

Study of Spray and Combustion Using Fast Response Common Rail Injector with Micro Orifice Nozzle

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ABSTRACT

Reduction of the nozzle hole diameter was investigated with the aim of further reducing harmful exhaust emissions from diesel engines. This investigation used a prototype injector with an injection rate gradient at the start of fuel injection that is approximately 80% greater than that of a conventional mass-production injector. Three types of nozzles were prepared that have the same hydraulic flow rate and only different numbers of holes and hole diameters from 0.08 mm to 0.12 mm. Spray measurement results confirmed that as the hole diameter becomes smaller, the spray penetration drops and the droplet Sauter mean diameter also becomes smaller. In addition, the results of engine combustion tests showed that reducing the nozzle hole diameter and increasing the number of holes increased the rich portion of the air-fuel mixture, so this does not contribute to the reduction of harmful exhaust emissions.

1. Introduction

As a recent trend in automobile exhaust emission regulations, various countries are implementing CO₂ emissions regulations as a global warming countermeasure in addition to conventional harmful exhaust emission regulations, and are considering more stringent regulations in the future. Diesel engines generally have higher thermal efficiency than gasoline engines, making them more advantageous in reducing CO₂ emissions. However, an issue with diesel engines is increasing the overall cleanness of emissions. Diesel combustion is basically lean combustion, so NO_x cannot be sufficiently reduced using a three-way catalyst in the manner of a gasoline engine. NO_x emission after-treatment technology for lean combustion continues to advance⁽¹⁾, but the conversion performance cannot yet be considered adequate. In order to reduce NO_x it is necessary to introduce EGR gas during engine combustion to lower the combustion temperature. However, the introduction of EGR gas increases the rich portion of the air-fuel mixture, which leads to an increase in particulate matter (PM). In this manner, NO_x and PM have a trade-off relationship in diesel combustion, but simultaneous reduction is demanded. New turbocharger technology and fuel injection technology using a common rail system⁽²⁾⁻⁽⁴⁾ are being introduced to meet these demands. As a result, exhaust emissions are becoming cleaner and are complying with the strict emission regulations of each country. Numerous studies have been reported on premixed charge compression ignition (PCCI)

combustion and low-temperature combustion that aims to simultaneously reduce NO_x and PM⁽⁵⁾⁻⁽⁸⁾ in order to support more stringent exhaust emission regulations in the future. However, these methods have issues such as increased emission of unburned gases such as HC and CO, and limited operating ranges.

The authors focused on the rate of injection of the injector with the aim of increasing fuel diffusion in order to reduce exhaust emissions in conventional diesel combustion without increasing the period of ignition delay in the manner of PCCI combustion. Increasing the injection rate gradient at the start of fuel injection (achieving high-speed injection) was confirmed to effectively reduce exhaust emissions^{(9), (10)}. In addition, the influence of high-speed injection on exhaust emissions reduction was clarified by observing the flame behavior and using the two-color method to analyze the flame temperature and soot concentration in a glass piston engine⁽¹¹⁾.

Reduction of the nozzle hole diameter can be given as a method for further reducing exhaust emissions using the prototype injector in this study. Reduction of the nozzle hole diameter is known to promote spray atomization and contribute to the simultaneous reduction of NO_x and PM⁽¹²⁾. However, when the nozzle hole diameter is reduced but the injection pressure is held constant, the spray penetration decreases, so it is thought that the air-fuel mixture distribution of the spray becomes richer. Therefore, reduction of the nozzle hole diameter is predicted to produce the two contrary phenomena of reduced exhaust

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emissions due to the promotion of fuel spray atomization and increased exhaust emissions due to a drop in spray penetration.

This study aimed to quantitatively clarify the influence of reducing the nozzle hole diameter from the perspectives of spray and combustion. Nozzles with different hole diameters were prepared, and spray tests were performed to investigate the influence of reducing the hole diameter on the spray shape and droplet size distribution. In addition, the influence of reducing the hole diameter on combustion characteristics was investigated by combustion tests using a diesel engine.

2. Tested Injectors

2.1. Structure and Flow Characteristics

Table 1 shows the specifications of the injectors used in the tests, and Fig. 1 shows a cross-section drawing of the prototype injector structure. To obtain high-speed injection performance unavailable with the conventional method, the prototype injector uses a piezo element to control the common rail hydraulic pressure by means of a valve, and directly opens and closes the nozzle needle using that controlled high fuel hydraulic pressure. In addition, to further increase the rate of injection at the start of injection, the seat angle of the nozzle needle was broadened in comparison to that of the mass-production injector in order to expand the aperture area at low needle lift as shown in Fig. 2. Figure 3 shows the flow rate characteristics test results when nozzles with the specifications given in Table 2 are mounted. Figure 3 confirms that the broadened seat angle increases the fuel flow rate at low needle lift as intended.

