

Research Paper

Effects of spray pattern and piston shape on in-cylinder flow in a two-cylinder spray-guided gasoline direct-injection optical engine

Donghwan Kim^a, Jisoo Shin^a, Jisoo Kim^a, Junkyu Park^b, Yousang Son^b, Sungwook Park^{c,*}^a Department of Mechanical Convergence Engineering, Graduate School of Hanyang University, Seoul 04763, Republic of Korea^b Hyundai Motor Company, Republic of Korea^c School of Mechanical Engineering, Hanyang University, Seoul 04763, Republic of Korea

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ABSTRACT

In gasoline direct-injection engines, in-cylinder flow plays an important role in terms of fuel atomization, mixture, turbulence, combustion, and emission. The in-cylinder flow is affected by the air–fuel interaction and piston shapes, and it is required to understand and optimize these influences. In this study, visualization of in-cylinder flow was performed depending on the piston shape and spray pattern using particle image velocimetry measurements in a two-cylinder optical engine with high-compression ratio and spray-guided direct-injection. Effects of these parameters were evaluated quantitatively by calculating average flow speed, average tumble ratio, and average turbulent kinetic energy. The mixture was confirmed using numerical simulation. The piston shape had no significant effect on the structure and strength of the in-cylinder flow. The 7-hole injector with asymmetrical spray pattern was more effective to enhance the turbulence than the 10-hole injector with symmetrical spray pattern. As the injection timing was delayed up to before top dead center 120°, the average turbulent kinetic energy after fuel injection increased by up to 574 % in the 7-hole injector. However, considering stratification and possible knock region, ideal mixtures were found in the injection timing range of before top dead center 270° to 210° in both injectors.

1. Introduction

Pure electric vehicles (PEVs) [1] and hybrid electric vehicles (HEVs) [2,3] has been rapidly developed due to the low exhaust pollutants. The exhaust gas and fuel efficiency regulation for the internal combustion engines are being made more stringent [4]. However, it is hard to exclude the internal combustion engines from transportation because PEVs and HEVs have some problems. For example, the battery life, the battery weight, limited mileage range, and lack of charging stations can be problems [5]. Many countries are considering an assessment of the life cycle of vehicles across production, manufacturing, driving, and disposal to objectively evaluate the greenhouse gases generated by vehicles [6]. However, it is hard to satisfy future strict emission regulations using the spark ignitions (SI) engines with conventional combustion mode which operated under premixed and stoichiometric conditions. Therefore, new combustion strategies such as spray-guided stratified charge (SGSC) [7], homogeneous charge compression ignition (HCCI) [8,9], and spark-controlled compression ignition (SPCCI) [10] have been studied in gasoline direct injection (GDI) engines with high

compression ratio. In general, gasoline center injection is utilized because it has some advantages in terms of air–fuel mixing performance, wall-wetting, combustion stability, potential for stratified combustion, and possibility of various injection strategies application. Therefore, in SGSC and SPCCI combustion concept, the lean burn combustion strategy with gasoline center injection is being applied to improve fuel efficiency.

GDI engines have a relatively short time to form air–fuel mixture in the cylinder before ignition timing, resulting in the formation of a locally rich mixture. The mixture characteristics are dominantly influenced by design and control parameter such as piston geometry and injection strategy. When GDI engines fail to form an appropriate mixture for the purpose of combustion strategies due to improper injection strategies, the stratifications of the temperature distribution in the cylinder becomes sever due to the non-uniform concentration. Furthermore, the risk of NOx emission can increase owing to a local high-temperature region, and the thinner half of the stratified mixture's outer edge is prone to flame extinguishment [11]. In addition, exhaust pollutants can increase owing to the wall-wetting caused by improper injection strategies at cylinder liner and piston head [12,13]. Therefore, it is necessary to optimize the in-cylinder flow, spray development, and mixture

* Corresponding author at: School of Mechanical Engineering, Hanyang University, 222 Wangsimni-ro, Seongdong-gu, Seoul 04763, Republic of Korea.
E-mail address: parks@hanyang.ac.kr (S. Park).

Nomenclature

c	Number of cycles
i	x -coordinate
j	y -coordinate
k	Individual cycle
$\vec{r}_{i,j}$	Position vector at (i,j)
$\vec{r}_{t,c}$	Position vector at tumble center
S_{avg}	Average flow speed
TKE_{avg}	Average turbulent kinetic energy
TR_{avg}	Average tumble ratio
u	Velocity component in the x -coordinate direction
u'	Turbulent velocity component in the x -coordinate direction
$\bar{u}$	Average velocity component in the x -coordinate direction
v	Velocity component in the y -coordinate direction
v'	Turbulent velocity component in the y -coordinate direction
$\bar{v}$	Average velocity component in the y -coordinate direction
$\vec{V}$	Velocity vector
ω	Engine angular velocity

Abbreviations

ASOE After Start of Energizing

BP	Bowl Piston
BTDC	Before Top Dead Center
CFD	Computational Fluid Dynamics
CW	Continuous-wave
DBI	Diffuse Back-Illumination
GDI	Gasoline Direct Injection
HCCI	Homogeneous Charge Compression Ignition
HEVs	Hybrid Electric Vehicles
ICP	Inclined and Centralized Piston
IOP	Inclined and Open Piston
LIF	Laser Induced Fluorescence
MS	Moment Summation
PCX	Planar Convex
PCV	Planar Concave
PEVs	Pure Electric Vehicles
PIV	Particle Image Velocimetry
SGSC	Spray-Guided Stratified Charge
SI	Spark Ignition
SPCCI	Spark-Controlled Compression Ignition
TC	Tumble Center
TDC	Top Dead Center
TKE	Turbulent Kinetic Energy
TR	Tumble Ratio

characteristics in the GDI engines for design and control parameter.

Tumble intensity is one of the important factors representing in-cylinder flow characteristic in GDI engines because combustion stability and efficiency can be achieved through increasing initial flame propagation speed [14,15]. High tumble intensity can increase the velocity and turbulent kinetic energy (TKE) of flow between spark plug gap, resulting in rapid initial flame kernel [16,17]. However, not only the turbulence intensity around the spark plug, but also the equivalence ratio distribution, flow direction, and spark energy affect ignition stability. These factors that affect early flame propagation are also important for lean burn combustion in the GDI engines [18]. This is because turbulence has an effect in relation to the formation of wrinkles in the

flame front. Therefore, various factors must be considered for a good combustion. Especially, in-cylinder flow and mixture formation characteristics around the spark plug are decided by relationship between the spray and air during the intake and compression strokes, and engine design. In short, injection strategies (e.g., injection timing and spray pattern etc.) and shape of combustion chamber (e.g., piston and pent-roof etc.).

Many previous studies investigated effects of piston shape and injection strategies such as spray pattern on characteristics of mixture formation and in-cylinder flow. Harshavardhan et al. [19] investigated in-cylinder flow and air–fuel interaction caused by different piston shape using a computational fluid dynamics (CFD) in a four-stroke

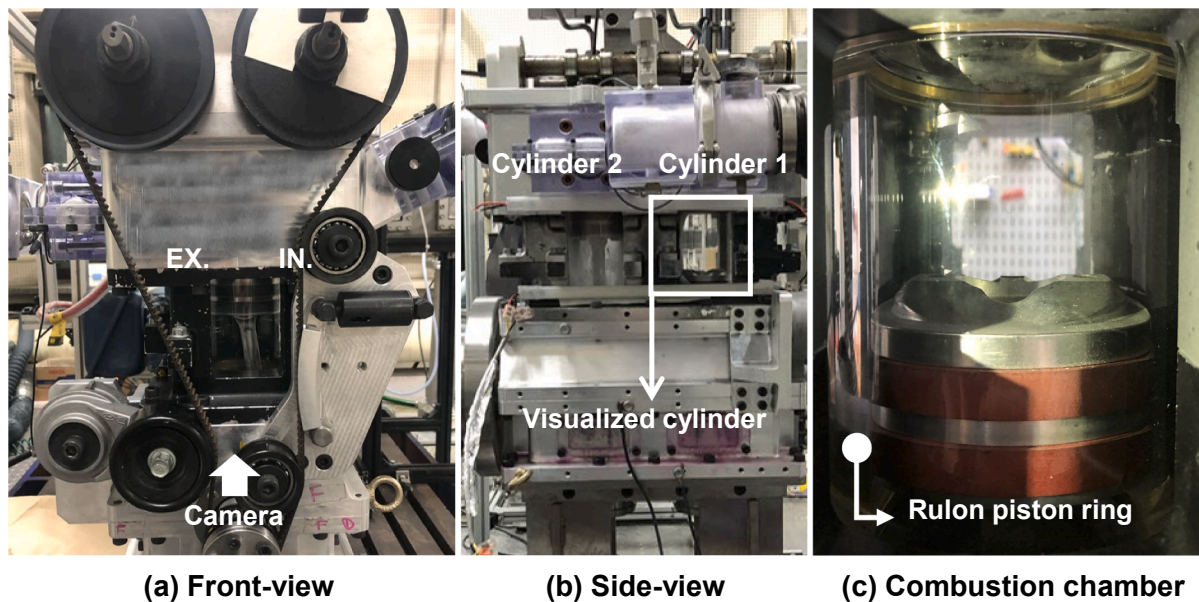


Fig. 1. Images of optical engine.

Table 1

Details of the optical engine.

Items	Details
Bore (mm)	75.6
Radius of crank (mm)	41.6
Stroke (mm)	83.38
Displacement volume of two cylinders (cc)	748.5
Length of conrod (mm)	147.8
Compression height (mm)	27.5
Height of block (mm)	217.0
Piston Topping (mm)	0.33
Crank offset (mm)	10
Piston pin offset (mm)	0.7
Stroke/Bore ratio (-)	1.10
Compression ratio (-)	14

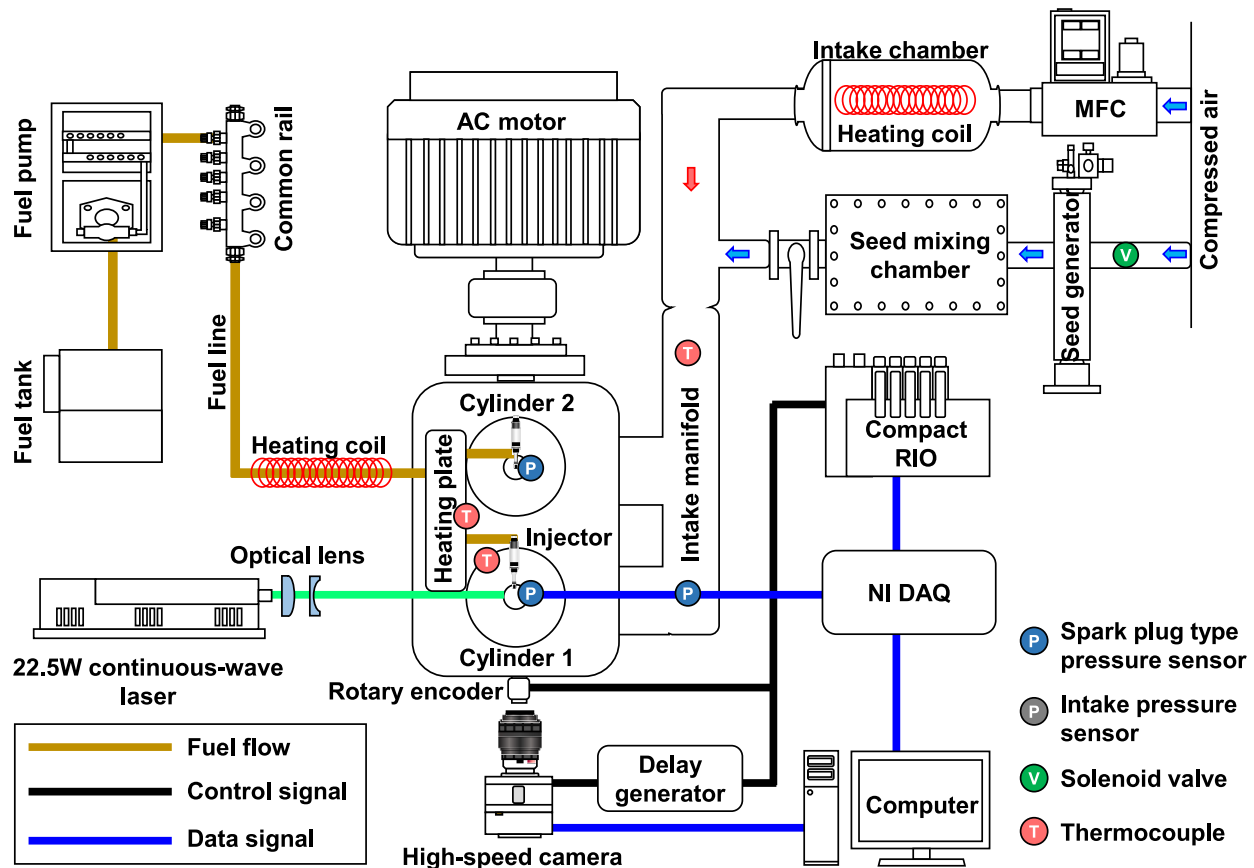
Table 2

Details of the experimental instruments.

