Comparison of the combustion characteristics of gasoline and gasoline–ethanol blend under gasoline compression ignition mode

Van Thanh Ngo¹ and Truong Giang Nguyen

Abstract: In this study, conventional gasoline with a RON of 95 and gasoline–ethanol blend (20% ethanol in volume [E20]) were experimentally compared in terms of combustion characteristics and performance. First, second injection timing (SOI₂) was investigated to study combustion characteristics while maintaining the dilution level at 25%. In this part, two fuels with high octane numbers were used to investigate the influence of ethanol on combustion characteristics under gasoline compression ignition (GCI) mode. The in-cylinder pressure and heat-release rate (HRR) indicated that the fueling of the engine with E20 leads to the maximum peak pressure and easier position of maximum HRR in comparison with pure gasoline in all cases of the second injection. The main HRR curve of E20 moves closer to the top dead center (TDC) compared with that of gasoline. This HRR behavior indicates that the overall HRR of E20 is more compact than that of gasoline. Second, SOI₂ was fixed at two cases of −6, −3 crank angle degree (CAD) after TDC (ATDC), charge dilution and boosted pressure were investigated to study the sensitivity of dilution on combustion characteristics with gasoline–ethanol blend fuel. The in-cylinder pressure and HRR indicated a two-stage combustion phenomenon for a low dilution.

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PUBLIC INTEREST STATEMENT
Conventional engines generally operate in two combustion modes: Spark Ignition (SI) and Compression Ignition (CI). SI engines have limited thermal efficiency due to knocking at high compression, whereas CI engines have been limited by high pollutants in exhaust emission. To overcome the limitations of conventional engines, gasoline compression ignition (GCI), which is a potential strategy to achieve high specific power, high fuel efficiency was introduced. In this study, two fuels with high octane numbers (Gasoline of RON 95 and E20) were used for comparison on combustion characteristics under GCI mode. The in-cylinder pressure and heat-release rate indicated that the fueling of the engine with E20 leads to the maximum peak pressure and easier position of maximum HRR in comparison with pure gasoline in all cases of the second injection. The main HRR curve of E20 moves closer to the top dead center compared with that of gasoline.
ratio (from 0% to 15%) and a single stage with a peak at approximately 5–6 CAD ATDC at high dilution.

Subjects: Mechanical Engineering; Power & Energy; Transport & Vehicle Engineering; Chemical Engineering

Keywords: charge dilution; ethanol; gasoline compression ignition; second injection timing; boosted pressure

1. Introduction
Conventional engines generally operate in two combustion modes: spark ignition (SI) and compression ignition (CI). SI engines have limited thermal efficiency due to knocking at a high compression (Zeng et al., 2012), whereas CI engines have been limited by high pollutants in exhaust emission (Li & Ogawa, 2012). To overcome the limitations of conventional engines, gasoline compression ignition (GCI), which is a potential strategy to achieve high specific power, high fuel efficiency, as well as low soot and nitrogen oxide (NOx) emissions, was introduced (Nguyen et al., 2019). The GCI combustion mode is an advanced low-temperature combustion (LTC) technique. LTC techniques take advantage of the high autoignition of SI engines and the high compression ratio of CI engines (Solanki et al., 2020; Wei et al., 2020), resulting in low soot and NOx emission in exhaust gas while maintaining high efficiency. GCI engines operate on fully premixed homogeneous combustion (e.g., homogeneous charge compression ignition [HCCI]), partially premixed combustion (PPC; G. Kalghatgi et al., 2010; Rezaei et al., 2013; Selimau et al., 2014), and diffusion-controlled combustion modes under low, medium, and high load conditions, respectively. The partially premixed charge of GCI enables the control of combustion phasing as it is closely coupled with the fuel injection timing, offering a practical advantage over HCCI in which the combustion phasing is decoupled from injection timing and dominated by the kinetics of chemical reactions.

Recently, studies on gasoline PPC mode have been prioritized because this mode overcomes high load limit and low fuel consumption and has lower pollution of emission (Putrasari et al., 2018; Tang et al., 2016; Woo et al., 2016) compared with other combustion regimes, such as HCCI engines fueled with gasoline and low-temperature combustion mode with diesel fuel (Natti et al., 2009; Zhao et al., 2018). In PPC engines, the injection timing is later than that of the HCCI engine and thus has a moderate heat-release rate (HRR); therefore, the high load limit of HCCI engines is overcome because of the additional time for fuel and air to mix, leading to a more stratified thermal/fuel. In addition, the PPC mode has lower NOx and particulate matter emissions than conventional engines (Rousselle & Foucher, 2013).

