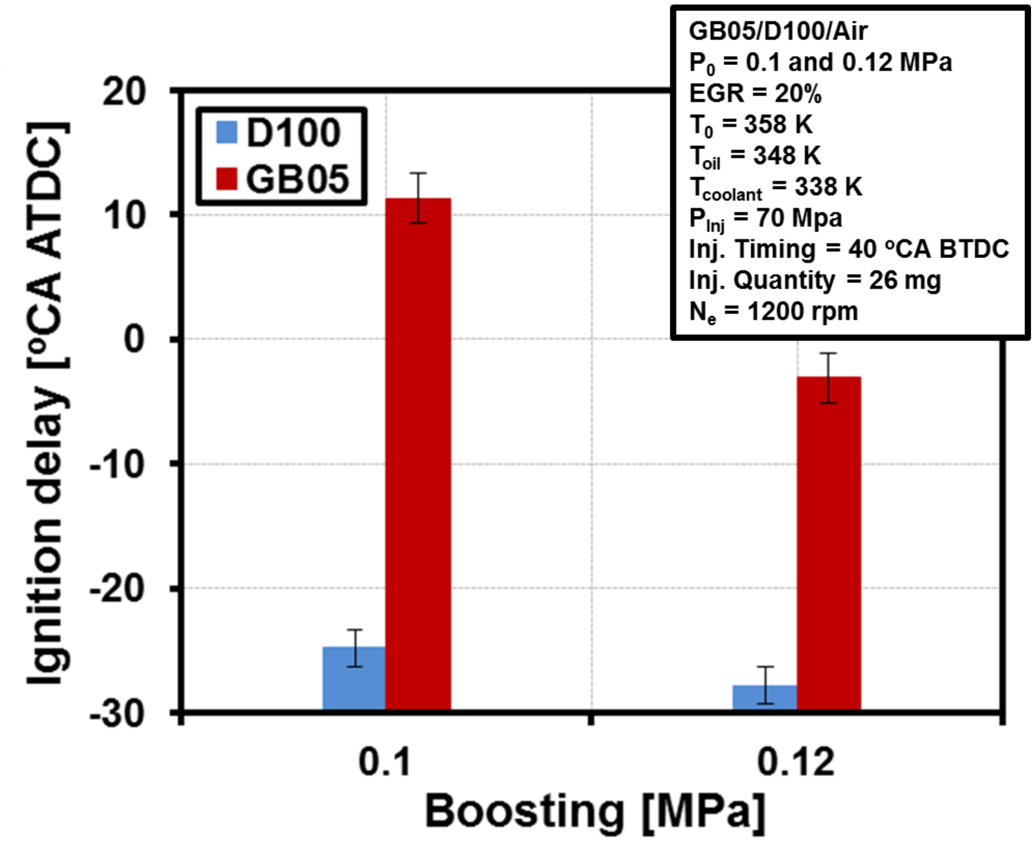


Fig. 5.43 Effect of boosting on Ignition delay of single injection mode

Fig. 5.44 shows the effect of various intakes boosting on maximum of in-cylinder pressure and Fig. 5.45 shows the effect of various intakes boosting on 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.

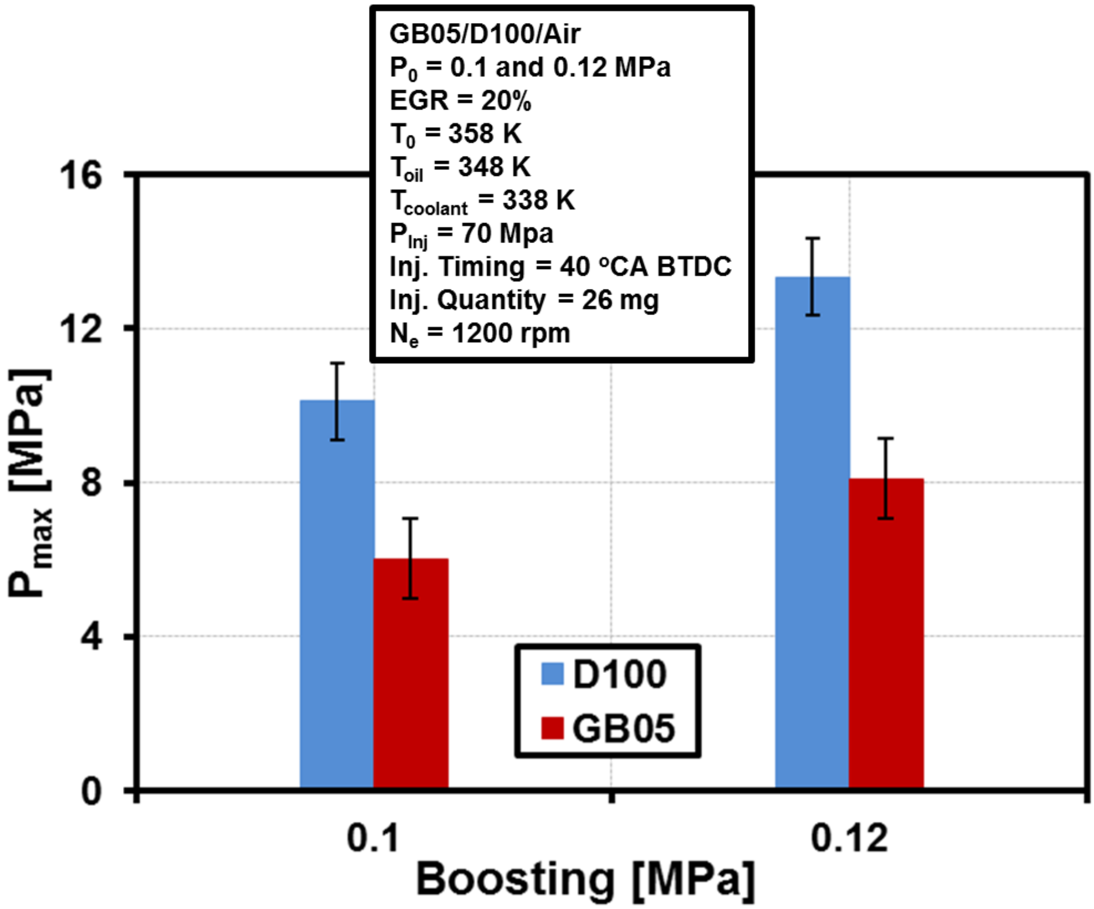


Fig. 5.44 Effect of boosting on in-cylinder max pressure of single injection mode

Effect of various intakes boosting on IMEP if GCI engine fueled with gasoline-biodiesel blends in single injection strategy can be seen in Fig. 5.46. 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 centane number of gasoline, which resulted higher-pressure rise rate as shown in Fig. 5.45. Fluctuate of in-cylinder pressure may lead to the unstable combustion and resulted the lower IMEP value.

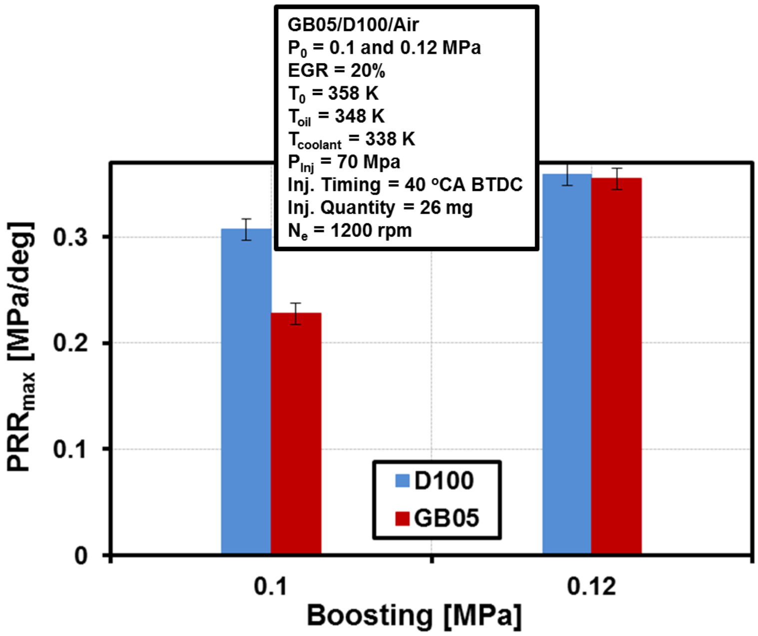


Fig. 5.45 Effect of boosting on peak pressure rise rate of single injection mode

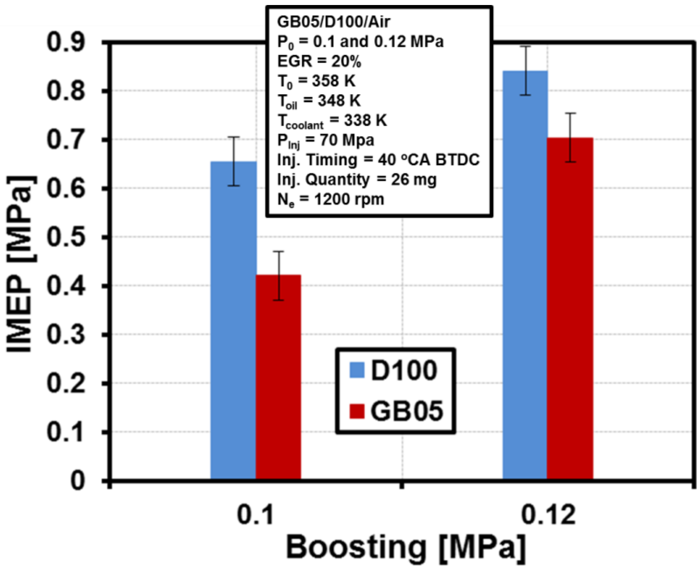


Fig. 5.46 Effect of boosting on IMEP of single injection mode

The indicated thermal efficiency of GCI engine using single injection strategy affected by various intake boosting is presented in Fig. 5.47. 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 the indicated thermal efficiency of GCI engine with diesel fuel is higher than gasoline-biodiesel blend.

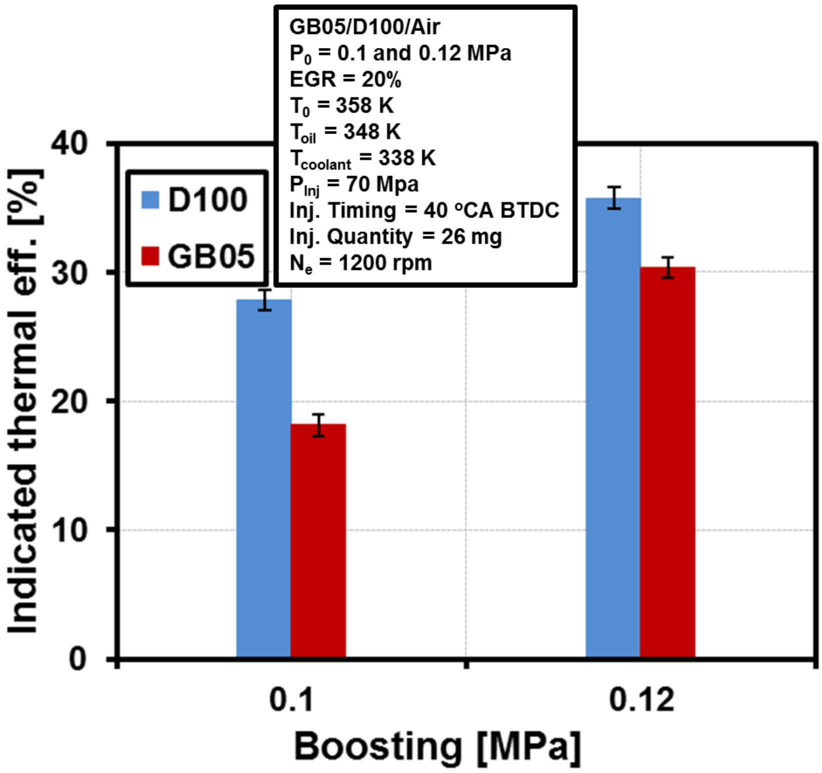


Fig. 5.47 Effect of boosting on indicated thermal efficiency of single injection mode

Fig. 5.48 shows the effect of intake boosting on CO emission of GCI engine using single injection 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 form 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 Fig. 5.48. 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 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.