Items	Details
Sensys PTD	Intake pressure (± 0.5 %FS)
Sensys PSH	Exhaust pressure (± 0.3 %FS)
LEO2	Injection pressure (± 0.1 %FS)
Kistler 6056A	In-cylinder pressure (± 0.3 %FS)
HFC-D-303B	Intake air flow rate (± 0.2 %FS)
USB 6341 (NI DAQ)	Data acquisition (residual gain error: 0.0072 %, residual offset error: 0.0013 %)
E50S8-360-3-T-24 (encoder)	Crank angle (max. response frequency: 300 kHz)
NI 9751 (NI module)	Direct injection injector control
NI 9757 (NI module)	Lambda
NI 9411 (NI module)	Crank angle

engine with compression ratio of 10 under the engine speed of 1,500 rpm. They found that higher tumble ratio and turbulent kinetic energy of in-cylinder flow were occurred in flat-with-center-bowl piston as compared to flat piston. This was because the piston shape affected the in-cylinder flow development process. Wang et al. [20] investigated characteristics of fuel stratification under different piston shapes and injection strategies based on the numerical analysis. They found that shallow bowl shape on the piston improved the velocity around spark plug, and late injection could generate a clear tumble flow near spark plug. Murali Krishna et al. [21] studied effects of piston shapes on tumble flow by using particle image velocimetry (PIV) measurements. They found that the pentroof-offset-bowl piston is useful to increase tumble intensity and average TKE at time of spark for SI engines. These studies suggested that a bowl-type piston, rather than a flat-type piston, was effective in enhancing the in-cylinder flow.

Geschwindner et al. [22] investigated flow and spray G characteristics in two optical engines. The combination of spray imaging using diffuse back-illumination (DBI) and Mie-scattering, and PIV imaging techniques were used to investigate the relation between the flow and spray. They found that the flow was strongly disturbed by fuel injection. If the tumble flow was weak, it was difficult to maintain its initial high momentum. Therefore, air entrainment during injection caused two counter-rotating vortices. As a result, initial flow structure show be maintained after fuel injection to keep the initial high momentum of in-cylinder flow. Regarding injection timing, they emphasized the need for an extended analysis of other ambient pressure conditions that are affected by injection timing. Wang et al. [23] investigated optimization of an injector spray pattern in a turbocharged GDI engine using laser induced fluorescence (LIF). Moreover, piston crown and glass liner



* AC: Alternating Current, MFC: Mass Flow Controller, I DAQ: National Instruments Data Acquisition System

Fig. 2. Schematic of optical engine system.

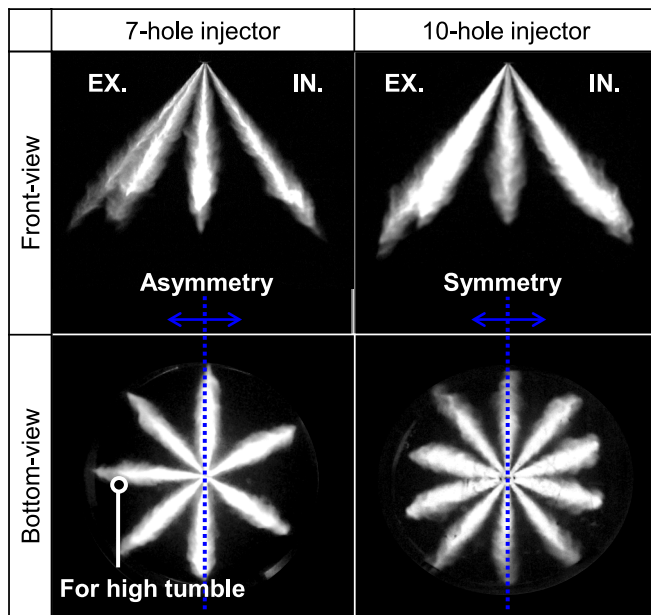


Fig. 3. Spray structure of test injectors in front-view and bottom-view.

images were used to evaluate fuel impingement. They found that optimized injector pattern can increase ignition and combustion stability for warm-up operation. The selected injector can also assure good air–fuel mixing and combustion, and less soot tendency.

It was clear that the piston shape and spray pattern affected the in-cylinder flow. In addition, the flow intensity and mixture formation are affected by the interaction between the spray and the in-cylinder flow, and a wide range of injection timing must be considered to derive the optimal injection strategy. Furthermore, since the application of a high compression ratio is being considered to improve combustion characteristics in GDI engines, it is important to reveal the correlation between spray and in-cylinder flow according to the piston shape and spray pattern at a higher compression ratio than the conventional compression ratio of 9 to 10. In addition, since one of the purposes of the applying a high compression ratio is improving efficiency though combustion in lean conditions [24], a strategy for mixture formation for combustion stability in lean condition must be prepared.

Therefore, in this study, two-cylinder optical GDI engine with spray-guided injection and high compression ratio of 14 was used to analyze

Table 3
Specifications of PIV system.

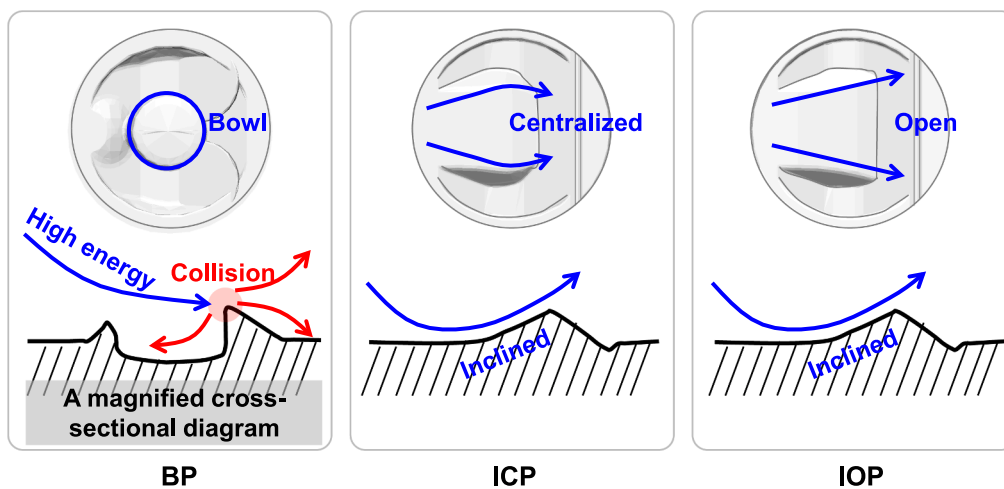
Items	Details
Camera	High-speed camera (VEO 710L, Phantom)
Laser	Continuous-wave laser (532 nm wavelength, 22.5 W)
Region of interest	65 × 100 mm
Resolution	320 × 344 pixels
Seed	Hollow sphere glass Diameter: 18 μm Density: 0.6 g/ml
PIV image time interval (dt)	20 μs
Camera frame rate	50,000 fps
Exposure time	19 μs
Interrogation area	Pass 1: 32 × 32 pixels (6.5 × 6.5 mm, 50 % overlap) Pass 2: 16 × 16 pixels (3.25 × 3.25 mm, 50 % overlap)
Correlation algorithm	Fast Fourier Transform

the correlation among piston shapes, spray patterns and in-cylinder flow through the visualization of combustion chamber. Three types of piston bowl geometries and spray patterns of asymmetric and symmetric cases were considered. The in-cylinder flow characteristics before and after injection were analyzed using PIV in a wide range of injection timings for injection quantity corresponding to λ 1.5. In addition, using Mie-scattering imaging, the relationship between the spray development and the in-cylinder flow which varied for the parameters, was investigated. To compensate for design limitations of the optical engine, the pent-roof region was investigated by using a CFD. Also, the mixture preparation was evaluated for different parameters based on CFD simulation results.

2. Methodology

2.1. Optical engine system

In this study, the two-cylinder optical engine was used to investigate effects of spray pattern and piston shape on flow characteristics. Detailed description can be found in the previous studies [25–27]. Front-view and side-view of optical engine are shown in Fig. 1(a) and 1(b). To reflect the effects of cylinder-to-cylinder, two-cylinder was applied as shown in Fig. 1(b). However, the experiment was only conducted at cylinder 1. The entire combustion chamber except for the pent-roof region was made as transparent quartz to visualize the in-cylinder flow (Fig. 1(c)). The optical engine had a bore of 75.6 mm and a stroke of 83.38 mm. A displacement volume of two cylinders was about 750 cc. A Rulon was used as a material of piston ring to reduce the friction force



* BP: Bowled Piston, ICP: Inclined and Centralized Piston, IOP: Inclined and Open Piston

Fig. 4. Three types of piston shapes.

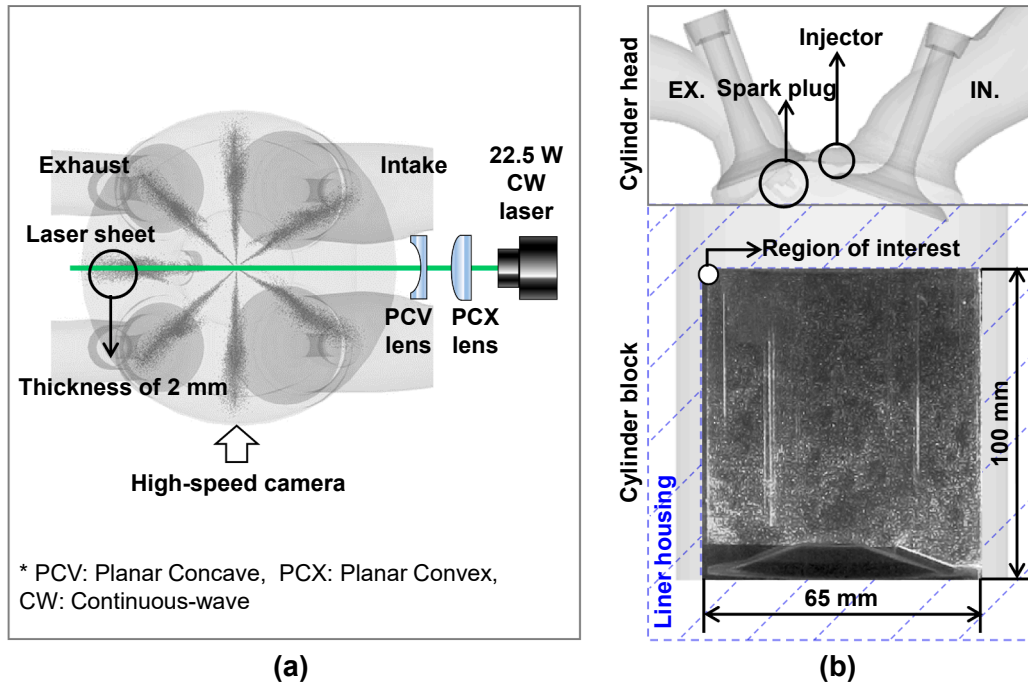


Fig. 5. Configuration of PIV system: (a) optical system, and (b) field of view.

Table 4
Experimental conditions.

Items	Details
Piston shape	BP, ICP, IOP
Test injectors	7-hole injector (asymmetry) 10-hole injector (symmetry)
Injection timing (BTDC°)	300, 270, 240, 210, 180, 150, 120
Intake air temperature (°C)	30
Fuel rail temperature (°C)	90
Fuel	gasoline
Lambda	1.5
Fuel mass (mg)	18.8
Injection duration (ms)	1.1 (7-hole injector), 0.9 (10-hle injector)
Engine speed (rpm)	1,500
Intake pressure (bar)	1.0 (wide-open throttle)
Injection pressure (bar)	700

*BP: Bowl Piston, ICP: Inclined and Centralized Piston, IOP: Inclined and Open Piston.

between a piston and cylinder. Therefore, the optical engine could be operated stable without breakage of quartz even though there was no oil supply into the combustion chamber. The detailed specifications of the optical engine are shown in Table 1.