Exhaust gas recirculation (EGR) is an important factor for controlling GCI combustion. The high level of EGR leads to a leaner fuel-air mixture caused delaying auto-ignition. Besides, increasing the EGR rate increased pumping and heat transfer loss, thus reducing the engine's thermal efficiency. The GCI combustion process can be optimized by adjusting EGR (Jiang et al., 2019) and injection strategies (Ngo, 2020; Putrasari et al., 2018).

The combustion phasing of GCI engines can be controlled by coupling a partially premixed charge with fuel injection timing (Goyal et al., 2019; G.T. Kalghatgi et al., 2006). The combustion phasing is decoupled from injection timing and dominated by the kinetics of chemical reactions (Shamun et al., 2016). The active control of combustion phasing in GCI engines has motivated the investigation of multiple injection strategies. Duan et al. (2018) reported that the double-injection strategy further reduces the pressure rise rate (PRR) and variation of the cycle. While a wide variety of multiple injection strategies have been applied, the overarching aim is similar to earlier injections being executed in the intake stroke or early compression stroke for the preparation of a premixed charge and to later injections being added near the top dead center to control the combustion phasing and the maximum PRR.
Previous studies have shown that GCI engines can operate efficiently on low octane gasoline-like fuels (Hildingsson et al., 2009; Köten et al., 2020). These studies suggested that the optimum research octane number (RON) for GCI engines lies from 75 to 85. However, Manente et al. (2011) explored the effect of RON ranging from 70 to 100 on the optimization of GCI combustion. Cung et al. (2017) investigated the effect of three different types of gasoline fuels with different chemical compositions but similar RON (RON 98) under GCI operation and showed that the octane number has greatly affected the combustion process. The EGR strategy is used to control the combustion process and NOx level in emission. The EGR strategy is used to control the combustion process and NOx level in emission (Ra et al., 2012). Increasing fuel reactivity (low octane rating) requires a high EGR rate to control GCI combustion (Kolodziej et al., 2016; Yao et al., 2015).

The GCI engine fueled ethanol blend can improve efficiency, fuel consumption, and soot, hydro-carbon (HC), carbon monoxide (CO) emissions (Köten et al., 2020; Mao et al., 2018; Pan et al., 2021) by advancing the timing of the first injections (SOI1). However, early SOI1 causes cyclic variations, knock, and NOx emission (Woo et al., 2016).

In this study, gasoline with a RON of 95 and gasoline–ethanol E20 blend with high volatility, which has low ignitability and high octane number and is called low reactivity fuels (Zelenyuk et al., 2014) are studied to investigate combustion characteristics under the GCI combustion mode. The gasoline–ethanol blend is a potential alternative fuel for GCI engines. It has high oxygen content, high resistance to autoignition, and high evaporative cooling (Noh & No, 2017; Zhou et al., 2019). The present study further explores the potential of double-injection strategies in gasoline and gasoline–ethanol blend (E20)-fueled GCI engines by varying SOI2 and dilution levels.

2. Materials and methods

2.1. Fuel selection
As a fundamental study, the GCI model is tested for a gasoline–ethanol blend using dilution in comparison with a reference fuel, i.e., conventional gasoline. Ethanol produced from renewable feed stocks has lower autoignition quality with a high evaporative cooling effect than gasoline, thus contributing to the increased ignition delay time. The properties of the fuels tested are provided in Table 1.

