



Effects of Mechanical Injector Replacement on Spray and Combustion Characteristics in an Agriculture Diesel Engine

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Abstract

The objective of this study was to analyze through spray and engine experiments the effects of replacing mechanical injectors. The spray characteristics of the mechanical injectors of a used diesel engine and those of new mechanical injectors of the same model were compared. The injection quantity of the new injector was approximately 2–4% higher than that of the used injector owing to the deposit on the nozzle surface. The opening pressure of the used injector was 225 bar on account of the deterioration of the spring inside the injector. This pressure was lower than that of the new injector. The average spray penetration length of both injectors increased similarly up to 2 ms after injection. Owing to deposits on the nozzle surface, the spray angle of the used injectors tended to be smaller than those of the new injectors. The results of engine experiments showed that the new injectors recovered over 90% of engine torque and engine power on account of the increased fuel injection quantity. However, employing the new injectors caused heat loss and fuel consumption rate deterioration. The new injectors effectively reduced NO_x and PM emissions by retarding the fuel injection timing and increasing the fuel injection pressure.

Keywords Injector replacement · Combustion variation · Operating parameter · Mechanical injector · Agriculture diesel engine

Abbreviations

CO	Carbon monoxide
CO ₂	Carbon dioxide
DPF	Diesel particulate filter
ECU	Engine control unit
EGR	Exhaust gas recirculation
EPA	Environmental protection agency
EU	European Union
HC	Hydrocarbon
NO _x	Nitrogen oxides
O ₂	Oxygen
P_{amb}	Ambient pressure
PM	Particulate matter
RPM	Revolutions per minute

SCR	Selective catalyst reduction
SI	Spark ignition
TASOI	Time after start of injection
THC	Total hydrocarbon
US	United States

1 Introduction

Diesel engines have been widely used in various industries, such as power generation, construction, and transportation, owing to their high thermal efficiency and power (Dhinesh et al., 2016; Sullivan et al., 2004). Moreover, they have been used in agriculture, where they have made immense contributions to the wide-scale production of food for human consumption.

However, nitrogen oxides (NO_x) and particulate matter (PM) emitted from diesel engines are known to cause environmental pollution and have harmful effects on human health (Hauser et al., 2001; Kimura et al., 2001; Twigg & Phillips, 2009). Accordingly, international discussions have been held to reduce NO_x and PM emissions. The United States and Europe have led the establishment and

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enforcement of emission regulations. In particular, compared to other engines, agricultural-machine engines have long operating times and old engine parts that are often not properly replaced. As a result, the performances of the aged agricultural-machine engines are deteriorated and their NO_x and PM emissions are dramatically increased (Zhu & Andersson, 2020).

To satisfy emission regulations, various combustion strategies and after-treatment devices have been developed and applied. The multi-stage injection strategies can be divided into pre-injection and post-injection methods in accordance with the fuel injection timing (Choi & Park, 2022; Hardy & Reitz, 2006; Tanaka et al., 2002). Several studies on multi-stage injection strategies showed that the post-injection method effectively reduces NO_x and PM emissions (Badami et al., 2003; Benajes et al., 2001; Chen, 2000; Choi & Park, 2022; Farhan et al., 2020; Hotta et al., 2005). In addition, research has been conducted on low-temperature combustion that dramatically reduces NO_x and PM simultaneously through increased exhaust gas recirculation (EGR) and delayed fuel injection timing (Imtenan et al., 2014). Although there are various combustion strategies, the combustion approach is essential for controlling the solenoid injector based on the engine control unit (ECU). Therefore, its application to an old agricultural-machine engine using a mechanical injector is practically impossible. A selective catalytic reduction (SCR) device and a diesel particulate filter (DPF)—post-treatment devices installed in the exhaust port of diesel engines—effectively reduce NO_x and PM emissions, respectively, without applying additional combustion strategies (Amiridis et al., 1996; Kang et al., 2018; Reşitoğlu et al., 2015). Nevertheless, the post-treatment system cannot inherently prevent the formation of NO_x and PM. Therefore, it is necessary to design a method that can fundamentally prevent the formation of NO_x and PM in old agricultural-machine engines.

Previous studies have shown that the formation of NO_x and PM deposits in diesel engines is the main cause of their injector spray deterioration. Research has shown that the spray of a deteriorated injector affects not only the engine performance but also the NO_x and PM emissions. Payri et al. (2020) reported that, in the case of an aged common rail diesel injector, the static flow rate was lower than that of a new injector because deposits on the orifice of the injector nozzle acted as an obstacle to the flow of fuel. D'Ambrosio et al. reported that coked injectors decreased the penetration length and spray cone angle owing to deposit formation. It was also reported that fuel consumption and exhaust emissions increased the same output from the engine owing to poor fuel atomization and injection dynamics (D'Ambrosio & Ferrari, 2012). Kidoguchi et al. reported that deposit formation in the injector hole made the spray cone angle smaller and the spray penetration longer. Consequently, the

fuel was wetted on the combustion chamber wall and produced soluble organic fraction (SOF) emissions (Kidoguchi et al., 2018). Richards et al. reported that the spray cone angle was narrow, and the spray penetration was short in the case of a fouled injector spray. As a result of an in-cylinder photographic investigation, it was reported that the fouled injector began combustion near the nozzle, resulting in poor combustion efficiency and incomplete combustion (Richards et al., 1997).

According to prior research, when a used injector is replaced with a new injector, or when the spray performance of the used injector is restored to the new injector level, engine performance improvement and emission reduction can be expected. McCormick et al. quantitatively measured the smoke opacity and exhaust gas emission of engines of heavy-duty diesel vehicles with high smoke opacity. After the engine was repaired, the exhaust gas was measured again to compare it before and after repair. Among the experimental results, there were effects of reducing smoke opacity, reducing exhaust gas, and improving fuel efficiency in the case of replacing some or all the injectors (McCormick et al., 2003). However, replacing used injectors with new ones is costly and not environmentally sound.

Conversely, the method of restoring injector performance through remanufacturing is not only economical but also environmentally friendly. Amaya et al. reported that, when all injector parts were remanufactured and reused, there was an environmental benefit of 46.10% over the classic lifecycle of an injector (Amaya et al., 2010). However, to maximize the benefits of remanufacturing, it is necessary to understand which injector parts are important factors in the deterioration of spray performance. Therefore, this study investigated the effects of the replacement of mechanical injectors by measuring the injection amount and rate and by conducting spray visualization experiments. In addition, an engine experiment was conducted to investigate the effect of the spray characteristics of each injector on the engine.

2 Experimental Method

2.1 Operation Mechanism of the Mechanical Injector

A schematic diagram of the operation mechanism of the mechanical injector is shown in Fig. 1. The mechanical diesel injectors are controlled by an injection pump connected to the engine and a pressure spring inside the injector. Injection pump pressurizes and delivers fuel through a plunger. The fuel pressurized by the plunger of the injection pump is delivered to the injector and increases the pressure of the accumulating chamber inside the injector. The

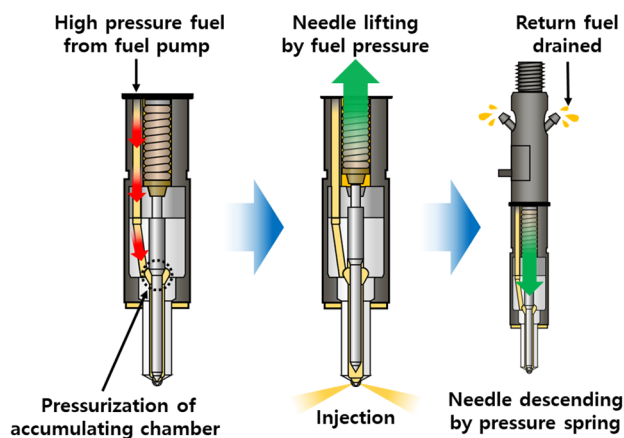


Fig. 1 Schematic diagram of the operation mechanism of the mechanical injector

increase in accumulating chamber pressure generates a force that lifts the injector needle and starts fuel injection. As fuel is injected, the pressure in the accumulating chamber decreases, and the force to lift the injector needle decreases. As a result, the needle is lowered by the pressure spring and fuel injection ends.