A fuel pump and common rail were used to realize the high-pressure injection system. The fuel was heated by winding a heating coil around the fuel line of about 1,000 mm. Moreover, the fuel rail with temperature sensor and cartridge heater was installed on cylinder head to maintain the fuel temperature. Intake pressure was maintained by using mass flow controller and heating intake camber. The AC motor with 37 kW was used to operate the optical engine. The test engine could be controlled by NI compact RIO system (National Instruments, USA). Injection timing, duration, and multiple injection were also controlled by compact RIO system. The details about sensors and NI modules which are used during experiments are tabulated in Table 2. A continuous wave laser with high-speed camera was used for PIV measurements. The detailed optical engine system is drawn in Fig. 2.

In this study, two injectors with different spray pattern as Fig. 3 were

used to investigate the effects of spray pattern on characteristics of in-cylinder flow. A 10-hole injector has symmetrical spray structure to form a stratified mixture efficiently, and a 7-hole injector has asymmetrical spray structure based on the horizontal line in bottom-view. 7-hole injector was made asymmetrically to enhance the tumble flow. Both the test injectors were of the solenoid and center-mounted type. In addition, each of these had a diameter of 100 μm and maximum operating injection pressure of 1,000 bar.

Fig. 4 shows the three types of piston head designs. The bowl piston (BP) was used as a base piston shape. In section 3.2, piston design was fixed as BP to evaluate effects of spray pattern. A bowl piston is commonly used for stratification. Even a stratification strategy is introduced, the tumble flow is still important. Because the tumble flow in GDI engines affects on air–fuel mixing performance, combustion stability. Therefore, to evaluate the levels of turbulence and tumble flow of the bowl piston, two pistons which are made for solely for the purpose of intensifying tumble flow without considering stratification were evaluated additionally. In three pistons with different shapes, the compression ratio was maintained as 14 to evaluate effects of piston shape on in-cylinder flow. The inclined and centralized piston (ICP) was designed to intensify the rotational in-cylinder flow along the top surface of the inclined piston. The inclined and open piston (IOP) was also designed for intensifying the tumble flow. Both the pistons were designed to reduce the disturbance of tumble flow. The difference between the two pistons is in the degree of clustering of the flow through the center of the piston's top surface.

2.2. PIV measurements

The tracking capability of particle was evaluated by calculating Stokes number. The Stokes number should be less than 1 for a particle to track a given flow. The Stokes number was calculated based on the following reference [28] and was tabulated in previous study [27]. The synchronization and timing errors were minimized using continuous-wave (CW) laser with high energy. The interrogation area was selected based on the maximum velocity region and crank angle (e.g. near intake valve open timing, the area located close to the intake valve

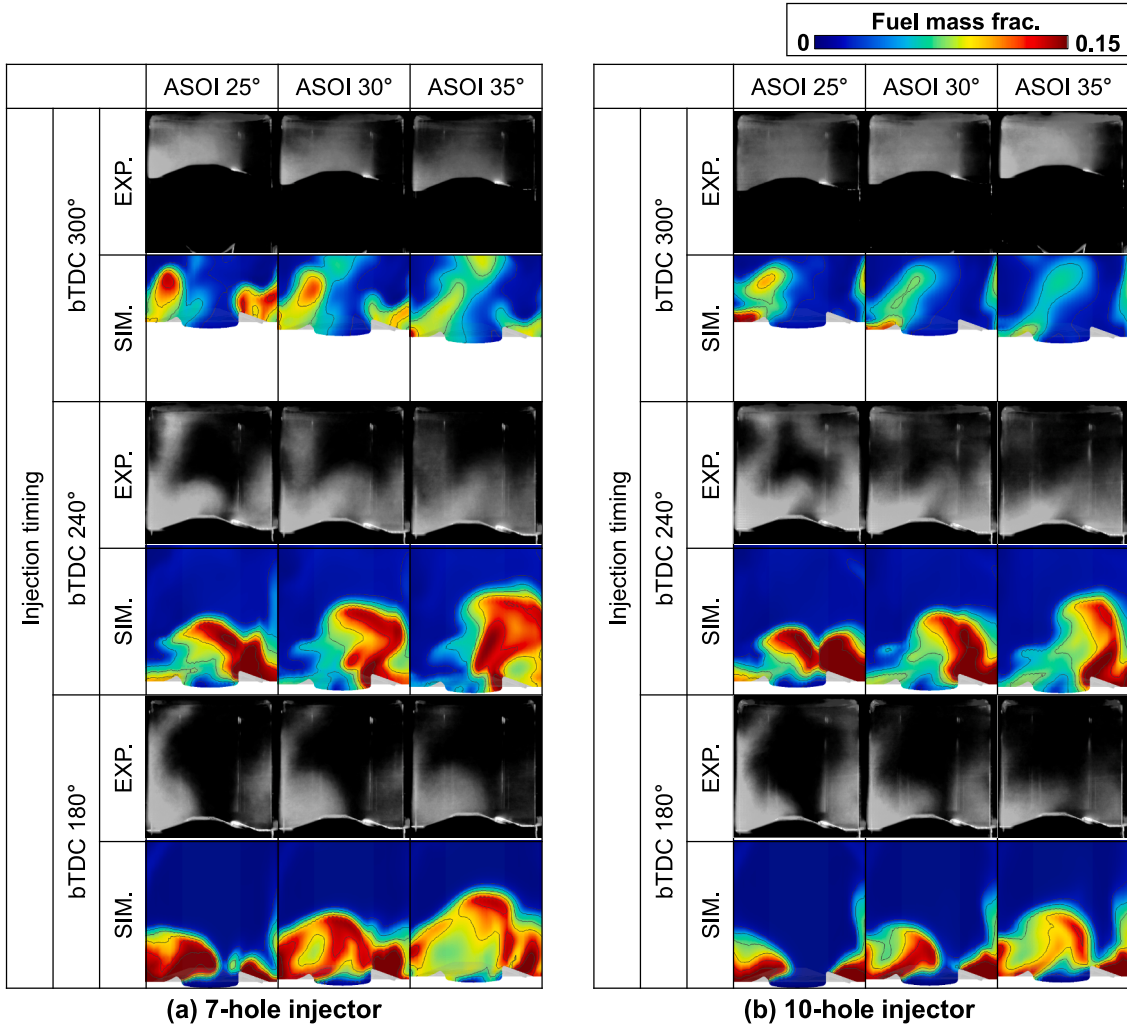


Fig. 6. Comparison of fuel distribution during engine operation between experiment and simulation: (a) 7-hole injector, (b) 10-hole injector.

in the two-dimensional image plane) through the trial-and-error. As a result, to satisfy the one-quarter rule [29], the first and second interrogation area was selected as 32×32 pixels and 16×16 pixels, respectively. The time interval between one pair of images was decided by the frame rate of the high-speed camera, so that the laser pulse triggering technique such a frame straddling did not be applied. The background noise was minimized by painting the back of quartz cylinder with the black. Errors caused by distortion of quartz surface was minimized by adapting a long distance between camera and vertical image plane, low magnification lens (35 mm prime lens), excluding edge of the quartz cylinder, high f-number of lens [30,31]. The tracer was used as a hollow sphere glass with a diameter of $18 \mu\text{m}$. It had a relatively large size compared to the silicone oil ($0.5\text{--}1.0 \mu\text{m}$) [32,33], but it had a low density. The detailed information of PIV system is given in Table 3.

Fig. 5 shows the configuration of PIV system. A laser sheet crossing the center of cylinder was generated, as shown in Fig. 5(a). The visualized region is depicted in Fig. 5(b). In this study, PIV measurements in pent-roof region were replaced by using CFD due to the design limitation. However, the pent-roof region was analyzed using CFD to compensate for design limitations of the optical engine. We used MATLAB software (PIVlab, MathWorks, USA) [34] to process raw images. Moreover, based on the velocity components, average tumble ratio (TR), average turbulent kinetic energy (TKE), and tumble center (TC) were derived. The in-cylinder flow was quantitatively evaluated depending on spray pattern and piston shape.

2.3. Quantitative indices

In this study, ensemble-mean flow fields, TKE distribution, average flow speed, average TR, average TKE and TC were mainly used to evaluate flow characteristics. Instantaneous images of 50 cycles were used for calculating ensemble-mean flow fields [26]. The details about calculating average flow speed, average TR, average TKE, and average TC are depicted in previous studies [25–27]. In an image plane, PIV measurements can provide velocity components, u with x -coordinate and v with y -coordinate, at a given location with i and j indices. Average of absolute magnitude of velocity, the spatially averaged speed, in an image plane can represent the momentum per unit density. And average flow speed can be derived following Eq. (1).

$$S_{avg} = \frac{1}{c} \sum_k \frac{1}{n \times m} \sum_i \sum_j \sqrt{u_{i,j,k}^2 + v_{i,j,k}^2} \quad (1)$$

where S_{avg} is average flow speed, c is number of total cycles, c is an individual cycle. And we tried to define the tumble center in the same method as the calculation process in the simulation. Therefore, the tumble center was defined following Eq. (2) [25]. The tumble center has the maximum moment summation (MS). The mean tumble center was calculated based on the ensemble-mean flow fields of 50 cycles.

$$MS = \sum_{i=1}^n \sum_{j=1}^m ((\vec{r}_{i,j} - \vec{r}_{r,c}) \times \vec{v}_{i,j}) \quad (2)$$

where (r, c) is a position in the image plane. And, in this study, the

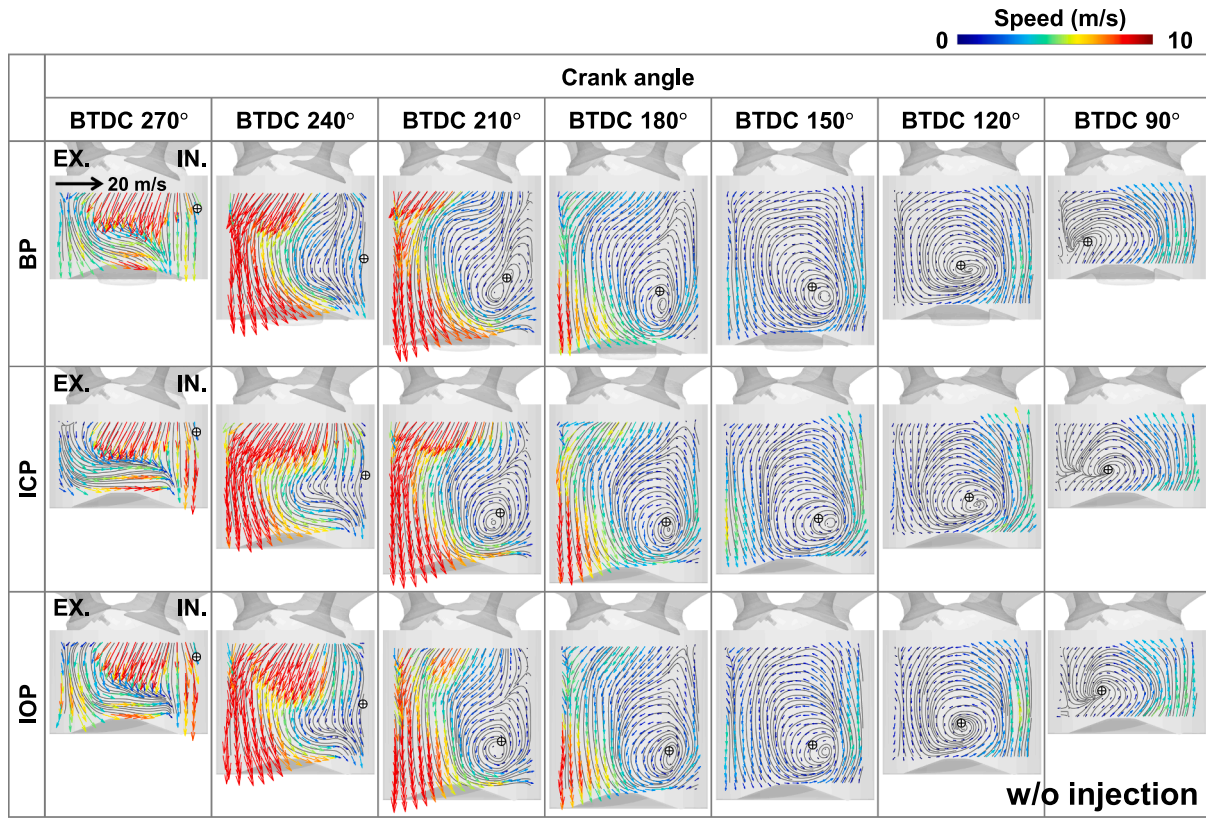


Fig. 7. Ensemble-mean flow field for piston shapes (@ w/o injection).

