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Investigation of In-Cylinder Flow Patterns in 4 Valve S. I. Engine by Using Single-Frame Particle Tracking Velocimetry

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The in-cylinder flow field of gasoline engine comprises unsteady compressible turbulent flows caused by the intake port, combustion chamber geometry. Thus, the quantitative analysis of the in-cylinder flow characteristics plays an important role in the improvement of engine performances and the reduction of exhaust emission. In order to obtain the quantitative analysis of the in-cylinder gas flows for a gasoline engine, the single-frame particle tracking velocimetry was developed, which is designed to measure 2-dimensional gas flow field. In this paper, influences of the swirl and tumble intensifying valves on the in-cylinder flow characteristics under the various intake flow conditions were investigated by using this PTV method. Based on the results of experiment, the generation process of swirl and tumble flow in a cylinder during intake stroke was clarified. Its effect on the tumble ratio at the end of compression stroke was also investigated.

Key Words : In-Cylinder Gas Flow, Tumble Ratio, Single-Frame Particle Tracking Velocimetry, Swirl and Tumble Flow, Tumble and Swirl Intensifying Valve

1. Introduction

The shapes of intake port and the gas flow characteristic in the engine cylinder have influenced the improvement of fuel-air mixing, combustion process, and the reduction of exhaust emissions. With the increasing demand for less harmful exhaust emissions and improved fuel economy, the combustion chamber and intake manifold shape in the reciprocating engines have been changed to the swirl and tumble induced structure in order to provide more effective mixture preparation and improvement of combustion for the engine operations. Thus the flow analysis in the spark ignition engine is an important factor for the optimum design of combustion chamber and intake manifold system. In reduction of HC and CO emissions during the engine operation, the improvement of swirl or tumble flow is an important factor for the promotion of gas flow characteristics in the engine. The combustion improvement is very dependent on the intake flow patterns and gas flow in the cylinder.

Early works on the influence of swirl and tumble flow in the engine were carried out by many researchers. Many studies on the measurement and prediction of in-cylinder flow in the internal combustion engine have been performed by Arcoumanis et al. (1990), Newman et al. (1995), Reuss (2000), and Shuliang (2000). Lee et al. (1992) and Khalighi et al. (1995) measured full-field flow motion around intake valve and inside cylinder respectively by use of particle image velocimetry (PIV) method. Lee et al. (1999) tried measurements of detailed flow motion to investigate turbulence scale near the

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spark plug with two-color PIV method. In the mean while, tumble intensified system was applied to most of lean burn engine in order to improve the turbulent enhancement and stratification of charged mixture using the tumble effect. Neuβer et al. (1995) have shown that the importance of particle tracking velocimetry (PTV) for the control of in-cylinder flow motion and fast burn lean combustion in a four valve SI engine by applying tumble intensifying system. Regarding the improvement of flow characteristics in a SI engine, Tabata et al. (1995) and Choi et al. (1999) investigated the lean combustion system by use of an in-cylinder flow control with assistance of PIV. However, in-cylinder flow and flow characteristics in a SI lean burned engine become more important, and the investigation of flow process is more difficult to analysis. The analysis of in-cylinder flow field still contains many uncertainties of flow characteristics.

In this work, the effects of the swirl and tumble intensifying system on the in-cylinder flow characteristics under the various intake flow conditions were investigated using single frame PTV for the engine design application such as intake port and combustion chamber geometry.

2. Experimental Apparatus

Figure 1 shows the schematic diagram of experimental apparatus for the visualization of gas flow in the cylinder. This system provided the effect of the swirl and tumble flow on the gas flow in the engine due to the change for the flow conditions.

