In-cylinder Air Swirl in an Overhead Valve Type Engine*

— Effects of Intake Ports —

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The effects of two types of intake ports on swirl characteristics were investigated in motoring and steady flow tests. One port was a directional type and the other was a spiral type. Two methods for predicting swirl speeds were also checked for their availabilities.

The conclusions are as follows: Velocity profiles of swirl at various places inside cylinder are not similar to each other but variable remarkably. The swirl speed distribution along the cylinder axis, however, is almost uniform. A swirl producing ability of intake port is not necessarily assessed by the result of a steady flow test using a paddle wheel anemometry. Predicted swirl speeds are affected by intake ports.

Key Words : Internal Combustion Engine, Swirl, Intake Port Design, Velocity Distribution, Motoring, Steady Flow, Simulation

1. Introduction

In recent years, a modeling of in-cylinder airflow of internal combustion engine has appeared on the scene because of the decisive effect of swirl on combustion. To the modeling, the assessment of an initial flow condition like a velocity profile of swirl at the beginning of compression process is indispensable. Detailed measurements of swirl velocity profiles at various places inside cylinder, however, have never been made in an actual operation of engine.

In addition, the effects of intake port system on swirl characteristics have been little investigated in an actual operation, too. Monaghan-Pettifer⁽¹⁾ measured velocity profiles of swirl by means of L.D.V. for three types of intake ports and compared the swirl speeds predicted using the results of a steady flow test. However, they did not necessarily describe the complicated behaviors of swirl.

In this study, using two types of intake ports whose swirl producing processes differed from each other, the effects of the intake ports on swirl characteristics were investigated in motoring and steady flow conditions. Two methods for predicting swirl speeds from the results of steady flow tests were also checked for their availabilities and the effects of intake ports on the predictions were investigated.

2. Experimental Apparatus and Procedure

A single cylinder four-stroke cycle engine ($\phi 100 \times 110$ mm) was used in a motoring condition. Schematic diagrams of intake ports used in this study are shown in Fig.1 (b) and (c). The former shows a directional type of port (designated D type) which had an intake duct directed tangentially to the cylinder in order to impart the intake air with an swirl moment. The latter shows a spiral type of port (designated S type) which was constructed spirally and in which air was compelled to make about one and a half rotation around intake valve stem inside the port. D type port had an intake valve with a shroud of 9 mm high. S type one had an intake valve without the shroud. Both valves have a diameter of 50 mm, a seat angle of 30 degrees and the maximum valve lift of 10 mm.

A schematic diagram for measuring airflow velocities of swirl in motoring the engine is shown in Fig.1(a). The velocities were measured by an electric spark discharge method⁽²⁾. A measuring plug was inserted into the cylinder to obtain velocity profiles of swirl, being set in a position in which the velocities were detected. All the measurements were made at a given period of intake valve closure, at which the velocities were presumed parallel to the horizontal plane that was normal to the cylinder axis. Since the measured velocities considerably varied from cycle to cycle, then a proper velocity at each station was represented by the mean value of frequency diagram of velocities which was obtained by computing 256 data of the duration time from the beginning of spark to the arrival of ionized path at a receiver. To prevent the collision of the measuring plug with the piston, an elongated piston having a

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slit (10×33×110 mm) was mounted on the engine piston. Velocity profiles were obtained on the radii of A, B, C and D in each measuring section of No.1-4 as indicated in Fig.1 (a) and (b). Top clearance was 21.5 mm and compression ratio was 6.12 without counting the slit volume. The surface of piston crown was flat except at the slit. The engine was motored at a given speeds of 600, 800 and 1000 rpm without throttling.

Fig.1(a) Schematic diagram of measuring method of swirl velocity in motoring

Fig.1(b) Directional type of intake port (D type)

Fig.1(c) Spiral type of intake port (S type)

In this study, steady flow tests were also made using a paddle wheel and a hot wire anemometer. Two types of paddle wheels shown in Fig.2 were used. The large paddle wheel was currently used to measure a swirl speed. The small one was to measure the rotational speed of a swirling jet issued from the intake valve, being installed axi-symmetrically to the intake valve and to the cylinder by shifting the cylinder. Air-flow velocities of swirl and jet were measured by means of a hot wire anemometer. The measuring techniques are ones described in the references of (3) and (4).