average TR was derived based on the TC. The average TR was defined by averaging TR of 50 cycles following Eq. (3).

$$\overline{TR}_{avg} = \frac{\frac{1}{c} \sum_{i=1}^c \sum_{j=1}^n \sum_{k=1}^m (\vec{r}_{ij} - \vec{r}_{tc}) \times \vec{V}_{ij}}{\omega \sum_{i=1}^n \sum_{j=1}^m (\vec{r}_{ij} - \vec{r}_{tc}) \cdot (\vec{r}_{ij} - \vec{r}_{tc})} \quad (3)$$

where $(\vec{r}_{ij} - \vec{r}_{tc})$ is the distance between the TC and the location with indices (i, j) , $\vec{V}_{ij}$ is the velocity, and ω is the engine angular velocity. Velocity components, u' and v' , can be calculated by Eq. (4). And then, the average TKE was defined following Eq. (5).

$$(\overline{u'_{ij,k}}, \overline{v'_{ij,k}}) = (\overline{u_{ij,k} - \bar{u}_{ij,k}}, \overline{v_{ij,k} - \bar{v}_{ij,k}}) \quad (4)$$

$$TKE_{avg} = \frac{1}{c} \sum_k \frac{1}{n \times m} \sum_i \sum_j \frac{1}{2} \times [(\overline{u'_{ij,k}})^2 + (\overline{v'_{ij,k}})^2] \quad (5)$$

where u and v are velocity components of an individual cycle k at position with indices i and j , $\bar{u}$ and $\bar{v}$, are averaged u and v of the number of total cycles c .

2.4. Experimental conditions

In this study, three pistons and two injectors were used to investigate effects of piston shape and spray pattern on in-cylinder flow. The optical engine was operated with 1,500 rpm. A high-pressure injection system was applied to the optical engine to improve atomization performance. Injection pressure affects physical phenomenon such as the plume-to-plume interaction, atomization performance, air entrainment, air-spray interaction, and wall-wetting. Therefore, the effect of injection pressure on the in-cylinder flow was evaluated in the previous studies [27]. Injection pressure was selected as 700 bar, and injection timing was set under various conditions including early injection and late injection. Fuel mass was a fixed as 18.8 mg to conduct the experiment

under the same load and lambda conditions. Therefore, injection duration was different in two test injectors. It is assumed that the same momentum per unit mass is supplied during fuel injection. In the experiment, the equivalence ratio was calculated based on the air mass and fuel mass. First, fuel mass was measured depending on injection pressure and energizing time. Air mass was controlled by mass flow controller with maintaining intake manifold pressure. Based on the measured fuel mass data, injection duration was accurately controlled by NI compact RIO system. Intake air temperature was maintained as 30 °C by using heating chamber. The specific conditions are arranged in Table 4.

2.5. Simulation model validation

The simulation was conducted using CFD program, CONVERGE v2.4, to evaluate mixture formation and flow characteristics in the pent-roof region. According to the grid independency results which considered the base grid size from 3.5 to 6.0 mm in previous study [35], the base grid size of 4.0 mm was appropriate considering the simulation time and accuracy. Thus, based on results of previous study, based grid size was set as 4.0 mm, and grid embedding applied inside the cylinder to have grid size of 2.0 mm. Additionally, adaptive mesh refinement was activated based on the velocity difference of 2.5 m/s.

For the turbulence model, RNG k-ε model that showed accuracy in predicting turbulence of spray and combustion in the swirling flow of the engines [36] was used. This model has been used and verified for GDI spray and engine simulation in many studies [37,38]. KH-RT model was used for both test injectors to simulate the spray and droplets break-up processes. Those models had a good agreement with experiment in the previous study [14]. The model was validated based on the comparison results of mixing process after injection between experiment and simulation. In Fig. 6, the result in the cross section where the laser sheet was formed were compared, and the simulation results were represented

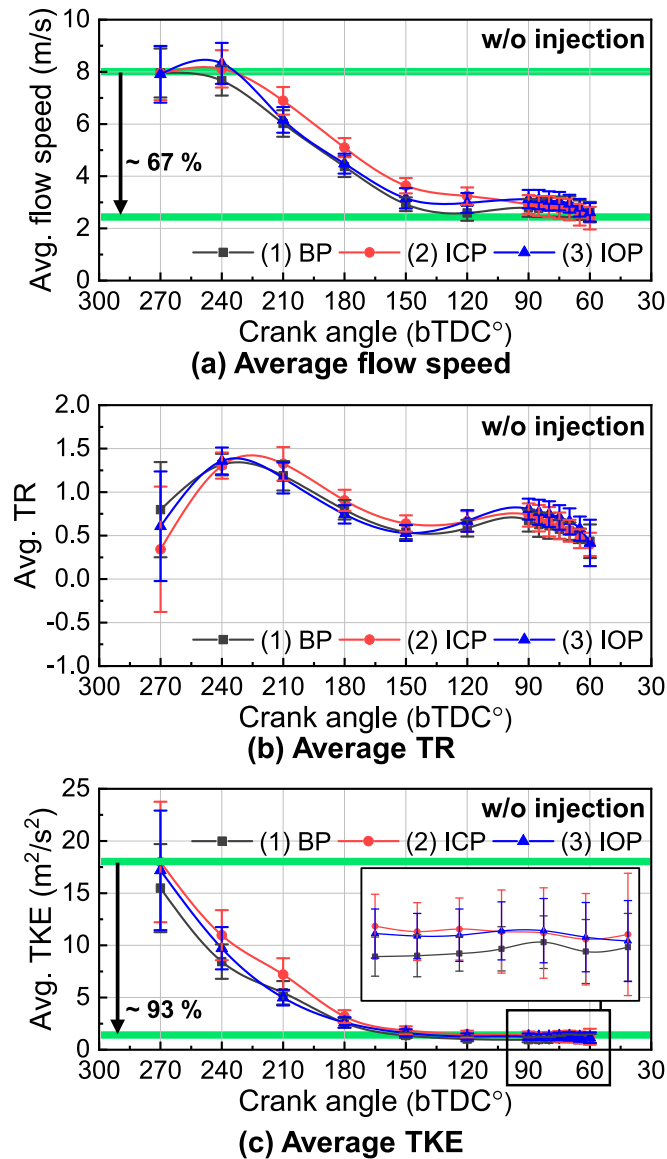


Fig. 8. Average flow speed, average TR, and average TKE for piston shapes (@ w/o injection).

by applying the same ROI as the ROI of experiment. From the results with various injection timing, the simulation model well follows the spray development process for different in-cylinder flow condition for the test injectors.

3. Results and discussions

3.1. Effects of piston shapes on in-cylinder flow

The in-cylinder flow affects on the air–fuel mixing performance, combustion efficiency, combustion duration, emissions, and knock mitigation. These are decided through the interaction process between in-cylinder flow and injected fuel. In short, the combustion and emissions characteristics are significantly linked with the in-cylinder flow. The piston shape is one of the important factors affecting the in-cylinder flow. The in-cylinder flow intensify can be increased or decreased by the upper surface of the piston design. Moreover, the injected fuel moves following the wall and piston head during the interaction between spray and air. Therefore, various air–fuel mixtures can be formed depending on the piston shape. The piston shape is designed to achieve various

objectives such as intensification of tumble or swirl flow, increase of compression ratio, and formation of stratified mixture. In this study, the piston with a bowl was selected as the base piston to form a stratified mixture. Moreover, two pistons with different shapes were used to ensure a high turbulence intensity near the ignition timing by intensifying the tumble flow.

Fig. 7 shows the ensemble-mean flow field depending on the crank angle under different types of pistons. There was no clear structural difference of ensemble-mean flow for the piston shapes. At the beginning of the intake stroke (BTDC 270°), the fastest flow field was formed around the intake valves. A strong in-cylinder flow at the left side of the cylinder occurred while intake valves opened. The in-cylinder flow speed decreased as an intake air was filled inside the cylinder, reducing the difference between the intake manifold and in-cylinder pressure. At around BTDC 180°, the flow momentum concentrated on the left side of the cylinder moved to the right side of the cylinder while rotating counterclockwise. The compression stroke after BTDC 180° showed a higher speed on the right side of the cylinder with the piston ascending. These results were the same regardless of the piston shape. In the case of the ICP, the rotational flow was marginally higher than those for the other piston shapes. This was because the momentum loss caused by the collision at the piston surface decreased compared with the BP. Furthermore, the flow momentum was concentrated at the cylinder center where the PIV measurements was performed. However, the detailed difference of in-cylinder flow structure according to piston shape could not be observed. This was because large-scaled PIV measurements were conducted in this study.

The average flow speed, average TR, and average TKE are shown in Fig. 8. The ICP had marginally higher average flow speed and average TR from before top dead center (BTDC) 210° to BTDC 150°. However, there was no difference according to the piston shape at BTDC 60°. It means that there was no significant difference in the later compression stroke because of the rapid and continuous momentum dissipation. The average TKE shows a tendency similar to that for average flow speed. The difference of average TKE between piston shapes was not evident during the later parts of the compression strokes. The bowl piston is generally used for stratified charge combustion because it is advantageous for fuel stratification and air–fuel mixture formation [39]. Although rotational flow formation was expected to be disadvantages because of the uneven piston surface, it showed similar average flow speed, average TR, and average TKE compared to other pistons. The fuel injection can determine the momentum and direction of the in-cylinder flow by interaction with the cylinder wall and piston surface. Therefore, liquid-phase fuel distribution and in-cylinder flow after fuel injection were observed depending on the piston shape.

Fig. 9 shows the liquid-phase fuel distribution after the injection timing 30° CA according to the injection timing under different piston shapes. In general, evaporated fuel distribution can be captured by LIF imaging technique. However, liquid phase fuel before evaporation can be easily capture by Mie-scattering technique with laser or light-emitting diode lamp. In this study, to observe fuel distribution right after end of injection in PIV image plane, CW laser was used as a light source. At injection timing of BTDC 300°, fuel atomization and the mixing performance could be increased owing to the interaction between the spray and strong intake air flow. The strong interaction between spray and in-cylinder flow promotes better air–fuel mixing performance. The injected fuel spray can be broken up into smaller droplets, increasing the surface area available for evaporation. Enhanced atomization can improve fuel vaporization and mixing performance, resulting in better combustion efficiency, reduced fuel consumption, and lower emissions. However, a direct piston impingement of the fuel can occur if the fuel injection is too early (injection timing of BTDC 300°). In this case, it provides a cause for spray momentum loss and soot emissions [40]. And it was difficult to form a complete rotational flow owing to the narrow space inside the cylinder. Meanwhile, when the injection timing was delayed, the in-cylinder flow weakened,

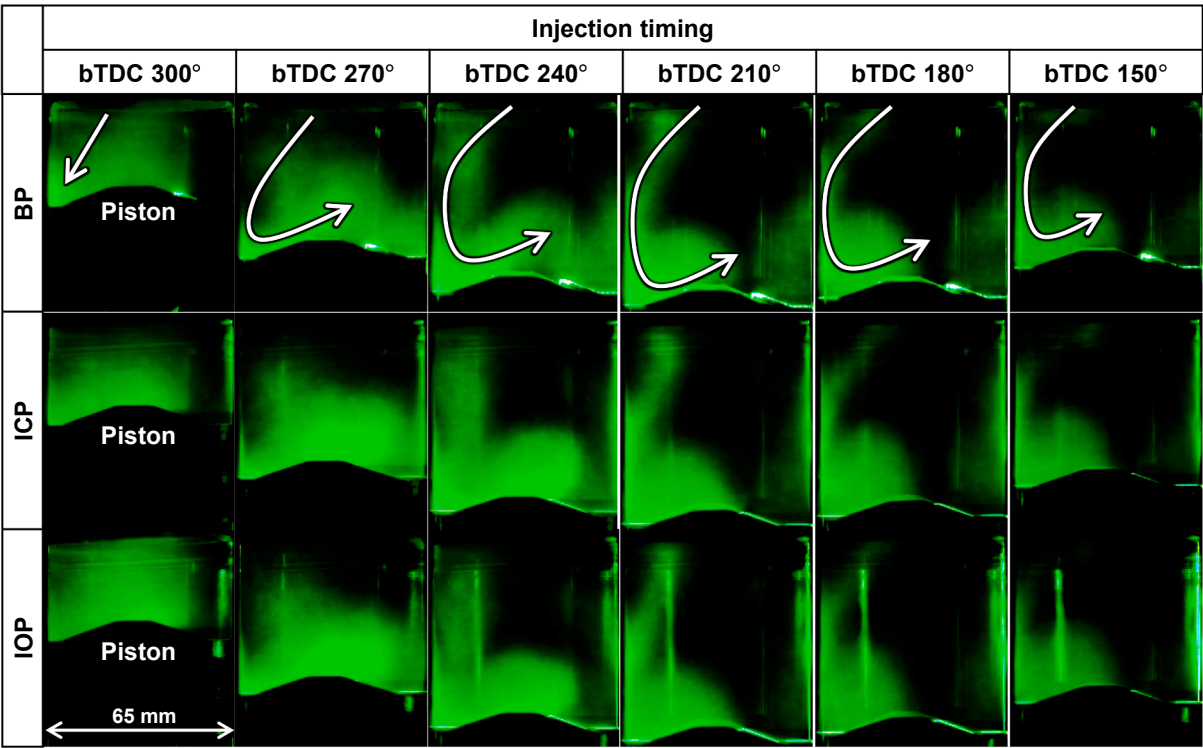


Fig. 9. Liquid-phase fuel distribution in two-dimensional image plane for piston shape under different injection timings (@ ASOE 30° CA).

