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Effect of intake manifold geometry on cylinder-to-cylinder variation and tumble enhancement in gasoline direct injection engine

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Abstract

In order to improve the performance of the gasoline direct injection engine system, it is fundamentally important to reduce the cylinder-to-cylinder variation which affected by the intake manifold geometry. Furthermore, the early tumble development which influences the characteristics of the mixture as followed by the atomization and evaporation of the fuel, also greatly affects engine performance. Thus, in this study, the cylinder-to-cylinder variation in volumetric efficiency and tumble for two different type of intake manifold (curved type and straight type) was investigated using computational fluid dynamic program, CONVERE v2.4. And influence of the intake manifold curve radius to the early flow intensity and tumble development was analyzed. As a result, it was advantageous for cylinder-to-cylinder variation in the straight intake manifold compared to the curved intake manifold. When the intake manifold curve radius was increased in the straight intake manifold, it was effective in strengthening the early flow and tumble intensity. At 3000 rpm, the distance from the intake manifold inlet to the port also had an effect. Therefore, it is possible to improve the intake manifold performance by increasing the intake manifold curve radius and adapting turbocharging at engine speeds above 3000 rpm.

Keywords

Intake manifold, Tumble enhancement, Gasoline direct injection engine, Computational fluid dynamics, cylinder-to-cylinder variation

NOMENCLATURE

k	Turbulent kinetic energy
ω	Angular speed of the flow around the center of mass in the tumble direction
$\omega_{crank\ shaft}$	Angular speed of the crank shaft

ABBREVIATIONS

GDI	Gasoline Direct Injection
CFD	Computational Fluid Dynamics
PIV	Particle Image Velocimetry
IVO	Intake Valve Open
IVC	Intake Valve Closed
EVO	Exhaust Valve Open
EVC	Exhaust Valve Closed

1 **Introduction**

2 In current transportation market, the role of internal combustion engines is still important¹ because
3 the electric vehicles have problems to overcome, such as battery problem and power generation^{2,3}.
4 And there is great scope for even further improvements in engines with advances in combustion
5 technologies^{4,5}. Among internal combustion engines, gasoline direct injection (GDI) engines have
6 the advantages of fuel economy, transient response, and precise air-fuel ratio control^{6,7}. Thus,
7 many studies have been focused on improving engine efficiency and reducing the exhaust
8 emissions of GDI engines⁸⁻¹¹.

9 It is known that the engine system needs to be optimally designed and to be adjusted adequate
10 engine operating parameters to ensuring engine performance¹²⁻¹⁵. And reducing the cylinder-to-
11 cylinder variation should be basically secured to realize the optimized engine performance for a
12 multi-cylinder engine.

13 The cylinder-to-cylinder variation is closely associated with the intake process in GDI engines,
14 and it is inevitable, but it has a crucial impact on engine performance¹⁶. Additionally, in the GDI
15 engine system, the tumble flow in the cylinder generated by intake process determine the turbulent
16 intensity, mixing process, and further the combustion process¹⁷. Therefore, the intake system needs
17 to be developed because the difference of charged air mass and turbulent intensity by intake
18 manifold geometry between cylinders causes the variation of combustion characteristics¹⁸.

19 The intake manifold geometry can be varied depending on the engine mounting space and the
20 number of cylinders in the engine design, and it determines the intake manifold performance¹⁹. The
21 in-cylinder flow characteristic is bound to be affected by the intake manifold geometry because the
22 flow path is formed by the intake manifold inlet to the in-cylinder. Thus, the cylinder-to-cylinder
23 variation can vary depending on the shape of the intake manifold due to the different flow paths.
24 Therefore, an appropriate intake manifold design that can minimize cylinder-to-cylinder variation
25 and form an optimal flow that ensures high tumble intensity is required²⁰. Some studies about
26 intake manifold design have been conducted to enhance intake manifold performance.