A schematic diagram of the experimental method is shown in Fig. 2. Mechanical diesel injector spray experiments were conducted using the injector drive system. In that system, a ten-horsepower three-phase induction motor was used to drive the injection pump. The injection duration was controlled using the fuel control lever of the injection pump, and the number of injections per minute was controlled through the pump rotation speed.

Figure 2a shows the setup of the injection quantity measurement. The measurement was conducted using the precision scale (EX6201G, OHAUS) of the injected fuel in the measuring vessel. The injection quantity measuring vessel was placed in front of the injector nozzle to collect the injected fuel.

Figure 2b shows the setup of the injection rate measurement. The Bosch tube method was used to measure the injection rate. The length of the Bosch tube was 30 m, and a damping chamber was attached to damp the pressure waveform occurring in the Bosch tube after injection. In addition, the pressure inside the Bosch tube was kept constant using a relief valve. The pressure sensor used to measure the injection rate was a piezoresistive absolute pressure sensor (4045A100, Kistler). To measure the injection rate immediately after injection, the pressure sensor was installed at a point 30 mm below the injector nozzle. In addition, the fuel pressure at the injector fuel inlet was measured to determine the injection start time. For this purpose, a piezoresistive pressure sensor (4067A1000, Kistler) was installed in front

of the injector fuel inlet. The injection rate data and the fuel pressure were acquired using a DAQ board (BNC-2110, NI).

Figure 2c shows the setup of the spray visualization. Macroscopic spray properties, such as spray angle and spray penetration length, are affected by ambient density (Arrègle et al., 1999; Hiroyasu & Arai, 1990; Taşkıran & Ergeneman, 2011). Therefore, the constant volume chamber was used in the spray visualization experiment to create an ambient density inside the engine cylinder. The dense atmosphere in the chamber was produced by supplying the chamber with nitrogen. The Mie scattering optic technique was used to identify the spray. As a light source to visualize the spray, two 100-W LED lamps were used, and they were installed on both sides of the constant volume chamber. To obtain the bottom-view images of the spray, a mirror was installed under the chamber, and the spray image reflected on the mirror was captured with a high-speed camera (VEO-1010L, Phantom). The start time of the camera shooting was controlled using a CompactRIO controller (National Instruments), and shooting was initiated when the fuel pressure was higher than the threshold pressure.

Figure 2d shows a schematic diagram of the engine experimental apparatus. In this study, Perkins 1104C-44T engines of more than 10 years of age were utilized to perform the engine experiments. The detailed engine specifications are summarized in Table 1. Various pieces of experimental equipment were attached to the engine. A starter motor and a 300 kW EC dynamometer were used to operate the engine under a steady state. The engine torque and power were measured on an EC dynamometer controller, and the engine data were saved on the NI-DAQ board. A temperature controller equipped with a feedback system maintained the respective temperatures of the intake air, coolant, and engine oil. A low-pressure fuel pump, fuel throttle valve, and return line were installed to provide a stable fuel supply during the engine experiments. An AVL MDS 350 device was connected to the exhaust port to measure the concentration of exhaust gas components.

2.2 Experimental Conditions

Figure 3 depicts the injector used in the experiment, and the detailed injector specifications are summarized in Table 2. The injectors (2645K011, Perkins) used in the experiment were the same type as the ones installed in the agricultural diesel engines (1104C-44T, Perkins) to which Tier 2 emission regulations were applied. The new injectors were the same model. The new injectors had no external damage. Used injectors are injectors that were installed in the 1104C-44T model diesel engine of an agricultural tractor that was used for more than 2000 h for rotary hoeing, loading, etc.; thus, there were deposits in the injector nozzles caused by combustion.