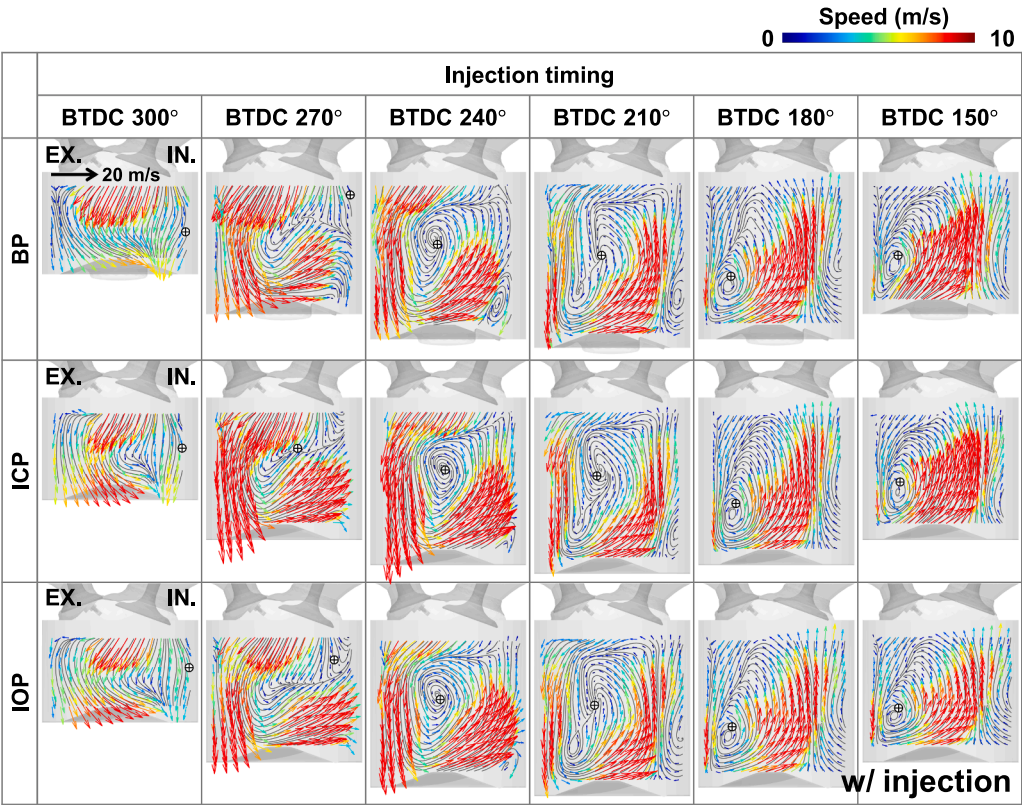


Fig. 10. Ensemble-mean flow field after fuel injection for piston shape under different injection timings (@ w/ injection, ASOE 30° CA).

whereby it was difficult for the injected fuel to move to the entire cylinder area along the in-cylinder flow. Therefore, the fuel was relatively stagnant on the upper-left side of the piston compared with the early

injection timing. The piston shape did not significantly affect the liquid-phase fuel distribution shape after fuel injection. Rather, the injection timing had a substantial influence on the fuel distribution shape after

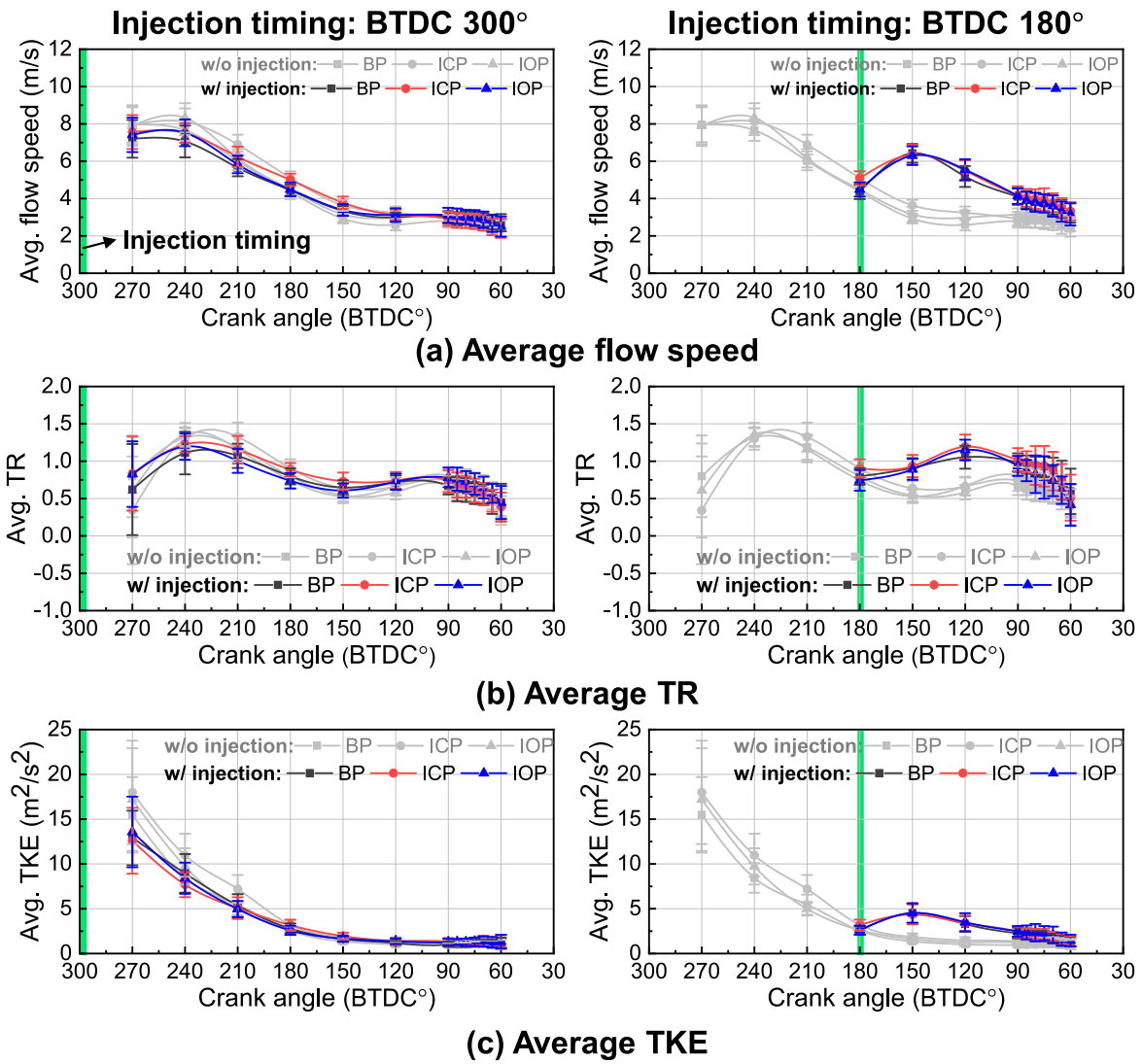


Fig. 11. Quantitative indices after fuel injection for piston shapes (@ w/ injection).

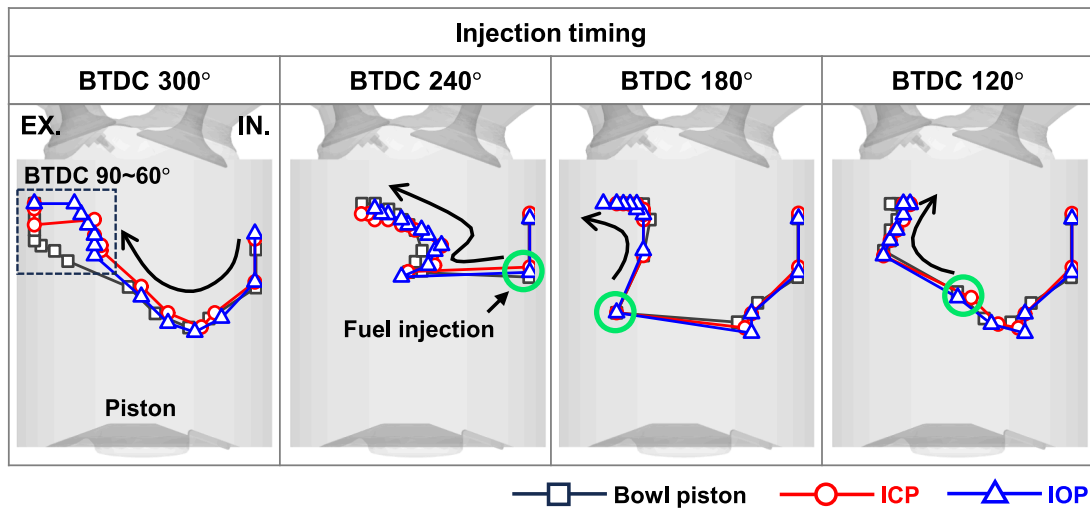


Fig. 12. TC travel path according to piston shape (@ w/ injection).

fuel injection, regardless of the piston shape. To summarize, the fuel distribution was affected more by the intensity of interaction between the spray and air.

Fig. 10 shows ensemble-mean flow fields after fuel injection for piston shape under different injection timings. The flow was intensified after injection having high momentum except for the injection timing of

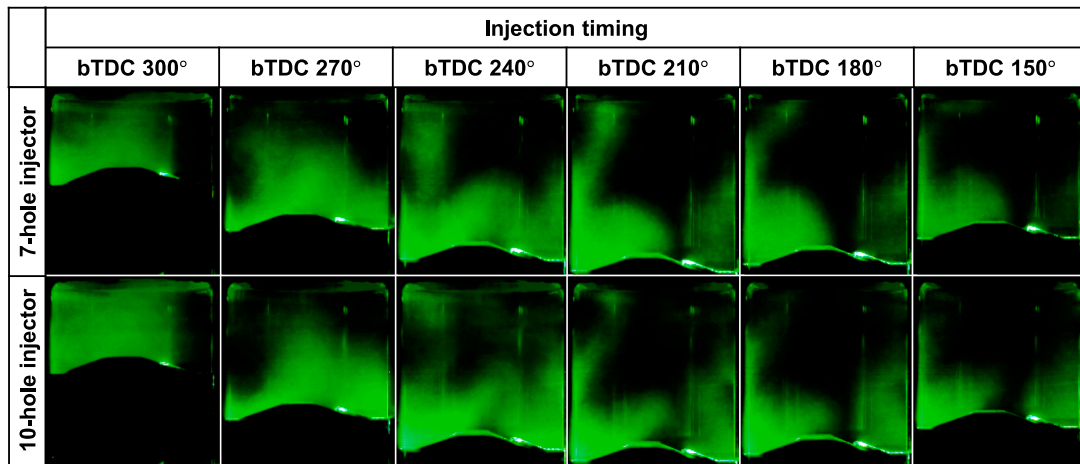


Fig. 13. Liquid-phase fuel distribution for spray patterns (@ ASOE 30° CA).