27 Ceviz, M.A. and Akin, M²¹ are studied using intake manifold with variable plenum length for

28 multipoint fuel injection system. They investigated the effect of intake manifold plenum length for
29 different engine speeds on engine efficiency and fuel consumption. And the variable plenum length
30 is highly effective in fuel consumption in multipoint injection systems.

31 Sung et al.²² have established a one-dimensional equation for investigating the design variables
32 about the intake system of four-cylinder diesel engine. The optimum manifold length became
33 shorter when engine speed was increased. And they found that the difference in inlet air mass
34 between cylinders was very small.

35 These previous studies have focused on the effect of intake manifold length on the charged mass
36 and manifold pressure in the multipoint injection systems and diesel engine systems. However, in
37 the GDI engine system, it is important to find tumble flow variations for different intake manifold
38 geometry. Further, the cylinder-to-cylinder variation needs to be analyzed with consideration of the
39 in-cylinder tumble characteristic because it is closely connected with mixture formation and
40 turbulent intensity. Therefore, in this study, the two kinds of intake manifold geometry which have
41 different flow path (curved intake manifold, straight intake manifold) were used to investigate the
42 cylinder-to-cylinder variation in GDI engine system using computational fluid dynamic (CFD)
43 program CONVERGE v2.4. The simulation results were validated with comparing the particle
44 image velocimetry (PIV) results. The cylinder-to-cylinder variation was analyzed based on
45 volumetric efficiency and tumble ratio. Furthermore, to enhance the in-cylinder flow and tumble of
46 straight manifold, the influence of intake manifold curve radius was investigated.

47 48 **Methods**

49 **Optically accessible engine.** An optically accessible 2-cylinder engine was used for this
50 experiment. A transparent quartz was used to visualize the in-cylinder flow using the particle
51 image velocimetry (PIV) technique. The cylinder bore was 75.6 mm and the stroke was 83.38 mm.

52 The detailed specifications of the optically accessible engine are presented in Table 1.

53 Figure 1 shows the schematic diagram of the engine operating system. The intake air was
54 controlled by a mass flow meter and an intake chamber was used to reduce the fluctuation of the

intake flow from the mass flow meter to the intake manifold. Engine data was acquired by using an encoder that was attached to the crank shaft directly, NI DAQ, and NI compact RIO.

Figure 1. Schematic diagram of the engine operating system

Table 1. Engine specification

Compression ratio	14.1
Bore	75.6 mm
Stroke	83.38 mm
Conrod length	147.8 mm

Displacement (2 cylinders)	748.5 cc
Rated RPM	6,300 rpm

Particle image velocimetry (PIV) system.

(a)

(b)

Figure 2. PIV system configuration and image plane: (a) top-view and (b) front-view

A schematic drawing of the PIV system and image plane is shown in Figure 2. A laser sheet of 2 mm was generated by a Nd:YAG pulse laser (Double-cavity, power: 26 mJ, wavelength: 532 nm). Planar convex and concave lenses were used to expand a laser beam entering the engine cylinder from the intake valve to the exhaust valve. The image plane had a width of 65 mm and a height of 100 mm except for the pent roof region. A hollow sphere glass (diameter: 10 μm) was used as a tracer of the intake flow. Particles were supplied through the particle mixing chamber connected to the particle generator. The illuminated particles were captured by a high-speed camera (Phantom, VEO 710) with a Nikon 105 mm macro lens. A delay generator was used to synchronize the high-speed camera shutter timing and the laser oscillation timing. The PIV images were processed using MATLAB software (PIV lab). Additional functions, such as calculating the tumble ratio, turbulent kinetic energy, and vortex core position, were added to analyze the characteristics of the in-cylinder flow quantitatively. To detect the particle displacement, FFT cross-correlation and the Gauss 2x3-point algorithm were used. The mean vector fields were derived from 40 images recorded at each crank angle position and each engine speed. The first correlation employed a 64 x 64 interrogation area with 50% overlap. A second interrogation area had 32 x 32 pixels. In this experiment, the time interval at the intake stroke was set as 0.1 ms.