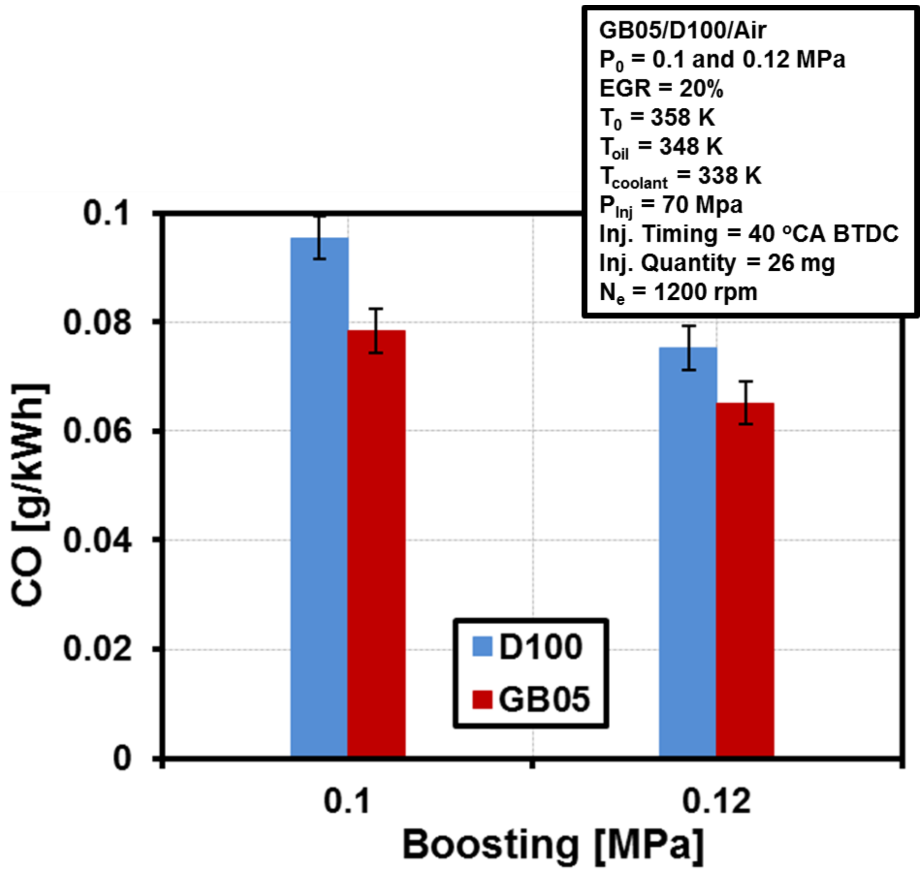


Fig. 5.48 Effect of boosting on CO emission of single injection mode

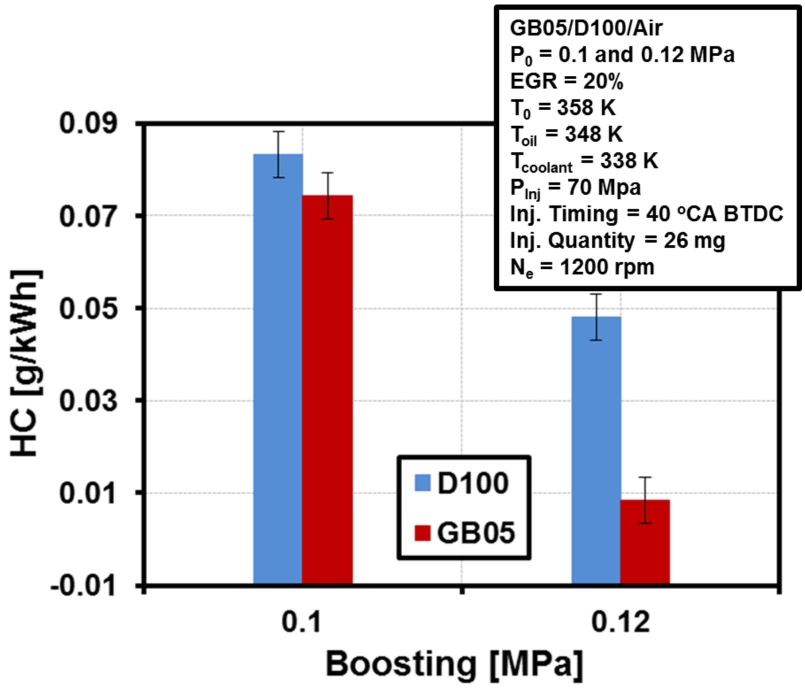


Fig. 5.49 Effect of boosting on HC emission of single injection mode

Fig. 5.50. shows the effect of intake boosting on NOx emission of GCI engine with single injection strategy. Overall, the NOx emission of GCI engine fueled with diesel is higher than GCI engine fueled with gasoline-biodiesel blend when using single injection 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 NOx 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 NOx 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 NOx emission to be around 0.2 mg/kWh.

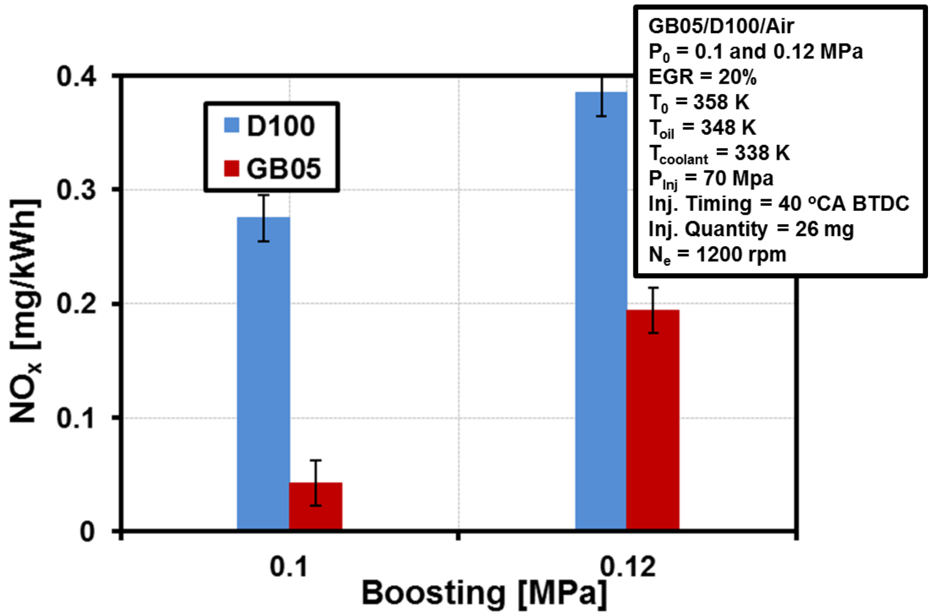


Fig. 5.50 Effect of boosting on NOx emission of single injection mode

The effect of intake boosting on smoke emission of GCI engine with single injection strategy can be seen in Fig. 5.51. Smoke emission of GCI engine fueled with diesel fuel is very high almost 6 g/m3 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/m3. Increasing intake boosting to be 0.12 MPa makes smoke emission of GCI engine fueled with diesel fuel decrease very significant around 3 g/m3. 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/m3. If the point of view of GCI engine fueled with gasoline-biodiesel blend running on single injection 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.

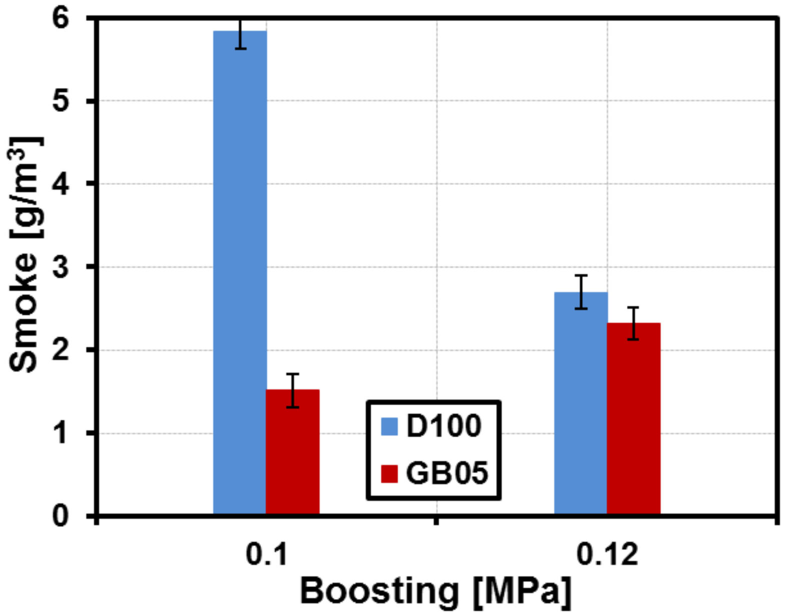


Fig. 5.51 Effect of boosting on smoke emission of single injection mode

5.3.3 Effect of EGR and multiple injection strategy

A same energy input was also implemented to the multiple injection mode. The total injected fuel including pilot and main injection is 16 mg every cycle. Fig. 5.52. shows the effect of various EGR rates on in-cylinder pressure, temperature and HRR of GCI engine using multiple injection strategy. The highest in-cylinder pressure is for GCI engine fueled with gasoline-biodiesel blend with 20% EGR rate. In multiple injection mode of GCI engine without EGR, the pilot injection provides 70% of the fuel amount, which was injected and satisfies the mixing criteria before auto ignition occurs. Then, the main injection was applied with 30% of the total fuel amount to control the combustion in shorter period. The effect of this strategy is premixed combustion of the pilot injection become weaker, which decreased temperature and pressure before the main injection. Furthermore, it was resulted the peak values of the in-cylinder pressure and heat release rate decreased. This strategy is advantageous for gasoline-biodiesel blend fuel. However, the utilization of various EGR rate combined pilot and main injection strategy showed the increasing of cylinder pressure, in-cylinder temperature and heat release rate, which is sometimes become disadvantage due to the emission deterioration.

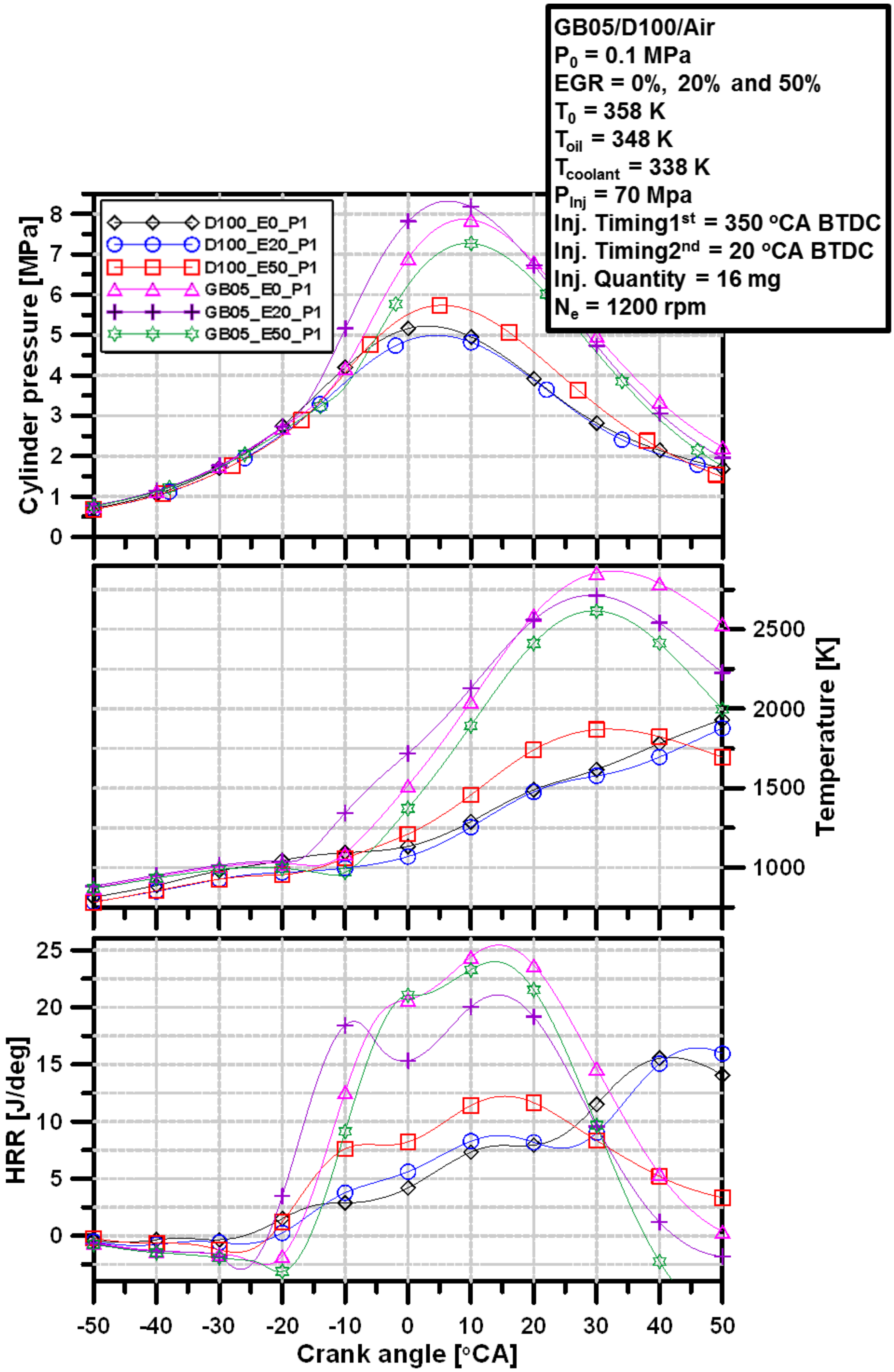


Fig. 5.52 Effect of EGR on cylinder pressure, temperature and HRR of multiple injection mode

The effect of various EGR rates on ignition delay of GCI engine using multiple injection mode can be seen in Fig. 5.53. Ignition delay of diesel fuel is longer than gasoline-biodiesel blend for GCI multiple injection combustion without EGR. This condition suspected due to the diesel fuel which has lower volatile property than gasoline. Then, when the 70% of fuel is injected earlier, a huge amount of fuel will be impingement on the wall of cylinder, which will leads to the difficulty of auto ignition, even though, the main injection was injected near before TDC. The narrower injection angle will be useful to solve this issue. For diesel fuel, the ignition delay more retarded when 20% of EGR was applied, then ignition delay become very early in the 50% EGR condition. Slightly different with diesel fuel, for gasoline-biodiesel blend, the ignition delay with 20% EGR is shorter than without EGR. However, when EGR rate increased to be 50% the ignition delay is back to be retarded.