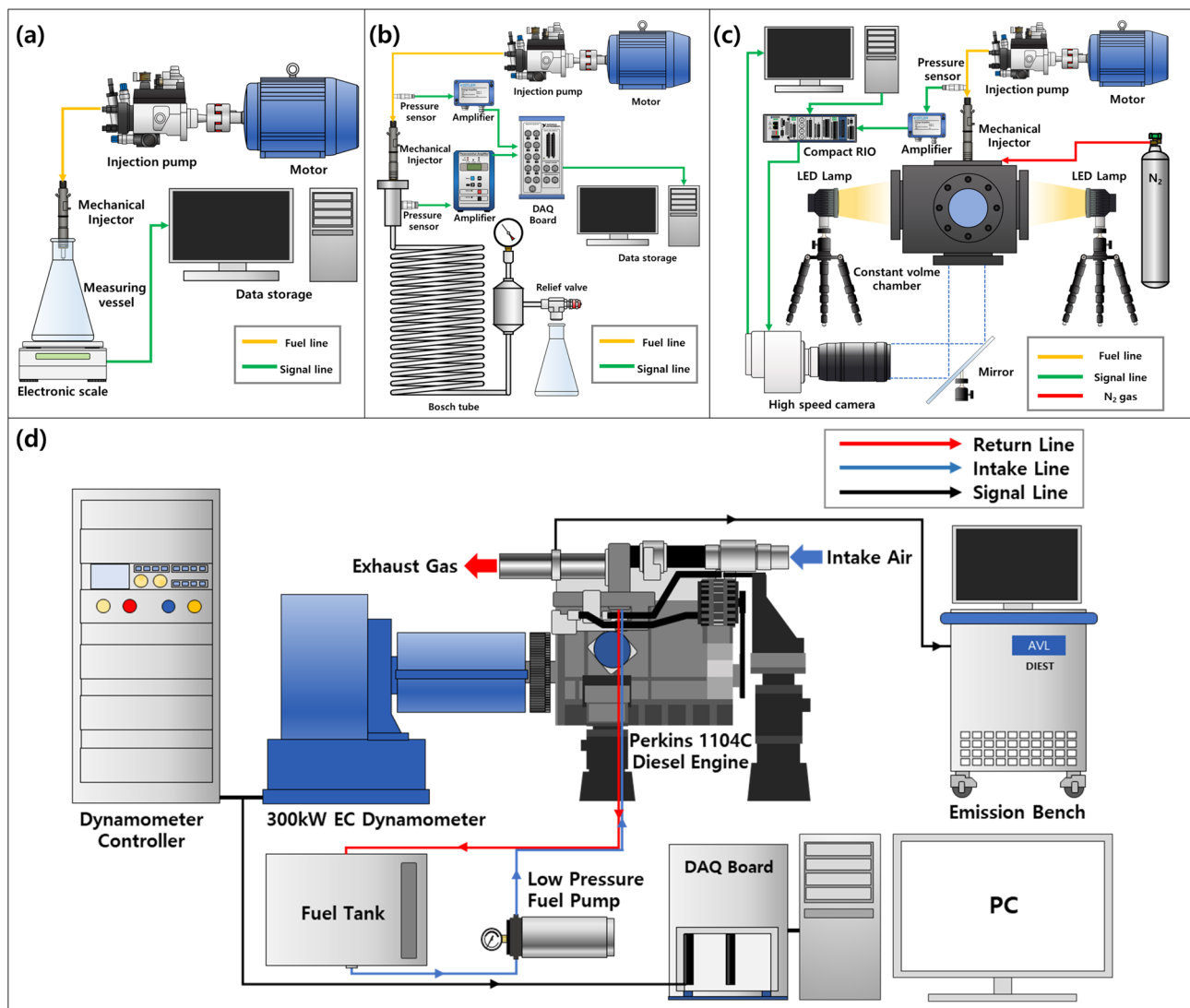


Fig. 2 Schematic diagram of experimental setup: **a** injection quantity measurement, **b** injection rate measurement, **c** spray visualization, and **d** Perkins 1104C-44T diesel engine setup

Table 1 Specifications of Perkins 1104C-44T engine

Parameter	Value
Number of cylinders (inline)	4
Cycle (stroke)	4
Bore (mm)	105
Stroke (mm)	127
Displacement volume (mm ³)	1100
Compression ratio	19.3:1
Target emission regulation	EU Stage II/ US EPA Tier 2

Table 3 summarizes the injection quantity measurement conditions. The speed of the injection pump was set at 600 RPM, which corresponded to the engine speed of 1200 RPM. As the driving condition of the injector, the circumstance corresponding to the full load in the engine was selected. To clearly verify the injection quantity in accordance with the injectors, the injection pump was operated under a full load condition. Therefore, the fuel control lever of the injection pump was pulled to its maximum.

The experiment was conducted at room temperature, and the injection quantity measurement result was calculated by dividing the mass of fuel injected from the injector for 1 min by the number of injections.

Fig. 3 Injectors used in the experiments**Table 2** Injectors used in the experiments

Parameter	Value
Number of holes	5
Diameter of hole (mm)	0.24
Umbrella angle (deg)	136
Opening pressure (bar)	290

Table 3 Injection quantity measurement conditions

Item	Description
Injection pump speed (RPM)	600
Operating load of injection pump (%)	Full load (100%)
Fuel temperature (°C)	25
Fuel	Diesel

Table 4 Injection rate measurement conditions

Item	Description
Injection pump speed (RPM)	800
Operating load of injection pump (%)	Full load (100%)
Fuel temperature (°C)	25
Pressure in the Bosch tube (bar)	10

Table 4 summarizes the injection rate measurement conditions. The speed of the injection pump was fixed at 800 RPM. An extension of the fuel line was required to connect the injector to the Bosch tube. As a result, the fuel pressure decreased compared to that before the fuel line was extended, and it was necessary to increase the fuel pressure. Increasing the fuel pressure is possible by increasing the rotational speed of the injection pump. However, if the rotation speed of the injection pump is faster than 800 RPM, a part of the fuel supplied to the injection pump is bypassed owing to the governor inside the pump, and the fuel pressure does not increase.

Table 5 Spray visualization conditions

Item	Description
Injection pump speed (RPM)	800
Operating load of injection pump (%)	Full load (100%)
Ambient gas temperature (°C)	25
Ambient gas pressure (bar)	16
Ambient gas density (kg/m ³)	18.13
Fuel	Diesel
Imaging frequency (fps)	10,000

The load conditions under which the injection pump was assessed were full load conditions, similar to the injection quantity measurement condition. The injection rate data were acquired while the fuel pressure in the Bosch tube was maintained at 10 bar and the temperature was at room temperature.

Table 5 summarizes the spray visualization conditions. The speed of the injection pump was 800 RPM, and the load condition of the injection pump was a full load condition.

The inside of the constant volume chamber was pressurized with nitrogen at room temperature at a pressure of 16 bar to reproduce the ambient density in the combustion chamber. Bottom-view images of the spray reflected from a mirror were recorded at a rate of 10,000 fps.

Table 6 summarizes the engine experimental conditions. The engine speed was varied from 1000 to 1200 RPM with an interval of 100 RPM. To clearly verify the combustion and emissions characteristics in accordance with the injectors, the engine was operated under a full load condition. The engine oil and coolant temperatures were maintained at 80 °C by the temperature control system. In addition, the intake air temperature at the front of the turbocharger was fixed at 30 °C. In engines using mechanical injectors, the fuel is injected when the pressure of fuel compressed by the pump becomes higher than the tension of spring. Thus, in this study, the fuel injection parameters for injection timing, injection pressure, and injection duration were varied

Table 6 Engine experimental conditions

Item	Description
Engine speed (RPM)	1000, 1100, 1200
Engine operating load (%)	Full load (100%)
Oil and coolant temperature (°C)	80
Intake temperature (°C)	30
Fuel injection timing, pressure, duration	Depending on spring elasticity of injectors

in accordance with the elasticity of the spring mounted on the injector.

2.3 Spray Image Post-processing

The spray image processing procedure using MATLAB codes is shown in Fig. 4. The obtained spray images were processed using the MATLAB program to quantitatively analyze the macroscopic properties of the spray. The first step in spray image processing was to obtain an average spray image by overlaying ten spray images under the same conditions. The superimposed images were binarized based on the threshold to identify the spray boundary.

For the binarization of the spray image, the pixel value was set to 1 in the case of spray, and the pixel value was set to 0 in the case of non-spray. Finally, the spray penetration length was determined by measuring the distance from the nozzle center to the farthest spray boundary coordinates, and the spray angle was calculated through the area formula of the circular sector.

3 Results and Discussion

3.1 Comparison of Injection Quantity

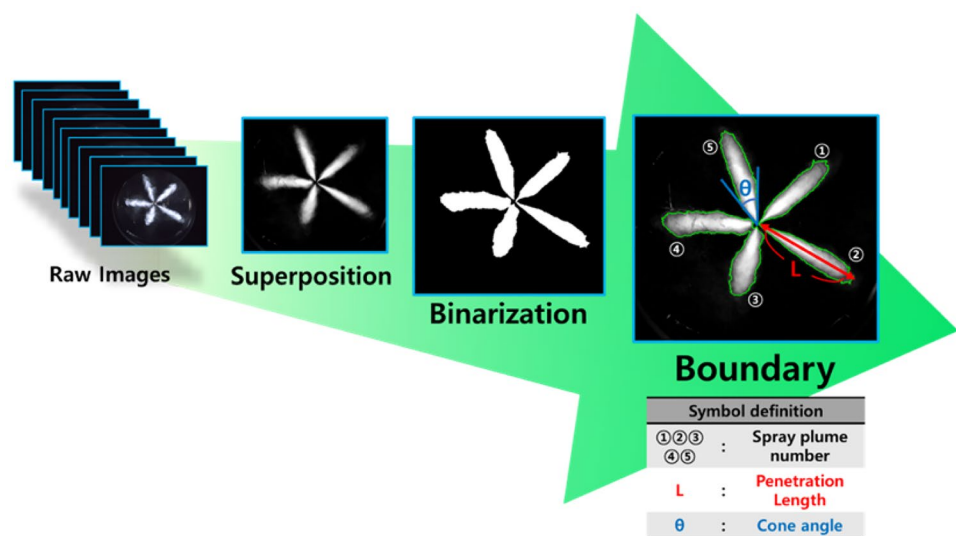
In this part of study, the injection quantities of the used injectors and new injectors were compared. For the measurement of the injection quantity, the injection pump speed was 600 RPM, and the measurement was performed under conditions corresponding to the speed of 1200 RPM. In addition, fuel was injected while the fuel control lever of the injection pump was pulled to the maximum, which corresponded to the maximum engine operating load condition.