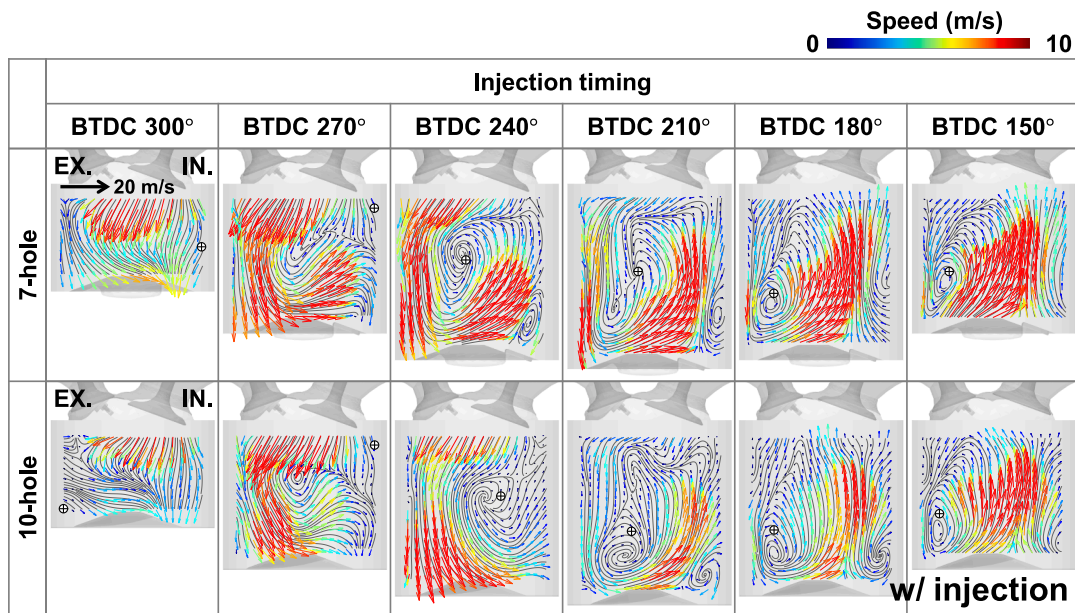


Fig. 14. Ensemble-mean flow fields after fuel injection under different spray pattern (@ w/ injection, ASOE 30° CA).

BTDC 300°. Moreover, although the piston was different, the in-cylinder flow showed a structurally similar shape regardless of piston shapes. The effect of enhancing the flow by fuel injection was relatively insignificant at the early injection timing because the fuel was injected in a state where the intake air flow was relatively strong. This occurred because the direct piston impingement caused momentum loss. Meanwhile, in the delayed injection during the compression stroke, when flow was relatively weak, a large momentum was supplied when the fuel was injected, so that the flow was intensified significantly. That is, the effect of fuel injection on the in-cylinder flow is determined by the difference in momentum between the flow and the spray. When the spray momentum is higher than the flow momentum, the strength and direction of the flow are decided by the spray. Therefore, the factors such as injection pressure, spray pattern, and injection duration can be major factors determining the air-fuel mixture formation and flow strength after fuel injection.

Fig. 11 shows quantitative indices after fuel injection for piston shapes. In the injection timing BTDC 300° conditions, strong in-cylinder flow occurred, and the piston was positioned high. Thus, high air-spray interaction intensity and fuel impingement occurred. Rotational flow is

difficult to form due to the relatively narrow inner space of the cylinder. The injected fuel collided with the upper surface of the piston and lost their momentum resulting in stagnated fuel, so that the average flow speed decreased. This is because the fuel had a high density compared to air. However, the effect of fuel injection decreased, and the average flow speed was similar to that of the conditions without fuel injection. In the later part of compression stroke (after BTDC 90°), similar average flow speeds were measured regardless of the fuel injection and piston shape. The average TR also did not show a clear difference depending on the presence or absence of fuel injection. The average short TKE decreased after fuel injection as did the average flow speed. In, early injection disturbed in-cylinder flow development after fuel injection but did not affect on the in-cylinder flow in the later part of compression stroke.

In the injection BTDC 180° conditions, the piston was located at bottom dead center and the in-cylinder flow was weak. Therefore, direct piston impingement of fuel did not occur, and air-spray interaction strength was relatively low. The spray with high momentum determined the strength and direction of the in-cylinder flow. After fuel injection, the average flow speed increased markedly. The average TR and the average TKE also increased significantly after fuel injection. However,

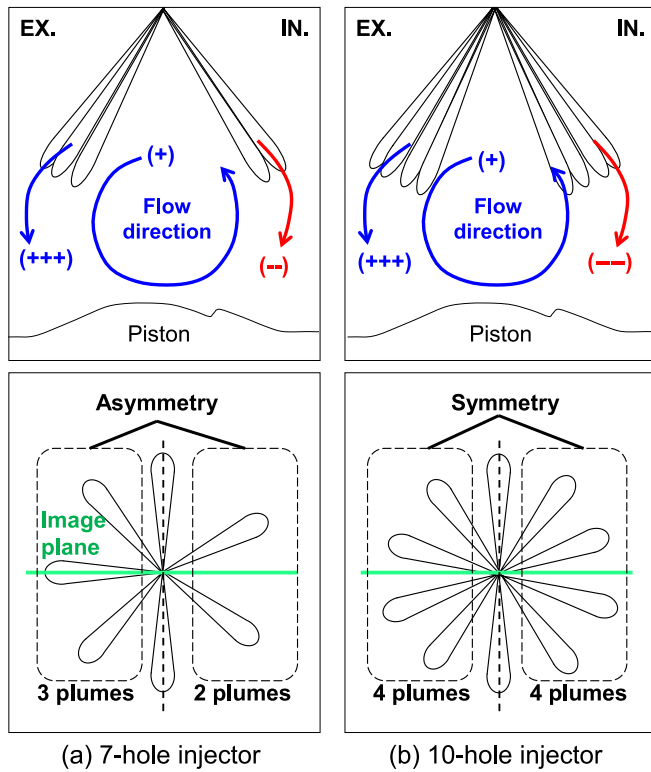


Fig. 15. Comparison of 7-hole and 10-hole injectors.

the difference between piston shapes became more insignificant. This meant that the fuel injection had a greater effect on the in-cylinder flow than the piston shape. The structural change of the in-cylinder flow depending on the piston shape after fuel injection can be confirmed by the TC travel path, as shown in Fig. 12.

Fig. 12 shows the TC travel path from BTDC 270° to BTDC 60° under different piston shapes and injection timing. The TC, by definition, is located on the opposite side of the high-velocity zone. Thus, regardless of the piston shape, it was located on the right side of the cylinder in the intake stroke and on the left side of the cylinder in the compression stroke. In the previous results, it was confirmed that the effect of fuel injection disappeared in the later part of compression stroke in case of early injection timing. Although the effect of fuel injection was insignificant, a difference in the TC travel path occurred between BTDC 90° and BTDC 60°. However, in this region, the average flow speed and average TKE were very low, so that it cannot be considered as significant difference. Moreover, regardless of the piston shape, TC was located at the same location at BTDC 60°. As the injection timing was delayed, the TC travel path was determined by the fuel injection. Despite the different piston shapes, TC travel path after fuel injection was the same. This, like the previous results, emphasizes the importance of the fuel injection strategy. In other words, the fuel injection was more effective than the piston shape in terms of enhancing tumble flow or turbulence intensity. Therefore, in section 3.2, two test injectors with different spray pattern (symmetry and asymmetry) were used to analyze effects of spray pattern on the in-cylinder flow.

3.2. Effects of spray patterns on in-cylinder flow

The spray pattern in GDI engines plays a vital role in atomization performance, air–fuel mixing performance, wall-wetting, combustion, and emission characteristics. In this section, two test injectors with different spray patterns are used. The test injectors were a 7-hole injector and a 10-hole injector. The 7-hole injector had an asymmetric spray pattern to intensify the tumble flow, and the 10-hole injector had a

symmetric spray pattern to form a stratified mixture.

Fig. 13 shows the liquid-phase fuel distribution after the injection timing of 30° CA according to the spray pattern. In the early injection conditions (BTDC 300°), the fuel distribution was affected mainly by the intake flow supplied with a large momentum because momentum of injected fuel was lost due to the direct piston impingement. As a result, the fuel distribution of the two test injectors was similar. However, the flow intensity decreased when the injection timing was retarded. Therefore, the injected fuel was hard to be scattered because of weak in-cylinder flow. As a result, injected fuel locally stagnated near piston head. This trend was observed for both the injectors. However, in the 10-hole injector, more fuel stagnated on the right position of the cylinder compared with the 7-hole injector. This occurred because the injected fuel toward the right position of the cylinder moved into the PIV image plane following the cylinder wall and piston head.

Fig. 14 shows the ensemble-mean flow fields after fuel injection under different spray patterns. In the injection timing of BTDC 300° conditions, the tendency wherein the fuel injection disturbed the flow development could be observed more clearly in the 10-hole injector. This was because more plumes impeded the flow path as these collided with the inflowing air from the right to the left of the cylinder. Moreover, direct piston impingement occurred on a wider area to result in momentum loss. In the case of the 7-hole injector, a strong flow was formed in the upper right direction on the cylinder after the injection timing of BTDC 270°. On the other hand, in the 10-hole injector, a strong upward flow was formed after the injection timing of BTDC 210°. Also, vortices were formed on both sides of the cylinder. In the case of no flow, only the spray pattern determines the flow after fuel injection. However, in a real engine, the flow always exists so that the rotation direction and strength of the flow should be considered. In the case of the 7-hole injector, more plumes developed toward the left side of the cylinder. Therefore, considering the sum of momentum of flow and spray, the flow was formed towards the upper left side of the cylinder after fuel injection. In the case of 10-hole injector, the flow after fuel injection was determined by the rotation direction and strength of the flow. This is because the same momentum was supplied symmetrically to the left and right side of the cylinder. As a result, the updraft did not occur under the relatively strong flow conditions, but as the flow intensity decreased, the updraft was formed by fuel injection.

The direction of flow and direction of spray are expressed in Fig. 15. In the 7-hole injector, the direction of flow rotation and that of spray development were identical in the PIV image plane (see Fig. 15(a)). Therefore, the in-cylinder flow generated by the spray and that generated during the intake stroke rotated in an identical direction. Thereby, the tumble flow could be strengthened. Meanwhile, in the case of the symmetrical spray pattern of the 10-hole injector (see Fig. 15(b)), because the flow and the spray collided to result in loss of momentum. Moreover, the flow induced by the spray on the left side could collide with the flow induced by the spray on the right side. Therefore, the 10-hole injector had a disadvantage to reduce tumble flow. These results could be observed more clearly from the quantitative indices.