2.2. Experimental setup and matrix test
The engine was modified from an original four-cylinder CI engine (PSA DW10 engine), which is a typical light-duty diesel engine. The engine was driven by an electrical motor to control the engine speed at 1500 rpm and the indicated mean effective pressure (IMEP) at 4.5 bar. The specifications of the modified engine are listed in Table 2. The schematic of the experimental system is shown in Figure 1.

| Properties                     | Gasoline | Ethanol | E20 |
|--------------------------------|----------|---------|-----|
| Oxygen content (%wt)           | 0        | 34.7    | 6.94|
| Density at 15°C (g/ml)         | 0.736    | 0.79    | 0.748|
| Lower heat value (MJ.kg⁻¹)     | 44       | 26.78   | 40.8|
| Research octane number (RON)   | 95       | 107     | 99  |
| A/F stoichiometric             | 14.7     | 9.0     | 13.96|
Table 2. Engine specifications

| Parameters                  | Descriptions                             |
|-----------------------------|------------------------------------------|
| Engine model                | Peugeot PSA DW10                         |
| Bore (mm)× Stroke (mm)      | 85 × 88                                  |
| Connecting rod length (mm) | 145                                      |
| Displacement (l)            | 0.499                                    |
| Geometric compression ratio | 16:1                                     |
| Fueling system              | Gasoline direct injection (GDI)          |

In this experiment, three parameters such as SOI$_2$, dilution ratio, and boosted were investigated, while other operating conditions were held constant. The engine speed was fixed at 1500 rpm. The intake air, supplied from an air compressor, was heated up to the desired temperature by electrical heaters at the plenum intake. Because it was difficult to operate the engine with turbo-charge, the boosted operation was also simulated by the air compressor. The common-rail pressure was also fixed at 400 bar. The timing of the first injection and the proportion of the first and second injections were fixed at −35 CAD ATDC and 30%–70%, respectively. The experimental conditions are summarized in Table 3.

The total of mass injection was constant when SOI$_2$ was varied from −6 to 0 CAD ATDC, whereas the dilution ratio was fixed at 25%. In addition, the dilution level was varied from 0% to 25%, which is the maximum achievable level of maintaining stable combustion, whereas SOI$_2$ was fixed in two cases: SOI$_2$ at −3 and −6 CAD ATDC, as shown in Table 4.

The intake nitrogen, simulated dilution, was provided by a compressor. Each gas supplied was controlled by a Brooks gas mass flow controller. For operation without dilution, the airflow was adjusted to achieve the desired intake pressure, as measured by a pressure transducer on an intake runner. For operation with dilution, the airflow was reduced from the amount required to achieve the desired intake pressure with air alone, and the valve on the dilution line was opened.

With this system, the engine parameters can be controlled by a closed-loop control to keep the desired conditions. After a steady period with a coefficient of variation of IMEP below 3%, a piezoelectric pressure transducer (Kistler 6043A), mounted on the cylinder head with an accuracy of ± 2%, measured the in-cylinder pressure and recorded it using an optical crank encoder with a resolution of 0.1 CAD. For all the experimental results, in-cylinder pressure data were recorded and averaged for 100 consecutive

Table 3. Experimental conditions

| Parameters                              | Values |
|-----------------------------------------|--------|
| Intake temperature Tin (°C)             | 165    |
| Intake Pressure pin (bar)               | 1 and 1.2 |
| Injection pressure (bar)                | 400    |
| Injection strategy                      | Double |
| Injected fuel ratio (%m)                | 30/70  |
| 1st start of injection SOI$_1$ (CAD ATDC)| −35    |
| Engine speed (rpm)                      | 1500   |
| IMEP (bar)                              | 4.5    |
| Fuels                                   | Gasoline, E20 |
cycles to calculate combustion characteristics using thermodynamic analysis. Other results were obtained by analysis of all the different measures through those sensors.

The intake nitrogen, simulated dilution, was provided by a compressor as follows:

$$EGR = \frac{N_2}{Air + N_2} \quad [\%]$$ (1)

The total of mass injection was constant when SOI$_2$ was varied from −6 to 0 CAD ATDC, whereas the dilution ratio was fixed at 25%. In addition, the dilution ratios were varied from 0% to 25%, which is the maximum achievable level of maintaining stable combustion, whereas SOI$_2$ was fixed in two cases: SOI$_2$ at −3 and −6 CAD ATDC, as shown in Table 4.