The specifications of single cylinder visualization engine are listed in Table 1. The engine used for this work was a single-cylinder four-valve engine with pentroof combustion chamber. The visualization system consists of an elongated piston with quartz window, quartz cylinder liner, engine block, and cylinder head including modified valve system. The visualization system of gas flow in the cylinder is composed of 5W Ar-ion laser, cylindrical lens, 35 mm camera, and

Table 1 Specifications of visualization engine

Engine		85mm
Displacement		88mm
Bore × Stroke		499cc
Compression ratio		7
Combustion chamber		Pentroof
Valve timing	**IVO**	18°BTDC
	IVC	54°ABDC
	EVO	47°BBDC
	EVC	17°ATDC

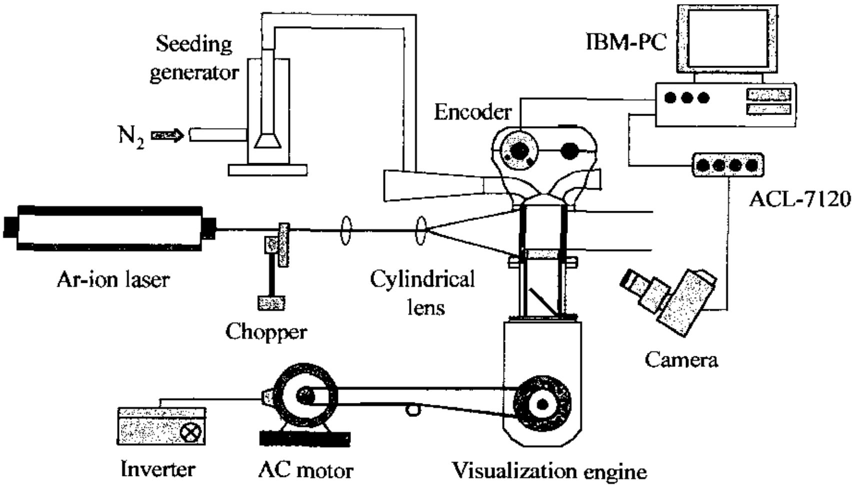


Fig. 1 Schematic diagram to measure in-cylinder flow

processing system of image data for the gas flows. And a rotary-type encoder and timer board were used to synchronize the crank angle position and camera shots.

3. Experimental Procedures

In order to measure the in-cylinder flow characteristics, the test engine is driven by an electric motor with the capacity of 11kW. In this experiment, the gear ratio between flywheel and drive shaft of motor is 2:1 and timing belt mechanism transmits the driving power. The visualization engine is operated by an electric motor and its speed was controlled by an inverter system. Ar-ion laser of 5W is used as a light source and a mechanical chopper was installed to produce coded pulse of continuous laser beam. Figure 2 shows an illuminating coding pulse used to determine the velocity direction and gas flow.

For the sake of detecting the particle trace, the original image of gas flow field was recorded and transformed by the 35mm camera and image grabber system. The flow images from 35mm negative film were digitized through an image data processing board driven by a microcomputer. The processes for obtaining velocity vector such as original image, filtering and thresholding, boundary detection, centroid calculation, and final vector are shown in Fig. 3.

Upon completion of the particle image processing, the particle trajectories are replaced by flow vectors whose vector lengths are determined to the particle velocities at the corresponding location in the gas flow field in the cylinder. The engine speed used for this work was 600rpm. Figure 4 shows the configuration of intake port and combustion chamber used to experimental engine. Figure 5 shows the test section for measurements in-cylinder flow fields. In-cylinder tumble flows were measured at central axis of cylinder, and swirl flows were measured at 25mm down from TDC as shown in this figure. In-cylinder flow fields were taken at 4 degree of crank angle degrees and downward distance from the top dead center are listed in Table 2.

To generate several different swirl and tumble effects in engine cylinder, the swirl intensifying valve (SIV) and tumble intensifying valve (TIV) with 30% opening area of intake port were installed between the intake manifold and the cylinder head. Figure 6 shows the inlet flow conditions

