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Swirl in a Four-Stroke Cycle Engine Cylinder*

By Seiichi TANABE**, Yoshisuke HAMAMOTO*** and Shunichi OHIGASHI****

Air swirl speeds were measured in a motoring and firing run of a direct injection diesel engine, and the results were compared with those in steady flow tests. Experimental results show that, during the combustion period, the swirl speed in the firing condition is almost the same as that in motoring. There is a linear correlation between the swirl speed predicted from the results of the steady flow tests and that measured at top dead center in motoring. Therefore, it is recognized that the steady flow test is available to evaluate a swirl producing ability of inlet system of engine.

The reduction in swirl speed during the compression period was estimated by considering the friction forces acting at the air-solid interfaces.

1. Introduction

In an internal combustion engine, particularly in a direct injection diesel engine, the rotational movement of cylinder charge called swirl has a predominant effect on combustion process, and thus on exhaust emissions. For a spark ignition engine, swirling flow of cylinder charge is now to be adopted to minimize the drivability penalty with EGR and charge dilution which are effective for controlling nitrogen oxide emission⁽¹⁾⁽²⁾. Therefore, at present, the evaluation of the swirl producing ability of an inlet system of engine is becoming the key subject in a design and development of engine.

Up to the present, the swirl producing ability of inlet system has been generally determined by the results of a steady flow test using a paddle wheel⁽³⁾⁻⁽⁶⁾. Fitzgerald and Allison⁽⁴⁾ proposed a method for predicting the swirl speed of an actual engine from the results of steady flow test. No attempt, however, has been made to confirm experimentally the usefulness of their proposal.

In this paper, air flow velocities of swirl in an engine cylinder were measured in both conditions of motoring and firing and also in a steady flow condition. The test results show that there is a linear correlation between the swirl speed in the engine cylinder and that predicted in the steady flow test, and also that the swirl speed during the combustion period in the firing condition shows no remarkable difference from that in the motoring one.

The reduction in swirl speed during the compression stroke of engine was also estimated by considering the shearing force due

to the skin friction acting at the air-solid interface.

2. Experimental method and set-up

The inlet system used in this study was of a four-stroke cycle direct injection diesel engine ($\phi 125 \times 110$ mm, shrouded inlet valve, opening at 24 degs. before TDC and closing at 48 degs. after BDC). Schematic diagrams of the engine port are shown in Fig. 1. By changing the shroud position α , various intensities of swirl were able to be produced. In this test, α was changed 0, 30 and 60 degs.. All of the tests were made by keeping the engine speed constant at 900 rpm.

Fig.1 Schematic diagram of test engine

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** Lecturer, Faculty of Engineering, Tottori University, Koyama-cho, Tottori.

*** Assistant Professor, Faculty of Engineering, Kyoto University.

**** Professor, Faculty of Engineering, Kyoto University.

2.1 Steady flow test of swirl

Figure 2 shows the steady flow test rig. Air was blown into a lucite cylinder having the same diameter as the actual engine and having an extended length of 400 mm. Swirl speeds were measured by two methods. One used a paddle wheel anemometer and the other did a hot wire anemometer (constant temperature method using a tungsten wire of $\phi 5\mu\text{m} \times 1\text{mm}$).

The configuration of paddle wheel (Fig. 2) is similar to that used by Pischinger⁽³⁾. The paddle wheel consisted of plastic vanes and a ball bearing. The rotational speed of paddle wheel was measured by a photo-electric technique. The paddle wheel was mounted at a distance of 190 mm (1.52 times as large as the cylinder diameter) from the cylinder head. At that position, the maximum rotational speed of the wheel was obtained.

Using the hot wire anemometer, the air flow velocity and its direction were determined by the following procedure. The hot wire probe was inserted into the cylinder towards the diametric directions which were indicated by A and B in Fig.2. The probe was able to rotate around its axis. By the rotating angle, at which the maximum output of the anemometer was obtained, the flow direction β could be determined, and by the maximum output, the flow velocity u was determined (β and u are defined in Fig.3(b)).