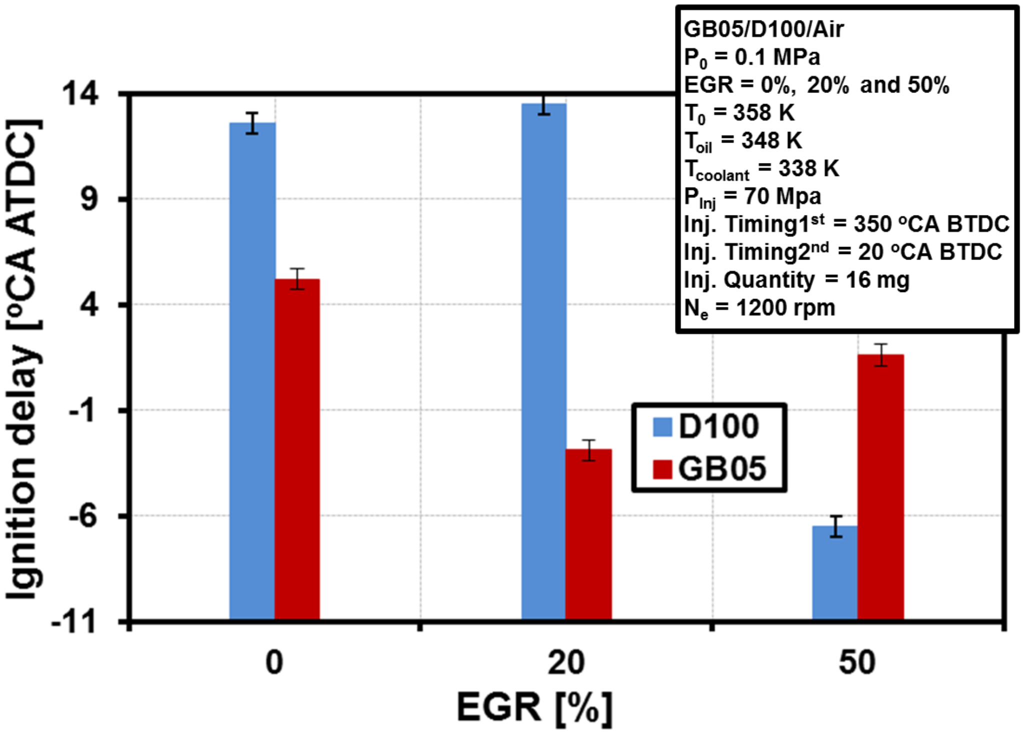


Fig. 5.53 Effect of EGR on ignition delay of multiple injection

Fig. 5.54 shows the effect of EGR on maximum in-cylinder pressure and Fig 5.55 shows the effect of EGR on maximum pressure rise rate of GCI engine using multiple injection combustion strategy. The increasing 20% EGR rate for GCI engine fueled with diesel with multiple injection mode does not gives any effect on maximum in-cylinder pressure. However, maximum in-cylinder pressure slightly increasing when the EGR rate increase up to 50%. While, for gasoline-biodiesel blend the maximum in-cylinder pressure a little increase when EGR rate 20%, and more or less same with 0% EGR when 50% EGR was applied. In case of PRRmax, the increasing EGR rate, the increasing PRRmax of GCI engine fueled with diesel using MPCI mode. However, for GCI engine fueled with gasoline-biodiesel blend using MPCI mode, there is almost no effect on PRRmax when the EGR rate is increases.

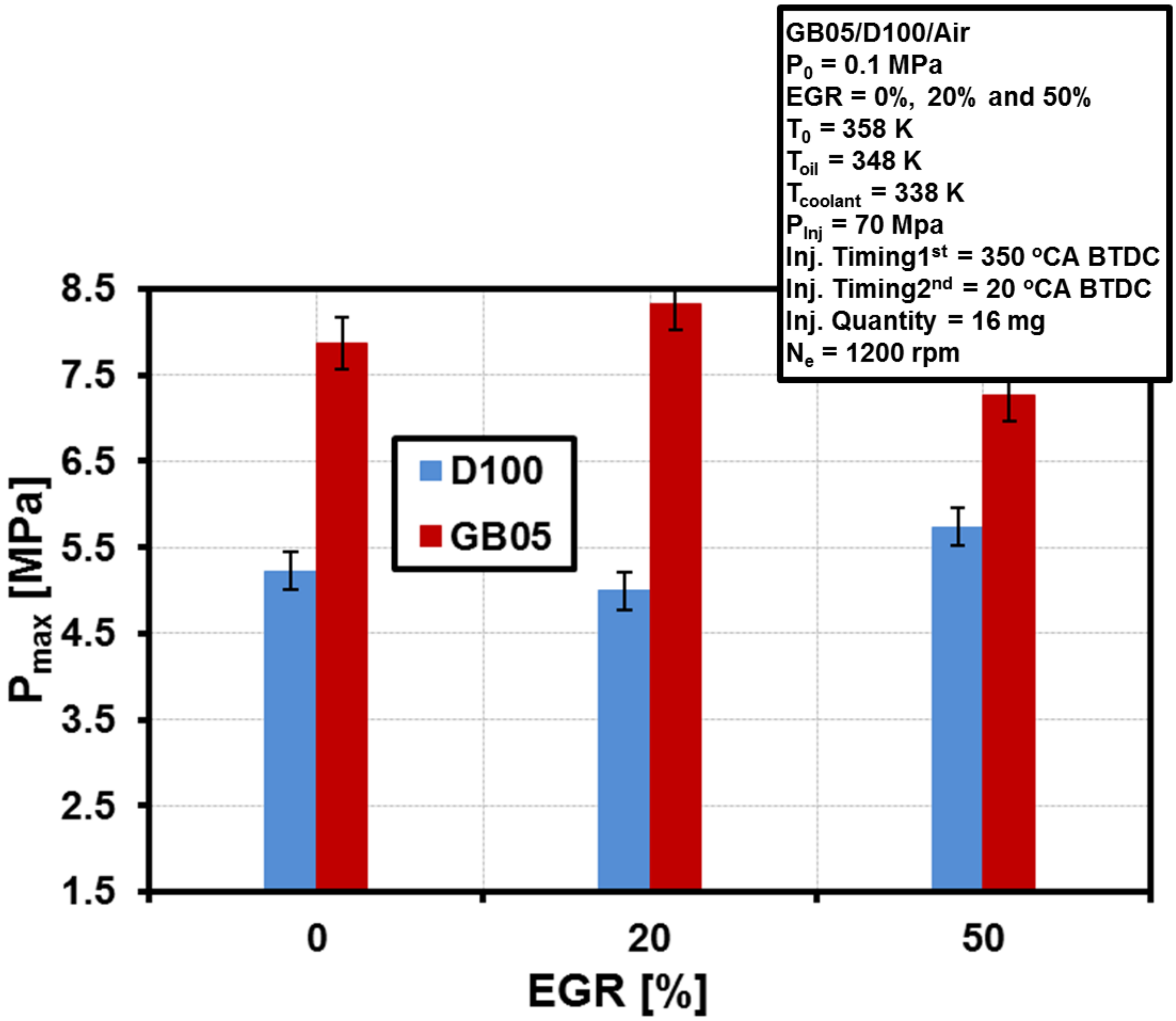


Fig. 5.54 Effect of EGR on In-cylinder max pressure of multiple injection

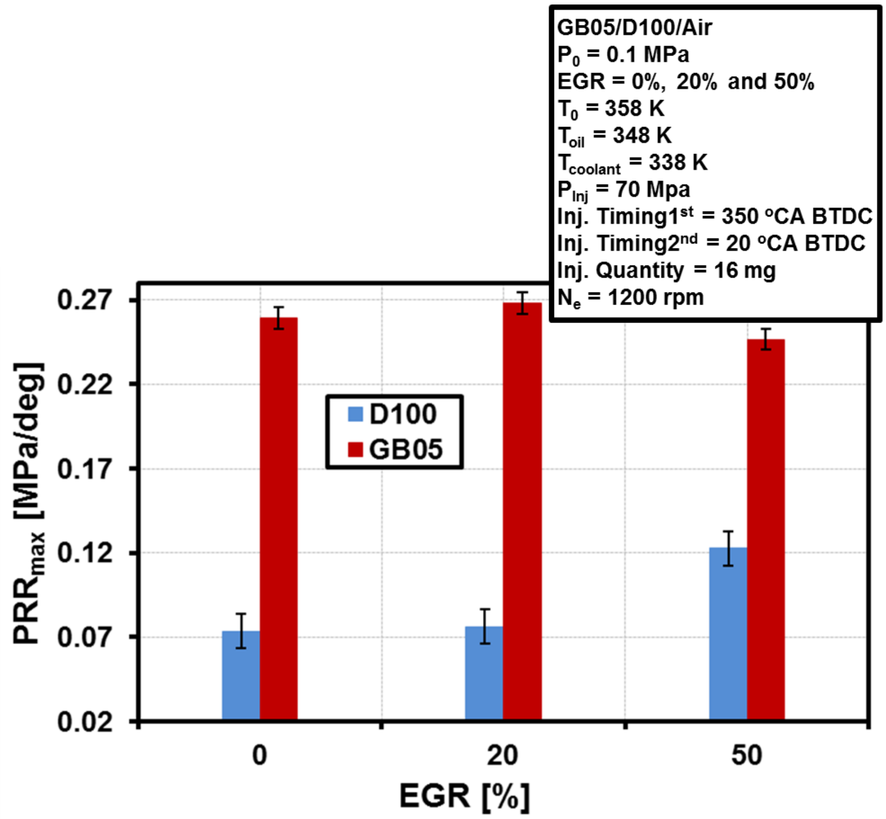


Fig. 5.55 Effect of EGR on peak pressure rise rate of multiple injection

The effect of EGR rate on IMEP of GCI engine running on multiple injection mode can be seen in Fig. 5.56. For GCI engine fueled with diesel, there is no effect on IMEP when the EGR rate is increased. However, the increasing EGR rate makes the IMEP of GCI engine fueled with gasoline-biodiesel blend decreased. The multiple injection application combined with the increasing the temperature of air intake, engine oil and engine coolant, the obtained IMEP of the GCI engine fueled with gasoline-biodiesel blend more and less higher than diesel fuel. Even though gasoline-biodiesel blend has lower energy content and lower ignition sensitivity compared to diesel fuel, the combustion quality of gasoline-biodiesel blend can be improved by using multiple injection mode, agreed with that already reported previously [97]. Furthermore, by using EGR application, the IMEP can be maintained and emission can be improved.

