The Effect of Swirl on Spark-Ignition Engine Combustion*

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In order to investigate the effect of swirl on combustion in a spark-ignition engine, a new experimental apparatus, namely, the single-event engine was prepared. The swirl flow in the cylinder was visualized by a smoke-wire method, and a laser Doppler anemometer and a hot-wire anemometer were adopted for the velocity measurement. The turbulence intensity was determined on the basis of the hot-wire data. A stoichiometric propane-air mixture was ignited at the center of the combustion chamber; thus, the flame was propagated almost axisymmetrically in the flow field at various swirl intensities with no squish flow. The burning velocity, S_T , and the thickness of burning zone, δ_T , were determined in the spark-ignition engine cylinder. The result shows that S_T and δ_T increase with increasing the turbulence intensity, and that δ_T is correlated linearly with S_T except under the condition of weak turbulence.

Key Words: Internal Combustion Engine, Combustion, Swirl, Turbulence, Flow Measurement, Turbulent Burning Velocity, Burning Zone, Turbulent Flame

1. Introduction

It is well-known that the turbulent flow field in a spark-ignition engine cylinder plays an important role in determining the combustion characteristics, thermal efficiency and exhaust emissions⁽¹⁾. Exhaust gas recirculation (EGR) reduces NOx emissions, and the use of lean mixtures improves the thermal efficiency of the engine. However, these methods cause cycle-to-cycle fluctuations in combustion and the deterioration of driveability. In spark-ignition engines, faster burning can raise thermal efficiency and reduce NOx emissions by extending the range of stable engine operation, thus allowing very dilute fuel/air mixtures to be used⁽²⁾. It is important to clarify the influence of turbulence and mean fluid motion on combustion, and a model is needed to describe these interactions⁽³⁾. The entrainment eddy burning model is a reasonable phenomenological one^{(4)~(7)} for turbulent combustion

in the spark-ignition engine. For further development of this combustion model, it is necessary to make clear the turbulence characteristics and the interaction between turbulence and combustion in the engine cylinder.

In order to clarify the effect of turbulent swirl on combustion, a single-event spark-ignition engine was employed. A propane-air mixture was ignited at the center of a pancake-type combustion chamber. Thus, the flame was propagated almost axisymmetrically in the flow field at various swirl intensities with no squish flow. In this paper, we describe the swirl and its turbulence intensity, the burning velocity and the burning zone thickness of a turbulent flame in the cylinder.

2. Experimental Apparatus

Figure 1 shows a schematic diagram of the experimental apparatus, the main dimensions of this test engine and the experimental conditions. The engine cylinder and the mixture tank are connected with a pipe. When the engine is driven by an electric motor, the fuel gas (propane) and air pass back and forth between the cylinder and the tank through the intake

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valve, which is normally open, and fuel gas and air are mixed homogeneously. After preparing the homogeneous mixture in this way, the intake valve is closed at BDC, and then the mixture in the cylinder is compressed and expanded repeatedly by the piston motion. During the intake stroke, a swirl flow is produced in the cylinder. There is no squish flow because the combustion chamber is of pancake type.

The swirl velocity at TDC of the compression stroke decreases with the increase of the number of compression strokes, N , after the closure of the valve. The combustion phenomena under various intensities of swirl flow can be examined as follows: the mixture is ignited near end of compression after a various number of compression strokes following valve closure.

It is possible, in this test engine, to observe the flow pattern in the cylinder. This engine has an elongated piston with a glass crown. An inclined mirror inside the piston could be fixed to the cylinder block by providing a slot in the side of the elongated piston. The spark gap for ignition is located in the center of the combustion chamber in the midplane of the clear-

ance volume.

Operations of this single-event engine are controlled by a computer. The valve closure, TDC, and BDC signals are produced with photosensors, and a pulse train of crank angles is produced by an electromagnetic rotary encoder (360 pulses/rev.). When the valve-close signal is added to the SCR-circuit in accordance with the computer command, an electric discharge occurs through the solenoid coil. Then, the stopper piece is removed and the valve closes by a spring.

3. Mean Velocity and Turbulence

3.1 The flow pattern in the cylinder measured with a smoke-wire method

The air swirl in the cylinder was visualized in the case of noncombustion by a smoke-wire method (SW)⁽⁸⁾. Nickel wires were set up in the midplane of the clearance volume and in the direction of the cylinder diameter. The smoke was visualized from below the piston using a reflex mirror and a high-speed camera. Figure 2 shows the overlapped tracing of the smoke front during the third compression

Fig. 1 Schematic diagram of a single-event engine

stroke after the valve closure (N3) at an engine speed, n , of 600rpm and a compression ratio, ϵ , of 4.8. The angular velocities of several points of the smoke front relative to the cylinder axis were measured, and swirl velocities in the tangential direction were determined. The tangential velocity distribution along the cylinder radius is shown in Fig.3, where $\bar{U}$ and r denote the mean velocity in the crank angle interval of 110 and 55 degrees before TDC and the distance from the cylinder axis, respectively. It is recognized from these results that the swirl flow in the cylinder during the compression stroke was similar in velocity distribution to that of a solid body rotation. The figure also shows the results of measurements with a laser Doppler anemometer (LDA) and with a hot-wire anemometer (HWA). The velocities measured by using three kinds of methods, SW, HWA and LDA were close in value.