Fig. 16 shows a comparison of quantitative indices after fuel injection under different the spray pattern. In the injection timing of BTDC 300° conditions, average flow speed, average TR, and average TKE decreased after fuel injection in both injectors. This is because the high momentum loss of spray caused by air–fuel interaction and piston impingement of fuel. In the 10-hole injector, the average flow speed decreased by nearly half and the average TKE decreased by about 38 %. This showed a reduction rate of about 3 times and about 4 times, respectively, compared to the average flow speed and average TKE of the 7-hole injector. In the injection timing of BTDC 180°, average flow speed, average TR, and average TKE increased after fuel injection in both injectors. In the 7-hole injector, the in-cylinder flow was intensified effectively due to the asymmetry spray pattern. In the 10-hole injector, more momentum loss occurred by collision between the left flow and right flow. As a result, the average flow speed and average TKE of the 7-

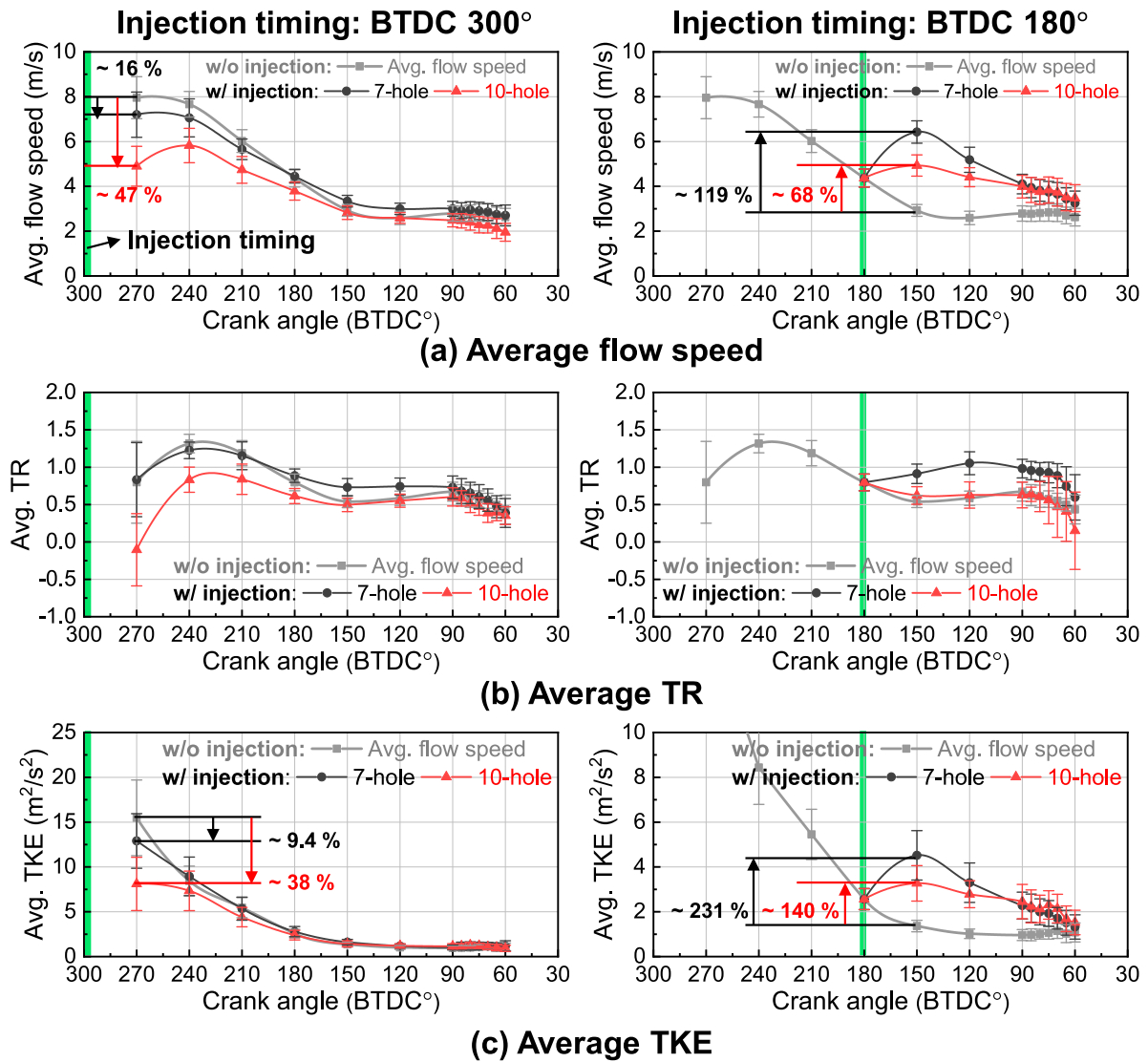


Fig. 16. Quantitative indices depending on spray pattern after fuel injection (@ w/ injection).

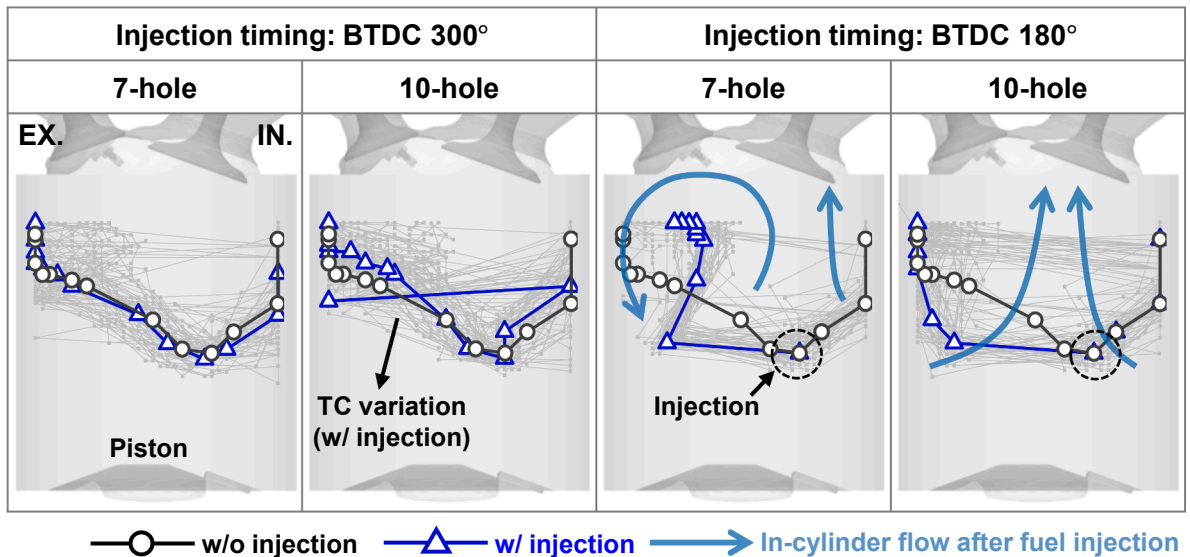


Fig. 17. TC travel path after fuel injection for spray patterns (@ w/ injection).

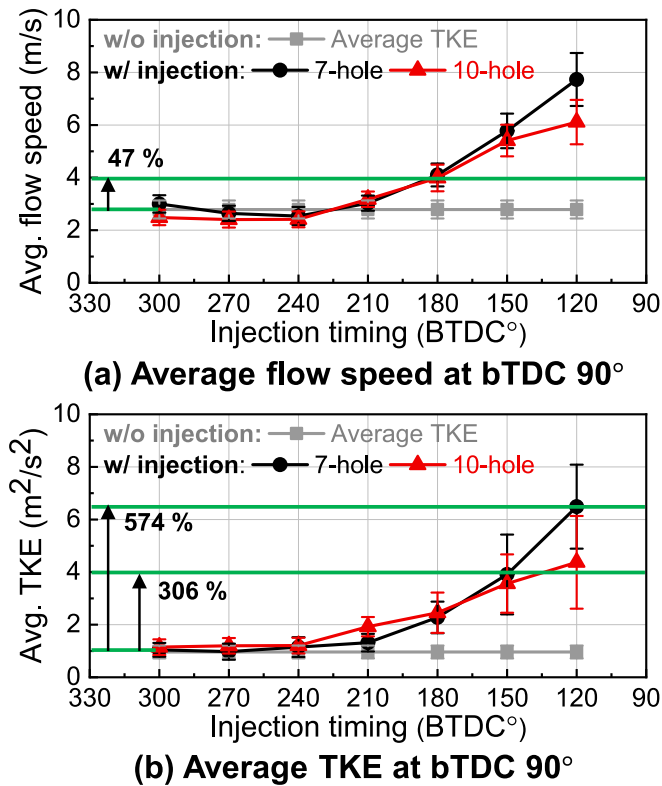
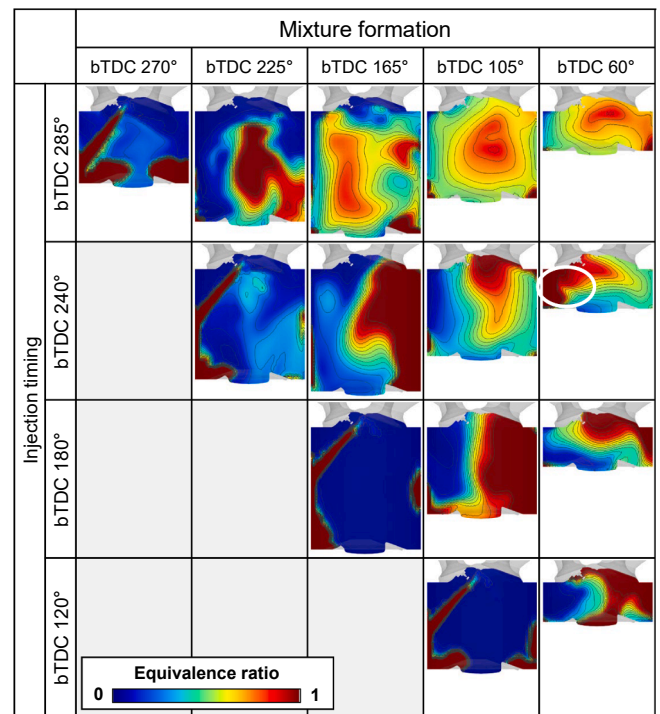


Fig. 18. Average flow speed and average TKE for injection timings in different test injectors (@ w/ injection, BTDC 90°).

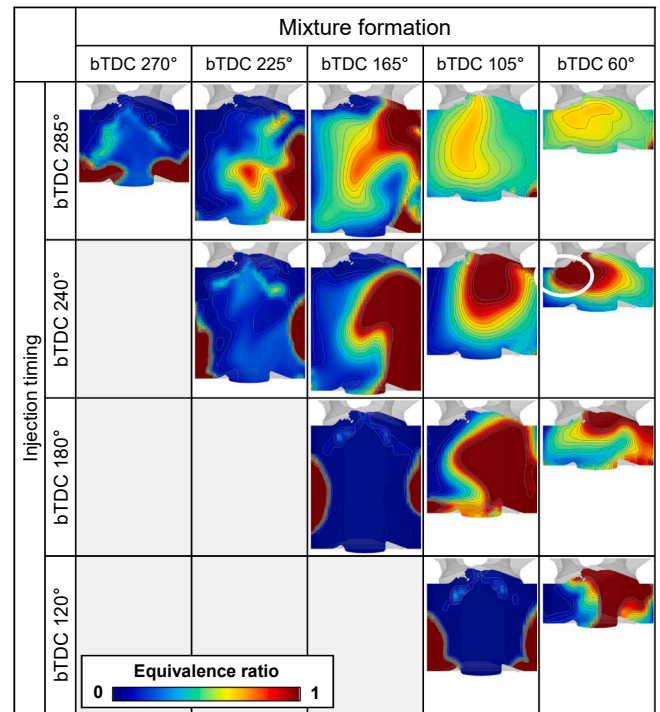
hole injector increased by 119 % and 231 %, respectively, while the 10-hole injector increased by 68 % and 140 %, respectively. As confirmed in the previous results, the 10-hole injector was disadvantageous in forming rotational flow after fuel injection. The 10-hole injector displayed a significantly lower average TR in both early and delayed injection timings compared with the 7-hole injector.

Fig. 17 shows TC travel path after fuel injection for spray patterns. The flow structure did not be affected by the early injection in the test injectors. This occurred because the spray underwent a large momentum loss during the injection. Therefore, TC travel path followed similar path regardless of the spray pattern. However, when the injection timing was delayed, the large momentum supply altered the flow structure. In the 7-hole injector, the TC was located on the upper side near the cylinder center because the flow rotated. In the 10-hole, an upward flow was generated, whereby the TC was located on the left side.

Fig. 18 shows the average flow speed and average TKE for the injection timings and spray patterns. For both injectors, fuel injection before BTDC 240° did not enhance the flow after BTDC 90°. The average flow speed showed a similar level even at the injection timing of BTDC 210°, and after the injection timing of BTDC 180°, both injectors showed an increase over 47% in the average flow speed. After injection timing of BTDC 150°, which has a relatively short momentum dissipation time, the effect of fuel injection was maintained even after BTDC 90°, showing high average flow speed in an asymmetrical 7-hole injector. The average flow speed was similar between the two injectors under BTDC 210° and BTDC 180° fuel injection conditions. However, the average TKE was slightly higher in the 10-hole injector. This is because the turbulence slightly increased due to the flow collision caused by the symmetrical spray development. The average TKE of the 7-hole injector increased as the injection timing was delayed and the difference of average flow speed between the injectors occurred. In the case of the 7-hole injector, the injection timing of BTDC 120° increased about 574% compared to the without fuel injection conditions at BTDC 90°. As a result of the PIV measurement, the 7-hole injector was more advantageous than the 10-



(a) 7-hole injector



(b) 10-hole injector

Fig. 19. Simulation results of mixture formation process for different injection timing depending on spray pattern.

hole injector in terms of turbulence and tumble enhancement. However, since combustion is greatly affected by the characteristics of the air-fuel mixture, it is necessary to evaluate the air-fuel mixture.