3. Results

3.1 Swirl in steady flow tests

Figure 3 shows the test results obtained by using the large paddle wheel. In D type the paddle speed increases with an increasing X/D and reaches the maximum at X/D of about 1.5. In S type, however, it becomes the maximum at X/D of 0.6 and decreases steadily with an increase of X/D . In the diagram X is the distance measured from the surface of cylinder head and D is the cylinder diameter. Angular momentums of swirl Ω are also indicated, which are calculated from the following equation by using axial components of airflow velocities measured by the hot wire anemometry at X/D of 1.5.

$$\Omega = \int_0^R 2\pi \rho_s \omega_1 \bar{u}_z r^3 dr \dots \dots \dots (1)$$

where, ρ_s is the density of in-cylinder air, R is the cylinder radius, r is the distance from cylinder axis, ω_1 is the angular speed of large paddle wheel and $\bar{u}_z$ is the mean value of the axial components of velocities

Fig.2 Paddle wheels used in steady flow test

Fig.3 Paddle speed distribution along cylinder axis and angular momentum (steady flow, valve lift 10 mm)

measured at four positions on each circumference having a radius of r . The distributions of $\bar{U}_z$ are shown in Fig.4, in which a reverse flow is observed in the neighborhood of cylinder axis in D type port.

Although, as shown in Fig.3, the maximum paddle speeds in both types of ports are not distinguishable from each other, the angular momentum passing the paddle wheel plane in D type is 1.26 times larger than that in S type. This is caused by the higher velocities appearing in the vicinity of cylinder wall in D type as shown in Fig.4.

The small paddle wheel installed at the valve exit has no rotation in D type, but in S type it has a high rotational speed and rotates at 1.59 times higher speed than that of the large paddle wheel installed at X/D of 0.6. The angular momentum of a swirling jet passing the intake valve is calculated by the following equation and in Fig.3 the calculated result is indicated at X/D of 0.05 where the small paddle wheel is located.

Fig.4 Profile of axial velocity (steady flow, air flow rate 28.8 L/s)

Fig.5 Effect of intake valve lift on paddle speed (steady flow)

$$Q = \dot{m} R_v^2 \omega_s \quad (2)$$

where, $\dot{m}$ is the airflow rate, R_v is the radius of pitch circle of the small paddle wheel and ω_s is the angular speed of the small paddle wheel. The angular momentum of the swirling jet determined by Eq.(2) is higher than that obtained by the large paddle wheel installed at X/D of 1.5 in the original arrangement of valve and cylinder. These facts imply that the swirl in S type is caused by a swirling jet passing the valve and so the angular momentum of swirl is the highest at the valve exit. Therefore, the lower paddle speeds appearing in the range of $X/D < 0.5$ in Fig.3 result from the fact that the large paddle wheel is not effectively driven by the swirling jet because of no invasion of the jet into the whole cylinder.

On the contrary, in D type, the air jet issued from the valve in the direction tangential to the cylinder has no swirl component, but it contributes to a swirl production after its impingement upon the cylinder wall(3). Therefore, the most effective swirl is formed farther downstream than in S type. Figure 5 shows the effects of valve lifts on paddle speeds. The paddle wheel is installed at X/D of 1.5 and 0.6 for D and S types, respectively. In D type the paddle speed changes with the change of the valve lift. In S type, however, it is little affected by the valve lift.

3.2 Swirl in motoring test

Figures 6 and 7 show the velocity profiles of swirl. The velocity U_c is the mean value determined from the frequency diagram described in Chapter 2. As clearly shown in the diagrams, the velocity profiles are considerably changed and they suggest that the velocity profile of swirl can not be represented by only a single profile.

A swirl center on each measuring section is estimated in the following way. In Fig.8, denoting the swirl center by the point of O, then the air flows passing the sections of OX and OY having a unit height are equal to each other. When the velocity profiles on the lines of OX and OY are presumed by the linear curves of OP and OQ, the velocities of XP and YQ are determined from the relation of $\triangle OXP \cong \triangle OYQ$. Subsequently, the velocity at the point of Ai will be approximated as follows. Assuming a circular flow tube passing the circle of $X_i A_i Y_i$ along which a flow quantity is constant, then the velocity at the point of Ai is expressed by $(OY_i/OA_i)Y_i Q_i$ and the velocity component normal to the diameter of AC is $(OY_i/OA_i)Y_i Q_i \cos \alpha$. The resultant velocity profiles on the diameters of AC and BD are shown in Fig.8. The center of an actual swirl is able to be estimated by referring to many profile diagrams determined by changing the position of O.