2.2. Rate of Injection and Spray Characteristics

Figure 4 shows the results of injection rate measurement when a fuel injection quantity of $18 \text{ mm}^3/\text{st.}$ is injected at an injection pressure of 100 MPa, which were the injection conditions during the spray characteristics and engine combustion tests described hereafter. The injection rate gradient of the prototype injector at the start of fuel injection was 78% greater than that of the mass-production injector, which confirmed that the method used in this study increased the rate of injection at the start of injection.

Figure 5 shows photographs of the spray 0.5 ms after the start of injection with a fuel injection quantity of $18 \text{ mm}^3/\text{st.}$ as used to measure the rate of injection. In addition, Fig. 6 shows time series numerical representations of the spray penetration and the spray angle obtained from the recorded high-speed video images. The experimental apparatus is described in the following section.

The spray angle was obtained by measuring the angle at the position 30 mm downstream from the nozzle outlet in the axial direction of the spray. Figure 6 confirms that the different injection rate characteristics of the two injectors produce large differences in the spray, and that the spray

Table 1 Specifications of common rail injection system

Injector	Mass production	Prototype
Max. pressure (MPa)	200	200
Max. injection events	5	5
Actuator	Piezo	Piezo
Needle actuation	Back pressure control	Fuel pressure direct control

Table 2 Specifications of injector nozzles

	Mass production	Prototype
Hole diameter (mm)	0.12	0.12
Number of holes	8	8
Spray hole angle (deg)	160	160
Flow rate (cm^3/min)	740	740
Sac volume (mm^3)	0.17	0.40
Seat angle (deg)	30	60

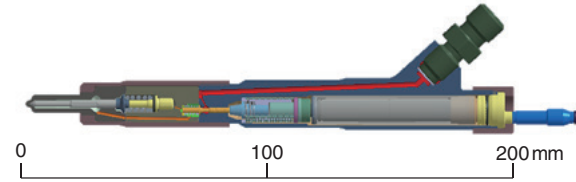


Fig. 1 Cross-section of prototype injector

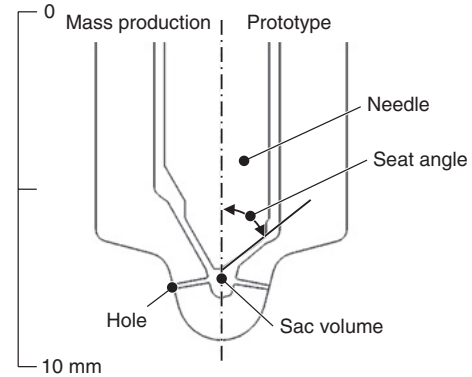


Fig. 2 Comparison of cross-section of special nozzle for fast common rail injector

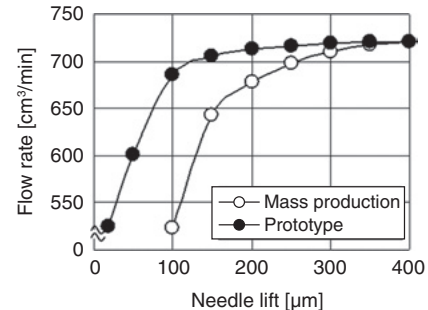


Fig. 3 Nozzle flow rate of two types of injector

penetration of the prototype injector 0.5 ms after the start of injection increased by 23% compared to that of the mass-production injector. However, in the reverse of the above-noted spray penetration increase, the spray angle

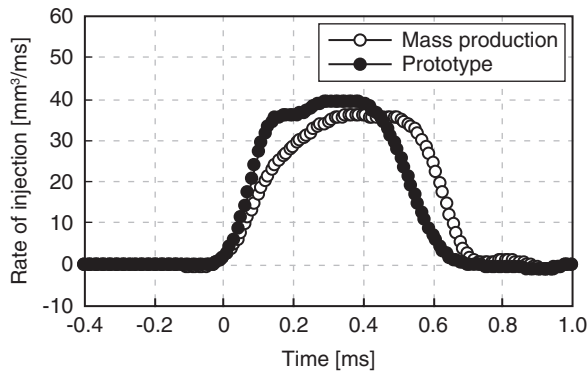


Fig. 4 Comparison of fuel injection rate profiles (1/4 engine load condition)

