Title	Optimization of gasoline compression ignition combustion with ozone addition and two-stage direct-injection at middle loads
Author(s)	Kobashi, Yoshimitsu; Dan Da, Tu Dan; Inagaki, Ryuya; Shibata, Gen; Ogawa, Hideyuki
Citation	International journal of engine research, 23(2), 232-242 https://doi.org/10.1177/1468087420984574
Issue Date	2022-02-01
Doc URL	http://hdl.handle.net/2115/83734
Rights	Kobashi Y, Dan Da TD, Inagaki R, Shibata G, Ogawa H. Optimization of gasoline compression ignition combustion with ozone addition and two-stage direct-injection at middle loads, International journal of engine research 23(2) pp. 232-242. Copyright © [2022] Institution of Mechanical Engineers (IMEchE). DOI: 10.1177/1468087420984574.
Type	article (author version)
File Information	Manuscript.pdf

[Instructions for use](#)

1 *Optimization of Gasoline Compression Ignition Combustion with Ozone*
2 *Addition and Two-stage Direct-Injection at Middle Loads*

3
4 Yoshimitsu Kobashi^{1*}, Tu Dan Dan Da¹, Ryuya Inagaki¹, G. Shibata¹, H. Ogawa¹
5

6 ¹ Division of Energy and Environmental Systems, Graduate School of Engineering, Hokkaido
7 University, Japan
8

9 **Abstract**

10 Ozone (O₃) was introduced into the intake air to control the ignition in a gasoline
11 compression ignition (GCI) engine. An early fuel injection at -68°CA ATDC was adopted
12 to mix the fuel with the reactive O-radicals decomposed from the O₃, before the reduction
13 of the O-radicals due to their recombination would take place. The second injection was
14 implemented near top dead center to optimize the profile of the heat release rate. The
15 engine experiments were performed around the indicated mean effective pressure (IMEP)
16 of 0.67 MPa with a primary reference fuel, octane number 90 (PRF90), maintaining the
17 15% intake oxygen concentration with the EGR. The quantity of the first injection, the
18 second injection timing as well as the ozone concentration were changed as experimental
19 parameters. The results showed that the GCI operation with the ozone addition makes it
20 possible to reduce the maximum pressure rise rate while attaining high thermal efficiency,

* Corresponding author at: Kita 13, Nishi 8, Kita-ku, Sapporo, Hokkaido, Japan

E-mail address: kobashi@eng.hokudai.ac.jp (Yoshimitsu Kobashi)

compared to that without the ozone. Appropriate combinations of the ozone concentration and the first injection quantity achieve low smoke and NO_x emissions. Further, the ozone-assisted GCI operation was compared with conventional diesel operation. The results showed that the indicated thermal efficiency of the ozone-assisted GCI combustion is slightly lower than that of the conventional diesel combustion, but that GCI assisted with ozone is highly advantageous to the smoke and NO_x emissions.

Keywords: Gasoline Compression Ignition Engine, Ozone, Two-stage Injection, Thermal Efficiency, Exhaust Emissions

Nomenclature:

CDC	Conventional diesel combustion
GCI	Gasoline compression ignition
HCCI	Homogeneous charge compression ignition
HTOR	High temperature oxidation reaction
	“First HTOR” stands for the HTOR of the first injection fuel
	“Second HTOR” stands for the HTOR of the second injection fuel
LTOR	Low temperature oxidation reaction
RCCI	Reactivity controlled compression ignition
ROHR	Rate of heat release
TDC	Top dead center
$(dp/d\theta)_{\max}$	Maximum pressure rise rate [MPa/°CA]
p_{inj}	Injection pressure [MPa]
$Q_{\text{inj.1st}}$	Mass ratio of first injection [%]

47 X_{O_3} Concentration of ozone (O_3) in the intake air [ppm]

48

49

50 **Greek symbols:**

51 ϕ_{exh} Exhaust loss [%]

52 ϕ_{other} Other losses [%] equivalent to cooling loss

53 η_i Indicated thermal efficiency [%]

54 η_{comb} Combustion efficiency [%]

55 $\theta_{inj,1st}$ First injection timing [°CA ATDC]

56 $\theta_{inj,2nd}$ Second injection timing [°CA ATDC]

57 $\tau_{ign,2nd}$ Ignition delay of second injection [°CA]

58 **1. Introduction**

59 **1.1 Background**

60 Compression ignition engines offer superior fuel economy. However, in diesel
61 engines, achieving reductions in exhaust emissions remains a problem, as a turbulent
62 diffusion flame surrounds the fuel spray, generating nitrogen oxide (NO_x), and as there is
63 soot formation in the fuel rich central reaction zone [1-2]. The low reactivity of gasoline
64 is advantageous to extend a no flame region between the tip of the fuel injector and the
65 uppermost stream in a turbulent diffusion flame. The length of this zone (from the injector
66 to the flame, elsewhere termed the lift-off length or as here the set-off length) plays a
67 significant role in the soot formation as air is entrained and mixed with the fuel upstream
68 of the set-off length [3-7]. Gasoline has the further benefit to form leaner mixtures due to

its high volatility. Considering these features, compression ignition engines using gasoline have attracted increasing attention as alternatives to diesel engines due to the potential for low emissions and high thermal efficiencies [8-11].