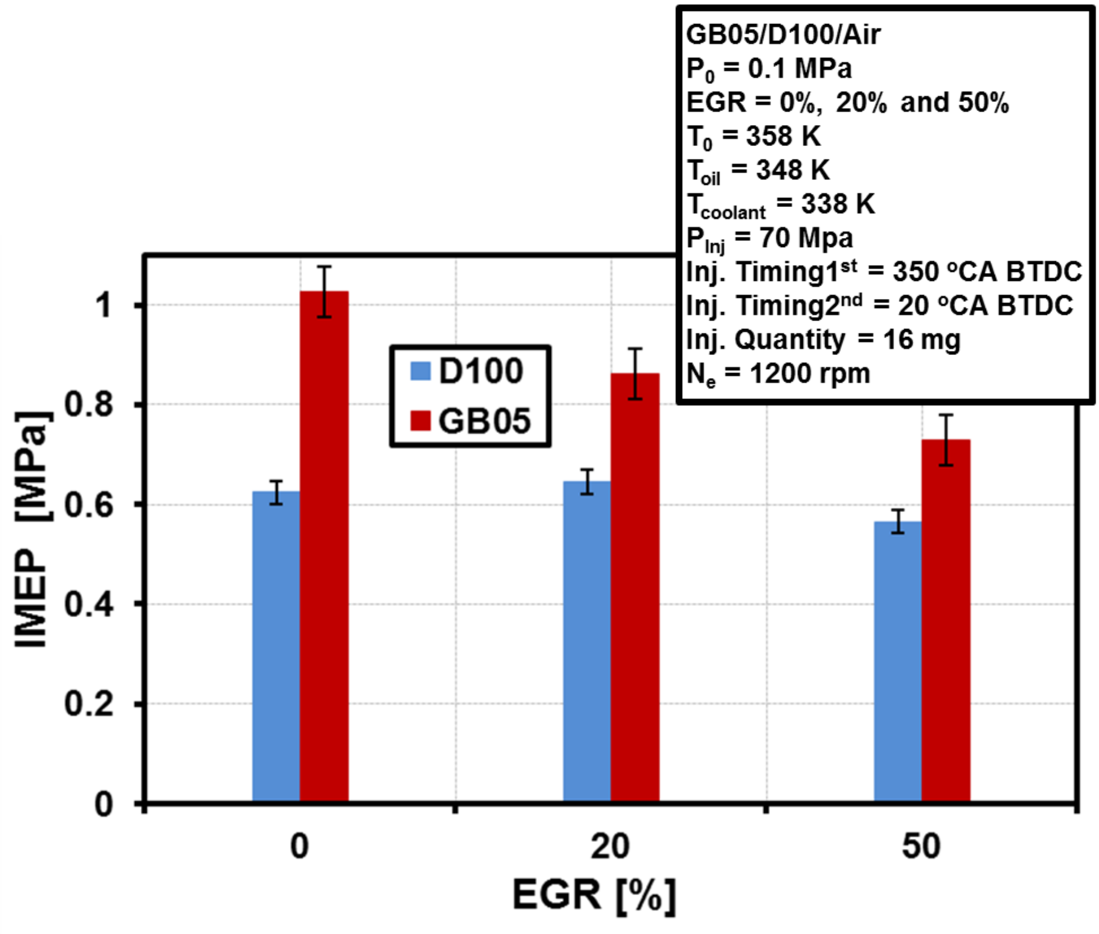


Fig. 5.56 Effect of EGR on IMEP of multiple injection

Indicated thermal efficiency of GCI engine using multiple injection mode that obtained from various EGR rates can be observed in Fig. 5.57. For diesel fuel, the increasing of EGR rate to be 20% caused a slightly increasing indicated thermal efficiency. However, higher EGR rate to be 50% lead to the decreasing of indicated thermal efficiency. A different trend obtained from gasoline-biodiesel blend, the increasing EGR rate, the decreasing of indicated thermal efficiency. For GCI engine without EGR, the higher thermal efficiency of diesel is due to the higher LHV of diesel compared to gasoline-biodiesel blend. However, using multiple injection mode and increasing the intake air, oil, and coolant temperatures could indicate the ability to improve combustion and efficiency of the engine. As reported previously[97], multiple injection mode, which consist of a pilot injection that supplies approximately 70% of the fuel amount at earlier timing, could provide sufficient mixing time for the gasoline-biodiesel blended fuel with a longer ignition delay. Then, supplying the main injection of approximately 30% of the fuel amount could induce combustion. Therefore, improvement in combustion will be obtained, resulting in a higher indicated thermal efficiency. However, by using EGR rate the indicated thermal efficiency on the contrary, not getting better, but getting worse. This can be related to a higher specific heat of dilution in the EGR mixture that prevent the normal combustion process which deteriorates the burning rate, but leads to reduction in combustion temperature inside the combustion chamber.

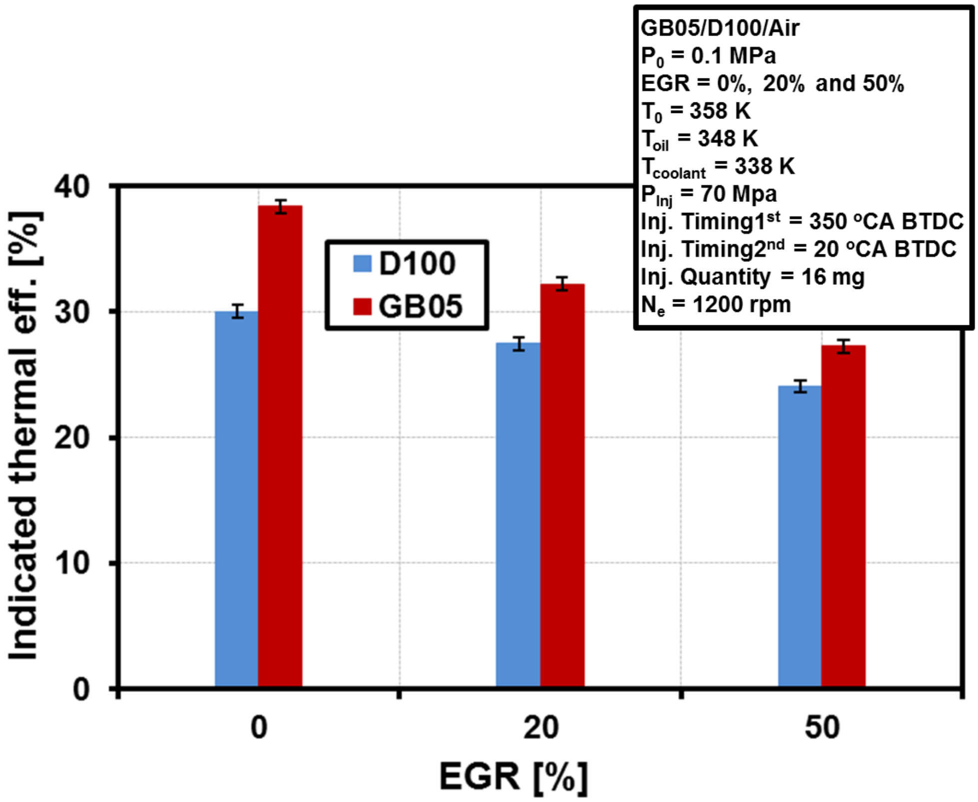


Fig. 5.57 Effect of EGR on indicated thermal efficiency of multiple injection

The effect of EGR rate on CO emission of GCI engine using multiple injection mode can be seen in Fig. 5.58. Increasing EGR to be 20% for GCI engine fueled with diesel fuel, increasing of CO emission. However, when 50% EGR rate was applied the CO emission of GCI engine is almost same when without EGR. Meanwhile, for GCI engine fueled with gasoline-biodiesel blend, the CO emission trend is that the increasing EGR rates, the increasing CO emission. CO emission of GCI engine fueled with diesel fuel is lower than GCI engine fueled with gasoline-biodiesel blend when EGR rates are below 20%. However, the CO emission of GCI engine fueled with diesel fuel lower than gasoline-biodiesel blend when the EGR rate increase to be 50%. This condition is agreed with the previous study[97], that CO emissions are generated mainly by incomplete combustion of fuel. In case of diesel with EGR below 20% compared with gasoline-biodiesel fuel, this fuel has a higher cetane number and is a so-called reactive fuel, has a shorter ignition delay, therefore more incomplete combustion occurs. While, when EGR rate more than 50%, diesel fuel probably more complete combustion. However, contrary, for gasoline-biodiesel blend 50% EGR rate, may lead the incomplete combustion due to its higher volatile, lower cetane number and longer ignition delay with high possibility misfiring and knocking.

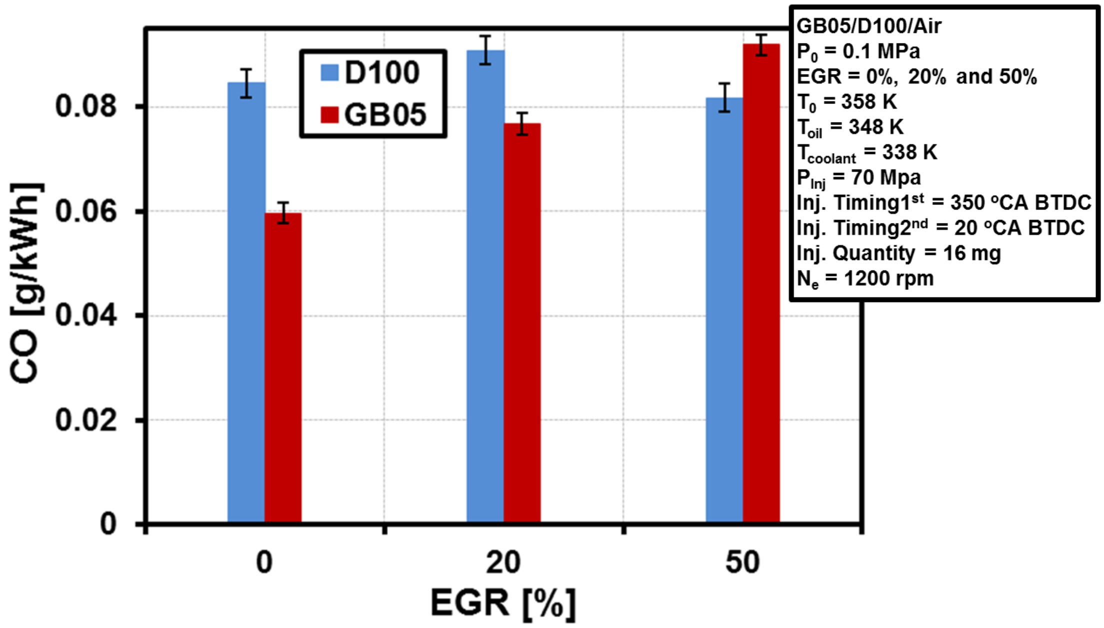


Fig. 5.58 Effect of EGR on CO emission of multiple injection

HC emission of GCI engine using multiple injection mode, which obtained from various EGR rates is revealed in Fig. 5.59. Both for GCI engine fueled with diesel fuel or gasoline-biodiesel blend show that the HC emission trend increases with the increasing of EGR rates. Similar with CO emission, a higher amount of HC emissions is a result of incomplete combustion of the fuel. In In the multiple injection strategy, the higher HC emissions indicate that the injection timing and mass distribution may be far from its optimum point[97]. An early pilot injection may cause the over-lean mixture region, which is the main source of HC emissions, this condition agreed with the previous study by Kim and Bae[85]. Ra et al. [6] showed that a significant UHC originated from the pilot injection and entered the crevice region, and optimum injection timing should be determined by the trade-off between NOx and UHC/CO emissions. There is also a possible interaction between the 70% pilot injected fuel and the oil film on the cylinder liner walls during compression stroke caused the UHC emissions might be over-predicted. Furthermore, by implementation of EGR it is also very possible the exhaust air dilution makes more deterioration in the HC emission of GCI engine.

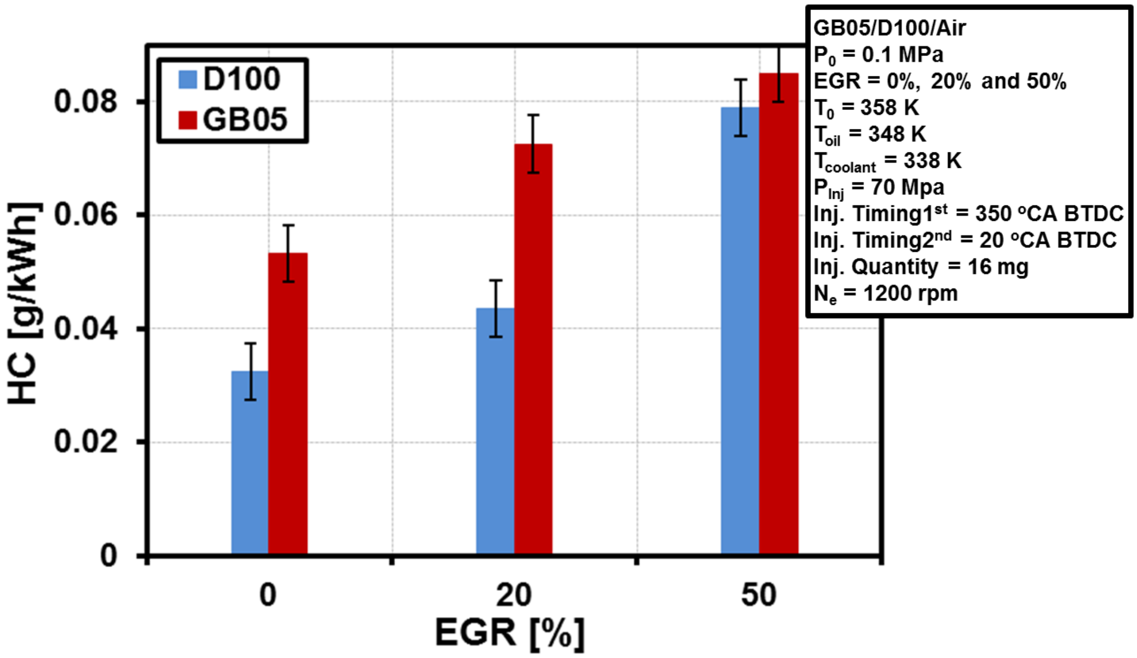


Fig. 5.59 Effect of EGR on HC emission of multiple injection

The effect of EGR rates on NOx emissions of GCI engine using multiple injection strategy can be observed in Fig. 5.60. Meanwhile, the effect of EGR rates on smoke emission of GCI engine using multiple injection mode is presented in Fig. 5.61. Commonly, in CI engine, the analysis of NOx is always related to smoke emission. The normal relation between NOx and smoke emission in CI engine is when NOx emission high, then the smoke emission will always low. To understand this relation easier the graph of "soot or NOx Island" can be used as reference[1]. Therefore, currently the advanced technology on CI engine is how to obtain an optimum relation between soot/smoke and NOx as lowest as possible for both of them. The results from Fig. 5.60 and 5.61, reveal that the optimum strategy in multiple injection mode for GCI engine fueled with gasoline-biodiesel is when the EGR rate at 50% was applied, while for GCI engine fueled with diesel without EGR is the best strategy to obtain low NOx and as well as smoke emission.