Vector field computation and analysis. To analyze the in-cylinder flow quantitatively, the tumble ratio and turbulent kinetic energy were computed using the MATLAB program. In this experiment, the tumble ratio and turbulent kinetic energy were calculated from an image plane except for the pent roof region. The details about deriving the tumble ratio and turbulent kinetic energy are specified as follows ²³:

$$Tumble\ Ratio = \frac{\sum_{i=1}^n \sum_{j=1}^m (\vec{r}_{i,j} - \vec{r}_c) \times \vec{V}_{i,j}}{\omega \sum_{i=1}^n \sum_{j=1}^m (\vec{r}_{i,j} - \vec{r}_c) \cdot (\vec{r}_{i,j} - \vec{r}_c)} \quad (1)$$

$\vec{r}_{i,j} - \vec{r}_c$: Distance of a given location with indices (i, j) to the center of rotation,

$\vec{V}_{i,j}$: Velocity in that location, and

ω : Engine's angular (crank) speed.

The tumble ratio is defined as the ratio of dividing an angular velocity of flow by an angular velocity of engine rotation. Although the real in-cylinder flow has a three-dimensional structure, the main motion of tumble flow is in a vertical cross section. Therefore, two-dimensional PIV data measured from an image plane can be used to analyze the characteristics of the tumble flow as follows ²⁴:

$$\text{Turbulent kinetic energy} = 0.5 \times (\overline{(u')^2} + \overline{(v')^2}) \quad (2)$$

u' : Total radial velocity of turbulent vector field in an image plane, and

v' : Total axial velocity of turbulent vector field in an image plane.

Turbulent kinetic energy was computed based on the fluctuation component of the turbulent vector field. The turbulent vector field was calculated by subtracting the mean flow vector field in the flow vector field of each cycle.

Computational details. The engine simulations were carried out using CONVERE v2.4 as the three-dimensional CFD program. To investigate the effects of the intake manifold geometry on the in-cylinder flow characteristics, the simulation was conducted in conditions as shown in Table 2. The valve event was presented based on the valve lift of 1 mm and the timing was based on the top dead-center of the compression stroke. The two cylinders were considered to investigate the cylinder-to-cylinder variation.

Table 2. Simulation condition

Engine speed	1,500 rpm	1,500 rpm	2,000 rpm	3000 rpm
Intake manifold inlet pressure	1.0 bar*	1.3 bar	1.0 bar	1.0 bar
IVO	bTDC 332°			
IVC	bTDC 114°			
EVO	aTDC 152°			

EVC	aTDC 370°
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Figure 3. Computational domain for simulation

Initially, the two types of intake manifold geometry were used to investigate the effects of the intake manifold geometry on cylinder-to-cylinder variation in intake manifold and cylinders (Figure 3). The computation domain has been set including the intake manifold, two cylinders, and exhaust manifold for multi-cylinder calculation. These were basic designs for the intake manifold. The curved manifold consisted of one flow path supplied to each intake port, and the flow direction at the manifold inlet and the port direction were vertical. The straight manifold had a shape in which the flow direction at the manifold inlet and the port direction were in-line, and the flow path split when it was supplied to each intake port.

Figure 4. Intake manifold geometry for evaluating the effect of intake manifold curve radius

Based on straight manifold design, the influence of the intake manifold curve radius on the intake process through the intake manifold and intake port was evaluated. The intake manifold geometry for different intake manifold curve radius is shown in Figure 4.

To quantify the in-cylinder turbulence level, the tumble ratio and turbulent kinetic energy were used. The tumble ratio was defined as the ratio of the angular speed of the flow about the center of mass in the tumble direction, ω , to the angular speed of the crank shaft, $\omega_{crank\ shaft}$, as shown in Eq. 3 as follows:

$$tumble\ ratio = \omega / \omega_{crank\ shaft} \quad (3)$$

The turbulent kinetic energy, k , was defined as half of the trace of the stress tensor as shown in Eq. 4 where $\tilde{u}_i$ is defined as Eq. 5.

$$k = \frac{1}{2} \tilde{u}_i \tilde{u}_i \quad (4)$$

$$\tilde{u}_i \equiv \overline{\rho u_i} / \bar{\rho} \quad (5)$$

For the turbulent kinetic energy of the region, the mass averaged turbulent kinetic energy was used.

