Fundamental Characteristics of Squish Flow in D. I. Diesel Engine*

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Fundamental characteristics of air motion in the combustion chamber induced only by squish flow were investigated by using a model research engine. The spatial distributions of mean velocity and turbulence intensity, and turbulence scales measured by hot-wire anemometry are presented for variations of inlet orifice diameter of the chamber and engine speed. From these results, the following were concluded. Spatially-averaged mean velocity at TDC is greater than half of the maximum theoretical squish velocity for an aspect ratio of orifice diameter of 0.4. Turbulence intensity and turbulence scales are less than those of swirl flow. Mean velocity and turbulence intensity increase proportionally with engine speed, but flow pattern and turbulence length scales are unchanged.

Key Words: Internal Combustion Engine, Diesel Engine, Squish Flow, Hot-wire Anemometry, Turbulence Characteristics.

1. introduction

The direct-injection diesel combustion system with a reentrant type bowl on the piston crown which utilizes the squish air flow induced by the piston movement to promote mixture formation and combustion in diesel engines seems to, if acknowledging its useful characteristic of shorter ignition delay and higher burning rate, provide an advantageous property for reducing the emission of nitric oxides by selecting a retarded injection timing. In addition, since the smoke density of exhaust gas is difficult to increase even at a retarded injection timing, and since energy loss due to gas movement and cooling is fairly low, this combustion system is considered to have a potentiality of lower pollutant emission without large sacrifice in fuel economy. However, as this property is inclined to fade out gradually with an increase in the amount of fuel supplied, it may be necessary to strengthen the function of air motion which is

effective for promoting the combustion.

Up to the present, making use of hot-wire anemometry, LDV and the electric-discharge method, many studies concerning the determination of air motion within the cylinder of diesel engines have been published. A large majority of these have dealt with the swirling flow field, and the squish flow has been presented only as a radial component of velocity in the combined flow field of swirl and squish⁽¹⁾.

Recent results obtained from numerical simulation for in-cylinder flow have illustrated that the air motion induced only by squish differs from that by the swirl-combined squish⁽²⁾⁽³⁾. Therefore, it is necessary to clarify the air motion in the piston bowl induced only by the squish motion for its effective use.

In this study, fundamental characteristics of squish-air motion in a piston bowl are investigated on the basis of experimental results obtained by hot-wire anemometry under motoring operations. A model research engine cylinder was used to avoid other types of air motion such as induction swirl and turbulence imparted during the intake stroke. Intensity of squish flow into the piston bowl was varied by changing the diameter of the inlet throat orifice. Spatial distribu-

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tions of mean velocity, flow direction and turbulence intensity, and turbulence scales are examined for each grid point set up on an axial section in the space of bowl and for variation of engine speed.

2. Experimental Apparatus and Method

2.1 Model engine cylinder

Figure 1 shows a schematic of the model engine cylinder. The piston bowl which was toroidally shaped on the bottom surface and had an orifice plate mounted on its throat was arranged coaxially in the cylinder so as to make an axisymmetric vertical vortex in the space. Intensity of squish motion was varied by changing the diameter of the throat orifice. Dimensions of the cylinder bore, stroke and compression ratio were 140 mm, 50 mm and 4.6, respectively. In order to reduce contamination by oil mist which causes a breakage in the sensor wire, an O-ring seal was used instead of piston rings. Leaked air from the cylinder was supplemented through a non-return valve installed on the cylinder head.

2.2 Measurement of velocity

Hot-wire anemometry may be one of the most useful means to measure the turbulent air motion at any location in the piston bowl, especially in one with an entry lip.

Output signals from the constant-temperature-type hot-wire anemometers must be corrected for changes in the temperature and gas properties in the measuring field. The temperature and velocity of the flow field can be estimated at the same time through a difference between heat-transfer rates of the sensor at two temperature levels, because the anemometers

respond to both the velocity and temperature of the surrounding gas. This method was applied successfully by Koso et al. to a flow field containing small temperature fluctuations under atmospheric conditions⁽⁴⁾. In this study, this method was improved by expanding the dynamic ranges of temperature and pressure.