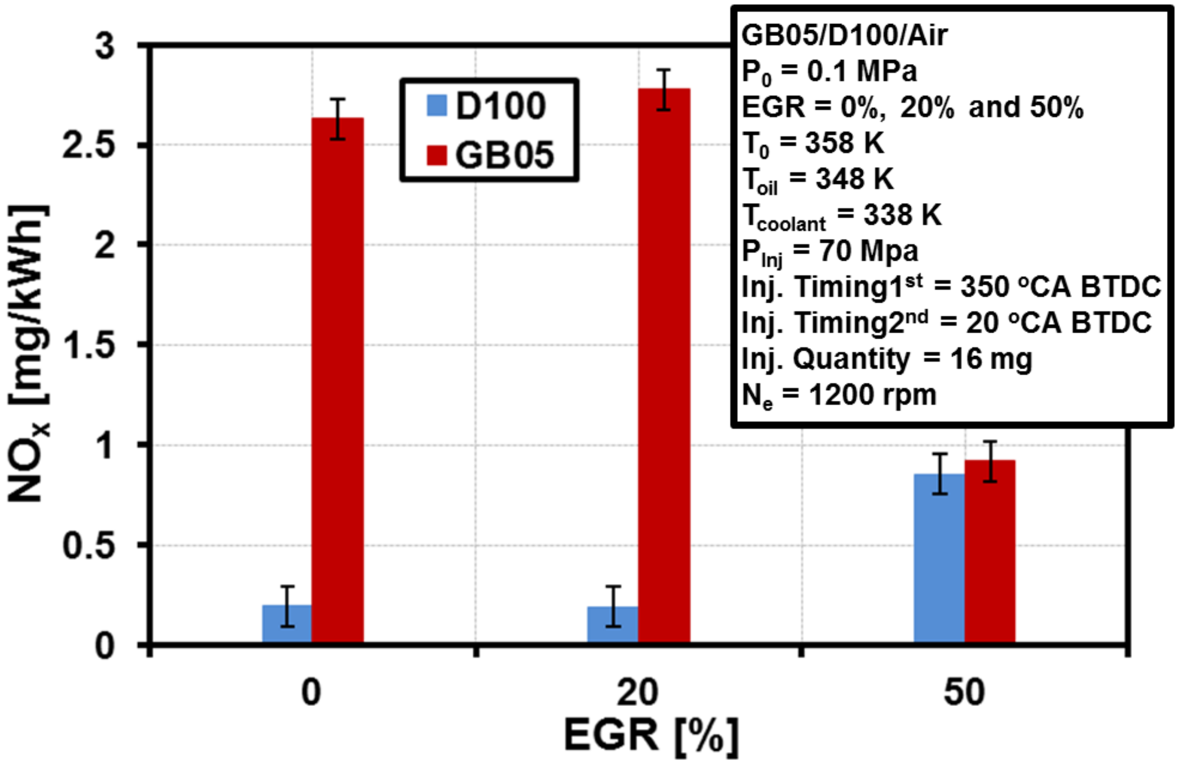


Fig. 5.60 Effect of EGR on NOx emission of multiple injection

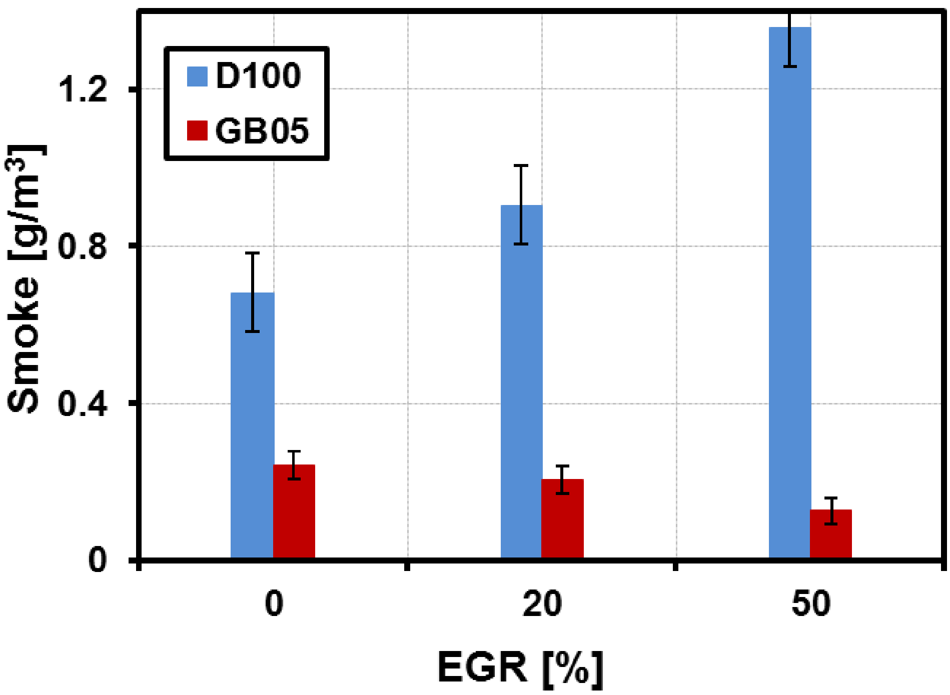


Fig. 5.61 Effect of EGR on smoke emission of multiple injection

5.3.4 Effect of boosting and multiple injection strategy

As explained previously that same energy input was used in the multiple injection mode, which is represented by total injection quantity of fuel around 16 mg every cycle. Fig. 5.62 shows the effect of intake boosting on in-cylinder pressure, temperature and HRR of GCI engine using multiple injection strategy. The highest in-cylinder pressure is for GCI engine fueled with gasoline-biodiesel blend with 0.12 MPa of intake boosting. While, the lowest in-cylinder pressure was happened on GCI engine fueled with diesel fuel running on 20% EGR and ambient pressure of intake boosting. In CI engine, a multiple injection strategy resulted a significant reductions of the in-cylinder peak pressure and heat release rate, as this is the main feature of multiple injection strategy with early pilot injection compared to single injection based on the previous research study[84]. Similar to the cylinder pressure, the highest in-cylinder temperature was for gasoline-biodiesel blend. It was normal, since the in-cylinder temperature is calculated using pressure data, the higher in-cylinder pressure then will be the higher in-cylinder temperature. Since the amount of injected fuel in the combustion chamber was constant, the injected fuel for both diesel and gasoline-biodiesel blend was adjusted to the same energy content; in addition, the fuels also unintentionally had the same weight. The utilization of EGR without increasing intake boosting caused to the less fuel was burned, which led to a lower indicated mean effective pressure (IMEP). The combustion phasing was also changed to be more retarded, but it had a small effect on the IMEP. Thus, reduction of the amount of fuel burned will contribute to lower combustion temperatures and lower residual temperatures. From the heat release rate the highest HRR obtained from GCI engine fueled with diesel fuel with 0.12 MPa of intake boosting, meaning that the combustion is potentially more unstable compared to the other condition. However, the heat release rate of GCI engine fueled with gasoline-biodiesel blend both by using 0.1 MPa and by 0.12 MPa of intake boosting show a lower value, which mean more stable of combustion inside the cylinder was happened.

Fig. 5.63 shows the effect of intake boosting on ignition delay of GCI engine using multiple injection mode. As already discussed in the section of effect of EGR rate, the ignition delay of GCI engine fueled with gasoline biodiesel blend is much earlier than diesel fuel when using the ambient pressure of intake boosting due to the multiple injection mode. The 70% of diesel fuel which is injected earlier caused a huge amount of fuel will be impingement on the wall of cylinder, thus will leads to the difficulty of auto ignition, even though, the main injection was injected near before TDC. Then, by increasing intake boosting it can be seen that there is no significant effect on the ignition delay both for gasoline-biodiesel blend and for diesel fuel. Slightly more retarded of ignition delay was happened on GCI engine using MPCI mode both for gasoline-biodiesel blend and for diesel fuels when intake boosting increase to be 0.12 MPa.

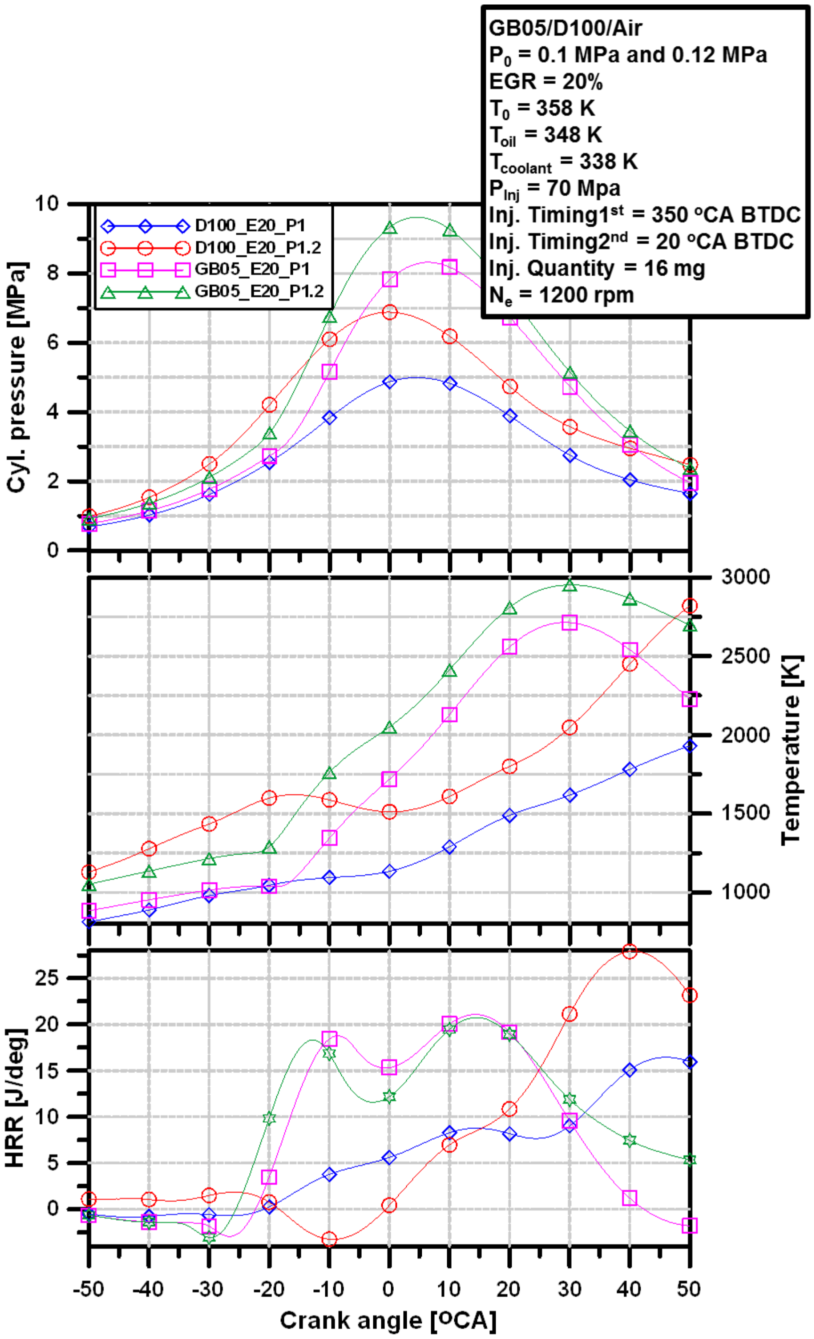


Fig. 5.62 Effect of boosting on cylinder pressure, temperature and HRR of multiple injection mode

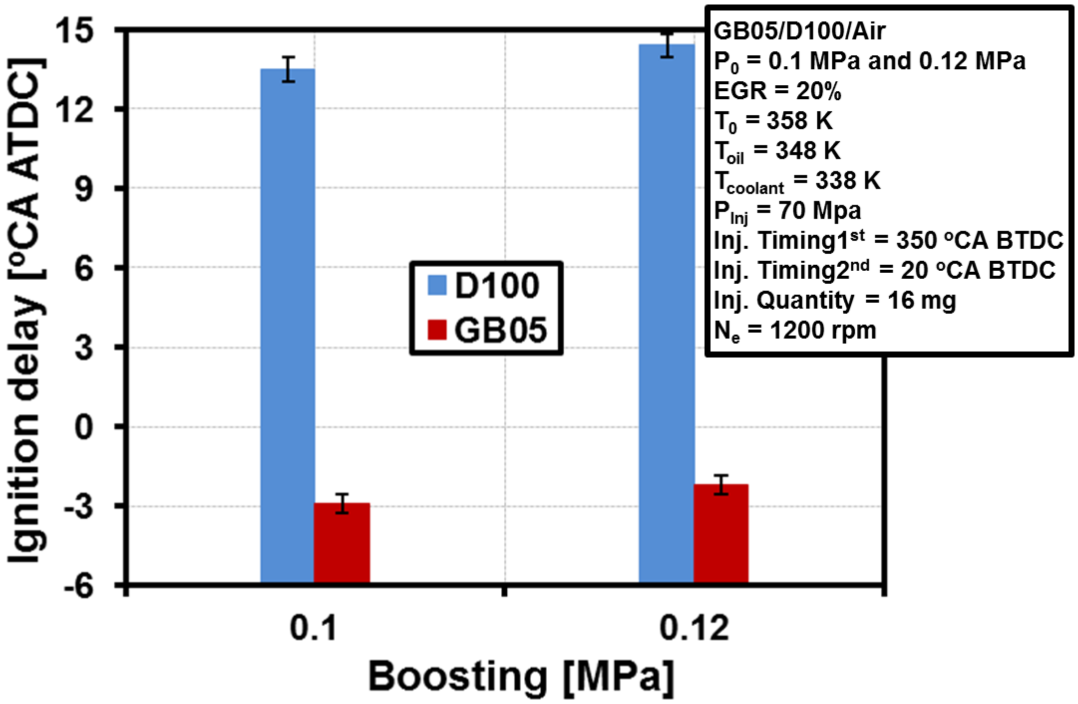


Fig. 5.63 Effect of boosting on ignition delay of multiple injection mode

Fig. 5.64 shows the effect of intake boosting on maximum in-cylinder pressure and Fig. 5.65 shows the effect of intake boosting on maximum pressure rise rate of GCI engine using multiple injection strategy. The trend of maximum in-cylinder pressure and maximum pressure rise rate of GCI engine using multiple injection strategy are same both for gasoline-biodiesel blend and for diesel fuel, which is increase when the intake boosting increased. Furthermore, both for Pmax and PRRmax of gasoline and biodiesel-blend are much more higher than diesel fuel in ambient pressure of intake boosting and 0.12 MPa. Usually, high volatility fuel like gasoline used in CI engines has higher in-cylinder gas pressure rise rate (thus engine noise) and unburned hydrocarbon (UHC) emissions than conventional diesel engines[12]. Then, using multiple injections, PPRR can be reduced, meaning that the engine noise is also reduced [97]. However, in this study the utilization of EGR and boosting, even increasing boosting lead to the increasing PRR.