2.2 Prediction of swirl speed in engine cylinder

Prediction of the swirl speed of an actual engine from the results of the steady flow tests was made by the following procedure. During the inlet stroke of engine, the mass flow, the valve opening, and consequently the swirl production vary from instant to instant. If two identical heads were fitted, one to a rig for steady state testing and the other to the engine, then for a given inflow and valve opening, the ability to produce the angular momentum would be assumed the same in each case. Since the velocity profile of swirl is to be estimated as a solid body rotation (refer to the results shown in Figs.10(b) and (c)), then in the steady flow the angular momentum of Ω_i passing a transverse plane of cylinder per unit time is given by

$$\Omega_i = \int_0^{R_1} \int_0^{2\pi} \rho_c u_{\theta} r^2 d\varphi dr \quad (1)$$

where $u_r = u \cos \beta$, $u_{\theta} = u \sin \beta$ and ρ_c is air density in cylinder and φ is rotating angle around cylinder axis and R_1 and r are denoted in Fig.3(a). The angular momentum Ω_i would be equal to the instantaneous rate of acquisition of angular momentum possessed by the engine cylinder charge at the crank angle position at which the corresponding condition of flow and valve opening takes place⁽⁴⁾, i.e.

$$d(I_c \omega_c)/dt = \Omega_i \quad (2)$$

where I_c is a total inertia moment of cylinder contents and ω_c is an angular velocity of swirl in an engine cylinder.

$I_c = \frac{1}{2} m_i (R_1^2 S + R_2^2 h) / (R_1^2 S + R_2^2 h)$. In this expression R_2 , S and h are denoted in Fig.3(a), and m_i is the total amount of mass of the cylinder charge at the time t , i.e. $m_i = \rho_0 V_c + m$,

Fig.2 Schematic diagram of steady flow test rig

Fig.3 Definition of symbols

where

$m = \pi R_1^2 \int_0^t \rho_c u_p dt$. Here, V_c and U_p are a clearance volume at TDC and a piston speed, respectively, and the density of cylinder charge at TDC of inlet stroke is assumed to equal the atmospheric air density ρ_0 . ρ_c was determined approximately by using the following hypotheses. The mass flow rate of the air aspirated at an arbitrary crank-

angle in the inlet stroke of engine is equal to that aspirated steadily by a piston moving at the constant speed, the same as the value of u_p appearing at the particular crank-angle. The change in state of the air passing the inlet valve is also presumed to be isentropic. Thus,

$$\rho_c = \rho_0 \left\{ 1 - \frac{\kappa-1}{2\kappa} \frac{\rho_0}{P_0} \left(\frac{\pi R_1^2 u_p}{\mu f} \right)^2 \right\}^{1/(\kappa-1)} \quad (3)$$

where κ is a ratio of specific heats and P_0 is an atmospheric pressure and μ is a flow coefficient of inlet system and f is a valve opening area. The cylinder pressure P_c is expressed by

$$P_c = P_0 \left\{ 1 - \frac{\kappa-1}{2\kappa} \frac{\rho_0}{P_0} \left(\frac{\pi R_1^2 u_p}{\mu f} \right)^2 \right\}^{\kappa/(\kappa-1)} \quad (4)$$

Comparison of the results calculated by Eq. (4) with those measured in a motoring run of engine is made in Fig. 4, which shows that Eq. (3) is approximately available for the establishment of the steady flow test condition, excepting at a high speed of engine. μ was measured in a steady state. In this study, the inlet period is regarded as the time interval between TDC (t_T) and BDC (t_B). Therefore, the angular velocity of swirl ω_c^* predicted at the end of inlet is able to be obtained by integrating Eq. (2) from t_T to t_B . Namely, $\omega_c^* = \int_{t_T}^{t_B} \Omega_c dt / I_c$, where

the angular momentum possessed by the cylinder content at TDC of inlet stroke is assumed to be negligible. Hence,

$$\omega_c^* = \frac{4\pi \int_{t_T}^{t_B} \int_0^{R_1} \rho_c \bar{u}_\phi \bar{u}_z r^2 dr dt}{\left(\rho_0 V_c + \pi R_1^2 \int_{t_T}^{t_B} \rho_c \bar{u}_\phi dt \right) (R_1 S_B + R_2^2 h)} \quad (5)$$

where $\bar{u}_\phi$ and $\bar{u}_z$ are the mean values of $u_\phi (= u \cos \beta)$ and $u_z (= u \sin \beta)$ measured in a steady state, respectively (refer to Fig. 3(b)), and S_B is a distance from the cylinder head to the piston face at BDC. By steady flow testing under various inlet conditions of valve openings and mass flow rates $\dot{m}$ ($= dm/dt$) corresponding at each crank-angle position of engine, ω_c^* is obtained by Eq. (5).