Computational model. To predict the turbulent characteristics of the in-cylinder flow effectively, a proper turbulence model is required. In this study, the RNG k- ϵ model^{25,26} was used for closure, which has shown high accuracy in many prior internal combustion engine simulation studies²⁷⁻³⁰.

Validation. To confirm the consistency of the experiment and simulation, validation was conducted. For the validation of the in-cylinder flow characteristics, PIV results and simulation results using the curved intake manifold were compared at an engine speed of 1500 rpm where the intake manifold pressure was 1.0 bar. The cross sections of velocity during the intake process are compared in Figure 5. The planes were clipped at the center of the cylinder. The flow supplied from the intake valve at the beginning of the intake process was directed toward the exhaust valve, hit the piston to form a tumble, and then went through a compression process to form a flow rising toward the intake valve in the experimental results. The simulation results had a similar trend in velocity distribution. Furthermore, the qualitative data were compared in Figure. 6. There was a difference between the experimental and simulation results for the tumble ratio in bTDC 270°, at which the flow had a strong turbulence due to fast intake flow. However, the simulation results showed high accuracy in the subsequent results. And the turbulent kinetic energy and maximum velocity in the same analysis domain as the PIV domain showed similar results in experiments and simulation.

Figure 5. Comparison of the cross-sectional velocity distribution between the experiment and simulation

Figure 6. Comparison quantitative data between experiment and simulation : (a) Tumble ratio, (b)

Figure 7. Image comparison of the intake manifold flow in steady-state conditions between the experiment and simulation

For the flow in the intake manifold, the simulation result was compared with the Mie-scattering experimental results under steady-state conditions in which cylinder 2 was open and the inlet air flow rate was 8 g/s. The Mie-scattering was conducted using a metal-halide lamp as a light source and BnB-1 middle fog as a flow tracer. Three conditions were used with varying intake throttle valve opening angles. The simulation images showed the stream-lines from the intake manifold inlet, which was reasonable to compare with the Mie-scattering image using fog. As a result of validation of the intake manifold flow, the flow travel distance and shape in the intake manifold showed similar results with various intake throttle valve opening angles (Figure 7).

Results and discussion

Performance of curved manifold and straight manifold. One of the crucial factors in enhancing intake manifold performance is reducing cylinder-to-cylinder variation for stable combustion between cylinders. Thus, to evaluate the performance of the curved manifold and straight manifold, the intake process was investigated from the aspect of the cylinder-to-cylinder

variation. And further, the tumble motion was investigated. Because strong tumble during intake process is important for GDI engine system to enhance the spray atomization preventing wall wetting and promoting evaporation of fuel droplets³¹.

Figure 8. Cross-sectional velocity distribution for the curved manifold during intake process : (a) Engine speed : 1500 rpm, Intake pressure : 1 bar, (b) Engine speed : 2000 rpm, Intake pressure : 1 bar

To understand the cylinder-to-cylinder variation due to the intake manifold geometry, first, the intake manifold flow development during intake process was analyzed. The curved intake manifold had a shape in which the direction of the air flow supplied from the manifold inlet and the direction of the port were vertical. The distance from the intake manifold inlet to the intake port inlet differed

between cylinders 1 and 2 due to the structural features. Since the position of the intake port of cylinder 2 was located closer to the intake manifold inlet, the velocity of the flow at the port inlet of cylinder 2 was faster and the momentum was larger during intake process (Figure 8). The velocity distribution inside the intake manifold showed a maximum velocity at the manifold inlet, so that the induction speed at the port close to the manifold inlet, that is, the port where the swirl valve is located, appeared fast in intake process of both cylinders.