### 2.3. Combustion Analysis

A water-cooled Kistler cylinder pressure sensor was mounted on the cylinder head for in-cylinder combustion analysis. The inlet pressure pegged the mean value of 100 cycles of in-cylinder pressure signals at a certain operating point. The pegged mean cylinder pressure was used for HRR calculation. HRR was calculated with an energy balance where Woschni’s model was used to consider the heat transfer that occurs during the engine cycle, which was determined from the average pressure data as (Nguyen et al., 2019):

$$\frac{dQ}{d\alpha} = \frac{\gamma}{\gamma - 1} p \cdot dV + \frac{\gamma}{\gamma - 1} V \cdot dp$$ (2)

where $Q$ is the net heat release, $\gamma$ is the ratio of specific heats ($\gamma = C_p / C_v$), $p$ is the instantaneous in-cylinder pressure, $V$ is the instantaneous combustion chamber volume, and $\alpha$ is the crank angle.

The accumulated apparent heat release was calculated by integrating HRR, CA10, CA50, and CA90, which were defined as the crank angles of 10%, 50%, and 90% of the max accumulated apparent heat release. The combustion duration was defined as CA90-CA10.

The cycle-to-cycle variations of IMEP were employed to evaluate the combustion stability at low load, and it was defined as the standard deviation in IMEP ($\sigma_{\text{IMEP}}$) divided by the mean IMEP (IMEP), which is given in formula (Gharehghani, 2019):

$$\text{COV}_{\text{IMEP}} = \frac{\sigma_{\text{IMEP}}}{\text{IMEP}} \times 100\%$$ (3)

### 3. Results and discussions

#### 3.1. Effect of SOI$_2$ on combustion characteristics

Figure 2 compares the in-cylinder pressure and HRR for gasoline and E20 fuel at SOI$_2$ of −6, −4, −2, and 0 CAD ATDC. At a fixed SOI$_1$ and fuel proportion 70/30, SOI$_2$ is varied to investigate the effect

| Table 4. Matrix test | SOI$_2$ | Dilution |
|----------------------|--------|----------|
| **Fuel**             |        |          |
| *Gasoline*           | −6, −4, −2, 0 | 25     |
| *E20*                | −6, −4, −2, 0 | 25     |
| *E20*                | −6, −3  | 0, 5, 10, 15, 20, 25 |
of SOI$_2$ on combustion characteristics of GCI mode. SOI$_2$ strongly affects the in-cylinder pressure and HRR while fixed mass fuel flow rate with each fuel. A prominent advantage of GCI combustion in controlling the combustion phasing is demonstrated in Figure 2. For instance, all the peak in-cylinder pressure and HRR occur at later crank angles, and their magnitude increases when SOI$_2$ is advanced. This occurrence leads to increased noise with high dP/da and dHRR/da.

For $\sim$2 and 0 CAD ATDC SOI$_2$ with gasoline and E20, the combustion process is typical conventional diesel combustion, including premixed and diffusion combustion, as shown in the HRR plots.
with two peaks. The mixed-mode combustion shows the first peak in the HRR traces due to the reaction of the first injection fuel, which is followed by the second injection peak corresponding to the second injection.

As SOI2 advances to −6 CAD ATDC, the peaks of in-cylinder pressure and HRR are highest and latest, this seems only the premixed heat-release (one peak of HRR) due to faster combustion reaction rate caused by more premixed charge before the start of combustion with high fuel/thermal stratifications.

**Figure 2** exhibits that the highly premixed charge condition with advanced SOI2 leads to a high HRR peak, thereby increasing the in-cylinder pressure. All of SOI2, the peaks HRR of gasoline are later than of E20. With an early SOI2 (i.e., −4 and −6 CAD ATDC, the peak HRR of gasoline becomes higher than that of E20. By contrast, with a later SOI2 (i.e., −2 and 0 CAD ATDC), the HRR peak of gasoline becomes lower than that of E20. Fuel provides a cooling effect in-cylinder gas due to SOI2, leading to a decrease in the in-cylinder pressure near TDC, thus providing additional time for fuel and air to be homogeneously mixed before the start of combustion.

**Figure 3** shows the HRR at the low-temperature reaction (cooling flame). For SOI2, set at −6 CAD ATDC, the curve displays the delayed HRR peak likely due to the evaporative cooling effect caused by the second injection. This phenomenon is also called “low-temperature gasoline combustion” (LTGC; Dec et al., 2015), which is demonstrated by the local peaks in HRR curves at approximately −4 and −2 CAD ATDC in **Figure 3**. For the E20 case, it can be observed that adding 20% ethanol to the gasoline not only suppresses the LTGC but also advances intermediate-temperature gasoline combustion.