In this study, three types of ω_c^* were considered, they were denoted by $(\omega_c^*)_1$, $(\omega_c^*)_2$ and $(\omega_c^*)_3$ determined in the following three cases of (1) - (3), respectively.

(1): Using the results measured by means of the hot wire anemometer (refer to the section 3.1)

(2): $\bar{u}_\phi = \omega_s r$, where ω_s is the angular speed of the paddle wheel. $\bar{u}_z = (3\dot{m}/2\pi R_1^3 \rho_c) r$, where $\bar{u}_z$ is proportional to the cylinder radius.

(3): $\bar{u}_\phi = \omega_s r$ and $\bar{u}_z = \dot{m}/\pi R_1^2 \rho_c$, where $\bar{u}_z$ is uniform in the transverse plane of cylinder.

In the cases of (2) and (3), the angular momentums calculated by Eq. (1) are expressed by $(3/5)\omega_s \dot{m} R_1^2$ and $(1/2)\dot{m} \omega_s R_1^2$, respectively. The angular momentum in the case of (2) becomes 1.2 times as large as that in (3), namely $(\omega_c^*)_2 = 1.2(\omega_c^*)_3$.

2.3 Measurement of swirl speed in motoring

A schematic diagram for measuring the flow velocity of swirl in a motoring condition is shown in Fig. 5. Two types of pistons as shown in Figs. 9 (a) and (b) were used; one (C type) has a cavity and the other (F type) has no cavity. The air flow velocity was measured by means of the electric discharge method⁽⁷⁾⁻⁽⁹⁾. A bakelite disk plate installed

Fig. 4 Pressure changes in cylinder during inlet stroke

Fig. 5 Schematic diagram of measuring method of swirl velocity in motoring

Fig. 6 Combustion chamber of C' type piston at TDC (for measurement of swirl motion in firing)

with four sets of detectors measuring the flow velocities was mounted at the piston crown. Each set consisted of two discharge electrodes and a probe for detecting the discharge path. By changing the mounting position of the disk plate, the velocities were measured at sixteen positions in total as shown in Fig.9. The discharge electrodes and the probe are steel needles having a diameter of 0.6 mm. The lead wires connected to the discharge electrodes and the probe were taken out through the windows open at the piston skirt and the cylinder wall. The measurements were made at BDC and TDC of the compression stroke.

2.4 Measurement of swirl speed in firing

Swirl speeds were determined by analysing a series of flame photographs taken by a high-speed camera through a quartz window mounted on the piston crown. By measuring the angle δ of flame shifting around the cylinder axis during the crank-angle θ , the swirl ratio could be determined by δ/θ . Figure 6 shows the dimensions of combustion chamber at TDC. The firing test was made by rearranging the apparatus shown in Fig.5. The combustion chamber was observed in a mirror fixed at the lower part of the cylinder, and also illuminated by a xenon lamp for taking pictures of fuel spray. The fuel injection nozzle was of an automatic valve type having a single hole of 0.34 mm diameter. Controlling the amount of fuel supplied, the excess air factor was kept constant at four. Combustions were made for several cycles before and after taking the pictures. The film speed was 5000 frames per second. Hereafter, the combustion chamber shown in Fig.6 is named the type C'.