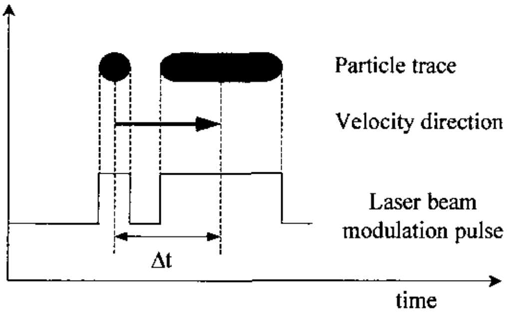


Fig. 2 Illumination coding pulse used to determine velocity vector

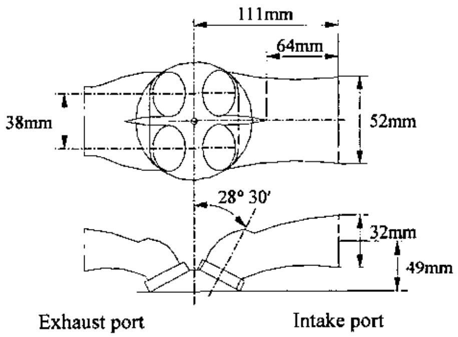
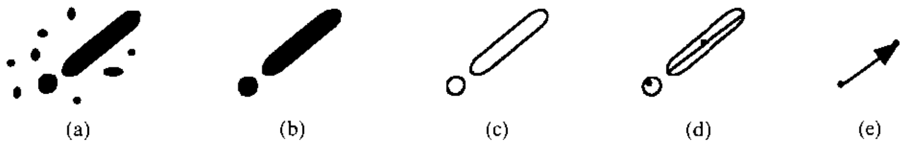
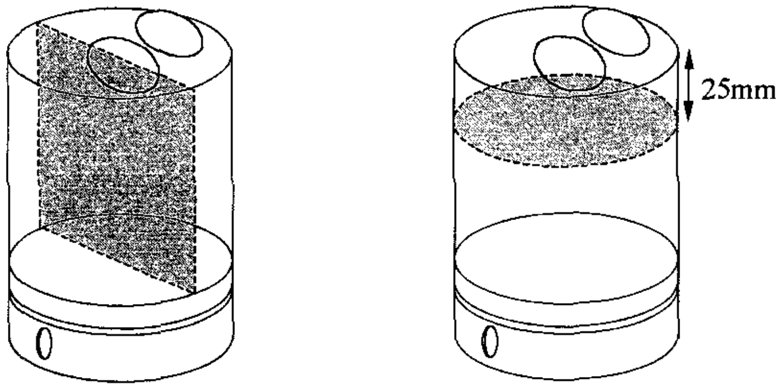


Fig. 4 Configuration of intake port


(a) (b) (c) (d) (e)

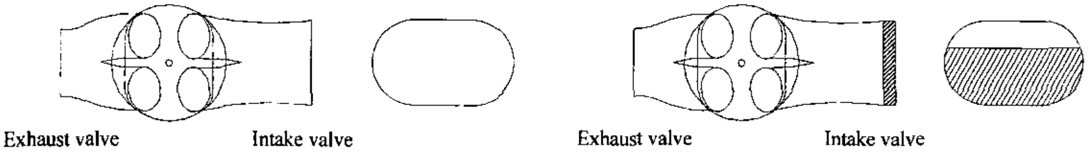
(a) Original image (b) The image after filtering and thresholding
(c) Boundary detection (d) Calculation of centroid
(e) Obtaining velocity vector

Fig. 3 Procedures for obtaining velocity vector

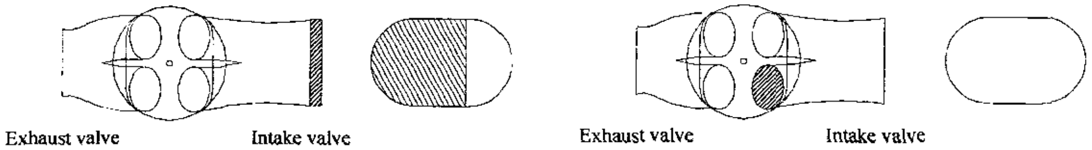


(a) Tumble measurement section (b) Swirl measurement section

Fig. 5 Test section for measurements



(a) Base engine (b) 30% TIV condition



(c) 30% SIV condition (d) 1 valve deactivated condition

Fig. 6 Inlet flow conditions of test engine

Table 2 Test section for measurements

Crank angle	Downward distance from TDC
BBDC 60°	71mm
BDC	88mm
ABDC 60°	71mm
ABDC 120°	27mm

of the test engine. In this study, four conditions of intake flow such as base engine, SIV and TIV condition, and one valve-deactivated condition were applied to investigate the improvement of swirl and tumble flow in the engine cylinder.