In lean burn combustion with a high compression ratio, the important thing along with turbulence and flow enhancement is the mixture preparation. In lean burn combustion, the combustible equivalent ratio around the spark plug at the spark timing is required for stable

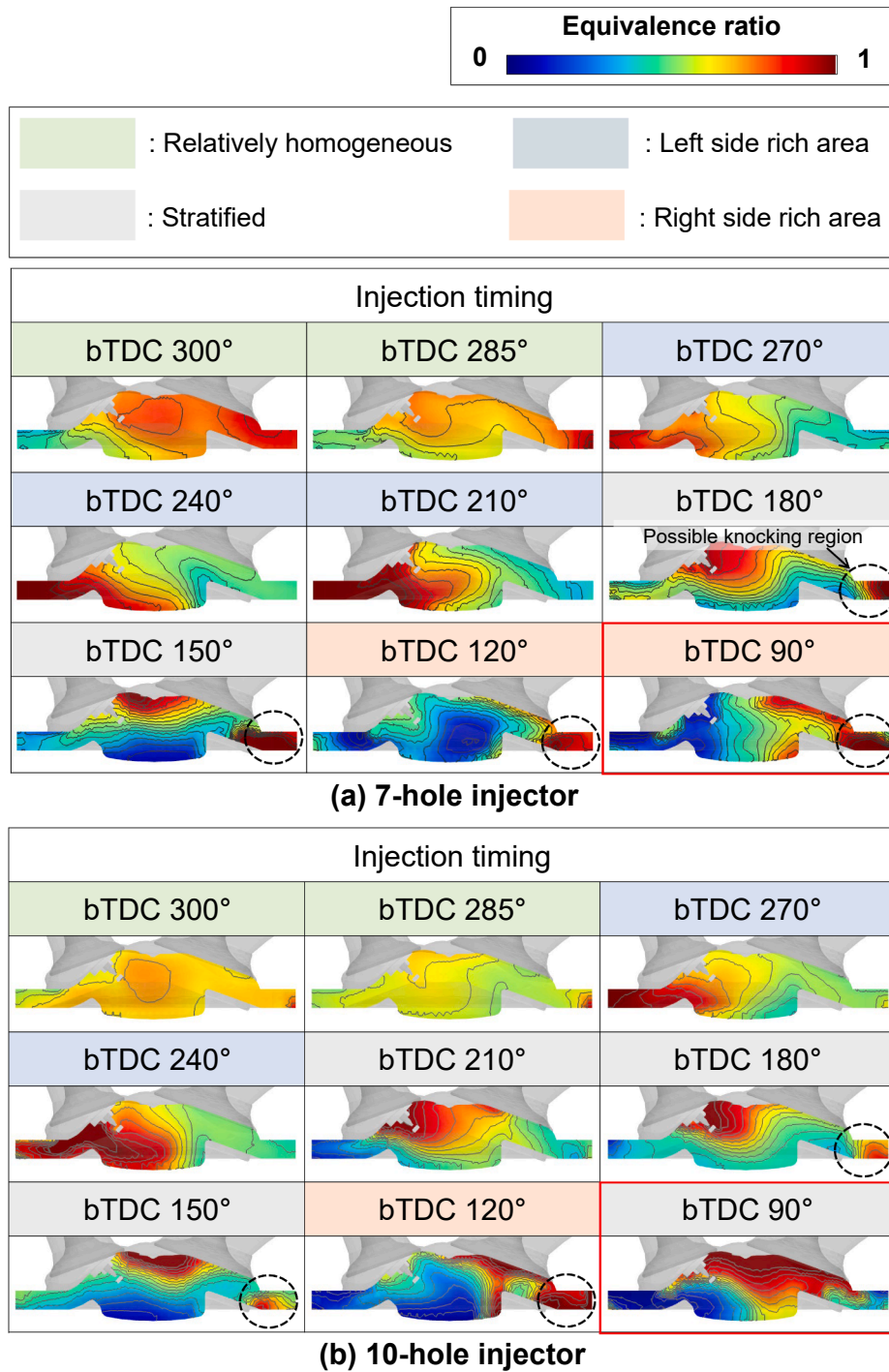


Fig. 20. Fuel distribution for different injection timing depending on spray pattern at BTDC 20°.

combustion [41]. This means ideal stratification of the fuel. In addition, due to the application of a high compression ratio, the formation of a rich region in the end gas region which causes knocking [42]. and deteriorates engine performance should be avoided.

The fuel mixing process depending on the spray pattern was investigated in Fig. 19. When fuel was injected at earlier injection timing with strong air flow (BTDC 285°), the continuous mixing along the flow was revealed. In this case, the fuel development process had not a significant difference between spray patterns. At the more retarded injection timing (BTDC 240°), the evaporated fuel moved by tumble flow from the right side to left side of the cylinder in the compression stroke. At this

injection timing, due to the tumble flow enhancement from the asymmetry of 7-hole injector, the fuel progressed more left side of the cylinder when using 7-hole injector. For the injection at BTDC 180°, where compression process begins, the time for fuel to undergo tumble flow was relatively shorter than that at BTDC 240°, so that at the end of compression, the fuel was richly distributed near spark plug. In both case of injectors at injection timing of BTDC 120°, the fuel was distributed to the right side of cylinder at the end of the compression due to weak flow and insufficient mixing time for both spray pattern.

Since the mixture characteristics near the spark timing are important for combustion process, the mixture distributions at BTDC 20° for

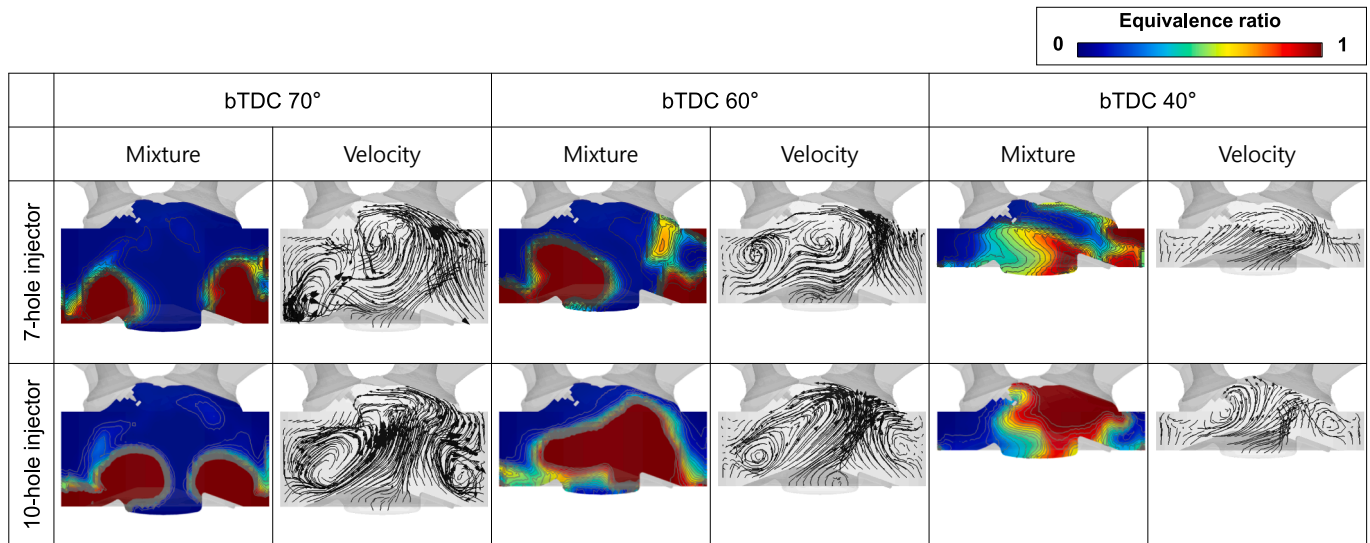


Fig. 21. Mixing process and velocity vectors at injection timing of BTDC 90° depending on spray pattern.

different injection timing depending on the spray pattern were represented in Fig. 20. As the injection timing retarded, the mixture characteristic evolved in order of relatively homogeneous, left side rich area, stratified, and right side rich area. When fuel injected at early injection timing of BTDC 300° and 285°, the fuel experienced fast intake flow, so that the homogeneous mixture was formed with continuous mixing. As the injection timing more retarded, the mixing time was shorter, and flow intensity weakened than early intake process. The fuel moved along with tumble flow. Therefore, with injection timing retarded the fuel concentrated in left side of cylinder, spark plug, and right side of cylinder. However, at injection timing of BTDC 210°, due to the tumble enhancement from the asymmetry of the 7-hole injector, the mixture characteristic for 7-injector showed left side rich area but stratified for 10-hole injector. And mixture characteristics at injection timing of BTDC 90° need to be considered mixture development affected by not only the spray pattern but also piston movement.

As the injection timing retarded until BTDC 90° where the air flow intensity is weak, the mixture distribution was mainly affected by spray behavior and piston. In the case of the 10-hole injector, two spray plumes impacting the piston collided and rise upward, and the fuel almost stagnated in the center of the cylinder due to the momentum loss caused by the collision (Fig. 21(b)). As a results, the stratified mixture was formed at BTDC 20°. On the other hand, in the case of the 7-hole injector, the single plume in the tumble plane moved right-upward after impacting the piston so that it made the right side rich in fuel distribution (Fig. 21(a)).

As a result, when the stratified and left rich portion was formed, it was possible to obtain combustion efficiency for lean combustion by securing the combustion equivalent ratio near the spark plug. However, mixtures with many squish areas away from the spark plug location should be considered areas where knocking may occur. Considering these effects, ideal mixtures were found in the injection timing range of BTDC 270° to 210° in both cases for the spray patterns. In the 10-hole injector, a stratified mixture was obtained even at the injection timing of BTDC 90°. The 7-hole injector improved turbulence more than the 10-hole injector as the injection timing was delayed. However, the 10-hole injector was advantageous in forming a mixture because it could form a stratified mixture at the injection time of BTDC 90°. In order to utilize the turbulence strengthening effect of the 7-hole injector, a multiple injection strategy that injects a small amount of fuel near top dead center (TDC) can be applied.

4. Conclusion

In this study, the optical engine with spray-guided injection had two-cylinder to reflect the cylinder-to-cylinder interference. The oil supply system was replaced with a bearing system, so that PIV images with high-quality were captured. The in-cylinder flow was visualized by large-scaled PIV measurements to evaluate effects of the piston shape and spray pattern on in-cylinder flow. A bowl piston and two additional pistons made for intensifying tumble flow were compared quantitatively. Moreover, performance of asymmetric spray pattern and symmetric spray pattern was evaluated in aspects of intensifying in-cylinder flow. The following conclusions were drawn:

- The piston shape had no significant effect on the structure and strength of the in-cylinder flow. The average flow speed, average TR, and average TKE all showed differences within the error range (level of cycle-to-cycle variation) according to the piston shape.
- Fuel injection further reduced the difference between piston shapes. As the injection timing was delayed, the effect of fuel injection on the in-cylinder flow increased. That is, the importance of the injection strategy in terms of turbulence and tumble flow enhancement is emphasized.
- As a result of PIV measurements, the asymmetrical 7-hole injector was more advantageous than the symmetrical 10-hole injector in terms of turbulence and tumble flow enhancement. After fuel injection, the average flow speed and average TKE increased by 119% and 231% for the 7-hole injector, respectively, while they increased by 68% and 140% for the 10-hole injector, respectively. (Based on injection timing BTDC 180°, where piston impingement of fuel is minimized)
- Considering stratification and possible knock region, ideal mixtures were found in the spray timing range of BTDC 270° to 210° in both cases for the spray patterns. In the 10-hole injector, a stratified mixture was obtained even at the injection timing of BTDC 90°. To utilize the turbulence enhancement effect of the 7-hole injector, a multiple injection strategy that injects a small amount of fuel near TDC can be applied.

Conclusions regarding in-cylinder flow are hard to be generalize to all engine operating conditions. Because engine operating parameters such as engine speed significantly affects the flow dynamics including the intake and exhaust flow, swirl and tumble ratio, and turbulence level. Therefore, additional evaluation is required under various engine operation conditions.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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