3. Results

3.1 Swirl speed predicted from the results of steady flow test

An example of u and β determined by the hot wire anemometer in the steady flow test is shown in Fig.7, where the valve lift is the maximum value of 13.01 mm. It will be seen in the diagram that u and β increase with an increase in radius from the cylinder axis. The air jet from the inlet valve streams downwards obliquely along the cylinder wall. The observation of the flow by means of a tuft method manifested the existence of upward motion in the neighbourhood of the cylinder axis, and its direction fluctuated from time to time. The output of

Fig.7 Distributions of air flow velocity and its direction (measured by hot wire anemometer steady flow, $\alpha=30$ degrees, valve lift 13.01 mm)

Fig.8 Changes of Ω_i during inlet stroke (steady flow)

the hot wire anemometer was also fluctuated, and then a reliable measurement of velocity and its direction was difficult in that place. Subsequently, for determining $(\omega_c^*)_1$, the velocity profile of $\bar{u}_\phi$ had to be modified so that the value of $\bar{u}_\phi$ at the cylinder axis might fall to zero, and the axial velocity $\bar{u}_z$ at the cylinder axis was determined such that the flow quantity calculated from the flow distribution in the transverse plane of cylinder was equal to that measured by a flow meter. $\bar{u}_z$ was determined to have a

Table 1

Shroud position α degrees	Type of piston	Steady flow test			Motoring test				
		$(\omega_c^*)_1$ rad/s	$(\omega_c^*)_2$ rad/s	$(\omega_c^*)_3$ rad/s	$(\omega_c)_{BDC}$ rad/s	$(\omega_c)_{TDC}$ rad/s	$(AM)_{BDC}$ $\times 10^{-6}$ kgms	$(AM)_{TDC}$ $\times 10^{-6}$ kgms	$(AM)_{TDC}$ $(AM)_{BDC}$
0	C	332.6	248.6	207.2	386.8	312.0	122.5	68.4	0.518
	F	302.8	226.3	188.6		209.5	136.8	75.1	0.549
30	C	350.7	264.0	220.0	395.2	340.5	125.1	68.3	0.546
	F	319.2	240.3	200.3		232.4	139.8	82.5	0.590
60	C	277.4	227.3	189.4	352.3	292.8	111.6	57.3	0.514
	F	252.5	206.9	172.4		187.0	124.6	64.8	0.520

linear distribution curve passing through the value measured near the cylinder wall. By using the values of $\bar{u}_\varphi$ and $\bar{u}_z$, the angular momentum flow Ω_i could be obtained, the results being shown in Fig.8. By integrating Ω_i from TDC to BDC and calculating Eq.(5), $(\omega_c^*)_1$ was determined. $(\omega_c^*)_2$ was also determined by the same procedure. $(\omega_c^*)_3 = (5/6)(\omega_c^*)_2$. The results are shown in Table 1.

(a) C type piston at TDC

(b) F type piston at TDC

3.2 Comparison between the swirl speeds in engine and those predicted in steady flow test

The flow velocities U_c measured in motoring are shown in Fig.9, in which the diagrams (a) and (c) show the velocity distributions measured in the C type piston at TDC and BDC, respectively. The diagram (b) shows the results with the F type piston measured at TDC. By taking the average of velocities measured at four positions on each circle, the velocity profile of swirl was determined as shown in Fig.10. In the C type, the flow velocities of swirl measured at BDC increase with an increase in the value of r/R_1 like a solid body rotation. And at TDC the flow inside the cavity has also the velocity profile like a solid body rotation, but out of the cavity, the rate of velocity increment with r/R_1 becomes small. Velocity profiles in the F type piston measured at TDC are also like a solid body rotation, but the velocities are lower than those in the cavity of C type piston.

The swirl speeds, $(\omega_c)_{BDC}$ and $(\omega_c)_{TDC}$, in motoring were determined by Eq.(6), which implies the angular velocity of an equivalent solid body rotation⁽¹⁰⁾.

$$(\omega_c)_{TDC}, (\omega_c)_{BDC} = \frac{4}{r_2^4 - r_1^4} \int_{r_1}^{r_2} \bar{u}_c r^2 dr \quad (6)$$

(c) C type piston at BDC

Fig.9 Velocity distribution of swirl (in motoring, $\circ \alpha = 30$ degs., $\bullet 0$ deg., $\square 60$ degs.)

where r_2 and r_1 are the longest and the shortest radius at which the velocities were measured, respectively. For the calculation of the swirl speed at TDC in the C type piston, r_2 was 34 mm. At BDC in the F type, however, the flow velocity and its direction strongly fluctuated from cycle to cycle excepting the neighbourhood of the cylinder wall. Hence, the reliable information of velocity could not be obtained at BDC in the F type. This fact implies that a flow inside cylinder is fairly influenced by a shape of piston. But the velocities in the neighbourhood of the cylinder wall were almost equal to those in the C type, therefore the swirl speed of the F type was assumed to be little different from that of the C type. Hereafter, the value of $(\omega_c)_{BDC}$ in the F type was assumed equal to that in the C type.