Unlike homogeneous charge compression ignition (HCCI) in which a homogeneous mixture is ignited, gasoline compression ignition (GCI) is believed to be advantageous to enable better ignition control as the injection is implemented near top dead center (TDC), and as the ignition timing is closely coupled to the injection timing. In practice, however, achieving ignition control over a wide range of engine loads and rotation speeds is challenging as the low reactivity of gasoline increases the dependence of the ignition process of GCI on the time scale of chemical reactions. One approach to control the ignition process of GCI is to employ a multiple-injection strategy, where the fuel injected at an early timing produces reactive radicals, and increases the in-cylinder temperature before TDC, decreasing the ignition delays of the fuel injected subsequently [10-13]. A more effective approach is to introduce reactive chemical species into the intake. Ozone (O_3) may be a potential candidate for this, and it has been investigated for ignition control in HCCI engines [14-18]. Chemical kinetic studies have shown that ozone decomposes at around 600 K, and that O-radicals are generated via the decomposition as follows [19-20]:

88 The O-radicals reduce the onset temperature of the low temperature heat release, and
89 promote the ignition. The generated O-radicals, however, immediately recombine, to
90 form O_2 as:

92 This diminishes the impact of the ozone addition on the ignition process of the fuel. The
93 experimental study by Ekoto and Foucher have elucidated the reaction mechanism in a
94 single cylinder engine with an optical diagnostic approach [21].

95 **1.2 Methodology and Purpose of the Present Study**

96 Employing a two-stage injection strategy with an early first injection makes it possible
97 to utilize the reactive O-radicals [22], as shown in Fig.1. (a) The ozone-enriched air is
98 introduced at the intake stroke. (b) The first fuel injection is implemented before the ozone
99 decomposition takes place. (c) The fuel reacts with the O-radicals decomposed from the
100 ozone, and produces reactive radicals. (d) Near TDC, the fuel is injected into the
101 combustion chamber when the reactive radicals are present. This would cause a reactivity
102 stratification in the chamber as proposed in the reactivity controlled compression ignition
103 (RCCI) strategy [23]. The presence of the reactive radicals is advantageous in terms of

the combustion efficiency as examined in previous work [24]. A further potential advantage with this strategy is the ability to control ignition delays and set-off length of the second injection, enabling soot formation control.

A similar two-stage injection strategy with ozone addition has been proposed by Pinazzi and Foucher aiming to improve low load operation of GCI engines [25], in the present study, however, the GCI operation is extended to higher engine loads.

By examining the effects of the ozone concentration and the mass ratio between the first and second injections, the advantage over the conventional GCI combustion is elucidated. Smoke emissions are correlated with the ignition delays of the second injection to evaluate the possibility of set-off length control. Finally, the performance of GCI with the ozone addition is compared with that of diesel combustion employing a high pressure injection strategy.

2. Experimental Setup and Procedures

An illustration of the experimental setup detailed in the previous paper [22] is shown in Fig.2. The test engine was a single cylinder diesel engine equipped with a low pressure EGR pathway and a roots type blower (OGURA CLUTCH, TX04) for supercharging. A common-rail fuel injection system was used, employing a commercially available injector

(DENSO, G4S) and an 8-hole nozzle with nominal hole diameters of 0.125 mm and an umbrella angle of 155°. The specifications of the test engine are detailed in Table 1. The bore and stroke are 98 mm and 110 mm, and the displacement volume is 830 cm³, the compression ratio is 17.6.

A piezoelectric pressure sensor (KISTLER, 6125B) was installed in the cylinder head, and the electrical charge produced by the piezoelectric sensor was converted into a voltage signal with a charge amplifier (KISTLER, 5011B). The voltage signal proportional to the cylinder pressure was recorded with a transient memory board at a resolution of 0.2°CA for 150 cycles.

Ozone (O₃) was produced with an ozonizer (EcoDesign, ED-OG-S4AD) while introducing pure oxygen (O₂) from a gas cylinder, and it was delivered into the intake pipe (diameter: 52.7 mm), located 450 mm upstream of the intake port. The volume from the gas inlet to the intake port was larger than the displacement volume, and it was assumed that the ozone-containing oxygen were homogeneously dispersed in the combustion chamber.

The ozone concentration was measured with a UV ozone monitor (EBARA, EL-600) before the ozone-containing oxygen was introduced into the intake pipe. The volume-based ozone concentration X_{O_3} in the intake air was determined with the measured O₃

concentration and the flow rates of the oxygen, and the intake air which was determined using a differential pressure type flowmeter.

The concentrations of the total hydrocarbon (THC), carbon monoxide (CO), carbon dioxide (CO₂), nitrogen oxide (NO_x), and oxygen (O₂) were measured at the intake and exhaust pipes with an exhaust gas analyzer (Best Sokki, BEX-5100D). The smoke emissions were measured with a Bosch-type smoke meter (ZEXEL, DSM-20AN).