Figure 9. Cross-sectional velocity distribution for the straight manifold during intake process : (a) Engine speed : 1500 rpm, Intake pressure : 1 bar, (b) Engine speed : 2000 rpm, Intake pressure : 1 bar

On the other hand, the straight manifold was in line with the direction of the flow supplied from the inlet of the intake manifold and the direction of the intake port. There was a curved path near the intake port inlet. It resulted in a faster flow velocity towards the inner intake port than towards the outer intake port (Figure 9)³². The straight manifold has a symmetrical shape compared to the curved manifold. However, there was a difference between cylinders in the existence of a swirl valve at the inner port where flow was strengthened.

Figure 10. Pressure fluctuation in intake manifold for different intake manifold : (a) Curved manifold, (b) Straight manifold

Figure 10 shows the trend of pressure fluctuation for different intake manifold geometry during the intake process. The curved intake manifold cases appear a difference in pressure fluctuations depending on the cylinder during the intake process for all cases. And pressure of cylinder 1 lagged behind cylinder 2 due to the difference in distance from the intake manifold inlet. On the other hand, in the straight manifold, fluctuations occur during the intake process of cylinder 2 at 2000 rpm, causing a difference between cylinders. This is because there was more suction in cylinder 2 due to

the absence of the swirl valve on the inner port where the high velocity was formed, and this effect increased as the engine speed increased.

Figure 11. In-cylinder charged mass of each cylinders : (a) Curved intake manifold, (b) Straight intake manifold

Figure 12. Volumetric of each cylinders : (a) Curved intake manifold, (b) Straight intake manifold

Figure 11 shows the charged mass in cylinder for different intake manifold. In the case of curved manifold, a variation of the charged mass flow rate occurred between cylinders during the intake process due to the difference in distance from the manifold inlet to the port of each cylinder (Figure 11 (a)). Since the distance from the manifold inlet to cylinder 2 is shorter than the distance to cylinder 1, after the intake process, the charged mass in the cylinder was greater in cylinder 2. As a

result, the volumetric efficiency of cylinder 2 was greater than that of cylinder 1 for curved intake manifold under all conditions (Figure 12 (a)). In the case of a straight intake manifold, the charged mass flow rate during the intake process tended to be almost the same for cylinder 1 and cylinder 2 (Figure 11 (b)). After the intake process, the charged mass in cylinder 2 appeared slightly greater. This result can also be seen from the volumetric efficiency results for cylinders in Figure 12 (b). It resulted from the swirl valve located on one side of the intake port. In the straight intake manifold, the flow to the inner port has strengthened during the intake process for both cylinders, but for cylinder 1, the swirl valve has located at the inner port, preventing flow.

Table 3. Difference of volumetric efficiency between cylinders for different intake manifold

Simulation condition	Curved manifold	Straight manifold
RPM : 1500 rpm, $P_{\text{intake}} = 1.0$ bar	1.4%	0.1%
RPM : 1500 rpm, $P_{\text{intake}} = 1.3$ bar	1.7%	0.2%
RPM : 2000 rpm, $P_{\text{intake}} = 1.0$ bar	0.7%	0.6%

In both intake manifold geometry, volumetric efficiency of cylinder 2 was greater. In the curved intake manifold, the reason was the difference in the distance between the intake port and the manifold inlet, and in the straight manifold, the reason was the presence or absence of a swirl valve in the inner port where the flow is strong. As shown in Table.3, the cylinder-to-cylinder variation from the asymmetry of the curved intake manifold was greater than the cylinder-to-cylinder variation from the existence of the swirl valve in port. Comparing the effect of the intake manifold geometry on the volumetric efficiency variation between cylinders, the straight manifold geometry with a symmetrical shape had the advantage of reducing the cylinder-to-cylinder variation of volumetric efficiency.