In Table 1 are shown the swirl speeds determined in the motoring condition and those predicted from the results of steady flow tests. The values of $(\omega_c^*)_1$, $(\omega_c^*)_2$ and $(\omega_c^*)_3$ obtained by Eq.(5) are smaller than those of $(\omega_c)_{BDC}$. The linear correlations, however, are established between $(\omega_c)_{TDC}$ and $(\omega_c^*)_1$, $(\omega_c^*)_2$ and $(\omega_c^*)_3$ as shown in Fig.11. These facts imply that the swirl producing ability of an inlet system of engine can be evaluated by the results of the steady flow test.

The angular momentum (AM) of cylinder charge obtained by the expression

$$2\pi\rho_c \int_0^{R_1} \bar{u}_c(S+h)r^2 dr$$

is also shown in Table 1. The angular momentum decreases to about 55 % during the compression stroke.

Figure 12 shows a series of photographs of flames inside the combustion chamber in the firing condition. In the photographs, the flames shifting around the cylinder axis can be observed well. Swirl speeds analysed by the shifting angles of the flame are indicated by ● in Fig.13.

3.3 Reduction in swirl speed during compression stroke

Providing that shear forces due to the skin friction acting at the fluid-solid interfaces reduce the angular momentum possessed by the cylinder charge⁽¹¹⁾, the restraining torque forces exerted by the walls

of the cylinder, the piston and the cylinder head would equal the instantaneous rate of reduction of the angular momentum. Then, presuming a solid body rotation of swirl, the following equation is obtained;

$$\frac{d}{dt}(I_c \omega_c) = -\pi\rho_c \omega_c^2 \left[(c_f)_{cyl} R_1^4 S + (c_f)_{cav} R_2^4 h + \frac{2}{5} (c_f)_{p-c} R_1^5 \right] \quad (7)$$

where $(c_f)_{cyl}$, $(c_f)_{cav}$ and $(c_f)_{p-c}$ are skin friction coefficients pertaining to the cylinder wall surface, the cylindrical wall surface of piston cavity and to the piston and cylinder head surfaces, respectively. The first and the second terms of the right side of Eq.(7) are the restraining torque forces due to shear stress exerted by the cylinder wall

Fig.11 Relation between $(\omega_c)_{TDC}$ in motoring and $(\omega_c^*)_1$, $(\omega_c^*)_2$ and $(\omega_c^*)_3$ in steady flow test

(a) C type piston at TDC

(b) F type piston at TDC

(c) C type piston at BDC

Fig.10 Velocity profile of swirl (in motoring operation)

and the cylindrical wall of the piston cavity, respectively. The third term consists of the torque forces by the cylinder head surface, the piston top surface and the bottom surface of the cavity, and is deduced as follows. These three surfaces are considered to be replaced with the surfaces of two circular walls having the same diameter as the engine cylinder. Then the restraining torque exerted by the wall surface is presumed to act on the circle of the radius $\bar{r}$, at which the geometrical mean shear stress $\bar{\tau}_w$ will be obtained :

$$\bar{\tau}_w \bar{r} = 2\pi \int_0^{R_1} \tau_w r^2 dr / \pi R_1^2 \quad (8)$$

The shear stress τ_w exerted on an arbitrary radius r is given by the equation,

$$\tau_w = \frac{1}{2} (c_f)_{p-c} \rho_c \omega_c^2 r^2 \quad (9)$$

Assuming $(c_f)_{p-c}$ and ρ_c to be uniform,

$$\bar{\tau}_w = \frac{1}{2} (c_f)_{p-c} \rho_c \omega_c^2 \bar{r}^2 \quad (10)$$