The definitions of the ignition timing and ignition delay of the second injection $\tau_{\text{ign.2nd}}$ are shown in Fig.3. In most cases in the present study, a two-stage combustion occurred. The first combustion, termed the first HTOR here, is due to the high temperature oxidation reaction (HTOR) of the first fuel injection, and the second, termed the second HTOR, is due to the HTOR of the second fuel injection. The profiles of the rates of heat release (ROHR) due to the first and second HTOR were often overlapping. To determine the ignition timing of the second injection, the ROHR due to the first HTOR (B) was subtracted from the as-recorded ROHR (A), and the ROHR due to the second HTOR (C) was inferred. Assuming a Gaussian distribution in the ROHR profile of the first HTOR, the ROHR profile was inverted at the peak as suggested by the red and green parts of the curves. The ignition timing of the second injection was determined as the derivative of the ROHR of the second injection exceeded 15 J/°CA², and the ignition delay of the

second injection $\tau_{\text{ign.2nd}}$ was defined to be from the start of the second injection $\theta_{\text{inj.2nd}}$ to the ignition of the fuel introduced in the second injection.

3. Results and Discussion

3.1 Experimental Conditions

The experimental conditions are detailed in Table 2. The fuel used in the present study is a mixture of primary reference fuels: i-octane ($\text{i-C}_8\text{H}_{18}$) and n-heptane ($\text{n-C}_7\text{H}_{16}$), and the mixed-in fraction of i-octane in the liquid phase is 90 vol.%, which is termed PRF90, as detailed chemical kinetic models of primary reference fuels have been developed, and it is possible to investigate the effects of chemical reactions due to the ozone addition in the future work.

The engine speed was maintained at 1200 rpm. In the previous studies, ozone was applied to improve low load operation of GCI [22, 25]. To examine the benefit of the ozone addition at higher loads, the total injection quantity was set to 29 mg/cycle, and the resultant indicated mean effective pressure (IMEP) was around 0.67 MPa, at the conditions here. As the ozone chemistry mainly affects the low temperature oxidation reaction (LTOR), its interaction depends on the equivalence ratio. It may be interesting to investigate the effects of injection pressure, affecting the equivalence ratio, on the ozone

chemistry interaction, but the present study set the injection pressure p_{inj} to 80 MPa, considering the current situation in GCI engines.

The ratio of the first injection quantity $Q_{inj.1st}$, defined as the ratio of fuel in the first injection with respect to the total fuel injected, as well as the ozone concentration X_{O_3} was changed as the experimental variables, to examine the effects on the ignition timings of the second injection. In the previous study, retarding the first injection timing $\theta_{inj.1st}$ decreases the quantity of heat release by the first injection, even remarkable so later than -60°CA ATDC, showing that the O-radicals decomposed from ozone missed an opportunity to react with the first injection [22]. Considering this and the fact that too early injection may cause wall-wetting, the first injection timing $\theta_{inj.1st}$ was set at -68°CA ATDC. The second injection timing $\theta_{inj.2nd}$ was advanced from 6°CA ATDC until the local maximum of the indicated thermal efficiency was identified. The operation was limited by the thresholds of the maximum pressure rise rate $(dp/d\theta)_{max}$ less than 1.0 MPa/°CA, or when operation became unstable.

The intake oxygen concentration, determined to attain NO_x less than 100 ppm in a typical condition, was maintained at 15% with the EGR. By reference to situations in automotive engines at middle loads, the intake air pressure was set at 120 kPa (abs.), maintaining the exhaust gas at the same pressure by throttling the valve at the exhaust

pipe. Assuming the increase in the temperature due to the turbocharging, the intake air temperature was set at 60°C.

3.2 Effects of the Ozone Concentration

Figure 4 shows the profiles of the in-cylinder pressure and rate of heat release (ROHR), changed with the ozone concentration X_{O_3} , where the ratio of the first injection quantity $Q_{inj,1st}$ and the second injection timing $\theta_{inj,2nd}$ was set at 35% and at TDC. The small heat release due to the low temperature oxidation reaction (LTOR) of the first injection fuel was identified before -20°CA ATDC. The heat release appearing near TDC was due to the high temperature oxidation reaction (HTOR) of the first injection fuel (first HTOR), and subsequently the HTOR of the second injection fuel (second HTOR) occurred. With the increase in the ozone concentration X_{O_3} , the ROHR in the first HTOR increased, causing an advance in the second HTOR. The ROHR in the second HTOR increased from $X_{O_3} = 30$ ppm to $X_{O_3} = 60$ ppm due to the advanced combustion phasing. A further increase of the ozone concentration decreased the ROHR in the second HTOR due to the decreased ignition delay of the second injection, coupled with the effects of the decrease in fuel available for the second HTOR.