Figure 5 shows the comparison results of the injection quantity for each cylinder of the used injectors and the new injectors at an injection pump speed of 600 RPM. In this figure, the injection quantity of new injectors is usually 2–4% higher than that of used injectors.

However, in the case of the used injector installed in cylinder No. 2, the injection quantity was greater than that of the new injector. This is expected to be the result of a combination of the injection quantity difference due to the manufacturing tolerance of the nozzle of the new injector and the injection quantity difference due to the difference in the degree of nozzle contamination of the used injector.

The image in Fig. 6 is a magnification with an optical microscope of the nozzle surface of each injector. It is observed that deposits have accumulated on the surface of the used injector. The deposits on the nozzle surface reduced the diameter of the hole through which the fuel was injected. As a result, the injection quantity of used injectors was usually lower than that of new injectors.

Fig. 4 Spray image processing procedure using MATLAB codes



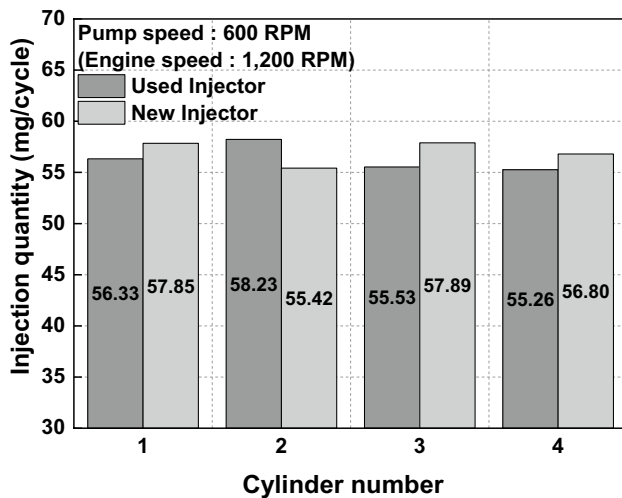


Fig. 5 Comparison of injection quantity of injectors by cylinder

3.2 Comparison of Injection Pressure and Injection Rate

Figure 7 shows the injection pressure and injection rate of the used and new injectors. The injection rate was measured when the speed of the injection pump was 800 RPM and the fuel control lever was pulled to the maximum.

Figure 7a shows the result of measuring the injection pressure of each injector. The fuel injection start point was the point at which the fuel pressure increased owing to the injection pump and then the pressure rapidly decreased as the injector valve opened.

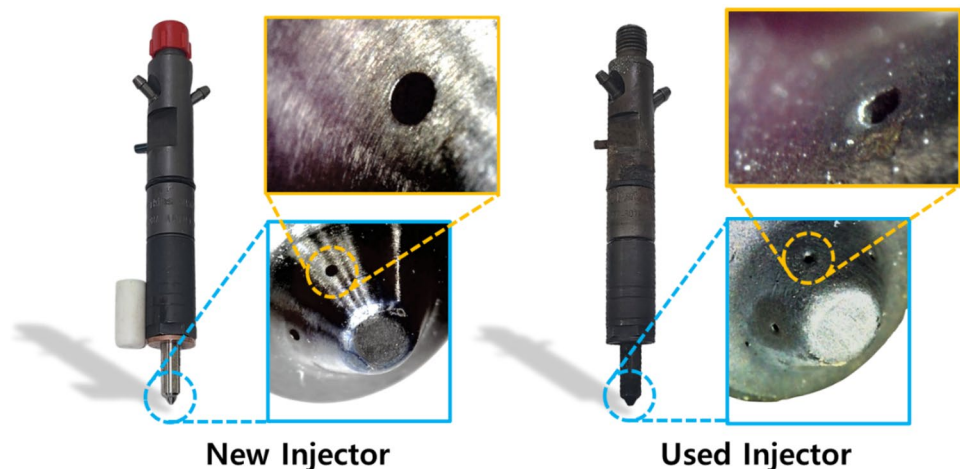
By comparing the injection start time of each injector in Fig. 7a, it is apparent that the used injector started injection earlier than the new injector. The injection end point was the point at which the injection pressure decreased on account of fuel injection and then increased again as the valve was closed.

By comparing the injection end time of each injector in this figure, it is observed that the injection end time of the used injector was later than that of the new injector. The opening pressure was the injection pressure when the injection began. The opening pressure of the used injector was 225 bar and the opening pressure of the new injector was 320 bar. Thus, the opening pressure of the used injector was lower than that of the new injector.

Figure 8 shows the effect of repeated combustion on the injector parts. Among these parts, the one that affected the injection pressure and injection timing was the spring inside the injector. This spring repeated compression and tension during the process of opening and closing the injector valve. And when the injector is not operating, it remains compressed. Therefore, the spring is expected to have a lower elastic modulus than the new spring due to fatigue caused by repetitive loads and creep caused by continuous loads. For this reason, the force with which the pressure spring of a used injector pushes the valve is expected to be less than that of a new injector. As a result, due to the low elastic modulus of the pressure spring, the used injector may have started injection at low pressure and ended the injection late.

Figure 7b shows the effect of spring deterioration on the injection rate. In this figure, the used injector injected 0.61 ms earlier than the new injector. This means that, when the engine rotation speed was 1600 RPM, the injection timing advanced by 2.9° . Moreover, the injection end of the used injector ended injection 0.68 ms later than that of the new injector. This means that, when the engine speed was 1600 RPM, the injection end was delayed by 3.3° . The injection duration of the used injector was, thus, twice that of the new injector, and the injection rate of the used injector was 55% of that of the new injector.