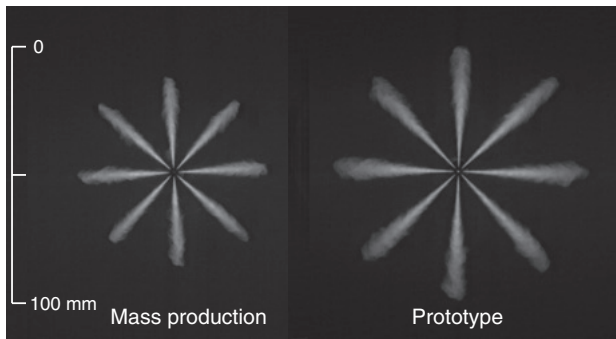


Fig. 5 Photographs of spray using high pressure vessel (0.5 ms after start of injection)

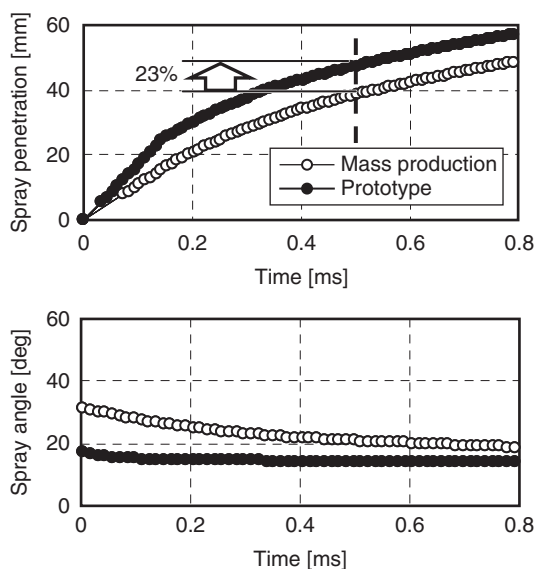


Fig. 6 Comparison of spray characteristics of two types of injector at 1/4 engine load

was confirmed to become narrower. This indicates that increasing the fuel injection rate has a large influence on the spray shape, and the injected fuel takes on a narrower spray shape with longer penetration.

In single-cylinder engine combustion tests using the prototype injector that was confirmed to have high-speed injection characteristics, the trade-off relationship between NO_x and soot was enhanced. Soot reduction effects of 35.2% at 1/4 engine load equivalent and 48.4% at 1/2 engine load equivalent were confirmed at the NO_x emission point that coincides with the setting point of the base four-cylinder engine. This is thought to indicate that increasing the fuel injection rate increased the spray penetration, which lowered the fuel concentration (lowered the richness) of the air-fuel mixture in the spray at the ignition timing.

3. Spray and Combustion Experiments

3.1. Injector Nozzles

Table 3 shows the specifications of the three nozzle types used in this study. The nozzle hole length was 1.05 mm for all three types. Only the nozzle hole diameter and number of holes differ, and all other parameters such as the hydraulic flow rate are the same. The nozzle sac shape including the needle was changed from the nozzle used in the previous chapter in order to machine 19 holes. This was done to secure sufficient distance between holes due to manufacturing technology constraints, and the sac volume increased by approximately three times compared to the prototype model used in the previous study. This is a factor increasing HC compared to a prototype nozzle, which is an issue. Note that the 0.08 mm nozzle hole machining is nearly the limit when using a normal electric discharge machine.

3.2. Tested Fuel

Table 4 shows the specifications of the fuel used for measurement. Diesel fuel with a cetane index of 43.2 was used in consideration of the U.S. and other countries where fuel with a relatively low cetane index is distributed in the market.

3.3. Spray Measurement Test

The spray characteristics test used a constant volume vessel and a high-speed video camera (FASTCAM-APX RS) made by Photron Limited. Figure 7 shows an overview of the measurement apparatus. A manual focus lens (Nikkor F f/1.2) with a focal length of 50 mm made by Nikon Corporation was mounted on the video camera, and recording was performed under conditions of a recording speed of 10,000 fps, image resolution of 512 × 512 pixels, and exposure time of 2 μs as shown in Table 5. The test was performed in the condition with N₂ gas charged to 1.4 MPa inside the constant volume vessel to achieve the same ambient density as that at the timing when fuel is injected during the engine performance test. The test