For measurement of gas temperature in the cylinder, 573 K and 773 K were adopted as sensor temperatures, and accumulative mean data over 256 cycles were sampled, for there were found small irregularities in the data of successive cycles concerning the temperature and properties of the gas.

Making use of the cut-and-try method, gas temperature was determined by solving a couple of simultaneous equations of heat-transfer rate expressed by Hamamoto's equation⁽⁵⁾ as follows:

$$Nu_m \cdot (T_g/T_f)^{-0.17} = A + B \cdot Re^{0.5} \quad (1)$$

where Nu_m : Nusselt number in which heat loss from a sensor wire to wire supports and the effect of temperature increase on the sensor surface are considered; T_f : mean film temperature of sensor; T_g : gas temperature; Re : Reynolds number; and A, B : coefficients determined from the result of calibration obtained by changing sensor temperature under atmospheric conditions. Gas density at any crank angle was calculated as a function of gas temperature and cylinder volume.

Outflow velocity from the piston clearance was measured to examine the accuracy of this correcting method. Figure 2 shows a comparison of results; namely, corrected data from the anemometer V_h and data by the electric-discharge method V_s which are theoretically independent of temperature and properties of the fluid medium. As the both experimental results correspond each other within a 5% of fluctuations, the practical usefulness of this correcting

Fig. 1 Model engine cylinder

Fig. 2 Result of verification of correcting method

method is sufficiently justified, because the former contains an error concerned with a process for estimating the properties of fluid such as viscosity and thermal conductivity, and the latter inherently contains an irregularity in measured values.

2.3 Hot-wire probe

Due to the axisymmetric configuration, air motion in the piston bowl has no tangential velocity component, hence, flow direction changes two-dimensionally at any location in the cross section of the bowl, according to the piston motion. As shown in Fig. 3, two types of hot-wire probes were used. A single sensor probe (type 1) having a thermocouple on the top of a support beam was used to measure absolute velocity, and a double sensor probe (type 2) having two sensors each inclined 45 degrees to the probe axis was used to detect the flow direction. These probes were symmetrically installed in the piston bowl.

The hot-wire sensor was made of a Pt-Ir wire of $12.7\text{ }\mu\text{m}$ in diameter, and its aspect ratio for length was approximately 200 for type 1 and 300 for type 2. The sensor operating temperature for measurement of instantaneous velocity was 773 K, which was higher by 350 K than the maximum temperature of air at TDC. Direction of flow was calculated assuming that the effective cooling velocity of the stream was subject to the cosine law.

2.4 Data acquisition and processing

Output signals from a hot-wire anemometer (KANOMAX model 1010), together with those from a pressure transducer and a thermocouple transducer, were digitized directly and processed by an on-line NEC PC-9801 microcomputer installed with a 12-bit A/D converter with four input channels. For measurement of instantaneous velocity, data were sampled over 40 successive cycles, and for measurement of flow direction and air temperature, mean data accumulated over 256 cycles were sampled. The data were sampled every 0.6 degrees of crank angle for the mean data acquisition, and every 0.15 degrees for the instantaneous data acquisition over ± 144 degrees.

Fig. 3 Forms of hot-wire probe

around TDC.

The time-averaging technique was used to determine the mean velocity, turbulence intensity, and turbulence scales⁽⁶⁾. For averaging purpose, a crank angle of 18 degrees, corresponding to the widest interval retaining a profile of turbulence intensity progress averaged under narrow intervals, was adopted.

In order to investigate air motion within the bowl, measurements were carried out at an engine speed of 600 rpm, changing the probe location and orifice diameter d , which was selected as 50, 58, 70 and 87 mm. The probe location was varied every 10 mm from $r=0$ to 40 mm in a radial direction and every 5 mm from $l=5$ to 30 mm in an axial direction from the top surface of the piston. To investigate the effect of engine speed on the flowing field, measurements were carried out at two probe locations for an orifice diameter of 58 mm.