The influence of fuel and SOI2 on the in-cylinder temperature is given in **Figure 4**. The temperature is calculated as the adiabatic core in the end gas in-cylinder pressure. The values of maximum temperature in the cylinder are similar to pure gasoline and E20 but decrease slightly with a later SOI2 (as shown in **Figure 5**). Decreasing in-cylinder temperature leads to a reduction in NOx formation.
Figure 5 shows the indicated efficiency, the maximum of the in-cylinder temperature, and the peaks of pressure rate with different values of SOI2 while keeping IMEP at approximately 4.5 bar. The indicated efficiency increases with advanced SOI2 because the advanced combustion phasing results in high positive work. This result is consistent with a high in-cylinder pressure peak and a long period of high-pressure condition but late in-cylinder pressure peak.

The value of maximum PRR indicates the combustion noise level of the cycle. In gasoline and E20, the PRR increases as SOI2 occurs earlier. When SOI2 occurs at −6 CAD ATDC, the PRR achieves above 5.3 bar/CAD, indicating a fast combustion process. For the tested conditions of this study, SOI2 is optimized at −5 CAD ATDC with E20 and −3 CAD ATDC with gasoline. Through SOI2 retardation, fuel is injected in the burned gas of the first injection, leading to an increased diffusion combustion period, which increases to the maximum of the in-cylinder temperature.
Figure 6 presents the phasing of combustion (CA50), combustion duration (CD), and ignition delay (ID) under different injection timings. The premixed combustion time (CA10-CA50) is defined as the crank angle difference between CA50 and combustion timing. Ignition delay is defined as the crank angle between the start of the first injection and CA0 (the position HRR through negative to positive). Figure 6 shows that in gasoline and E20 fuel, the ignition delay prolongs as the SOI2 retarder. Under the same SOI2, adding ethanol to gasoline can shorten the ignition delay, especially with advanced SOI2. E20 fuel with high oxygen content has a long premixed combustion time, as defined by (SOC-CA50), and high-temperature flame leads to easy fuel oxidation, resulting in low smoke emissions.

Figure 7 presents the start of combustion and temperature at the start of the second injection and also at the start of combustion under different injection timings. With the advance of SOI2, the start of combustion timing was retarded and then advanced.

An interesting trend observed in Figure 7 is that the start of combustion determined by the departure of the positive HRR value occurs at around TDC for the early second injection timings (SOI2 < −5 CAD ATDC) and earlier TDC while more late second injection timing (SOI2 > −5 CAD ATDC). This means two distinctively different charge conditions: one with the early second injection ranging of −6 CAD ATDC with gasoline fuel in which both the first and second injection complete before the start of combustion allowing for positive ignition dwell between the end of the second injection and the start of combustion, and the other with the late second injection ranging from −5 to 0 CAD ATDC in which the fuel is injected into the combusting gas.

The former would induce a highly premixed charge condition with high fuel/thermal stratifications, whereas the latter would cause a mixed-mode of premixed combustion and mixing-controlled combustion.

The −5 CAD ATDC second injection timing sits in between these two conditions such that the first injection reaction starts during the second injection.

3.2. Effect of dilution level on the combustion characteristics of E20

First, as the dilution level decreases at fixed intake pressure, the oxygen concentration in the intake increases. Therefore, the air–fuel ratio (global equivalent ratio) increases to keep the same amount of fuel injected. Figure 8 shows the in-cylinder pressure and HRR for E20 in two cases of SOI2 during
dilution sweep. At low dilution level conditions, in-cylinder pressure and HRR indicate a distinctive two-stage combustion phenomenon. This phenomenon occurs at the condition of a dilution level lower than 15% for SOI$_2$ at −6 CAD ATDC and lower than 20% for SOI$_2$ at −3 CAD ATDC.

At low dilution levels (<15%) with advanced SOI$_2$, the first stage of HRR occurs even before the second injection event. Therefore, it is reasonable to assume again that this stage corresponds to the premixed combustion from the first injection timing of −35 CAD ATDC. With an increase in
dilution level of up to 25%, the premixed combustion of pilot injection is suppressed (later peak value) and eventually merges with the combustion of the main injection with more premixed combustion.