From the experimental result of in-cylinder flow field measurements, the tumble ratio Rt was estimated by the relation between fluid angular momentum Mact and angular momentum induced from the angular velocity of the crankshaft Msolid as shown in Eq. (1).

$$R_t=\frac{M_{act}}{M_{solid}}=\frac{\sum m\cdot v_t\cdot r}{\int_0^{B/2}(2\rho\pi r)\,r^2\omega\cdot\Delta r}=\frac{\sum m\cdot v_t\cdot r}{\frac{m\omega}{2}(D/2)^2} \tag{1}$$

where, R_t is the tumble ratio, m is the mass of the each fluid element, v_t is the velocity in direction of tumble flow, and r is the distance from the center of the test section to fluid element. And ρ is the fluid density, ω is angular velocity of solid body, and D is cylinder bore. Thus the equation for tumble ratio R_t becomes

$$R_t=\frac{8\cdot\sum(vx-uy)}{n\omega D^2} \tag{2}$$

where, n is the number of grid.

4. Results and Discussion

Figure 7 shows the effect of intake flow conditions on the intake airflow rate at various engine speeds. From this result, it is known that the intake airflow rate increases as the engine speed increases and the amount of an air flow rate depends on the valve opening area. As the valve opening area is larger, the airflow rate increases. Even though the valve opening area of SIV and TIV cases is same, the airflow rate of SIV is more than that of TIV. Thus, SIV is more effective in a view of an airflow rate.

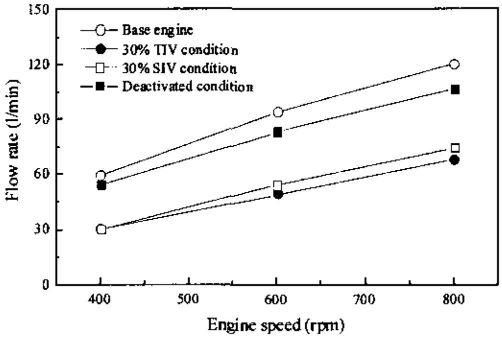
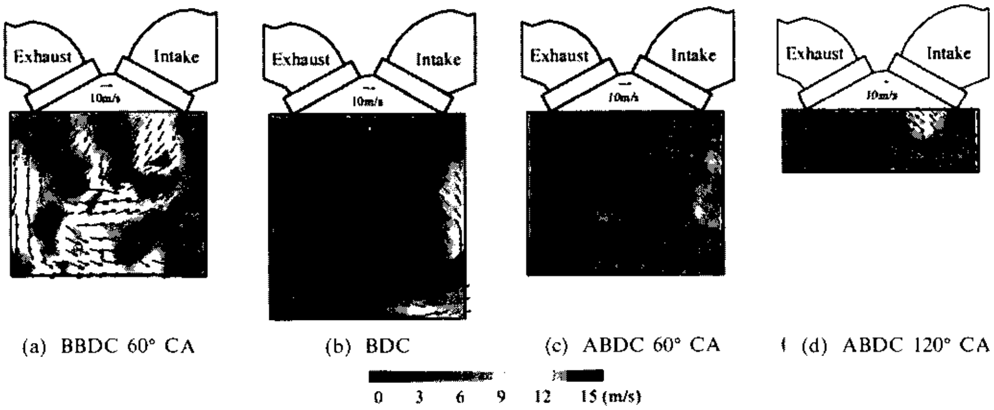


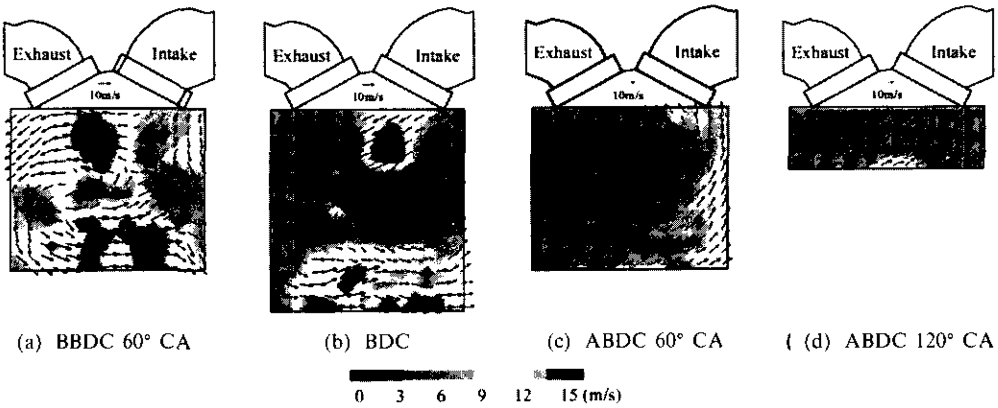
Fig. 7 Effect of inlet flow condition on the flow rate according to the engine speed