Table 3 Specifications of injector nozzles

	Nozzle A	Nozzle B	Nozzle C
Hole diameter (mm)	0.12	0.10	0.08
Number of holes	8	12	19
Spray hole angle (deg)	160	160	160
Flow rate (cm ³ /min)	740	740	740
Sac volume (mm ³)	1.23	1.23	1.23
Seat angle (deg)	60	60	60

Table 4 Measured properties of diesel fuel

Item	Test method	
Cetane index	ASTM D-976	43.2
Density (g/cm ³)	ASTM D-4052	0.852
Viscosity (mm ² /s)	ASTM D-445	2.2
T90 (°C)	ASTM D-86	306
Aromatic (vol/%)	ASTM D-1319	28.4
LHV (kJ/kg)	ASTM D-3338	42838

Table 5 Conditions of spray measurement experiment

Recording speed (fps)	10000
Shutter speed (μs)	2
Resolution (pixel)	512 × 512
Ambient gas	N ₂
Ambient pressure (MPa)	1.4
Ambient temperature (K)	298

consisted of injecting fuel to the inside of the constant volume vessel under conditions of a fuel injection quantity of 18 mm³/st. and injection pressure of 100 MPa, recording the spray with the high-speed video camera, and obtaining the spray penetration and spray angle from the resulting images.

3.4. Droplet Size Measurement Test

The droplet size of each nozzle was measured using a Phase Doppler Anemometry (PDA) system made by Dantec Dynamics A/S in order to compare the fuel spray atomization characteristics for different hole diameters. Figure 8 shows an overview of this test apparatus. The density, temperature and other ambient conditions inside

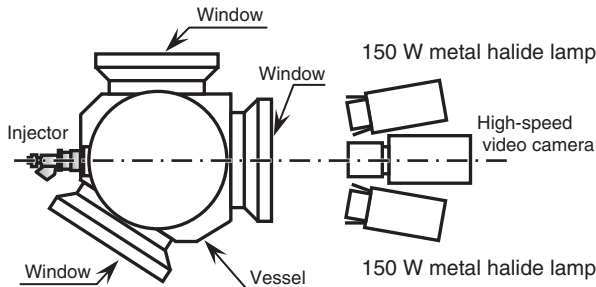


Fig. 7 Schematic diagram of spray measurement

Table 6 Specifications of single-cylinder engine

Engine type	4-st. single cylinder
Displacement volume (cm ³)	550
Bore × Stroke (mm)	85.0 × 97.1
Compression ratio	17.2
Fuel injection system	Piezo common rail
EGR rate	Continuously variable

the constant volume vessel were the same as those when measuring the spray, and measurement was performed at the position 60 mm downstream from the nozzle hole position.

3.5. Combustion Test Using a Single-cylinder Engine

Table 6 shows the specifications of the single-cylinder engine. This single-cylinder engine was manufactured based on a four-cylinder diesel engine that complies with the European emission regulation EURO 5⁽¹³⁾. Figure 9 shows the overall configuration of this single-cylinder engine, and Fig. 10 shows the shape of the engine combustion chamber. The intake is supercharged using a supercharger and the intake temperature is controlled to the designated temperature using a heater and intercooler