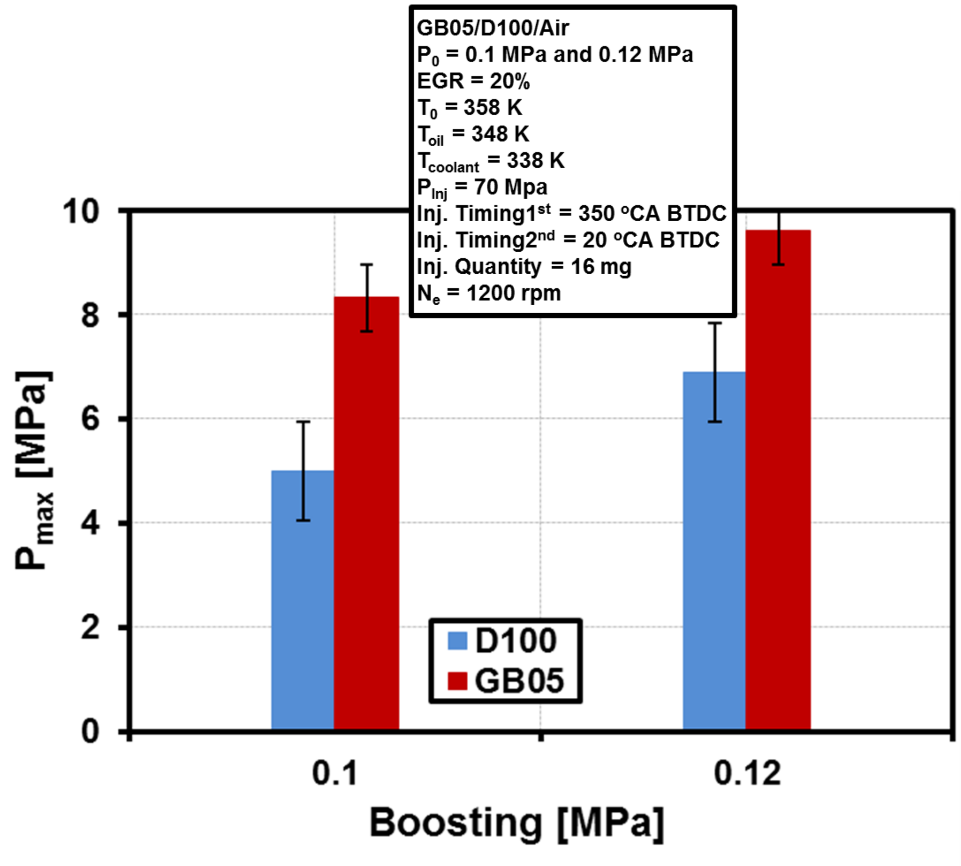


Fig. 5.64 Effect of boosting on max of in-cylinder pressure of multiple injection mode

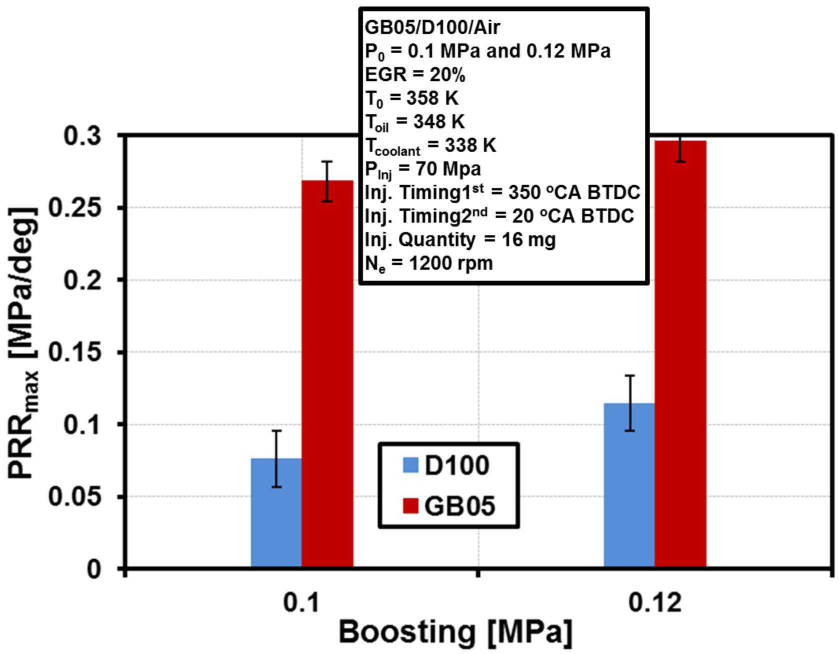


Fig. 5.65 Effect of boosting on max of pressure rise rate of multiple injection mode

The effect of intake boosting on IMEP of GCI engine using multiple injection mode can be seen in Fig. 5.66. In the previous discussion, the effect of EGR reveals that the increasing EGR rate lead to the reducing IMEP. To explain the effect of intake boosting, the 20% of EGR rate was chosen, then intake boosting 0.12 MPa was applied. It can be seen the normal effect that the increasing of intake boosting lead to the increasing of IMEP both for GCI engine fueled with gasoline-biodiesel blend and pure diesel fuel. The important phenomenon that should be highlighted is that the IMEP of GCI engine fueled with gasoline-biodiesel blend is higher than pure diesel fuel in both intake-boosting conditions 0.1 MPa and 0.12 MPa.

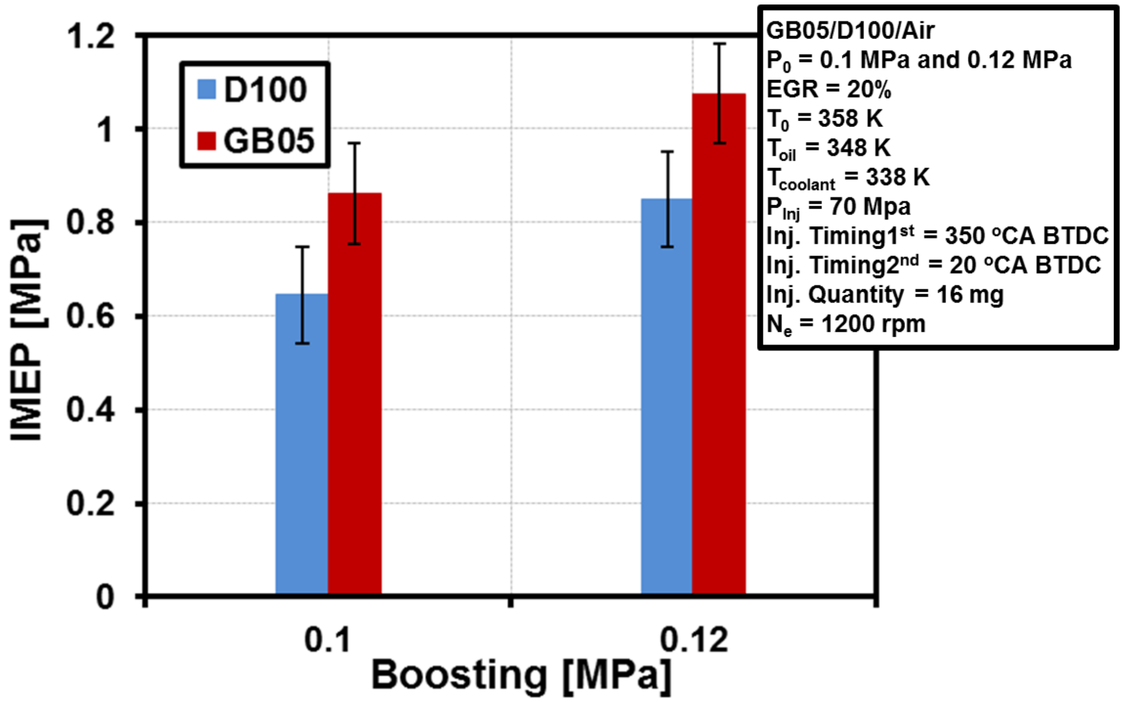


Fig. 5.66 Effect of boosting on IMEP of multiple injection mode

The effect of intake boosting on indicated thermal efficiency of GCI engine using multiple injection mode is presented in Fig. 5.67. There is also normal effect related to intake boosting, is that the increasing of intake boosting, the increasing of indicated thermal efficiency for both GCI engine fueled with gasoline-biodiesel and pure diesel fuel. However, the indicated thermal efficiency of GCI engine fueled with gasoline-biodiesel blend is higher than pure diesel fuel either with 0.1 MPa or with 0.12 MPa of intake boosting.

CO emission of GCI engine using MPCI mode, which resulted from 0.1 MPa and 0.12 MPa of intake boosting, is presented in Fig. 5.68. The increasing of intake boosting from 0.1 MPa to be 0.12 MPa caused the decreasing of CO emission for GCI both fueled with gasoline-biodiesel blend and diesel fuel. However, for all condition either 0.1 MPa or 0.12 MPa of intake boosting the CO emission of pure diesel is higher than gasoline-biodiesel blend. Meanwhile, the effect of intake boosting on HC emission of GCI engine using multiple injection mode is can be seen in Fig. 5.69. Similar with CO emission trend, the increasing of intake boosting leads to the decreasing of HC emission. However, HC emission of GCI engine with multiple injection mode fueled with gasoline-biodiesel blend is higher than pure diesel fuel both for 0.1 MPa and for 0.12 MPa of intake boosting. The effect of intake boosting on NOx emission of GCI engine using multiple injection mode can be seen in Fig. 5.70. There is a slightly reducing of NOx emission of GCI engine using multiple injection mode fueled with diesel fuel or almost no effect when increasing intake boosting from 0.1 MPa to 0.12 MPa on is applied. However, a significant effect was happened on GCI engine fueled with gasoline-biodiesel blend when using 0.12 MPa of intake boosting, the emission of NOx reduced almost a half of when 0.1 MPa was applied. The effect of intake boosting on smoke emission of GCI engine using multiple injection mode can be observed in Fig. 5.71. The increasing of intake boosting caused the significant reduction of smoke emission of GCI engine fueled with pure diesel fuel. However, the increasing of intake boosting form ambient pressure to 0.12 MPa leads to the increasing of smoke emission of GCI engine fueled with gasoline-biodiesel blend.