Figure 8 shows the flow field in the cylinder of base engine under the two valves operation under 600 rpm of engine speed. At an induction stroke, it can be seen that the intake air is introduced producing counterclockwise rotational flow and becomes main tumble motion when piston is at BDC. This main tumble motion maintains its direction at compression stroke, but the radius of rotation becomes smaller.



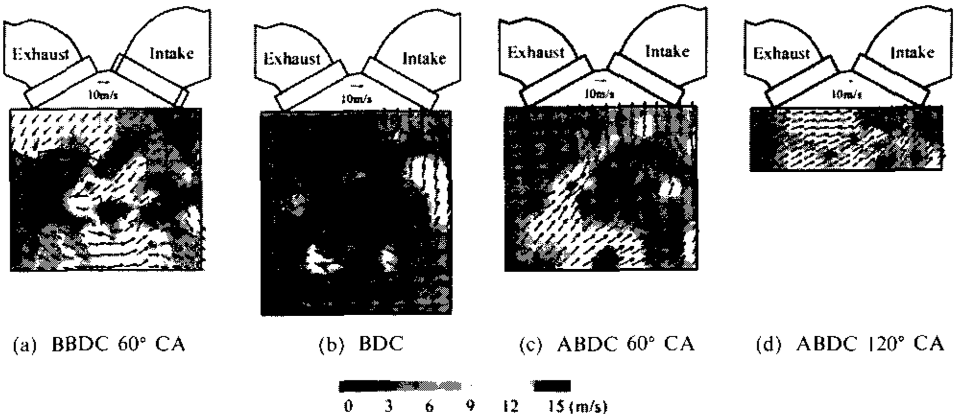
(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 8 In-cylinder tumble flow motin of base engine



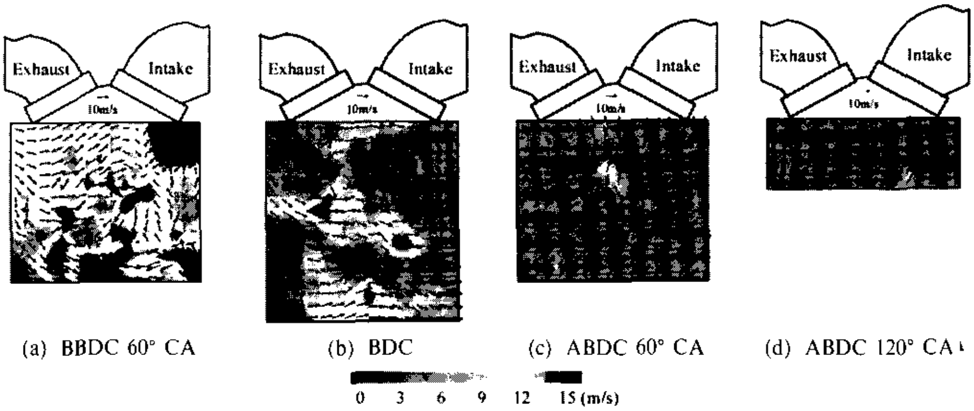
(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 9 In-cylinder tumble flow motion under 30% tumble intensifying valve condition



(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 10 In-cylinder tumble flow motion under 30% swirl intensifying valve condition



(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 11 In-cylinder tumble flow motion under 1 valve deactivated condition

Figure 9 indicated in-cylinder tumble motion at 30% TIV condition. In this condition, the counterclockwise rotational flow became intensified and the center of rotation moved to the center of vertical test section. This is due to that the reduced area of intake port induces faster airflow into the cylinder through the upper section of intake port. From the results, we could found out that the TIV is an effective tool to intensify in-cylinder tumble motions.