The swirl centers estimated at the engine speed of 1000 rpm are shown in the diagrams of Section 4 in Figs.6 and 7, where the figures 1 to 4 indicate the positions of the swirl centers in Sections 1 to 4, respectively. The swirl centers move in the

Fig.6 Velocity profile of swirl
(D type in motoring)

Fig.7 Velocity profile of swirl
(S type in motoring)

directions indicated by the arrows as the section descending downward from 1 to 4.

In D type the swirl center moves along an intake air jet. This will be explained by the fact that the jet width, that is measured along the circumference of valve head as shown later in Fig.13, becomes narrower due to the jet's impingement upon the cylinder wall and the width becomes narrower farther more downstream. Since the air is rotated around the position which corresponds with the inner side of the jet width, the swirl center moves in the direction along which the jet width becomes narrower.

In S type, since swirl is generated at valve exit as described in the preceding section, the swirl center in Section 1 is located in the vicinity of the valve center. Descending downward, however, the swirl center moves toward the cylinder center due to the swirl formation like a solid-body rotation. Much higher velocities revealed on the radius A in Section 1 result from the decrement of the flow passage between the cylinder wall and the intake valve.

Fig.8 Model of velocity profile for determining the swirl center

Fig.9 Mean profile of swirl velocity (n = 1000 rpm in motoring)

The distributions of swirl velocities of $(\bar{U}_c)$ are shown in Fig.9, where $(\bar{U}_c)$ is the average of four velocities of $\bar{U}_c$. In the upper part of the diagram, dimensionless velocity profiles are shown which are obtained by dividing $(\bar{U}_c)$ with $(\bar{U}_c)_m$ that is the maximum of $(\bar{U}_c)$ on each profile. It is recognized in the upper diagram that the swirls in Sections 2,3 and 4 in D type are more similar to a solid-body rotation than those in S type. In Section 1, the profile in S type looks more similar to a solid-body rotation than that in D type, but this is caused by the much higher velocities revealed on the radius of A. Therefore, the dimensionless velocity profile obtained by averaging the velocities except the radius of A is almost the same as that in D type.

The distributions of swirl speeds of ω_c determined by the following equation (4) are shown in Fig.10.

$$\bar{\omega}_c = \frac{4}{r_2^2 - r_1^2} \int_{r_1}^{r_2} (\bar{u}_c) r^2 dr \quad (3)$$

where, r_2 and r_1 are the longest and the shortest radii at which velocities are measured, respectively. ω_c hardly changes along the cylinder axis, but seems to have a tendency to decrease in the neighborhood of the cylinder-head.

Swirl speed of $(\bar{\omega}_c)$ which is the representative speed of an in-cylinder air swirl is determined by the following equation.

$$(\bar{\omega}_c) = \frac{\sum \bar{\omega}_{ci} H_i}{\sum H_i} \quad (4)$$

where, the in-cylinder air is divided into four layers each of which contains the measuring section, ω_{ci} is a swirl speed in each section and H_i is the height of each layer. The results are summarized in Table 1 and Fig.11. (ω_c) increases proportionally

Fig.10 Axial distribution of swirl speed (in motoring)

to an increasing engine speed. The ratios of $(\bar{\omega}_c)$ in D type to that in S type are around 1.26 which is the same as the ratios of the angular momentum of steady swirl in D type to that in S type as described in Fig.3. These results suggest that a swirl producing ability of intake port can not be assessed only by the steady flow test using a paddle wheel but it has to be examined by the angular momentum of swirl.

4. Checks of Swirl Speed Predictions

Two methods for predicting swirl speeds are checked for their availabilities. One is Fitzgeorge-Allison's method⁽⁵⁾ using a paddle wheel and the other is Tanabe et al's⁽³⁾ using a velocity distribution of jet issued from valve. In general, the swirl speed at intake closure is predicted by the following equation, neglecting the initial angular momentum at the beginning of intake process.