Figure 5 shows the indicated thermal efficiency η_i , combustion efficiency η_{comb} ,

exhaust loss ϕ_{exh} , and other losses ϕ_{other} at the fixed ratio of the first injection quantity $Q_{\text{inj.1st}}$ of 35%. Here, the other losses ϕ_{other} are equivalent to the cooling loss. The combustion efficiency was improved with the increase in the ozone concentration X_{O_3} . As the source of the unburned emissions may be the lean mixture formed by the first injection, this result suggests that the ozone addition improves the reactivity of the lean mixture. The retarded second injection timing $\theta_{\text{inj.2nd}}$ deteriorated the combustion efficiency, and increased the exhaust loss due to the retarded combustion phasing. With the increase in the ozone concentration X_{O_3} , the combustion phasing advanced as seen in Fig.4, followed by a decrease in the exhaust loss, and an increase in the other losses. As the balance between these effects, the maximum indicated thermal efficiency was achieved with $X_{\text{O}_3} = 60 \text{ ppm}$ and 120 ppm .

3.3 Effects of the Ratio of the First Injection Quantity

Figure 6 shows the profiles of the rate of heat release (ROHR), changed with the second injection timing $\theta_{\text{inj.2nd}}$, for different ratios of the first injection quantity $Q_{\text{inj.1st}}$. Here, the ozone concentration X_{O_3} attaining high thermal efficiency was selected with respect to each $Q_{\text{inj.1st}}$. The experiments were implemented under the following conditions; 120, 180, 240, 300, and 360 ppm for the $Q_{\text{inj.1st}}$ of 20%; 30, 60, 120, and 180

ppm for the $Q_{inj.1st}$ of 35%; 10, 30, 60, and 120 ppm for the $Q_{inj.1st}$ of 50%; 0, 10, and 30 ppm for the $Q_{inj.1st}$ of 60%, while changing the second injection timing. These conditions were chosen to find local maximum values of the indicated thermal efficiency with respect to the change in the ozone concentration. In the case of the $Q_{inj.1st}$ of 60%, there was no significant differences in the maximum indicated thermal efficiency between the ozone concentrations X_{O_3} , so that $X_{O_3} = 0$ ppm was selected here for a comparison purpose. The X_{O_3} decreased with the increase of the $Q_{inj.1st}$ from 300 ppm to 0 ppm, and stable GCI operation without ozone ($X_{O_3} = 0$ ppm) were achieved with the $Q_{inj.1st}$ of 60%. The second injection timing $\theta_{inj.2nd}$ achieving the maximum indicated thermal efficiency is that underlined in the figure.

The late combustion phasing was identified with the ratio of the first injection quantity $Q_{inj.1st}$ of 20%. This is because the ROHR in the first HTOR, which occurred near TDC, was very small. The ROHR in the first HTOR increased with the increase in the first injection $Q_{inj.1st}$ from 20% to 50%, advancing the second HTOR. With the $Q_{inj.1st}$ at 60%, in the absence of ozone, the first HTOR occurred after TDC, and the first and second injection fuels burned simultaneously with the second injections before the $\theta_{inj.2nd}$ of TDC, causing the sudden elevation in the ROHR.

Figure 7 shows the indicated thermal efficiency η_i , combustion efficiency η_{comb} ,

exhaust loss ϕ_{exh} , and other losses ϕ_{other} for different ratios of the first injection quantity $Q_{\text{inj.1st}}$. The data shown in this figure were acquired under experimental conditions identical to those in Fig.6, except for the data acquired at the $Q_{\text{inj.1st}}$ of 60% and $\theta_{\text{inj.2nd}}$ of -4°CA ATDC , which is added to show the trend including the knocking condition.

The advance of the second injection improved the indicated thermal efficiency as the combustion phasing was advanced, and the exhaust loss decreased. The $Q_{\text{inj.1st}}$ of 60% achieved the lowest exhaust loss as the combustion phasing was early (see Fig.6), and combustion periods were shorter than the others. The combustion efficiency with the $Q_{\text{inj.1st}}$ of 60% was lower than that with the $Q_{\text{inj.1st}}$ of 35% and 50% due to the absence of ozone. With the $Q_{\text{inj.1st}}$ of 60%, a further increase of the ozone concentration can increase the combustion efficiency and thermal efficiency at the second injection timings later than TDC, but the combustion phasing would be too far advanced against the early second injection timings, causing deterioration of the thermal efficiency. A further increase of the ozone concentration along with decreases of the intake oxygen concentration and intake air pressure may be a possible way to increase the combustion efficiency and to optimize the combustion phasing for the early second injections. With the $Q_{\text{inj.1st}}$ of 20%, the late combustion phasing deteriorated the combustion efficiency and exhaust loss, and caused the unstable operation, while advancing the second injection improved the combustion

efficiency due to the advanced combustion phasing, coupled with the combustion enhancement effects induced by the higher concentration of ozone.

The $Q_{inj.1st}$ of 60% yielded the higher other losses value. The larger quantity in the first injection may increase the cooling loss as the combustion temperature of the mixture which is widely spread throughout the combustion chamber increases. In addition, the rapid combustion of the $Q_{inj.1st}$ of 60% may increase the cooling loss due to increased wall heat transfer. The other losses with the $Q_{inj.1st}$ of 50% were slightly higher than that with the $Q_{inj.1st}$ of 35%, but the differences were rather small, and increased from $Q_{inj.1st}$ of 35% to 20%. This may be because the momentum of the spray flame increased with the increase of the second injection quantity, and more of the spray flame impinged on the piston wall, enhancing the heat transfer to the wall.