3. Experimental Results and Discussion

3.1 Spatial distributions of velocity and turbulence intensity

As an example of the obtained results, progresses of mean velocity $\bar{U}$, turbulence intensity u' and flow direction θ_d , for various crank angles using a 58 mm orifice diameter are shown in Fig. 4. It is apparent that a stream rushing into the bowl passes through measuring stations in the order of B, A and C, for in this order, peaks of $\bar{U}$ tend to be of greater height and longer duration. In proportion to the degree of stream development, turbulence intensity becomes stronger and the interval in which a higher value of u' con-

Fig. 4 Measured example of mean velocity, turbulence intensity and flow direction

tinues becomes longer. The flow direction changes with crank angle and location in the bowl. Concerning the flow direction, since the double sensor probe can only detects the relative angle held between the sensor wire and stream line, the flow direction is found in one of four directions on the measuring plane associated with this angle. Therefore, the flow direction was decided under the assumption of continuity of flow with respect to space and time.

In Fig. 5, differences of spatial distributions of mean velocity, flow direction and turbulence intensity for a change of crank timing are shown. The length and direction of arrow in this figure indicate the magnitude of mean velocity and the flow direction respectively, and the circle diameter indicates the magnitude of turbulence intensity. In the case of crank timing at 30°BTDC, flow velocity is higher in the vicinity of the bottom corner than in the central portion of the bowl. This means that a stream towards the bowl bottom is distorted along the side wall, and then finally reaches the lower side of the orifice plate. At TDC, squeezed incoming flow to the bowl stops, and a developed vertical vortex spreads throughout the space in the bowl, and turbulence intensity becomes uniform. At 30°ATDC, an outflow stream from the bowl overlaps a circulating flow retained by an inertial force acting on it. Under this situation, a stream having higher velocity rushes to the edge of the orifice plate, and more intense turbulence is created there.

Comparisons of spatial distributions of velocity and turbulence intensity at TDC for a change of orifice diameter are shown in Fig. 6. In the case of diameter $d=87$ mm, at which the throat of the bowl is not throttled, a small vertical vortex stays near the

edge of the orifice and air motion is extremely weak throughout the space in the bowl. As the throat diameter is reduced, i.e. the flow velocity into the bowl becomes higher, circulating velocity becomes higher and turbulence intensity becomes stronger throughout the space in the bowl.

In order to evaluate the intensity of air motion in the entire space of bowl, the spatially-averaged values of mean velocity $\bar{U}_s$ and turbulence intensity u'_s for the bowl volume V_c were calculated by the following equations.

$$\bar{U}_s = 1/V_c \int_{V_c} \bar{U} dV \quad (2)$$

$$u'_s = \left[1/V_c \int_{V_c} u'^2 dV \right]^{0.5} \quad (3)$$

where measured values were assumed as the representative values in each cylindrical space at any corresponding location of measuring station.

In Fig. 7, spatially-averaged values of mean velocity and turbulence intensity are plotted against the maximum theoretical velocity $W_{r\max}$ of radial inflow at the edge of orifice⁽⁷⁾ which indicates the effect of throat diameter on the squish flow intensity. Spatially-averaged mean velocity increases linearly with the theoretical velocity, and at TDC, it reaches up to 50% of the theoretical velocity with a range of throat diameter to cylinder bore ratios between 0.35 and 0.4 ($d=49\sim 56$ mm), which are the dimensions generally adopted for the squish-lip type combustion chambers. Spatially-averaged mean velocity decreases by approximately 20% in the interval between TDC and 30°ATDC. Since an appreciably high velocity stream still remains, it is expected that

Fig. 5 Difference of spatial distributions of velocity and turbulence intensity for crank timing

Fig. 6 Effect of orifice diameter on spatial distributions of velocity and turbulence intensity

this motion of air in the bowl contributes to the promotion of combustion. Spatially-averaged turbulence intensity increases with increase of the theoretical velocity, and its ratio to spatially-averaged mean velocity reaches about 0.11~0.15. Consequently, it is known that the turbulence induced only by squish motion is relatively weak compared to other types of air motion⁽¹⁾.

Though air motion in the piston bowl has not so strong a turbulence intensity, but is conserved to the end of the combustion period, since a stream having relatively high velocity also remains near the bowl bottom where a considerable amount of fuel is supplied, it is conceived that air motion in the bowl contributes to the promotion of combustion by transporting fresh air into burning regions and by mixing air with fuel evaporated from the wall.