$$\omega_c^* = \int_0^t \Omega dt / I_c \quad (5)$$

where, ω_c^* is the predicted swirl speed at intake valve closure, Ω is the angular momentum of air flowing into cylinder, t is time and I_c is the inertia moment of in-cylinder air at intake valve closure. In Fitzgeorge-Allison's method, $\Omega = \dot{m} \omega_1 R^2 / 2$ and the predicted value is designated by $(\omega_c^*)_{FA}$. In Tanabe et al's, $\Omega = 0.73 R W \cos \phi \cos \psi$

and the predicted one is by $(\omega_c^*)_T$. $\dot{m}$ is the flow rate of intake air at each crank angle, ω_1 is the paddle wheel speed, W is the momentum of the air passing the intake valve at a given valve lift, ϕ is the angle at which W is inclined to a horizontal plane that is normal to cylinder axis. ψ is the angle at which the horizontal component of W crosses the cylinder wall. $\dot{m}$ is determined by the following equation neglecting heat transfer from cylinder wall.

$$\dot{m} = \frac{\rho_0}{\kappa p_0} \left(\kappa p_c \frac{dV_c}{dt} + V_c \frac{dp_c}{dt} \right) \quad (6)$$

where, ρ_0 and p_0 are the density and the pressure of ambient air, respectively, κ is the ratio of specific heats and V_c and p_c are the in-cylinder volume and its pressure, respectively. p_c was measured by means of a piezo type transducer. The volumetric efficiency determined by integrating the distribution curve of $\dot{m}$ differed by only three percent from that measured by an orifice flow meter. Steady flow conditions were determined by $\dot{m}$ and valve lift curve.

The predicted results of $(\omega_c^*)_{FA}$ and $(\omega_c^*)_T$ are shown in Fig.12 where they are compared with the mean swirl speeds of $(\bar{\omega}_c)$ shown in Fig.11. In D type, $(\omega_c^*)_T$ falls into $(\bar{\omega}_c)$, but $(\omega_c^*)_{FA}$ is much smaller than $(\bar{\omega}_c)$. In S type $(\omega_c^*)_{FA}$ is a little smaller than $(\bar{\omega}_c)$, but $(\omega_c^*)_T$ becomes almost zero.

Fig.11 Relation between swirl speed and engine speed (in motoring)

Fig.12 Comparison of predicted swirl speed with actual speed

Table 1 Swirl speed of in-cylinder air (in motoring)

n	r.p.m.	600	800	1000
D type port $(\bar{\omega}_c)$, rad/s		260	342	420
S type port $(\bar{\omega}_c)$, rad/s		190	265	335
D type $(\bar{\omega}_c)$ / S type $(\bar{\omega}_c)$		1.37	1.29	1.25

Fig.13 Velocity distribution around intake valve
(steady flow, valve lift 9 mm)

As clearly shown in Fig.13 which shows jet flow distributions around the intake valves, in S type a swirling jet is observed. In D type, however, air is prevented from flowing out of the valve by the shroud but the main jet is expected to produce a swirl after impingement upon the cylinder wall. In the diagram, $U \cos \delta$ is a horizontal component of the velocity of U having the angle of δ to the horizontal plane.

5. Conclusions

By using the directional type and the spiral type of intake ports (designated by D and S types, respectively), the effects of the difference between swirl producing processes on swirl characteristics were investigated. The results are as follows;

(I) Swirl measured at the period of intake valve closure in a motoring condition.

(1) The velocity profiles of swirls at various places inside the cylinder are widely variable and they suggest that a single profile cannot express the representative profile of swirl. The mean swirl speed in each measuring section, however, is distributed almost uniform except in the neighborhood of cylinder-head.

(2) A swirl in D type is more similar to a solid body rotation than that in S type.

(3) The swirl center in D type moves along the jet flow issued from intake valve. In S type, it moves from the vicinity of valve center toward the cylinder axis.

(4) The swirl speed increases proportionally with an increasing engine speed.

(II) Swirl in the steady flow test using a paddle wheel.

(1) The paddle wheel plane where the paddle speed becomes maximum is located farther upstream in S type than that in D type because of the air rotating at the valve exit in S type.

(2) The valve lift has a considerable influence on the paddle speed in D type, but it has little influence in S type.

(III) Results of the motoring and the steady flow test.

(1) Even if there is no difference between the paddle speeds determined using different types of intake ports, swirl speeds in a motored engine are not equal to each other and depend on angular momentums passing a paddle wheel plane in a steady flow test. This implies that the swirl producing ability of intake port can not be assessed only in a paddle wheel test.

(2) The swirl speed predicted by using the jet flow distribution around the intake valve agrees with the swirl speed measured in D type, but in S type the predicted value is almost zero and so this prediction can not be applied to S type port. The swirl speed predicted by using a paddle wheel is lower than the swirl speed measured in both types of ports, but in S type the difference is a little.

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