In order to discuss the influence of swirl flow, Fig. 10 shows the flow behavior of in-cylinder gas for the swirl intensifying condition using the SIV as shown in Fig. 6(b). In these flow fields, the flow pattern in the cylinder is similar to that of tumble intensifying condition, but the flow field at compression indicated the upward flow behavior because of dissipation of tumble vortex by the action of piston.

Figure 11 shows the behavior of in-cylinder gas flow with the deactivated condition of one valve. For the case of swirl intensifying condition, the flow field in the cylinder gas showed the strong horizontal flow compared with the swirl intensifying condition as shown in Fig. 10.

Figure 12 shows the calculation results of tumble ratio due to the various intake flow conditions using the Eq. (2). During the induction stroke of piston, the flow behavior at 60 degree before BDC shows that the velocity vectors of the gas flow display the initiation of counter clockwise tumble flow. At the bottom dead center, the flow pattern shows the clear tumble flow through the cylinder vertical plane. It can be observed in Fig. 8(c) and

(d) that the tumble structure was reduced by the radius of tumble flow while the tumble structure continued to the 120deg crank angle after the BDC. From the inspection of Fig. 8, the base engine with two intake valves indicated that the tumble flow is generated in the cylinder even if the engine does not use tumble intensifying valve. As indicated in Fig. 12, the tumble ratio has the maximum value at the bottom dead center for the different four inlet conditions. In the case of tumble intensifying valve, the tumble ratio is 1. 35 times larger than that of base engine, and the TIV has an effect on the intensifying of tumble flow. It is observed that the one valve deactivated engine with a swirl intensifying valve shows the improvement of swirl flow compared to the base engine because it has lower tumble ratio. When the ratio of the stroke to bore was 0.32 at the period of 120deg crank angle after bottom dead center, the tumble ratio is reduced to 22% at the bottom dead center.

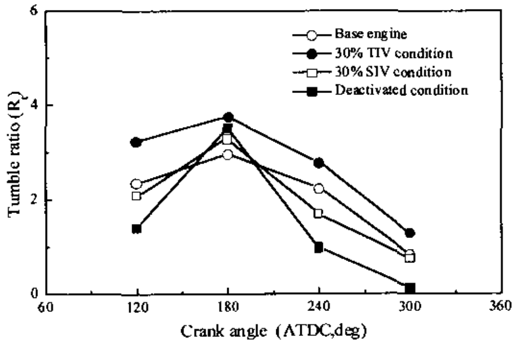


Fig. 12 Effect of inlet flow condition on the tumble ratio

Figure 13 shows in-cylinder swirl motion of the base engine under operation of two valves at 600 rpm of engine speed. The flow field was measured at 27 mm downward distance from the top dead center as shown in Fig. 5. The flow field of base engine represents the symmetric swirl motion at the horizontal centerline. And the rotational flow produced at the exhaust valve side fade away as intake and compression stroke proceeds.

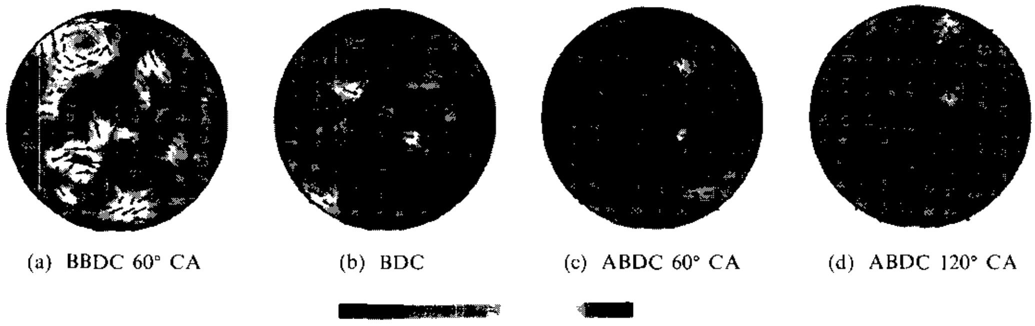

(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 13 Flow field in cylinder of the base engine