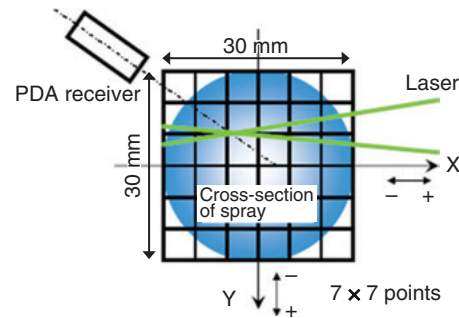


Fig. 8 Outline of droplet size measurement using PDA

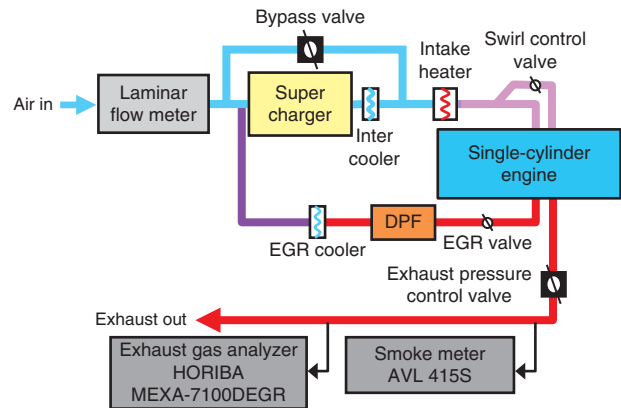


Fig. 9 Schematic diagram of single-cylinder engine system

Table 7 Conditions of engine combustion experiment

Nozzle hole diameter (mm)	0.12		0.10		0.08	
Engine speed (rpm)	1500		1500		1500	
Load	1/4	1/2	1/4	1/2	1/4	1/2
Injection pressure (MPa)	100	110	100	110	100	110
Intake air temperature (K)	343	333	343	333	343	333
Cooling water temperature (K)	353	353	353	353	353	353
Total injection quantity (mm ³ /st.)	18	32	18	32	18	32
Pilot injection timing (deg ATDC)	-8.0	-16.6	-8.5	-16.5	-8.7	-17.6
Main injection timing (deg ATDC)	5.8	4.0	6.0	4.1	6.5	4.0

as shown in Fig. 9. In addition, the exhaust is controlled to the set exhaust pressure using a pressure control valve. This configuration enables the single-cylinder engine system to reproduce the actual operating conditions in a four-cylinder engine. An exhaust gas analyzer (MEXA-7100DEGR) made by Horiba, Ltd. and a smoke meter (415S) made by AVL LIST GmbH were installed downstream from the above-noted exhaust pressure control valve to analyze the various exhaust gas components.

The combustion test was performed under conditions corresponding to an engine speed of 1,500 rpm and 1/4 and 1/2 engine loads, which have a high frequency of use in car mode tests. Table 7 shows the engine operating

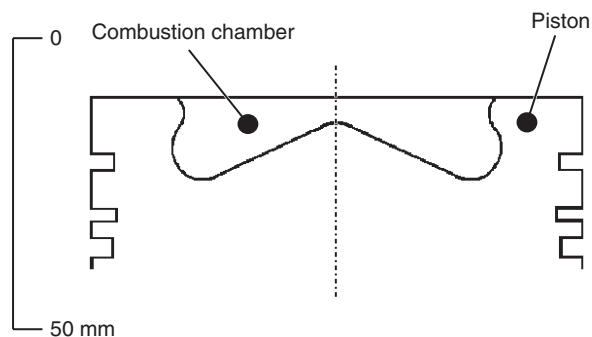


Fig. 10 Combustion chamber shape

conditions used for this test. The injection pattern used the two-stage pilot and main injection that is the standard injection method in conventional diesel combustion. The pilot injection quantity and timing were adjusted to achieve the same heat release value so that the $dP/d\theta_{max}$ used as the index of diesel engine combustion noise would be the same. In addition, the main injection quantity and timing were adjusted to align the main ignition timings. Note that increasing the fuel injection pressure increases the combustion noise and mechanical loss, so this variable was excluded from this study, and the tests using the three nozzles were performed with the same fuel injection pressure.

4. Results and Discussion

4.1. Spray Measurement Test Results

Figure 11 shows photographs of the spray taken with a high-speed camera, and Fig. 12 shows time series numerical representations of the spray penetration and

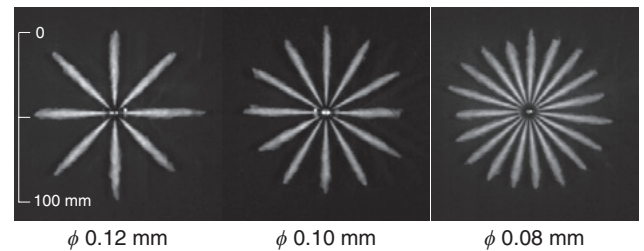


Fig. 11 Photographs of spray with three types of nozzles (0.5 ms after start of injection)

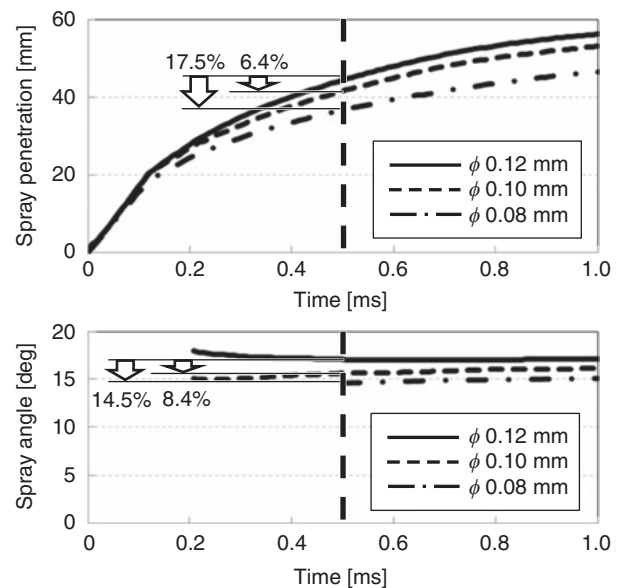


Fig. 12 Numerical comparison of spray penetration and spray angle

the spray angle. This measurement test was performed 10 times, and the spray data of all holes was averaged. The spray angle was measured at the position 30 mm downstream from the nozzle outlet in the axial direction of the spray.