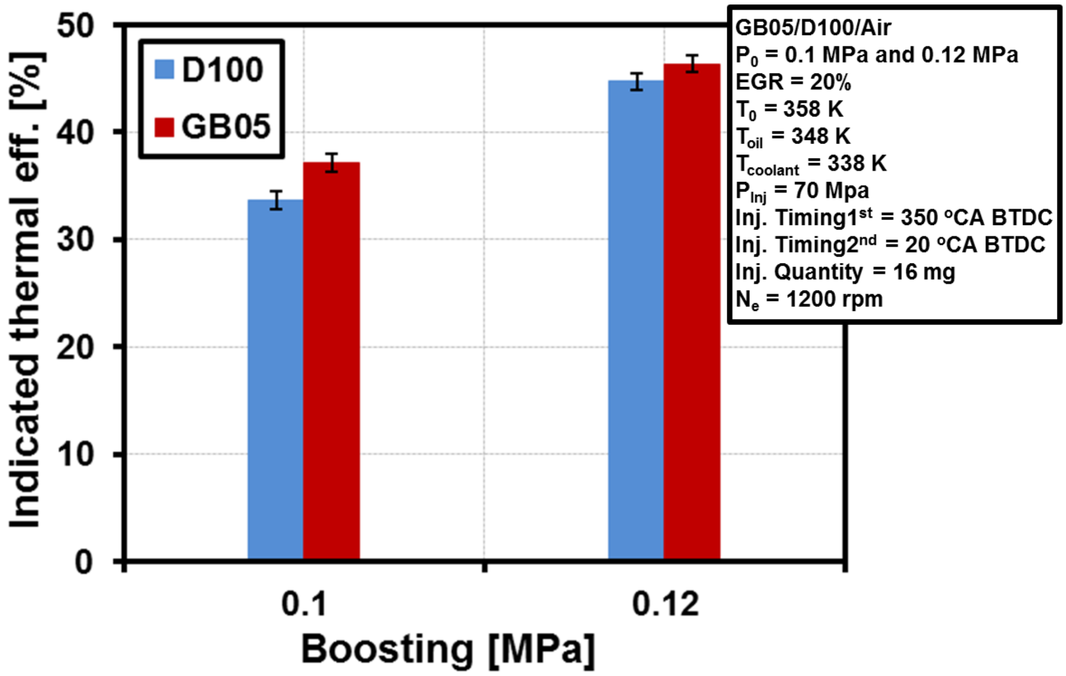


Fig. 5.67 Effect of boosting on indicated thermal efficiency of multiple injection mode

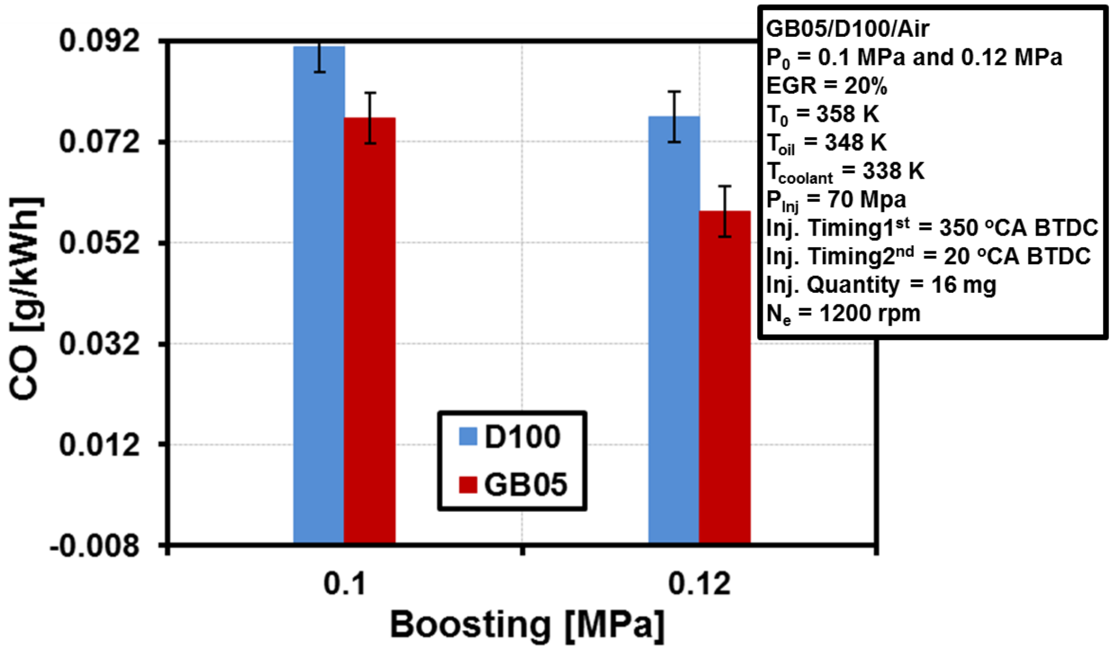


Fig. 5.68 Effect of boosting on CO emission of multiple injection mode

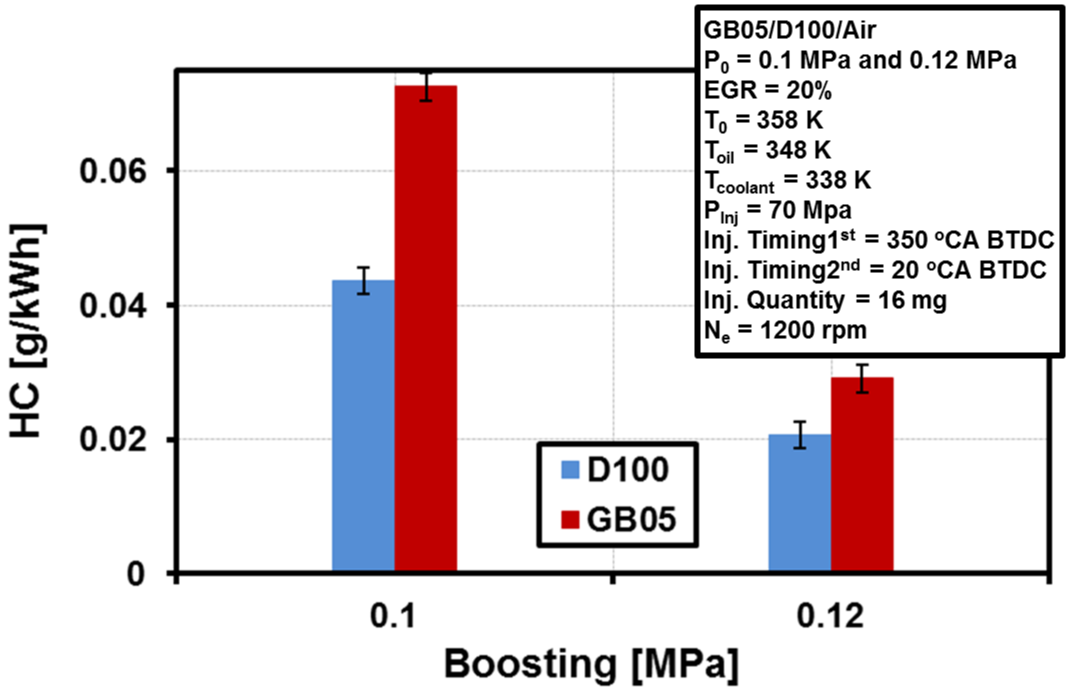


Fig. 5.69 Effect of boosting on HC emission of multiple injection mode

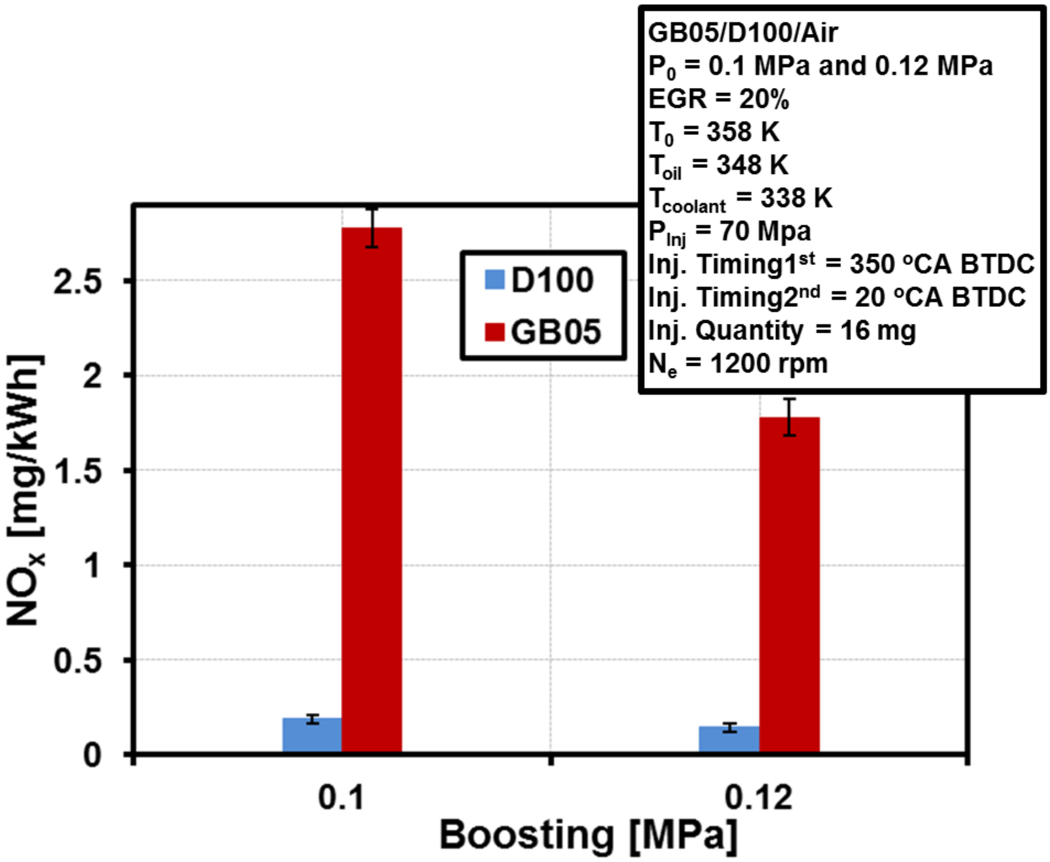


Fig. 5.70 Effect of boosting on NOx emission of multiple injection mode

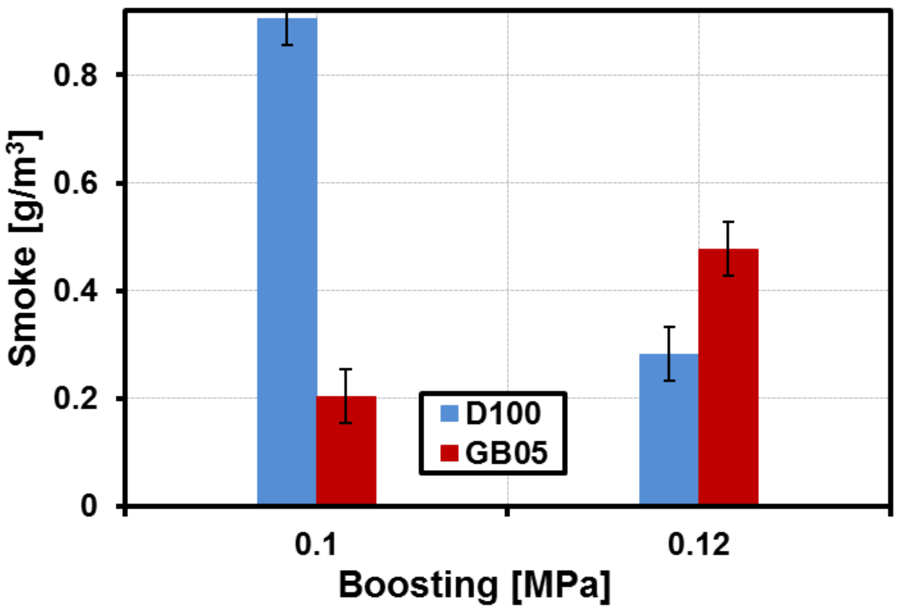


Fig. 5.71 Effect of boosting on smoke emission of multiple injection mode

5.3.5 Summary

This study of a GCI engine was conducted by adding 5% of biodiesel into gasoline and comparing the results with those from neat diesel in single-injection and multiple-injection modes combined with the application of EGR and intake boosting with the goal of obtaining high efficiency and low emissions. The engine testing used the same energy input for all experiments, an injected fuel amount of around 26 mg per cycle for single injection mode and 16 mg per cycle for multiple injection mode.

For single mode, increasing the EGR rate decreased the indicated thermal efficiencies with both diesel and the gasoline-biodiesel blend. The onward timing of injection at 40 °CA BTDC and 0% EGR resulted a high premixed fuel charge. Therefore, homogeneous combustion occurred, leading to a higher IMEP and thermal efficiency. Because of its higher volatility compared to diesel and their similar heat values, the GB05 at 0% EGR had a longer ignition delay and longer mixing period than diesel, producing more complete combustion and contributing to a rise in the engine thermal efficiency. However, the 20% and 50% EGR rates affected the air dilution inside the combustion chamber, which changed the completeness of the combustion and decreased the thermal efficiency. Using EGR reduced the NOx emissions with the GB to levels much lower than seen with diesel. Smoke emissions with diesel were high for all EGR rates. With the GB, lowest smoke was at the EGR 20%.

The thermal efficiency and NOx emissions of the GCI engine using diesel and GB increased along with the intake boosting, and then decreasing the smoke for diesel on contrary increasing smoke for GB. The intake boost increases the charge reactivity, which accelerates the combustion reaction velocity. When an intake boost is combined with a high compression ratio, the cylinder pressure becomes quite high at the maximum of the compression stroke. Thus, the IMEP increases as the amount of fuel stays the same, producing higher thermal efficiency.

In multiple injection mode, for both diesel and GB05, the increasing of EGR, indicated thermal efficiencies were decrease. The higher thermal efficiency may caused by the homogeneous mixing of air and fuel during the pilot injection. Furthermore, using multiple injection mode, which increases the temperature of air intake, oil, and coolant, could improve combustion and the efficiency of the engine. Multiple injection mode could supply enough mixing time for the biodiesel blended in gasoline, which has a longer autoignition timing than diesel. In that case, supplying approximately 30% of the fuel in the main injection could generate combustion. Hence, improved combustion will be achieved, yielding in a higher engine thermal efficiency. However, increasing the EGR rate worsened the indicated thermal efficiency, possibly because of the greater specific heat of dilution in the EGR mixture, which prevents the normal combustion process, deteriorating the burning rate and reducing the temperature inside the cylinder.