Overall, this suggests that with the $Q_{inj.1st}$ of 35% and 50%, the higher combustion efficiency and lower cooling loss are advantageous to the indicated thermal efficiency η_i . With the $Q_{inj.1st}$ of 60%, the lower exhaust loss is advantageous to the indicated thermal efficiency η_i . The combustion efficiency with the $Q_{inj.1st}$ of 60% is improved with the advanced second injection, but the knocking limits the improvements.

Figure 8 plots the maximum pressure rise rate $(dp/d\theta)_{max}$ for different ratios of the first injection quantity $Q_{inj.1st}$. The experiments were performed under the conditions in

Fig.7. At the second injection timings $\theta_{inj,2nd}$ before 2°CA ATDC, the maximum pressure rise rate with the $Q_{inj,1st}$ at 60% was the highest, followed by that with the $Q_{inj,1st}$ at 35%, 50%, and 20%, decreasing in this order. With the $Q_{inj,1st}$ at 60%, the heat release due to the first and second HTOR overlap as the second injection timing advances, resulting in the increase of the ROHR seen in Fig.6. The maximum pressure rise rate with the $Q_{inj,1st}$ of 20% was lower than the others due to the late combustion, but here a further advance in the second injection was limited by the appearance of knocking. As the amount of the heat release in the first HTOR was small (see Fig.6), the ignition delay of the second injection was long, and a somewhat large amount of fuel remained until the second HTOR. This causes a large heat release with the advance of the combustion phase associated with the advanced second injection. Overall, this suggests that an appropriate amount of heat release due to the first HTOR is necessary to control the maximum pressure rise rate as with the $Q_{inj,1st}$ of 35% and 50%.

3.4 Comparison of the Impact of Ozone Concentration and Ratio of Fuel Injection Quantity under the Optimized Conditions

Figure 9 shows the relation among the combustion efficiency η_{comb} , ozone concentration X_{O_3} , and the ratio of the first injection quantity $Q_{inj,1st}$. For all the data shown

here, the second injection timings achieving the maximum indicated thermal efficiency were selected. It can be seen clearly that the combustion efficiency increases with the increase of the ozone concentration for each level of the ratio of the first injection quantity.

Figure 10 shows the relation among the indicated thermal efficiency η_i , ozone concentration X_{O_3} , and the ratio of the first injection quantity $Q_{inj.1st}$. As in Fig.9, the second injection timings achieving the maximum indicated thermal efficiency were selected. The ozone concentration achieving the maximum indicated thermal efficiency decreased with the increase in the $Q_{inj.1st}$. The indicated thermal efficiency increases with the increase of the ozone concentration due to the increased combustion efficiency, but it decreases when the ozone concentration exceeds a certain level. This trend can be observed for each level of the ratio of the first injection quantity. This is due to a combustion phasing advancement and increased cooling loss. There were very limited differences in the maximum indicated thermal efficiency from the $Q_{inj.1st}$ of 35% to 60%. Recall that, however, in the case of $Q_{inj.1st}$ of 60%, the operation achieving high indicated thermal efficiency was accompanied by the high $(dp/d\theta)_{max}$.

Figure 11 shows the relation among the exhaust emissions, ozone concentrations X_{O_3} , and the ratio of the first injection quantity $Q_{inj.1st}$. The smoke and NO_x emissions increased with increases in the X_{O_3} and the $Q_{inj.1st}$. Increases in the X_{O_3} and $Q_{inj.1st}$ increased the heat

release in the first HTOR, and advanced the timing of the first HTOR, increasing the in-cylinder gas temperature and the NO_x formation rate. Figure 12 shows the smoke concentration vs. the ignition delay of the second injection $\tau_{\text{ign.2nd}}$, where all the data acquired in the present experiments are plotted. The data acquired under the same conditions as in Fig.11 are shown as filled (solid) symbols. Where the ignition delay was difficult to determine, when there was simultaneous ignition of the first and second injections, the experiment data is not plotted in the figure. The smoke emissions occurred with the $\tau_{\text{ign.2nd}}$ shorter than 6°CA ; as the larger quantity of the first injection resulted in the shorter $\tau_{\text{ign.2nd}}$, the smoke emissions increased with the increase of the X_{O_3} and $Q_{\text{inj.1st}}$.

3.5 Comparison with Conventional Diesel Combustion

The engine performance and the exhaust emissions of GCI with the ozone addition are next compared with those of conventional diesel combustion (CDC). The experimental conditions of the conventional diesel combustion are detailed in Table 3. The total injection quantity is slightly increased compared to the PRF90, to correct the differences in the lower heating values. The other conditions were identical to those in Table 2. As a preliminary experiment, the injection timings and the pilot injection amount which achieved the maximum indicated thermal efficiency were determined while

maintaining the crank angle interval between the pilot and main injections at 8°CA at the injection pressure p_{inj} of 80 MPa, at this condition the injection pressure was then changed from 80 MPa to 200 MPa, without other changes in the experimental conditions.