Fig. 6 Magnified image of the surfaces of new and used injector nozzles



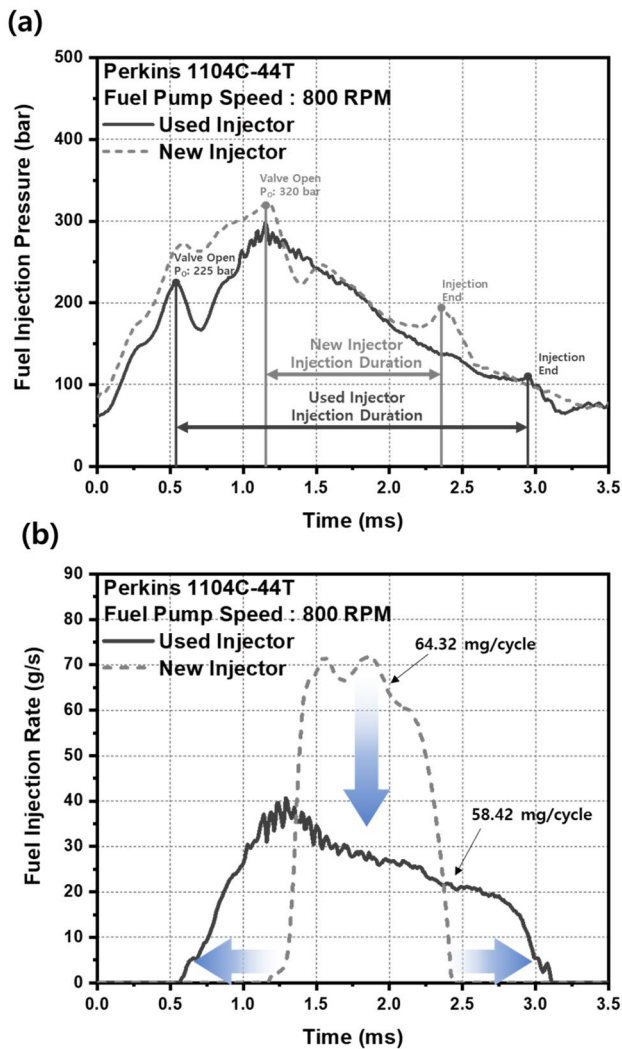


Fig. 7 Comparison of **a** fuel injection pressure and **b** fuel injection rate

3.3 Comparison of Spray Visualization Results

The spray images were obtained when the speed of the injection pump was 800 RPM and the fuel control lever was pulled to the maximum. To reproduce the ambient density of the engine combustion chamber, spray visualization was performed while the internal pressure of the chamber was maintained at 16 bar. Figure 8 presents the spray visualization images and spray penetration length of the used and new injectors.

In the spray image of the used injector and the new injector in Fig. 9a, the development velocities of the penetration lengths do not differ significantly despite the higher injection rate of the new injector.

This is because the new injector has a high injection rate, but the axial drag is large because the spray angle is large.

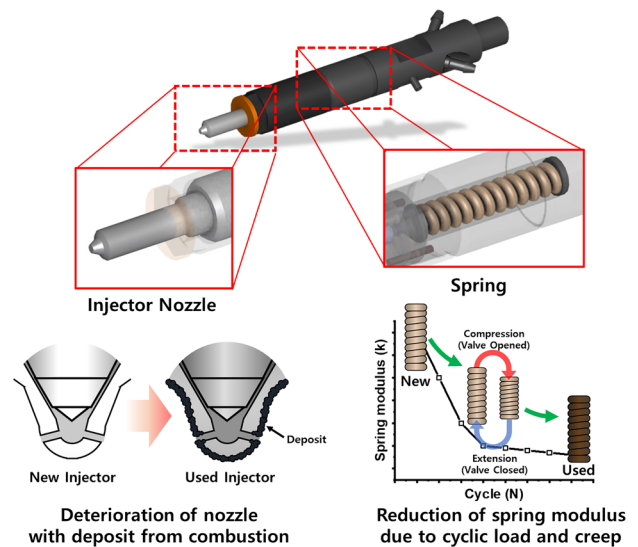


Fig. 8 Deterioration of injector parts due to repeated combustion

Conversely, the used injector has a low injection rate, but the drag force is small because the spray angle is small. Therefore, despite the difference in injection rates between the two injectors, they can have similar spray penetration lengths due to the difference in drag of the spray plume.

In Fig. 9b, the average spray penetration lengths of the used injector and the new injector are almost the same from the start of injection to the 2 ms point. However, after 2 ms, the average spray penetration length of the used injector continues to increase, whereas the average spray penetration length of the new injector does not increase. This is because the used injector continues injection 2 ms after the start of injection.

Figure 9c shows the spray penetration length for each spray plume. Unlike the trend of the average spray penetration lengths of the two injectors, a difference exists in the development velocity of the penetration length for each plume.

The development velocity of the penetration length of plume 2 is fast in the case of the new injector. According to a study conducted by Payri et al. (2020), deposits present in a diesel injector nozzle hole can obstruct the path of fuel flowing through the nozzle hole and reduce the effective nozzle hole area, thereby reducing the flow rate of the injected fuel.

Therefore, the difference in the development speed of plume 2 between the two injectors is expected to be due to the difference in the degree of deposit attached around the nozzle hole of the injector used. The development velocity of the penetration length of the remaining spray plumes is similar for the used and new injectors.

Figure 10 shows the results of the spray cone angle measurement and the effect of nozzle deterioration on the spray cone angle. Figure 10a shows the spray cone angle for each

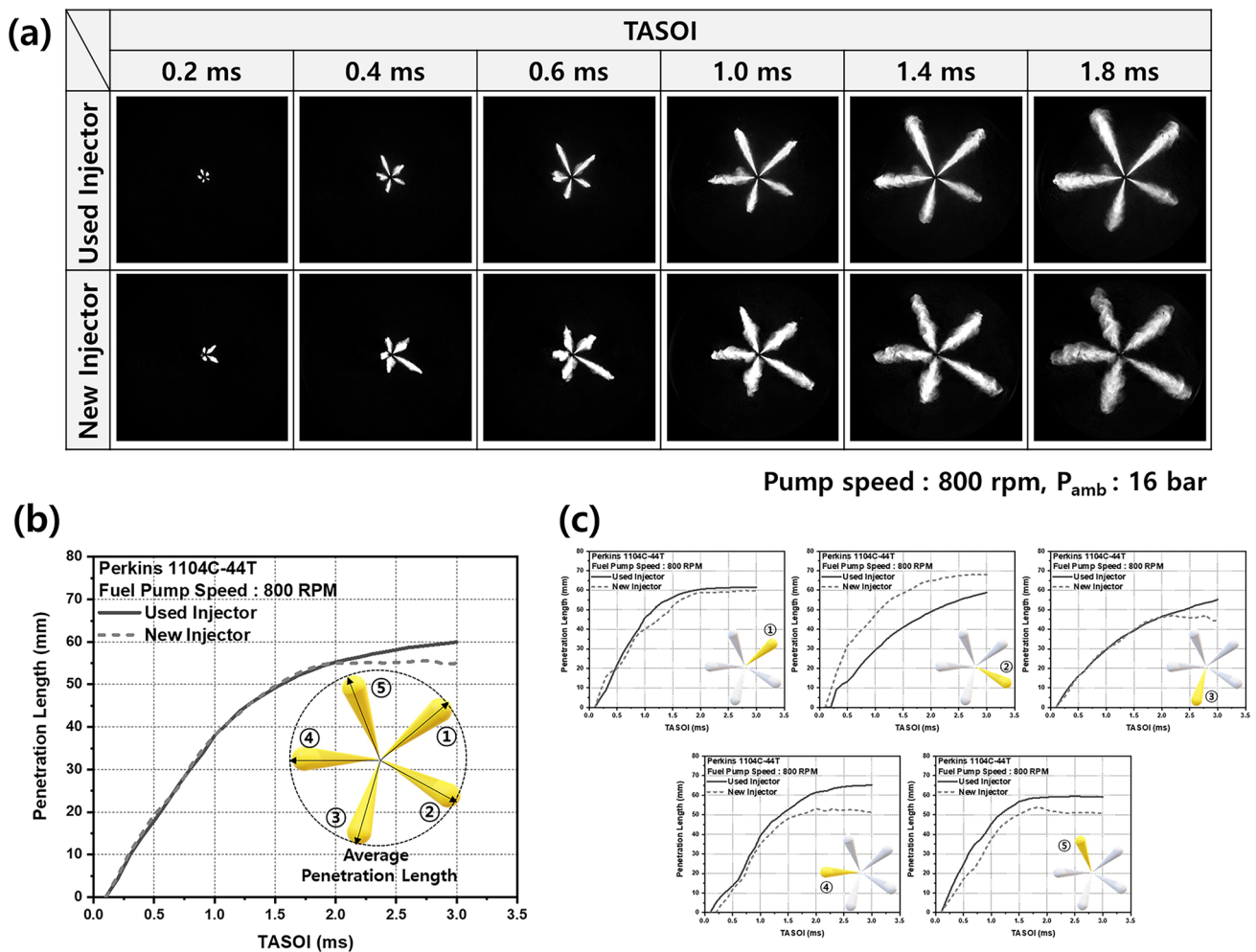


Fig. 9 Spray visualization images and spray penetration length: **a** spray images obtained after the start of injection, **b** average spray penetration length, and **c** spray penetration length by spray plume

plume at 1.0 ms after injection. The spray cone angles of each spray plume of the used injectors are generally smaller than those of the new injectors.