Figure 12 shows that the spray penetration decreases and the spray angle becomes narrower as the nozzle hole diameter becomes smaller. The 0.08 mm nozzle, which has the smallest hole diameter, exhibited 17.5% less spray penetration and a 14.5% narrower spray angle compared to the 0.12 mm nozzle at the point 0.5 ms after the start of injection. Note that measurement of the spray angle at the start of injection has large variance, so the data up to approximately 0.2 ms after the start of injection was excluded.

4.2. Droplet Size Measurement Test Results

Figure 13 shows the droplet size distribution measurement results and Fig. 14 shows the droplet Sauter mean diameter (d_{32}) for the three nozzle types. The droplet surface area is an important factor influencing droplet

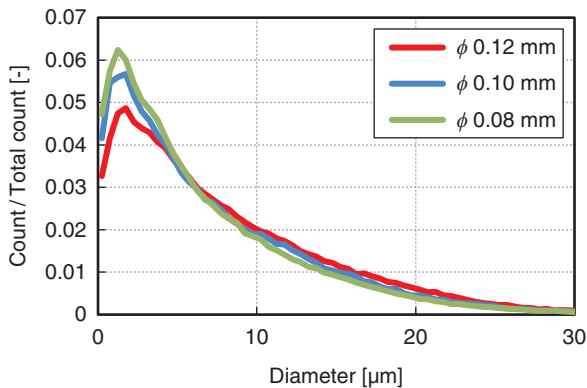


Fig. 13 Comparison of droplet size distribution with three types of nozzles

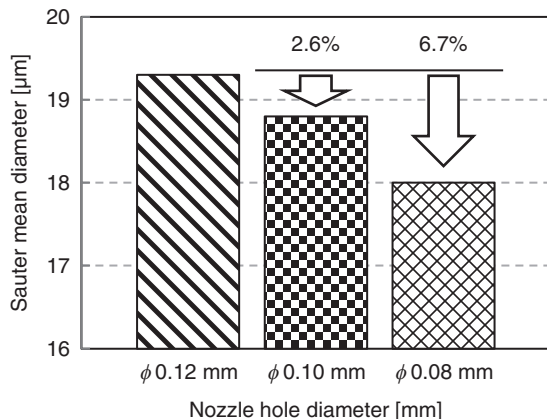


Fig. 14 Comparison of droplet Sauter mean diameter with three types of nozzles

vaporization and volatilization, so the droplet Sauter mean diameter, which takes into account the surface area, was used as the average droplet size. Figure 13 shows that the droplet size distribution shifts toward the smaller droplet size side as the nozzle hole diameter becomes smaller. The droplet Sauter mean diameter was confirmed to decrease as the nozzle hole diameter becomes smaller. The droplet Sauter mean diameter decreased by 2.6% when the hole diameter was reduced from 0.12 mm to 0.10 mm, and by 6.7% when the hole diameter was reduced to 0.08 mm. This confirmed that reducing the nozzle hole diameter increased fuel atomization.

4.3. Engine Combustion Test Results

Figure 15 (1/4 engine load; fuel injection quantity of 18 mm³/st.) and Fig. 16 (1/2 engine load; fuel injection quantity of 32 mm³/st.) show the results of combustion tests using the single-cylinder engine. These experiments were performed by gradually opening the EGR valve from the fully closed condition to increase the EGR rate. The figures show NOx on the horizontal axis, and each measurement data on the vertical axis. Comparison of each measurement data at the NOx value (the position of the gray lines in Fig. 15 and Fig. 16) that coincides with the matching point of the mass-production engine shows that in the 1/4 engine