Figure 13 plots the profiles of the ROHR between GCI with the ozone addition and CDC with the different injection pressures. The operation with the $Q_{inj,1st}$ of 35%, the X_{O_3} of 60 ppm, and the $\theta_{inj,2nd}$ of TDC was selected as the best GCI condition in this study. Compared at the p_{inj} of 80 MPa, GCI with the ozone addition presented a ROHR that was higher than CDC, representing the premixed combustion nature. The increase of the p_{inj} in CDC increased the ROHR of the pilot and main injections, reducing the combustion periods.

Figure 14 shows the indicated thermal efficiency η_i of CDC, compared with the best GCI operations derived from Fig.10. Compared at the same injection pressure of 80 MPa, the differences of the indicated thermal efficiency between CDC and GCI with the $Q_{inj,1st}$ of 35% and 60% are smaller than 1%, but increased with the increase of the injection pressure of CDC.

Figure 15 shows the trade-off relationship between the smoke and NO_x emissions. Here, the data shown in Fig.11 are plotted for comparison purposes. CDC with the injection pressure of 80 MPa generated the very high smoke concentration as well as the

slightly higher NO_x . Increasing the injection pressure up to 200 MPa in CDC, the smoke decreased dramatically, accompanied by the increase of NO_x . This result demonstrates that GCI assisted by ozone offers operation with the very low smoke and NO_x emissions.

4. Conclusions

The present study added ozone (O_3) to a gasoline compression ignition (GCI) engine, aiming to control the ignition timing and combustion progress. A two-stage fuel injection with very early injection and late injection close to top dead center was adapted to utilize O-radicals decomposed from the O_3 before the recombination reaction reduces them to O_2 . The first injection quantity, the second injection timing as well as the O_3 concentration were varied to explore the optimum operation conditions at medium loads with a single cylinder GCI engine. Further, the engine performance and exhaust emissions with GCI were compared with conventional diesel combustion operation. The conclusions may be summarized as follows:

1. The GCI operation with a smaller amount of first injection results in combustion with lower combustion efficiency, so that higher ozone concentrations are required to increase combustion efficiency and to attain higher indicated thermal efficiency.
2. For each level of the ratio of the first injection quantity, with the increase of the ozone

concentration, the combustion efficiency increases, and the combustion phasing advances, accompanied by the increase of the cooling loss. Due to the balance between the combustion efficiency and the cooling loss, he indicated thermal efficiency has a local maximum value with respect to the ozone concentration.

3. The GCI operation without ozone is accompanied by a higher maximum pressure rise rate due to the simultaneous ignition of the first and second injection fuels. With an appropriate amount of first injection and ozone addition, the high temperature oxidation reactions of the first and second injection fuels occur in two stages, decreasing the maximum pressure rise rate.

4. The increase of the ozone concentration and the first injection quantity increase the rate of heat release caused by the high temperature oxidation of the first injection fuel. This increases the in-cylinder temperature and NO_x emissions, and decreases the ignition delay of the second injection, accompanied by the increase of the smoke emissions.

5. The indicated thermal efficiency of GCI assisted with ozone is slightly lower than that of the conventional diesel combustion, but GCI assisted with ozone is advantageous to the smoke and NO_x emissions.

The present study produced O₃ from pure oxygen on a trial basis. According to the previous study by Inoue et al. who developed an intake in-line ozonizer, the ozone enrichment in the order of 60 ppm is technically viable with air [26]. Estimated by reference to this study [26], the energy loss due to the ozone production is about 1% in the engine operation condition employed in this study. Further improvements in the efficiency are still required.

Acknowledgements

This work was supported by JSPS Grants-in-Aid for Scientific Research (c) Number 19K04208.

5. References

- [1] Dec, J. E., "A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging," SAE Technical Paper, No.970873, 1997.
- [2] Kosaka, H., Aizawa, T. and Kamimoto, T., "Two-dimensional Imaging of Ignition and Soot Formation Processes in a Diesel Flame," International Journal of Engine Research, Vol.6, Issue 1, pp.21-42, 2005.
- [3] Siebers, D. and Higgins, B., "Flame Lift-Off on Direct-Injection Diesel Sprays under Quiescent Conditions," SAE Technical Paper 2001-01-0530, 2001.
- [4] Higgins, B. and Siebers, D., "Measurement of the Flame Lift-Off Location on DI Diesel Spray using OH Chemiluminescence," SAE Technical Paper 2001-01-0918, 2001.