For dilution levels of 15% (blue line) and 20% (red dot line), when SOI2 is set at −6 CAD ATDC, the HRR is more premixed combustion, and two peaks appear. This result indicates that advanced SOI2 provides additional time for fuel to be homogeneously mixed with air.

Figure 9 shows that decreasing the dilution level leads to the IMEP and the maximum PRR increase in both cases of SOI2. The addition of nitrogen reduces the oxygen available for combustion. The combustion reaction rate becomes slower, and the combustion process becomes longer (the HRR peaks move to farther TDC), leading to a reduction in PRR. Given the decrease in the concentration of the combustible mixture, the combustion process occurs later and becomes longer, resulting in decreased combustion and indicated efficiency. The COVIMEP is approximately 1% for all EGR ratios and SOI2.

3.3. Effect of dilution and boosted pressure on the combustion characteristics of E20
The variations in in-cylinder pressure and HRR occur near TDC are shown in Figure 10. It is clearly seen that the boosted pressure has a significant impact on combustion. Under the boosted condition, the maximum value of HRR is decreased while decreasing dilution, but pressures in-cylinder are higher. With SOI2 at −9 CAD ATDC, increasing dilution, peaks of HRR are later and higher. This is because the increase in intake air pressure leads to an improved combustion process in the cylinder. Furthermore, boosted pressure decreases the in-cylinder temperature leads to the timing for fuel and air mixture is shorter. This is caused by a decrease in the peak heat release and can be extended to high load of GCI engine.

The effects of EGR and boosted pressure on combustion phasing with two cases of SOI2 can be seen in Figure 11. Under the boosted condition, for SOI2 at −9 CAD ATDC, the combustion phasing CA50 is earlier, the combustion duration is shorter, and CA50-CA10 is lower compared to the SOI2 at −6 CAD ATDC. However, for SOI2 at −6 CAD ATDC, the effect of boosted pressure on combustion

Figure 9. Effect of EGR level on IMEP, indicated efficiency, COVIMEP and PRR.
phasing CA50 is weak. In both cases of the SOI2, the ignition delay is negative with the low dilution. The combustion process is less sensitive to dilution than without the boosted. Although the SOI affects the combustion development duration is in the same order, indicated the burning of the charge in diffusion mode.

4. Conclusions
In this study, the effects of charge dilution and SOI2 on the combustion of GCI engines fueled by gasoline RON 95 and E20 blends were investigated through experiments. The conclusions obtained from this current study can be summarized as follows.
(1) The variations in SOI2 lead to the regime change in GCI combustion such that advanced SOI2 induces a highly premixed charge with a positive ignition to dwell between EO12 and SOC. Moreover, a late second injection leads to fuel injection into the combusting gases of the first injection. However, the engine performance parameters vary directly as evidenced by the increased engine efficiency and advanced combustion phasing.

(2) In the case of E20, the SOC is strongly delayed when the dilution is increased to 25%. The maximum in-cylinder pressure decreases, and the peak value of HRR slightly decreases as the total quantity of fuel is introduced through the two injection auto ignites simultaneously. Under low dilution conditions, in-cylinder pressure and HRR indicate a distinctive two-stage combustion phenomenon. This phenomenon appears under a dilution lower than 15% for SOI2 at -6 CAD ATDC and lower than 20% for SOI2 at -3 CAD ATDC.

(3) Under the boosted condition, the peaks of HRR are decreased while decreasing the dilution ratio, but the in-cylinder pressure is higher than without the boosted, which leads to extending to a high load of GCI engine. The GCI combustion process is less sensitive to the dilution ratio.

Abbreviations
ATDC: After Top Dead Center; CAD: Crank Angle Degree; CI: Compression ignition; EGR: Exhaust gas recirculation; GCI: Gasoline Compression Ignition; HCCI: Homogeneous charge compression ignition; HRR: Heat Release Rate; IMEP: Indicated Mean Effective Pressure; LTHR: Low Temperature Heat Release; PPC: Partially premixed combustion; PRR: Pressure Rise Rate; RON: Research octane number; SI: Spark ignition; SOI1: Start Of 1st Injection; SOI2 : Start Of 2nd Injection; TDC: Top Dead Center

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