Figure 10b shows the difference in spray cone angles for each spray plume. The spray cone angles for each plume of the used injectors, except for that of plume 2, are smaller than those of the new injectors, and they are approximately 70% of the spray cone angles of the new injectors.

Figure 10c shows the cause of the difference in spray cone angle between the used injector and the new injector. In the used injector, deposits from repeated combustion appear on the nozzle surface and clog the nozzle hole. Subsequently, the diameter of the nozzle is reduced owing to the clogging of the nozzle hole, resulting in a narrow spray cone angle.

3.4 Comparison of Combustion and Emission Characteristics

In this part of the study, the effects of injector replacement on the combustion characteristics were investigated. For the engine experiments, the test engine was operated under the condition of a full engine operating load between the engine speeds of 1000 RPM and 1200 RPM. Various experimental data from the used and new injectors were measured.

Figure 11a shows the trends of engine torque in accordance with the injector under the conditions of engine speeds from 1000 to 1200 RPM. As shown in this figure, the used injector deteriorates the engine torque, which is almost maintained regardless of the engine speed. As a result, it is found that the used injector reduces the engine torque by more than 20% of the target specification.

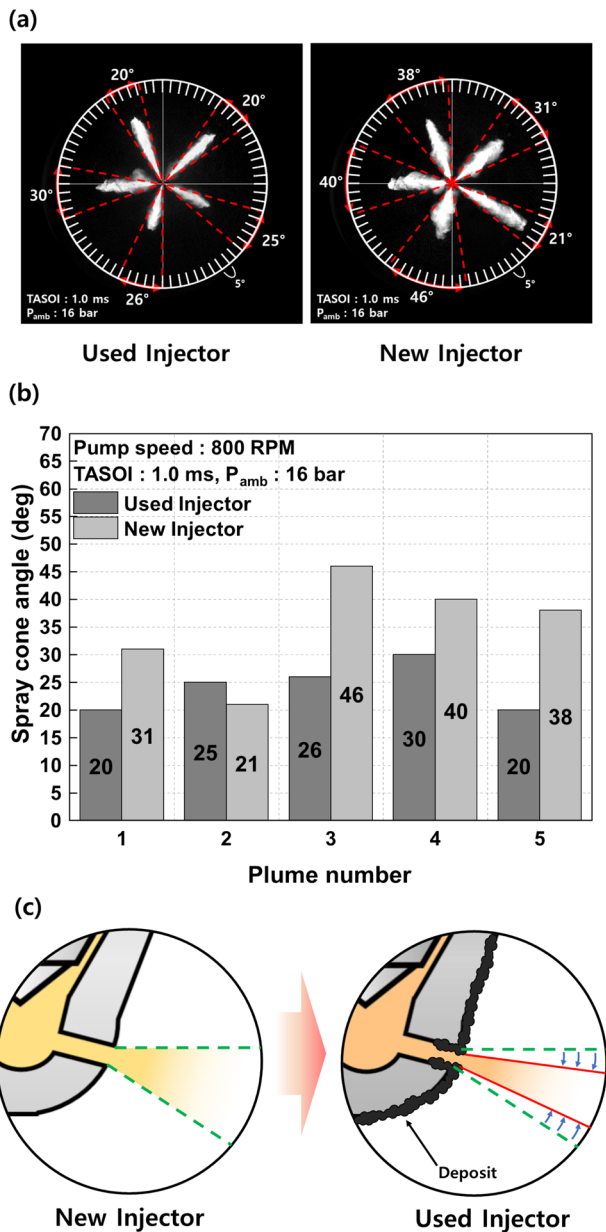


Fig. 10 Spray angle measurement result: **a** spray image at 1.0 ms after spray start, **b** measurement result of the spray cone angle at 1.0 ms after the injection start, and **c** spray angle reduction effect owing to nozzle deterioration

On the other hand, the engine torque is recovered to 90–96% of the target specification by replacing the used injectors with the new injectors. Moreover, with a faster engine speed, a higher engine torque is observed under the condition of the new injectors. The effect of injectors on engine torque is closely related to the fuel injection quantity.

As mentioned above, the used injector deteriorates the fuel injection quantity owing to the low elasticity of the spring mounted on the used injector. With the lower fuel injection quantity, the lower fuel energy is changed to the

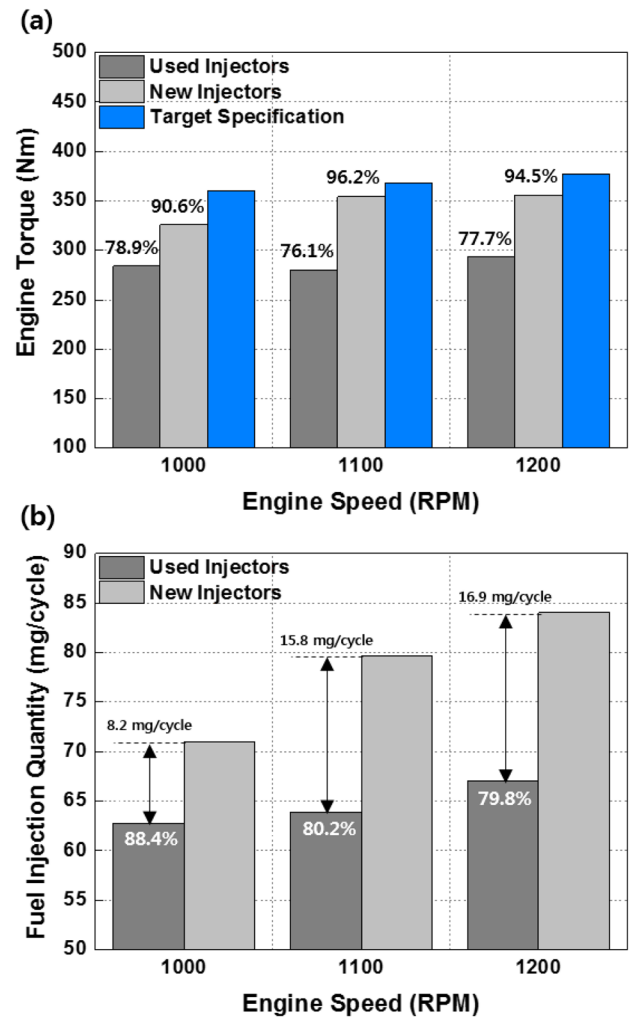


Fig. 11 Effects of injector replacement on **a** engine torque and **b** fuel injection quantity under different engine speeds

engine torque. To confirm the relationship between the engine torque and fuel injection quantity, the fuel injection quantity is calculated by obtaining the intake air flow rate and lambda value.