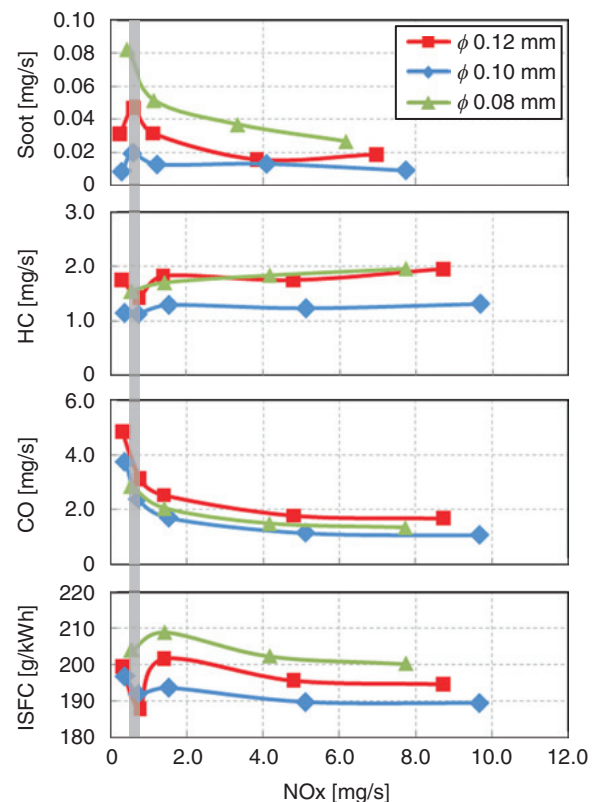


Fig. 15 Exhaust gas emissions at 1500 rpm, 1/4 engine load using three types of micro hole nozzles

load condition, the 0.10 mm nozzle exhibited the lowest soot emission quantity, and soot emission increases for the 0.08 mm nozzle, which has an even smaller nozzle hole diameter. However, in the 1/2 engine load condition, the 0.12 mm nozzle exhibited the lowest soot emission, and the 0.08 mm nozzle exhibited the highest soot emission, approximately four times that of the 0.12 mm nozzle, and also exhibited increased HC, CO and Indicated specific fuel consumption (ISFC).

Figure 17 shows the rate of heat release (ROHR) at the NO_x value (the position of the gray line in Fig. 15) that

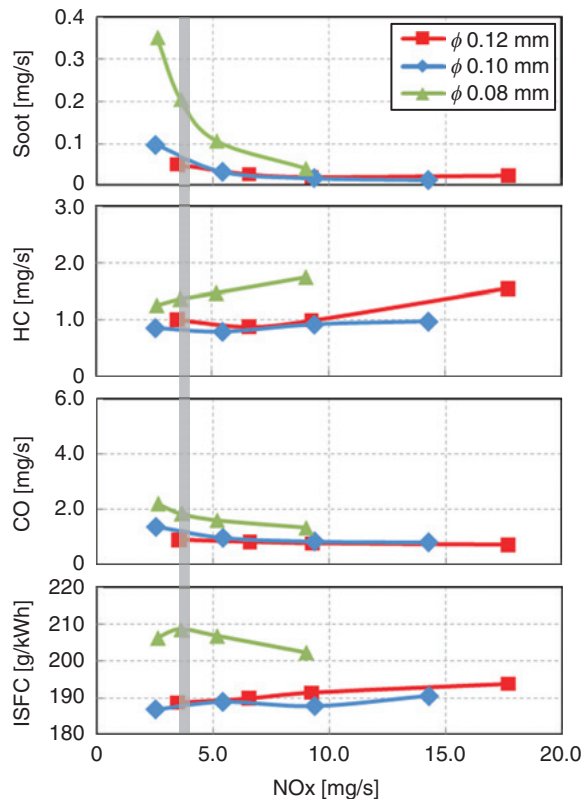


Fig. 16 Exhaust gas emissions at 1500 rpm, 1/2 engine load using three types of micro hole nozzles

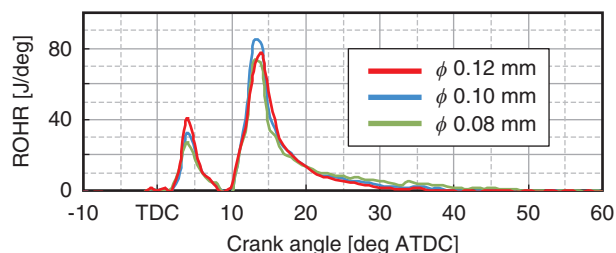


Fig. 17 Comparison of heat release at 1500 rpm, 1/4 engine load using three types of micro orifice nozzles

coincides with the matching point of the mass-production engine under conditions of 1/4 engine load and a fuel injection quantity of 18 mm³/st. According to this ROHR, the heat release peak of the main combustion, which is the dominant factor influencing exhaust emission performance, is largest for the 0.10 mm nozzle and smallest for the 0.08 mm nozzle, which has the smallest hole diameter. In addition, the delayed combustion was confirmed for the 0.08 mm nozzle, with combustion continuing in the latter part of the combustion period from an ATDC of 30° onward.