3.2 Turbulence characteristics

Variations of turbulent energy spectrum $E(f)$ for changes of crank timing and throat diameter are shown in Fig. 8. In the case of crank timing before TDC or for small orifice diameter, the turbulent energy spectrum decreases with increase of frequency, according to Kolmogoroff's $-5/3$ power law. From this result, it is estimated that the character of turbulence is almost same in these cases. At 30° ATDC, the decreasing inclination of slope for frequency becomes

Fig. 7 Relation between theoretical squish velocity and spatially-averaged values of mean velocity and turbulence intensity

larger, and as the orifice diameter is increased, changes in the inclination of the slope become more evident. As the explanation for this inclination, the following can be mentioned. For the former case, air motion is reduced by the expansion of cylinder charge, and for the latter case, low squish velocity induces a weak air motion.

Distributions of integral time scale L_t and integral length scale L_x , and microtime scale λ_t and microlength scale λ_x for a change of crank angle around TDC are shown in Fig. 9. Integral time scale is more sensitive to a change of the throat diameter of the bowl than a change of crank timing. As the orifice diameter is increased, namely, as air motion becomes weaker, values of the integral time scale increases. On the other hand, integral length scales remain almost constant, irrespective of changes of throat diameter and crank angle. These tendencies in the integral scale for these parameters are also found in the microscale.

Correlations between mean velocity and time scales are shown in Fig. 10. Both integral time scale L_t and microtime scale λ_t are reversely proportional to mean velocity, and the former is greater than the

Fig. 8 Comparisons of turbulent energy spectrum for orifice diameter and crank timing

latter by 2 to 2.5 times. Since the value of microtime scale is slightly smaller than that of the similar results obtained by Wakisaka et al.⁽⁶⁾ and Lancaster⁽⁸⁾, the magnitude of eddy-dissipating energy in a field of squish air motion should be smaller than that in a field of air motion accompanied with swirl and turbulence induced during the intake stroke.

3.3 Influence of engine speed

Variations of normalized mean velocity $\bar{U}/V_{pm}$ for the mean piston speed V_{pm} and flow direction for a change of engine speed n are shown in Fig. 11. They are almost same within a range of engine speeds from

Fig. 9 Distributions of turbulence scales for crank angle

Fig. 11 Influence of engine speed on mean velocity and flow direction

400 to 1000 rpm. Similar results were obtained at any measuring station in the space of the bowl. Thus, it is apparent that intensity of air motion in the bowl increases proportionally to engine speed, but flow pattern in the space is hardly affected by a change of engine speed.

Variation of turbulence characteristics at TDC with a change of engine speed are shown in Fig. 12. Turbulence intensity increases proportionally with an increase of engine speed; hence, its ratio to mean velocity remains constant. Not only time scales L_t , λ_t which are expressed by a unit of crank angle but also

Fig. 10 Relation between mean velocity and turbulence time scales

Fig. 12 Influence of engine speed on turbulence characteristics

length scales L_x, λ_x keep each constant value irrespective of variation of engine speed. Therefore, it can be concluded that in spite of proportional increase in turbulence intensity with engine speed, the magnitude of turbulence eddy hardly changes.

4. Summary

Fundamental characteristics of air motion in the piston bowl induced only by squish motion were investigated by means of hot-wire anemometry. From the results obtained by examining spatial distributions of mean velocity and turbulence intensity, and turbulence scale, the following were revealed.

(1) Spatially-averaged mean velocity at TDC reaches more than 50% of the theoretical squish velocity for an orifice diameter ratio to cylinder bore of 0.4. Even after the crank angle has progressed 30 degrees, it decreases merely by about 20%.

(2) Spatially - averaged turbulence intensity increases with the theoretical squish velocity, and its ratio to spatially-averaged mean velocity is about 0.11 to 0.15.

(3) The length scale of turbulence induced only by squish flow is less than that by swirl flow.

(4) Both mean velocity and turbulence intensity increase proportionally with engine speed, but flow pattern and turbulence length scale are unchanged.

Through this work, it was confirmed that the correcting method for the variation of temperature and pressure of the fluid used in this study to determine the velocity and temperature of the fluid at the same time through a difference of heat loss at

different sensor temperatures is practical for the measurement of air motion in the cylinder of reciprocating engines.

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