Figure 11b provides the trends of the fuel injection quantity in accordance with the injectors under the conditions of engine speeds of 1000 RPM to 1200 RPM. In addition, this figure shows that the effect of injector replacement on the fuel injection quantity in the engine experiments is greater than that of the spray experiments.

Unlike the spray experiment without a boost compensator, the fuel pump of the Perkins 1104C-44T engine is equipped with a boost compensator, and the fuel injection quantity is controlled by the pressurized intake pressure of the manifold in addition to the rotation speed of the fuel pump and the opening amount of the throttle lever. Injector replacement increases the speed of the turbine and turbo charger in proportion to the fuel injection quantity, and the

high intake air pressure moves the boost compensator control sleeve to further increase the fuel injection quantity.

As a result, the difference in the fuel injection quantity in accordance with the injectors is 8.2 mg/cycle under the condition of the engine speed of 1000 RPM. As the engine speed increased, the difference in the fuel injection quantity was increased to 16.9 mg/cycle.

The fuel injection quantity in accordance with the injectors also can be found from the results of engine power. Figure 12 illustrates the effects of the injector replacement on engine power under the condition of engine speeds of 1000 RPM to 1200 RPM. The experimental results from the used injectors show that the lower engine power is approximately 19–22% lower than that of the target specification. In addition, this figure shows that the new injectors recovered the engine power to 90% or more of the target specification.

Moreover, it is shown that the engine power is almost the same as the target specification under the condition of equipping the new injectors at an engine speed of 1200 RPM. The results of Figs. 11 and 12 demonstrate that the engine torque and power are restored by replacing the used injectors with the new injectors. However, it is impossible to completely recover the engine performance only by replacing the injectors on account of the diminished performance of the other components, such as the fuel pumps installed in the engines. Nevertheless, the above results demonstrate that injector replacement can improve engine performance more effectively than replacing other components.

Figure 13 shows the effects of the injectors on the fuel consumption rate and exhaust gas temperature under the condition of engine speeds of 1000 RPM to 1200 RPM. Since the engine operating at full engine load shows a combustion efficiency of more than 99% under all experimental

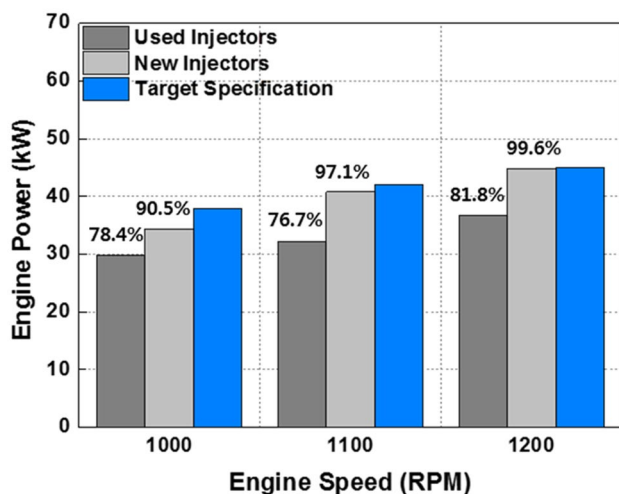


Fig. 12 Effects of injector replacement on engine power under different engine speeds

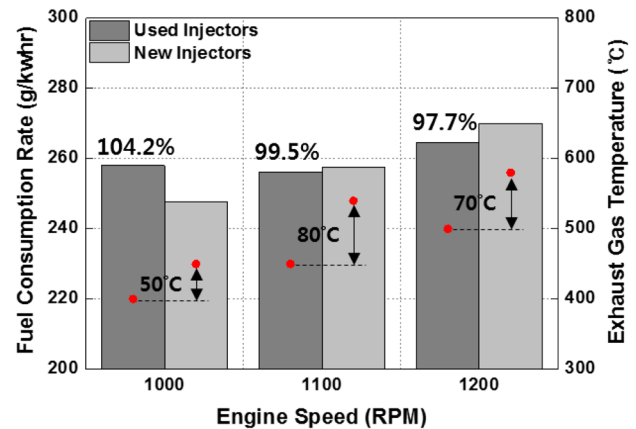


Fig. 13 Effects of injector replacement on fuel consumption rate and exhaust gas temperature under different engine speeds

conditions, the lambda error caused by incomplete combustion is negligibly low. Therefore, this study calculated the fuel consumption rate based on the intake air flow rate and lambda value. Unlike the fuel consumption rate, which was not dramatically changed under the application of the used injectors, the new injectors apparently increased the fuel consumption rate. Under the condition of the engine speed of 1000 RPM, the used injectors made the fuel consumption rate higher than that of the new injectors. As the engine speed increased, the trends of the fuel consumption rate in accordance with the injectors were reversed. The reason for the increase in fuel consumption rate is expected to be the deterioration of thermal efficiency due to the increased combustion temperature.

In particular, it was impossible to measure the cylinder temperature. Instead of measuring the combustion chamber temperature, the combustion temperature was indirectly compared by measuring the exhaust gas temperature.

The right side of Fig. 13 depicts the exhaust gas temperature in accordance with the injectors. It is observed that the new injectors raised the exhaust gas temperature by 50–80 °C compared to the used injectors. The exhaust gas temperature is proportional to the combustion temperature, and the exhaust gas temperature results showed that the exhaust gas temperature was high when the new injector was installed in the engine. This can be seen that when a new injector was installed in the engine, the temperature inside the cylinder was higher because more fuel was injected into the cylinder than with a used injector. Although high combustion temperatures can effectively oxidize incomplete combustion products, high combustion temperatures can cause reduced thermal efficiency due to heat loss through the cylinder wall and piston surface.

The following results show that the injector replacement had significant effects on the exhaust gas materials as well as the engine performance.

Figure 14a and b shows the CO and HC emissions in accordance with the injectors under the condition of engine speeds of 1000 RPM to 1200 RPM, respectively. Although CO and HC were produced by the incomplete combustion, the figures show the opposite emission trends in accordance with the injectors. The above experimental results were obtained because the major factors affecting CO and HC formation were apparently different. According to previous research, CO is mainly produced under the condition of a rich equivalence ratio with a low O₂ concentration, whereas HC is produced in regions of locally low temperature in the combustion chamber (Imtanan et al., 2014). Here, CO emissions were compared with the O₂ concentration in the exhaust gas, as given in Fig. 14a. As mentioned above, a large amount of fuel was supplied to the combustion chamber by the new injectors, and O₂ participated in the combustion reaction in proportion to the amount of fuel present.