Substituting Eqs.(9) and (10) into Eq.(8) ;

$$\bar{r} = (2/5)^{1/3} R_1 \quad (11)$$

Hence, if τ_w is uniform over the wall sur-

face and equal to $\bar{\tau}_w$, the restraining torque forces exerted by the surfaces of two circular walls are expressed by $2\pi R_1^2 \bar{\tau}_w \bar{r}$. Substitution of Eqs.(10) and (11) into this expression gives $(2/5)\pi (c_f)_{p-c} \rho_c \omega_c^2 R_1^5$, that is the third term of the right side of Eq.(7). The local skin friction coefficient C_f was given by the following equation which applies for a flat plate.

$$c_f = 0.0577 R_e^{-0.2} \quad (12)$$

Reynolds number R_e was determined as follows ; for $(c_f)_{cyl}$, $(c_f)_{cav}$ and $(c_f)_{p-c}$, the velocity terms were $R_1 \omega_c$, $R_2 \omega_c$ and $(2/3) R_1 \omega_c$, respectively, and the length terms were

$\int_0^t R_1 \omega_c dt$, $\int_0^t R_2 \omega_c dt$ and $\frac{2}{3} \int_0^t R_1 \omega_c dt$, respectively. For the viscosity term of R_e , the eddy viscosity $\mathcal{E}$ was used. From Prandtl's mixing length theory, $\mathcal{E}$ is expressed by $l^2 |d\omega_c/dr|$. The mixing length l is given by $l = 0.4 (R-r)$ for a pipe flow. Applying this hypothesis to the flow inside the engine cylinder, the mixing length can be written as $l_2 = (2/15) R_2$ for inside the piston cavity and $l_1 = (2/15) R_1$ for inside the cylinder excepting the cavity. l_1

Fig.12 Observation of flame in firing operation (combustion chamber of C' type, $\alpha = 30$ degs., film speed of 5000 frames per sec.)

Fig.13 Reduction of swirl motion in compression and expansion stroke of engine

and l_2 are the geometrical mean values in the circular pipes having the radii of R_1 and R_2 , respectively. Because of a solid body rotation of swirl, $du/dr = \omega$. Substituting these expressions into Eq.(12),

$$(c_f)_{\omega_1} = (c_f)_{\omega_2} = 0.92577 \left(\int_0^t \omega_1 dt \right)^{-0.2} \quad (13)$$

and

$$(c_f)_{\omega_2} = 0.93031 \left(\int_0^t \omega_2 dt \right)^{-0.2} \quad (14)$$

By substituting Eqs.(13) and (14) into Eq.(7). Eq.(7) was numerically calculated under the initial condition that $\omega = (\omega_1)_{BDC}$ at $t = 0$.

Figure 13 shows comparisons between the theoretical analysis and the experimental data. In the diagram, the swirl ratio is defined by the ratio of the swirl speed to the engine speed. The experimental data in motoring agree with the calculated results. In Fig.13 the calculated results using the C' type piston applied to the firing test are also shown. In this case, the initial value of swirl speed in the combustion chamber was also assumed to be equal to that measured at BDC using the C type piston in motoring. Although the theoretical calculation was carried out in the motoring condition of C' type piston, yet the experimental data in firing show similar tendencies to the calculated results. Hence, there would be no large difference between the swirl speed in motoring and that in firing, and Eq.(7) would be available for firing operation.

4. Conclusions

The results of the investigation are as follows ;

- (1) Swirl in the engine cylinder has a

velocity distribution like a solid body rotation.

(2) The swirl speed predicted from the results obtained in the steady flow test is lower than that measured at BDC in motoring operation. There is a linear correlation, however, between the swirl speed predicted from the results of a steady flow test and that measured at TDC in motoring. This implies that the steady flow test is available to evaluate the swirl producing ability of the inlet system of engine. The evaluation is also made by a simple model in which the swirl has a velocity distribution of a solid body rotation and its axial component is uniform.

(3) The angular momentum possessed by cylinder charge is reduced to about 55 % during the compression stroke.

(4) Between the swirl speeds in motoring and in firing, there is no remarkable difference at the neighbourhood of TDC in the compression stroke.

(5) The theoretical model in which the reduction in air swirl motion is due to the restraining torque of shearing force acting at the air-solid interface will be available for estimating the change of swirl during the compression stroke of engine.

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