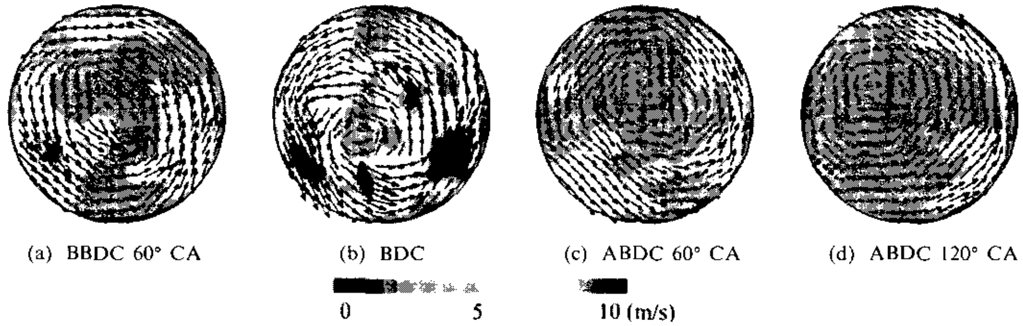

(a) BBDC 60° CA (b) BDC (c) ABDC 60° CA (d) ABDC 120° CA

Fig. 14 Flow field in cylinder under 1 valve deactivated condition

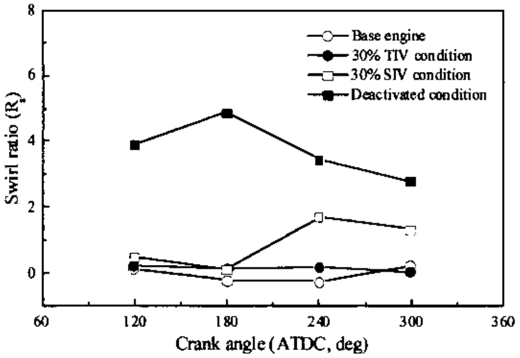


Fig. 15 Effect of inlet flow condition on the tumble ratio

Figure 14 shows in-cylinder flow field under 1 intake valve deactivated condition. In this inlet condition, the air is inducted with strong rotation flow motion. This swirl flow is increased at the bottom dead center and maintains to the 120 deg crank angle after bottom dead center. And the center of rotational flow corresponds to the center of cylinder. This result indicated that the 1 intake valve deactivated condition is effective to improve in-cylinder swirl motion.

Figure 15 shows the calculation results of swirl ratio from current PTV results according to the variation of intake flow conditions. As shown in this figure, when 30% SIV was installed, the swirl ratio was increased 3.6 times at 60deg crank angle before the BDC. At 120deg crank angle after the BDC, the swirl ratio was increased 3.5 times than at 60deg crank angle before the BDC. In the case of 1 valve deactivated condition, the swirl ratio is increased obviously and at 120deg crank angle after the BDC, it is 2.6 times than that of 30% SIV condition.

5. Conclusions

The in-cylinder flow characteristics under various intake flow conditions were investigated by using single-frame particle tracking velocimetry in four valves S.I. engine. Based on the experimental results, the following conclusions were obtained.

(1) In the case valve opening area of the swirl intensifying valve and tumble intensifying valve is same, the air flow rate of the swirl intensifying valve is more than that of tumble intensifying valve. Therefore, the swirl intensifying valve is more effective in terms of an airflow rate.

(2) The tumble ratio of in-cylinder flow field has the maximum value at the bottom dead center for the different four inlet conditions. In the case of tumble intensifying valve, the tumble ratio is 1.35 times larger than that of base engine, and the TIV has an effect on the intensifying of tumble flow.

(3) When the ratio of the stroke to bore was 0.32 at the point of 120 deg crank angle after bottom dead center, the tumble ratio is reduced to 22% of the tumble ratio at the bottom dead center.

(4) In the case of 1 intake valve deactivated condition, the swirl flow is increased at the bottom dead center and maintained to the 120 deg crank angle after bottom dead center. Thus this flow condition is effective to improve in-cylinder swirl motion.

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