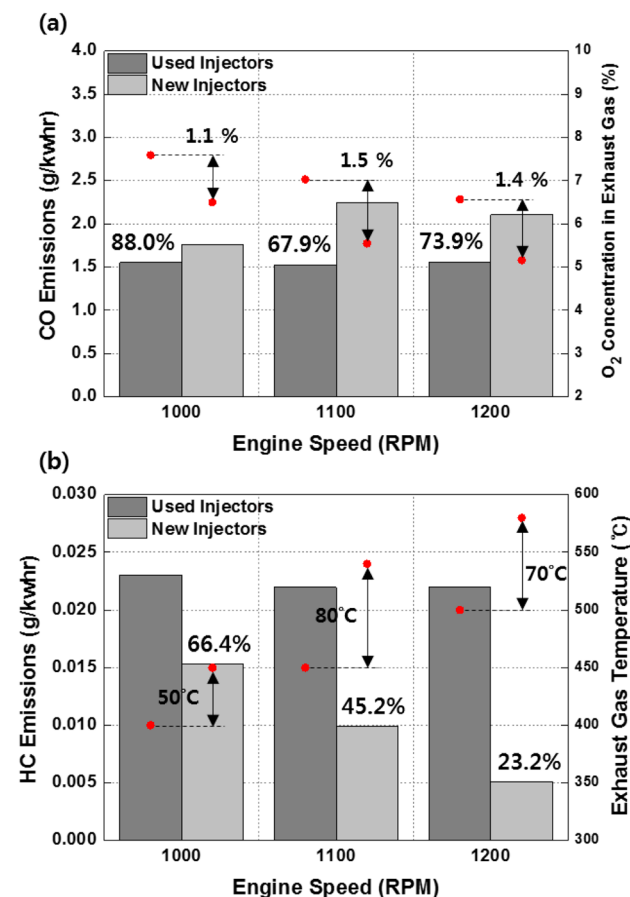


Fig. 14 Effects of injector replacement on **a** CO and O₂ emissions and **b** HC emissions and exhaust gas temperature under different engine speeds

Application of the new injectors promoted the CO owing to the lack of O₂ in the combustion chamber.

Figure 14b depicts the relationship of HC emissions to the exhaust gas temperature. This figure proves that increasing the exhaust gas temperature by equipping the new injectors suppressed the HC emissions. In addition, NO_x and PM emissions, which are the most critical issues for the emissions regulations of diesel engines, are predominantly affected by the injector replacement.

Figure 15a and b illustrates NO_x and PM emissions in accordance with the injectors under the condition of engine speeds of 1000–1200 RPM. Unlike the conventional results of NO_x and PM emissions, which present a trade-off relationship, this figure shows that not only NO_x emissions but also PM emissions are reduced by replacing the used injectors with new injectors. The spring and nozzle mounted on the injectors predominantly affect

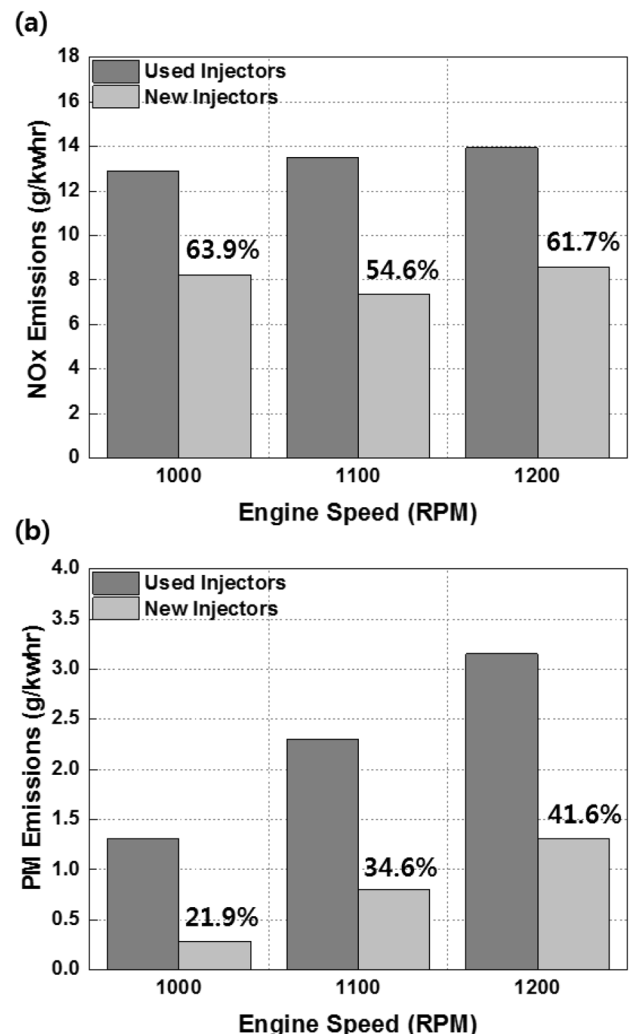


Fig. 15 Effects of injector replacement on **a** NO_x and **b** PM emissions under different engine speeds

NO_x and PM emissions. The NO_x emissions are closely related with the fuel injection timing and duration, which are determined by the elasticity of the springs mounted on the injectors. The low elasticity of the springs mounted on the used injectors advanced the fuel injection timing and increased the fuel injection duration, as given in Fig. 6. The used injector cases presented high NO_x emission despite a low combustion temperature.

Conversely, the new injectors retarded the fuel injection timing, resulting in low NO_x emissions. PM emissions are predominantly affected by the fuel atomization, which is related to the fuel injection pressure and nozzle geometry. The used injectors supplied the fuel under the condition of low pressure, deteriorating the fuel atomization. The region of a locally rich equivalence ratio promoted PM formation. In addition to the low injection pressure, the used injectors caused various spray plumes owing to PM precipitation contributing to high PM emissions.

4 Conclusion

The goal of this study was to investigate the effects of injector replacement on the spray and combustion characteristics. Spray and engine experiments were conducted, and various experimental data were measured. The main conclusions of this study are summarized as follows:

1. Owing to the deposits on the nozzle surface, the injection quantity measurement results of the new injectors were generally 2–4% higher than those of the used injectors. The deterioration of the spring inside the injector affected the injection pressure of the injector. As a result of the injection pressure measurement, the opening pressure of the used injector was 225 bar, which was 70% of that of the new injector. The deterioration of the spring also affected the injection timing. The injection timing of the used injector was advanced by 2.9° and the injection ending was delayed by 3.3° compared to the new injector when the pump speed was 800 RPM. Consequently, the injection rate measurement showed that the used injector had twice the injection period than the new injector, and the injection rate was approximately half that of the new injector.
2. According to the spray visualization, the average spray penetration length of the two injectors increased similarly up to 2 ms after injection. Moreover, after 2 ms, owing to the difference in injection duration, the spray penetration length of the used injector continued to increase, whereas the new injector did not. When the spray penetration length for each plume was compared, the new injector was shown to have increased more rapidly in the case of spray plume 2, while there was

no significant difference between the injectors for the remaining spray plumes. Owing to deposits on the nozzle surface, the spray angle of the used injectors was usually smaller than that of the new injectors. The spray cone angle of the used injector was approximately 70% of that of the new injector.

3. The engine experimental results showed that the new injectors recovered over 90% of the engine torque and engine power on account of the increased fuel injection quantity. Higher engine speed correlated with a larger difference in the fuel injection quantity in accordance with the injectors. However, employing the new injectors caused heat loss, and the fuel consumption rate was deteriorated.
4. The new injectors reduced HC emissions and increased CO emissions. This is because HC and CO were produced under the conditions of low temperature and low O₂ concentration conditions, respectively. The new injectors effectively reduced NO_x and PM emissions by retarding the fuel injection timing and increasing the fuel injection pressure. In addition, the heterogeneous nozzle geometry, which can cause varied spray plumes, can consequently cause high PM emissions.

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Data availability The datasets used and/or analysed during the current study available from the corresponding author on reasonable request.

Declarations

Conflict of interest The authors have no conflict of interest, financial or otherwise.

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