The optimum reducing smoke and NOx in GB is at EGR 50%. The increasing of intake boosting, the increase of indicated thermal efficiency for both GB and diesel Intake boosting make NOXxof GB reduce almost 0.5 from ambient pressure but increased smoke.

6. Summary and conclusions

This thesis was generally focused improvement the efficiency and emission characteristics for compression ignition engine operated in gasoline compression ignition mode fueled with gasoline-biodiesel blends. The study on GCI engine performance, combustion characteristics and emission characteristics fueled by gasoline-biodiesel blends was conducted in experimental series. The experimental series to simultaneously increasing engine efficiency while reducing exhaust emissions comprises of single injection strategy, multiple injection strategy, and the application of EGR and intake boosting. The following conclusions can be drawn from this study.

- The quality of fuel spray, atomization, and flow rate were influenced by several factors, one of which is backpressure. The variation in backpressure influences cavitation that may reduce the injection flow rate of the injector nozzle. However, there is no effect on the injection quantity of fuel when blending a small amount of biodiesel (from 5 to 20 %) into gasoline fuel, even though the back pressure was applied from 0.1 to 3 MPa. Normally the backpressure in a cylinder will affect the injection quantity when it is more than five MPa. The backpressures that formed, ranging from 0.1 to 3 MPa, are very small. Therefore, the effect of backpressure is negligible since it is relatively low compared with the great injection pressure.

- The shorter ignition delay of GB20 compared to diesel fuel and near TDC of combustion phasing besides are consistent with observations of earlier SOI of gasoline-biodiesel blends. This is due to the gasoline biodiesel stratification characteristics. Gasoline biodiesel blend fuel has fuel stratification, so when the higher biodiesel content of the gasoline mixture ignites faster than the other part of the fuel mixture, it will also influence the ignition delay, combustion phasing and combustion duration. In terms of combustion duration or mass burning fraction, the gasoline-biodiesel blends showed almost the same combustion duration as diesel fuel both for earlier and later SOI.

- In every SOI of gasoline-biodiesel blends in the GCI engine, the value of IMEP was identical to that of diesel fuel. The stability of combustion represented by the coefficient of variability (COV) of IMEP showed that all variation in SOI of the gasoline-biodiesel represented a higher confidence than diesel fuel.

- The thermal efficiencies of every SOI of gasoline-biodiesel blend can be obtained as high as that of diesel fuel. The high gasoline volatility in the GB20 combined with earlier or retarded injection timing allows the fuel air mixture to become a more homogeneous fuel-air mixture. Therefore, a complete combustion process can be achieved.
- The HC emission in a GCI engine can be reduced by using gasoline-biodiesel blends and dramatically decreases when the SOI is retarded. The NOx emission of a GCI engine using gasoline-biodiesel blends showed less satisfactory results than diesel fuel, however, there is a high possibility to reduce this by using retarded SOI.
- The improvement of combustion achieved for GB05 with multiple injections, at an in-cylinder temperature of approximately 1800 K, is better than that of 100% diesel. The heat release rate of multiple injections of GB05 is lower than that of a single injection of 100% diesel, but higher than that of multiple injections of 100% diesel and a single injection of GB05.
- The GCI engine fueled with a low cetane number fuel (GB05) in multiple-injection mode has the ability to control its combustion. The ignition delay of GB05 with multiple injections or a single injection are more advanced than 100% diesel with multiple injections. Gasoline fuel is usually more resistant to auto-ignition; therefore, the ignition delay will be increased. The combustion phasing of GB05 with multiple injections is almost the same as that of 100% diesel with multiple injections and a single injection.
- The highest PPRR occurred for the engine fueled with 100% diesel in single-injection mode. Using multiple injections, the PPRR can be reduced, meaning that the engine noise is also reduced. Based on the 5% combustion stability limit criteria from this experiment, the GCI engine fueled with GB05 for both single and multiple injections had good combustion stability.
- Using multiple injections and increasing the intake air, oil, and coolant temperatures could result in improvements in the thermal and combustion efficiencies of the engine. Meanwhile, the stability analysis using COV of IMEP and cyclic variations of Pmax shows satisfactory results. However, the cyclic variations analysis for the start of combustion (CA 10) show deteriorates effect in GB05 with multiple injection strategies.

- The pilot injection of GB05 could be the reason for decreasing CO emissions. The biodiesel content and gasoline as the highly volatile fuel in the GB05 showed the significant effect of lowering THC and CO emissions. The NOx emissions from GB05 for both multiple and single injections seem to be higher than that of 100% diesel with multiple injections and even higher than 100% diesel with a single injection. It is believed that this due to the oxygen content in the fuel.
- For single injection mode, 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 NOx emission. Even though, when 50% EGR was applied the NOx emission will also decreasing. The utilization of EGR gives effect on the drop of NOx emission value for gasoline-biodiesel blend much lower than diesel fuel. However, there are no effects of EGR rate variations on NOx 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 obtain its lowest value when EGR rates at 20%.
- For single injection mode, 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 NOx emission of GCI engine fueled with diesel is higher than GCI engine fueled with gasoline-biodiesel blend when using single injection mode for either ambient intake pressure or increasing intake pressure at 0.12 MPa. The increasing intake boosting also followed by increasing NOx 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/m3. 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.

- In multiple injection mode, for diesel fuel, the increasing of EGR rate to be 20% caused a slightly increasing indicated thermal efficiency. However, higher EGR rate to be 50% lead to the decreasing of indicated thermal efficiency. A different trend obtained from gasoline-biodiesel blend, the increasing EGR rate lead to the decreasing of indicated thermal efficiency. The optimum strategy in multiple injection mode for GCI engine fueled with gasoline-biodiesel is when the EGR rate at 50% was applied, while, for GCI engine fueled with diesel without EGR is the best strategy to obtain low NOx and as well as smoke emission.

- For multiple injection mode, the normal effect related to intake boosting was happened, that is the increasing of intake boosting, the increasing of indicated thermal efficiency for both GCI engine fueled with gasoline-biodiesel and pure diesel fuel. However, the indicated thermal efficiency of GCI engine fueled with gasoline-biodiesel blend is higher than pure diesel fuel either with 0.1 MPa or 0.12 MPa of intake boosting. A slightly reducing of NOx emission of GCI engine using multiple injection mode fueled with diesel fuel or almost no effect when increasing intake boosting from 0.1 MPa to 0.12 MPa is applied. However, a significant effect was happened on GCI engine fueled with gasoline-biodiesel blend when using 0.12 MPa of intake boosting, that is the emission of NOx reduced almost a half of when 0.1 MPa was applied. The increasing of intake boosting caused the significant reduction of smoke emission of GCI engine fueled with pure diesel fuel. However, the increasing of intake boosting form ambient pressure to 0.12 MPa leads to the increasing of smoke emission of GCI engine fueled with gasoline-biodiesel blend.

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APPENDICES

A. List of Publications

1. **Yanuandri Putrasari**, Ocktaeck Lim, "A study on combustion and emission of GCI engines fueled with gasoline-biodiesel blends", *Fuel, Elsevier,* Volume 189, pp 141-1541, 2017.
2. **Yanuandri Putrasari**, Narankhuu Jamsran, Ocktaeck Lim, "An investigation on the DME HCCI autoignition under EGR and boosted operation", *Fuel, Elsevier,* Volume 200, pp 447-457, 2017.
3. **Yanuandri Putrasari**, Ocktaeck Lim, "A study of a GCI engine fueled with gasoline-biodiesel blends under pilot and main injection strategies", *Fuel, Elsevier,* Volume 221, pp 269–82. Elsevier, doi:10.1016/j.fuel.2018.01.063., 2018.
4. **Yanuandri Putrasari**, Achmad Praptijanto, Widodo Budi Santoso, Ocktaeck Lim, "Resources, policy, and research activities of biofuel in Indonesia: A review", *Energy Reports*, *Elsevier*, Volume 2, pp 237-245, 2016.
5. **Yanuandri Putrasari**, Ock Taeck Lim, "Performance and Emission of Gasoline Compression Ignition Engine Fueled with 5 and 20% Gasoline-Biodiesel Blends under Single Injection Strategy", *Energy Procedia*, *Elsevier,* Volume 105, pp 1743-1750, 2017.
6. Narankhuu Jamsran, **Yanuandri Putrasari** and Ocktaeck Lim, "A computational study on the autoignition characteristics of an HCCI engine fueled with natural gas", *Journal of Natural Gas Science and Engineering, Elsevier,* Volume 29, pp 469-478, 2016.
7. Bambang Wahono, **Yanuandri Putrasari**, Ocktaeck Lim, "Construction of Response Surface Model for Compression Ignition Engine Using Stepwise Method", *The Korean Hydrogen & New Energy Society*, Volume 28, No. 1, pp. 98~105, 2017.
8. **Yanuandri Putrasari**, Kyeonghun Jwa and Ocktaeck Lim, "Influence of EGR and intake boost on GCI engine fueled with gasoline-biodiesel blend using early single injection mode", 10th International Conference on Applied Energy (ICAE2018), 22-25 August 2018, Hong Kong, China will be published in *Energy Procedia, Elsevier,* Volume:---, 2018.
9. Bambang Wahono, **Yanuandri Putrasari** and Ocktaeck Lim, "A Study on In-Cylinder Flow Field of Small Engine with Various Engine Speed, *Under review (R2)*

MEST-D-18-01030R1 in Journal of Mechanical Science and Technology, Springer, (Status date 2018-10-24).

10. **Yanuandri Putrasari,** Ocktaeck Lim, "A study of the effect of EGR and intake boosting on a GCI engine fueled with a gasoline-biodiesel blend," *Applied Energy-Elsevier, Under review (Status date 2018-10-29).*
11. **Yanuandri Putrasari,** Ocktaeck Lim, "A review of GCI: a promising technology potentially fueled with gasoline-biodiesel blends to meet future engine efficiency and emission targets", *Energies-MDPI, Under review (Status date 2018-11-25).*
12. **Yanuandri Putrasari,** Ocktaeck Lim, "DME as the next generation fuel for CI Engines: A review", Will be submitted to *Renewable & Sustainable Energy Reviews-Elsevier, (updated status 2018-10-24).*

B. List of Conferences

International Conferences

1. **Yanuandri Putrasari** and Ocktaeck Lim, 3rd International Conference on Sustainable Energy Engineering and Application (ICSEEA) 2015, Bandung, Indonesia, 2015.
2. **Yanuandri Putrasari** and Ocktaeck Lim, 4rd International Conference on Sustainable Energy Engineering and Application (ICSEEA) 2016, Jakarta, Indonesia, 2016.
3. **Yanuandri Putrasari,** Sung Jae Won and Octaeck Lim, International Conference of Applied Energy (ICAE) 2016, Beijing China, 2016.
4. **Yanuandri Putrasari** and Ocktaeck Lim, Faculty of Industrial Technology International Congress 2017 (FoITIC 2017), Bandung, Indonesia, 2017.
5. **Yanuandri Putrasari**, Widodo Budi Santoso, Achmad Prpatijanto and Ock Taeck Lim, International Conference on Advanced Automotive Technology (ICAT) 2018, Gwangju, Republic of Korea.
6. **Yanuandri Putrasari** and Octaeck Lim, International Conference of Applied Energy (ICAE) 2018, Hong-Kong, 2018.
7. **Yanuandri Putrasari** and Octaeck Lim, International Conference on Sustainable Energy Engineering and Application (ICSEEA), Indonesia, 2018.

Domestic Conferences

1. **Yanuandri Putrasari,** Achmad Praptijanto, Widodo Budi Santoso and Ocktaeck Lim, KSME 2015 Spring Conference at Ulsan Division, University of Ulsan, Ulsan, Korea, 2015.
2. **Yanuandri Putrasari,** Sakda Thongchai, Narankhuu Jamsran and Ocktaeck Lim, KSAE 2015 Annual Conference and Exhibition, Hwabaek International Convention Center, Gyeongju, Korea, 2015.
3. **Yanuandri Putrasari**, Sakda Thongchai, Narankhuu Jamsran and Ocktaeck Lim, KSAE 2016 Annual Conference and Exhibition, Ramada Plaza Jeju Hotel, Jeju, Korea, 2016.

4. **Yanuandri Putrasari** and Ocktaeck Lim, KSME 2017 Spring Conference at Ulsan Division, UNIST, Ulsan, Korea, 2017.
5. **Yanuandri Putrasari** and Ocktaeck Lim, KSAE 2017 Annual Conference and Exhibition, Havici Hotel and Resort Jeju, Jeju, Korea, 2017.
6. **Yanuandri Putrasari** and Ocktaeck Lim, KSME 2018 Spring Conference at Ulsan Division, University of Ulsan, Ulsan, Korea, 2018.
7. Y**anuandri Putrasari** and Ocktaeck Lim, KSAE 2018 Annual Conference and Exhibition, Busan, Korea, 2018.
8. Y**anuandri Putrasari** and Ocktaeck Lim, KSAE 2018 Conference Chapter Busan, Changwon and Ulsan, Changwon, Korea, 2018.

C. Soot emission samples