4.4. Discussion

The spray measurement tests confirmed that reducing the nozzle hole diameter reduces the fuel spray penetration, narrows the spray angle, and reduces the injected fuel droplet size, that is to say, promotes fuel atomization.

The engine combustion test results showed that the soot emission quantity at both 1/4 and 1/2 engine loads was largest during combustion using the 0.08 mm nozzle that was confirmed to increase fuel atomization performance. Based on these results, the smaller hole diameter is presumed to have increased the richness of the air-fuel mixture distribution within the fuel spray. The reason for this is thought to be that the smaller hole diameter resulted in a spray shape with reduced spray penetration and a narrow spray angle. A smaller nozzle hole diameter promotes fuel atomization within the spray, but also reduces spray penetration, reduces the mixture of fuel and air in the fuel spray at the start of injection, and reduces the ratio of air-fuel mixing that is thought to occur due to collision between the spray and the combustion chamber wall. This means that the injected fuel is present in a narrow range inside the combustion chamber, and combustion of this rich spray is thought to produce an increase in soot and HC. The delayed combustion of the 0.08 mm nozzle confirmed by heat release analysis also confirms an increase in the richness of the air-fuel mixture distribution within the fuel spray.

The 0.10 mm nozzle exhibited the smallest soot emission quantity at 1/4 engine load, but the fuel injection quantity was less than that at 1/2 engine load, so both the injection time and the spray penetration were shorter than those at 1/2 engine load. Therefore, the reason why the 0.10 mm nozzle exhibited the smallest soot emission quantity is thought to be that the air-fuel mixing promotion effect of increased atomization due to reducing the hole diameter had a larger influence on combustion than the air inclusion effect of increased spray penetration. In addition, the peak ROHR of the main combustion, which is the dominant factor influencing exhaust emission performance, was also highest for the 0.10 mm nozzle, which confirmed that early combustion is promoted by the atomization effect.

Based on the above, although reducing the nozzle hole diameter under conditions of the same injection pressure promotes fuel atomization, the spray penetration drops

and the richness of the air-fuel mixture within the spray increases, including the influence of wall collision, so there is thought to be an appropriate balance for reducing exhaust emissions. In addition, this balance is thought to change according to the injected fuel quantity and injection pressure, so it is necessary to determine the engine operating conditions to be given importance and set the nozzle hole diameter in consideration of this balance.

Regarding the influence of spray interference between the fuel injected into the engine combustion chamber due to increasing the number of nozzle holes to 19, interference between the spray from each hole of the 0.08 mm nozzle could not be confirmed in tests inside the constant volume vessel shown in Fig. 11, so the likelihood of spray interference is low.

5. Conclusion

Various experiments were performed using a prototype injector with high-speed injection performance that was confirmed to have exhaust emission reduction effects. These experiments used three types of nozzles with the same hydraulic flow rate but different nozzle hole diameters from 0.12 mm to 0.08 mm with the aim of further reducing exhaust emissions, and yielded the following findings.

- (1) Reduction of the nozzle hole diameter resulted in a narrow injected fuel spray shape with low penetration.
- (2) Reduction of the hole diameter reduced the fuel spray droplet size. The droplet Sauter mean diameter of the 0.08 mm nozzle was 6.7% smaller than that of the 0.12 mm nozzle.
- (3) The results of combustion experiments using a single-cylinder engine at medium and low engine loads confirmed an increase in exhaust emissions such as soot during combustion using the 0.08 mm nozzle as compared to the 0.12 mm and 0.10 mm nozzles. The increased soot emission of the 0.08 mm nozzle occurred because the smaller nozzle hole diameter resulted in lower spray penetration, which reduced mixture of the fuel with the air and increased the richness of the air-fuel mixture distribution within the fuel spray.
- (4) In combustion under low engine load conditions, the fuel atomization effect of a smaller nozzle hole diameter influences combustion, and the soot emission quantity of the 0.10 mm nozzle is smaller compared to the combustion results for the 0.12 mm and 0.08 mm nozzles. When the engine load is reduced, the injection time becomes shorter, so the spray penetration becomes shorter as well. However, the influence on combustion of atomization due to a reduced hole diameter exceeds the influence of reduced spray penetration, so the 0.10 mm nozzle exhibited the smallest soot emission quantity.

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