Figure 13. Tumble ratio during intake process : (a) Curved manifold, (b) Straight manifold

In curved manifold, the tumble ratio of cylinder 2 has increased faster because of the better air supply from the intake manifold to cylinder 2 at the early intake process from bTDC 300° to bTDC 240° as shown in Figure 13 (a). The pressure fluctuation during intake process of cylinder 1 lagged behind cylinder 2, so the tumble ratio also showed the same trend. Furthermore, at engine speed of 2000 rpm, the effect of the distance difference between the manifold inlet and cylinder was greater, with cylinder 1 having a two peak shape that decreases at bTDC 240°, but cylinder 2 continuously increasing to bTDC 240°. In the straight manifold, the tumble developed similar in cylinders 1 and 2 for engine speed 1500 rpm as shown in Figure 13 (b). However, for engine speed 2000 rpm, the tumble generation was promoted in cylinder 2 at the early intake process. This is because the influence of the presence or absence of the swirl valve of the inner port increased as the engine speed increased. Thus, the cylinder-to-cylinder variation for the tumble ratio was more pronounced compared to the straight manifold. In both intake manifold geometry, the variation of tumble was noticeable as the engine speed increased.

Figure 14. Velocity distribution in tumble plane of cylinder 1 at engine speed of 1500 rpm and intake pressure of 1.0 bar : (a) Curved intake manifold, (b) Straight manifold

Figure 15. Velocity distribution in tumble plane of cylinder 1 at engine speed of 2000 rpm and intake pressure of 1.0 bar : (a) Curved intake manifold, (b) Straight manifold

Through the velocity distribution in the tumble plane of cylinder 1, it can be seen that the straight intake manifold formed stronger flow intensity during intake process as shown in Figure 14 and 15. This is because the direction of flow supplied from the intake manifold inlet and the direction of the intake port was in line at the straight intake manifold, so the straight manifold had less flow momentum loss than the curved type.

Influence of the intake manifold curve radius. Based on the straight intake manifold design which had more advantage for cylinder-to-cylinder variation than curved type, the influence of intake manifold curve radius on intake charging process and tumble characteristics was investigated as an aim of optimization. The analysis was conducted based on the intake process and in-cylinder motion of cylinder 1.

Figure 16. Velocity distribution at intake manifold during cylinder 1 intake process for different intake manifold curve radius at engine speed of 2000 rpm

Figure 16 represents the velocity distribution at the intake manifold during the cylinder 1 intake process. As the intake manifold radius increased, the straightness of the flow inside the manifold is secured along the radius of the curved surface during rapid charging in early intake process. Therefore, it had a faster velocity distribution in the manifold due to a decrease in the momentum that the incoming flow collided with the surface and was lost.

308

309 **Figure 17. Velocity distribution in (a) valve planes and (b) valve plane 1 for different intake manifold**
310 **curve radius at engine speed of 2000 rpm**

311

312 The difference in flow velocity in the intake port was caused by the difference in flow velocity and
313 direction inside the intake manifold as shown in Figure 17 (a). When the plane cut with respect to
314 the inner port was referred to as valve plane 1 and the plane cut from the outer port was referred to
315 as valve plane 2, a higher speed distribution could be confirmed at the inner port due to the
316 characteristics of the straight intake manifold. Also, because of the influence of the flow inside the
317 intake manifold, the flow inside the intake port was strong and evenly distributed as the intake
318 manifold curve radius increased (Figure 17 (b)).

Figure 18. Velocity distribution in (a) intake manifold and (b) valve plane 1 for different intake manifold curve radius at engine speed of 3000 rpm

However, at the engine speed of 3000 rpm where rapid charging occurred, the distance between the intake manifold inlet and the port inlet also affected the intake process. At 3000 rpm, as the distance between the intake manifold inlet and the intake port increased, the response of the intake manifold deteriorated so that the velocity decreased in the region where flow was divided into each cylinder as shown in Figure 18 (a). Therefore, when comprehensively considered including the

effect with the intake manifold curve radius, the fastest air velocity within the intake port was shown at the intake manifold curve radius of 9 cm (Figure 18 (b)).