- [5] Siebers, D., Higgins, B. and Pickett, L., "Flame Lift-Off on Direct-Injection Diesel Fuel Jets: Oxygen Concentration Effects," SAE Technical Paper 2002-01-0890, 2002.
- [6] Pickett, L., Siebers, D. and Idicheria, C., "Relationship between Ignition Processes and the Lift-Off Length of Diesel Fuel Jets," SAE Technical Paper 2005-01-3843, 2005.
- [7] Yasutomi, K., Mueller, C. J., Pickett, L. M. and Skeen, S. A., "Investigation of the Spray and Combustion Characteristics of Four Multi-Component Diesel Surrogates Fuels Relative to Their Commercial Target Fuel," THIESEL 2018 Conference on Thermo- and Fluid Dynamic Processes in Direct Injection Engines, 2018.
- [8] Kalghatgi, G., Risberg, P., and Ångström, H., "Advantages of Fuels with High Resistance to Auto-ignition in Late-injection, Low-temperature, Compression Ignition Combustion," SAE Technical Paper 2006-01-3385, 2006.
- [9] Kalghatgi, G., Risberg, P., and Ångström, H., "Partially Pre-Mixed Auto-Ignition of Gasoline to Attain Low Smoke and Low NOx at High Load in a Compression Ignition Engine and Comparison with a Diesel Fuel," SAE Technical Paper 2007-01-0006, 2007.
- [10] Hanson, R., Splitter, D., and Reitz, R., "Operating a Heavy-Duty Direct Injection Compression-Ignition Engine with Gasoline for Low Emissions," SAE Technical Paper 2009-01-1442, 2009.
- [11] Manente, V., Johansson, B., and Cannella, W., "Gasoline Partially Premixed Combustion, the Future of Internal Combustion Engines?," International Journal of Engine Research, Vol.12, Issue 3, pp.194-208, 2011.
- [12] Ra, Y., Loeper, P., Andrie, M., Krieger, R., Foster, D., Reitz, R., and Durrett, R., "Gasoline DICI Engine Operation in the LTC Regime Using Triple-pulse Injection," SAE Int. J. Engines 5(3), 2012.
- [13] Sellnau, M. C., Sinnamon, J., Hoyer, K., and Husted, H., "Full-time Gasoline Direct-injection Compression Ignition (GDCI) for High Efficiency and Low NOx and PM," SAE Int. J. Engines 5(2), 2012.
- [14] Nishida, H. and Tachibana, T., "Homogeneous Charge Compression Ignition of Natural Gas/Air Mixture with Ozone Addition," Journal of Propulsion and Power, Vol.22, No.1, pp.151-157, 2006.
- [15] Mohammadi, A., Kawanabe, H., Ishiyama, T., Shoji M. and Komada, A., "Study on Combustion Control in Natural-Gas PCCI Engines with Ozone Addition into Intake Gas," SAE Technical Paper 2006-01-0419, 2006.
- [16] Schönborn, A., Hellier, P., Aliev, A. E. and Ladommatos, N., "Ignition Control of Homogeneous-Charge Compression Ignition (HCCI) Combustion Through

- Adaptation of the Fuel Molecular Structure by Reactions with Ozone,” *Fuel*, Vol.89, Issue 11, pp.3178-3184, 2010.
- [17] Foucher, F., Higelin, P., Mounaïm-Rousselle, C. and Dagaut, P., “Influence of Ozone on the Combustion of n-heptane in a HCCI Engine,” *Proc. the Combustion Institute*, Vol.34, Issue 2, pp.3005-3012, 2013.
- [18] Masurier, J.-B., Foucher, F., Dayma, G. and Dagaut, P., “Investigation of iso-Octane Combustion in a Homogeneous Charge Compression Ignition Engine Seeded by Ozone, Nitric Oxide and Nitrogen Dioxide,” *Proc. the Combustion Institute*, Vol.35, Issue 3, pp.3125-3132, 2015.
- [19] Heimerl, J. M. and Coffee, T. P., “The Detailed Modeling of Premixed, Laminar Steady-State Flames. I. Ozone,” *Combustion and Flame*, Vol.39, Issue 3, pp.301-315, 1980.
- [20] Yamada, H., Yoshii, M. and Tezaki, A., “Chemical Mechanistic Analysis of Additive Effects in Homogeneous Charge Compression Ignition of Dimethyl Ether,” *Proc. the Combustion Institute*, Vol.30, Issue 2, pp.2773-2780, 2005.
- [21] Ekoto, I. and Foucher, F., “Mechanisms of Enhanced Reactivity with Ozone Addition for Advanced Compression Ignition,” *SAE Technical Paper 2018-01-1249*, 2018.
- [22] Kobashi, Y., Wang, Y., Shibata, G., Ogawa, H. and Naganuma, K., “Ignition Control in a Gasoline Compression Ignition Engine with Ozone Addition Combined with a Two-stage Direct-injection Strategy,” *Fuel*, Vol.249, pp.154-160, 2019.
- [23] Kokjohn, S. L., Hanson, R. M., Splitter, D. A. and Reitz, R. D., “Fuel Reactivity Controlled Compression Ignition (RCCI): A Pathway to Controlled High-Efficiency Clean Combustion,” *International Journal of Engine Research*, Vol.12, Issue 3, pp.209-226, 2011.
- [24] Kobashi, Y., Tanaka, D., Maruko, T., Kato, S., Kishiura, M., and Senda, J., “Effects of Mixedness and Ignition Timings on PCCI Combustion with a Dual Fuel Operation,” *SAE Technical Paper 2011-01-1768*, 2011.
- [25] Pinazzi, P. M. and Foucher, F., “Potential of Ozone to Enable Low Load Operations of a Gasoline Compression Ignition (GCI) Engine,” *SAE Technical Paper 2017-01-0746*, 2017.
- [26] Inoue, T., Tamida, T., Hashimoto, T., Wada, N., Nakagawa, A., Sakashita, T., Wada, K. and Honda, T., “The development of intake in-line type ozonizer for application to HCCI combustion”, *Transactions of Society of Automotive Engineers of Japan*, Vol.48, No.4 (2017), pp.827-832 (in Japanese).