Figure 19. Tumble ratio for different intake manifold curve radius at engine speed of (a) 1500 rpm, (b) 2000 rpm, and (c) 3000 rpm

The difference in flow velocity from the intake manifold and intake port influenced the in-cylinder intake velocity, which governed early tumble development. At engine speed of 1500 rpm and 2000 rpm, the early tumble ratio became stronger as the intake manifold curve radius increased, and at 2000 rpm, the trend was more pronounced as shown in Figure 19 (a) and (b). On the other hand, at the engine speed of 3000 rpm, the distance between the intake manifold inlet and port also had an effect on the air charging process, so the early tumble was strongest at the intake manifold radius of

9 cm (Figure 19 (c)). Based on these results, in order to improve the intake manifold performance, while increasing the intake manifold curve radius, it can be considered to apply variable intake manifold inlet length with changing engine speed as in the study of M.A. Ceviz and M. Akin²¹ or utilize supercharging.

Figure 20. Volumetric efficiency for different intake manifold curve radius at (a) cylinder 1 and (b) cylinder 2

Figure 20 represents the volumetric efficiency for different intake curve radius and cylinders. In cylinder 1 without a swirl valve in the inner port, the volumetric efficiency increased by 0.5% as the intake manifold curve radius increased from 7 cm to 10 cm at engine speed of 2000 rpm. However, there was no significant difference in the volumetric efficiency for different intake curve radius under other conditions.

Conclusions

In this study, influence of intake manifold geometry on the cylinder-to-cylinder variation and in-cylinder flow characteristic were analyzed based on curved and straight intake manifold designs. Additionally, changes in the intake manifold radius were considered to enhance the in-cylinder flow for straight intake manifold. The main conclusions are as followed.

- (1) In curved intake manifold, the cylinder-to-cylinder variation was caused by the difference between the cylinder and intake manifold inlet. The cylinder-to-cylinder variation was a maximum deviation of 1.7% and a minimum of 0.7% between cylinders. On the other hand, in the straight intake manifold, the cylinder-to-cylinder variation occurred due to the presence (cylinder 1 intake port) or absence (cylinder 2 intake port) of the swirl valve in the inner port where the flow was concentrated. And there was a maximum deviation of 0.6% and a minimum deviation of 0.1% between cylinders. The straight manifold with symmetry shape was more efficient to reducing cylinder-to-cylinder for volumetric efficiency.
- (2) In curved intake manifold, the pressure fluctuation during the intake process of cylinder 1 lagged behind that of cylinder 2 so that the cylinder-to-cylinder tumble formation was larger than that of the straight intake manifold. In addition, the initial flow intensity in the straight intake manifold in which the flow supply direction from the intake manifold and the intake port direction were the same.
- (3) When the intake manifold curve radius was increased in the straight manifold, the velocity inside the manifold was strengthened so that the flow rate in the intake port was increased at engine speed of 2000 rpm. At engine speed of 3000 rpm, not only the intake manifold curve radius but also the distance between intake manifold inlet and the intake port affected to the intake process because of the rapid charging. Therefore, the intake manifold and port flow were the strongest at the intake manifold curve radius of 9 cm.
- (4) As a chain process, the flow intensity at the intake manifold and port influenced the early tumble development. At 2000 rpm, initial tumble ratio increased as the intake manifold curve radius increased, and at 3000 rpm, the initial tumble ratio was greatest at the intake manifold curve radius of 9 cm. Therefore, in order to enhance the intake manifold performance, it is considered to increase the intake manifold curve radius while turbocharging at an engine speed of 3000 rpm or higher.

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Author contributions

Jisoo Shin: Writing- Original draft preparation, Conceptualization, Methodology, Investigation, Experiments

Donghwan Kim: Writing- Original draft preparation, Investigation, Experiments

Yousang Son: Conceptualization, Methodology

Sungwook Park: Writing- Reviewing and Editing, Supervision

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