Table 1 Specifications of the test engine

Bore x Stroke [mm]		98 x 110
Displacement [cm ³]		830
Compression ratio [-]		17.6
Injection nozzle	Spray included angle [°]	155
	Number of nozzle hole	8
	Nozzle hole diameter [mm]	0.125

Table 2 Experimental conditions for gasoline compression ignition (GCI) with ozone addition

Fuel	PRF90
Engine rotation speed [rpm]	1200
Intake oxygen concentration [%]	15
Intake air temperature [°C]	60
Intake air pressure [kPa (abs.)]	120
Intake O ₃ concentration X_{O_3} [ppm]	0 to 360
Fuel injection pressure p_{inj} [MPa]	80
First injection timing $\theta_{inj.1st}$ [°CA ATDC]	-68
Second injection timing $\theta_{inj.2nd}$ [°CA ATDC]	-6 to 6
Ratio of first injection quantity $Q_{inj.1st}$ [%]	20 to 60
Total injection quantity [mg]	29
IMEP [MPa]	≈ 0.67

Table 3 Experimental conditions for conventional diesel combustion (CDC)

Fuel	JIS No.2 Diesel fuel
Fuel injection pressure p_{inj} [MPa]	80, 120, 160, 200
Pilot injection timing $\theta_{inj,1st}$ [°CA ATDC]	-10
Main injection timing $\theta_{inj,2nd}$ [°CA ATDC]	-2
Pilot injection quantity [mg]	1
Main injection quantity [mg]	29

Figure 1 Schematic of the combustion control methodology with ozone addition and two-stage injection, proposed in this study

Figure 2 Illustration of the experimental setup

Figure 3 Definition of the ignition delay of the second injection

Figure 4 Profiles of the in-cylinder pressure and rate of heat release (ROHR), changing with the ozone concentration X_{O_3} ($Q_{inj,1st} = 35\%$, $\theta_{inj,2nd} = TDC$)

Figure 5 Effects of the ozone concentration X_{O_3} on the indicated thermal efficiency η_i , combustion efficiency η_{comb} , exhaust loss ϕ_{exh} , and other loss ϕ_{other} ($Q_{inj.1st} = 35\%$)

Figure 6 Profiles of the rate of heat release (ROHR), changing with the second injection timing $\theta_{inj.2nd}$
Ozone concentrations achieving high indicated thermal efficiency were selected

Figure 7 Comparison of the indicated thermal efficiency η_i , combustion efficiency η_{comb} , exhaust loss ϕ_{exh} , and other losses ϕ_{other} between the ratio of the first injection quantity $Q_{inj.1st}$

Figure 9 Effects of the ozone concentration X_{O_3} and the ratio of the first injection quantity $Q_{inj,1st}$ on the combustion efficiency η_{comb}
The second injection timings achieving the maximum indicated thermal efficiency were selected

Figure 10 Effects of the ozone concentration X_{O_3} and the ratio of the first injection quantity $Q_{inj,1st}$ on the indicated thermal efficiency η_i
The second injection timings achieving the maximum indicated thermal efficiency were selected

Figure 11 Effects of the ozone concentration X_{O_3} and the ratio of the first injection quantity $Q_{inj.1st}$ on the smoke and NO_x emissions
The second injection timings achieving the maximum indicated thermal efficiency were selected

Figure 12 Smoke concentration vs. the ignition delay of the second injection $\tau_{ign.2nd}$
All the data points, except for the conditions where the ignition delay was difficult to determine (due to overlapping), were plotted

Figure 13 Comparison of profiles of the rate of heat release (ROHR) between the gasoline compression ignition (GCI) with ozone addition and conventional diesel combustion (CDC) with different injection pressures p_{inj}

Figure 14 Comparison of the indicated thermal efficiency η_i between the gasoline compression ignition (GCI) with ozone addition and conventional diesel combustion (CDC) with different injection pressures p_{inj} . The GCI data were derived from Fig.10, selecting the maximum indicated thermal efficiency with respect to each of the quantities of the first injection

Figure 15 Comparison of the smoke and NO_x emissions between the gasoline compression ignition (GCI) with ozone addition and conventional diesel combustion (CDC) with different injection pressures p_{inj} . The GCI data were derived from Fig.11.