Fig. 2 Time line of the smoke front ($\epsilon=4.8$, $n=600$ rpm, N3)

Fig. 3 Comparison of mean velocities with three kinds of measuring methods (SW, LDA, HWA) in the crank angle interval of 110 and 55 deg. BTDC

3.2 Mean velocity and turbulence intensity

A HWA has good frequency response and is useful for measuring turbulence. However, it is also sensitive to the temperature and to the pressure of the airstream; hence a serious problem regarding the use of hot-wire anemometry in an engine cylinder arises, where the temperature and pressure of gas fluctuate widely. In particular, the exact determination of the gas temperature near TDC is very difficult. In order to eliminate this uncertainty, in this study, the values of mean velocity measured with HWA were confirmed by those with LDA.

The platinum-iridium sensing element of HWA was 15 μm in diameter and 2 mm long. The wire temperature was kept constant at 800 K. Using a spark tuft method⁽⁹⁾, it was found that the swirl flow near TDC is almost horizontal. Therefore, the sensing element of HWA was set parallel to the cylinder axis. The output of HWA was compensated for the changes of gas temperature and pressure and the conductive heat loss to the wire supports⁽¹⁰⁾.

The LDA operated in the forward-scattering fringe mode and was incorporated with a frequency shifter of the Bragg cell type. The light source was a helium-neon laser of 15 mW. The total angle of intersection of laser beams was 0.2023 radians (11.59 degrees). JIS test dust No.11 (fine red clay) was used as particles for scattering the light. The Doppler signals were processed with a frequency tracking system.

Figure 4 shows an example of the tangential velocity of swirl flow, $\bar{U}$, measured with LDA. The measurement was made for the case with compression ratio $\epsilon=4.8$, engine speed $n=600$ rpm and the location of measuring point $r=30$ mm. It is recognized that the swirl velocity near TDC decreases with the number of compression stroke after the valve closure.

Mean velocity $\bar{U}$ and the turbulence intensity u'

Fig. 4 An example of swirl measurement with LDA

near TDC were defined in accordance with the time-average method which treated the flow in a given short time interval from TDC $-\Delta t/2$ to TDC $+\Delta t/2$ as a steady flow at each cycle⁽¹¹⁾:

$$\bar{U} = \frac{1}{z} \sum_{k=1}^z \bar{U}_k, \quad (1)$$

and

$$u' = \sqrt{\frac{1}{z} \sum_{k=1}^z (U_k - \bar{U}_k)^2}, \quad (2)$$

where U_k is the instantaneous velocity in the k -th run of the experiment, z is the number of runs of the experiments ($z=10 \sim 20$ in the measurement of the turbulence intensity) and

$$\bar{U}_k = \frac{1}{\Delta t} \int_{-\Delta t/2}^{+\Delta t/2} U_k dt, \quad (3)$$

and

Fig. 5 Mean velocity of swirl near TDC

Fig. 6 Turbulence intensity near TDC

$$\overline{(U_k - \bar{U}_k)^2} = \frac{1}{\Delta t} \int_{-\Delta t/2}^{+\Delta t/2} (U_k - \bar{U}_k)^2 dt. \quad (4)$$

The data sampling frequency was 20 kHz in this study. By using the finite time interval Δt , the fluctuating component of the velocity (turbulence) with a frequency lower than about $1/(2\Delta t)$ is cut off⁽¹²⁾. In this study, the cutoff frequency was about 100 Hz, because Δt was chosen as 5.0 ms. The choice of 100 Hz as the cutoff frequency was based on evidence that a turbulence frequency lower than about 100~150 Hz hardly affects combustion⁽¹³⁾.

The measurements of mean velocity near TDC were made under many kinds of experimental conditions using HWA and LDA, and both results were compared with each other. The measured values with HWA almost coincided with the LDA data. Consequently, it was confirmed that the HWA output was compensated with good accuracy. Figures 5 and 6 respectively show the mean velocity, $\bar{U}$, and the turbulence intensity, u' , near TDC obtained with HWA measurement.

The swirl ratio, SR , was defined as the ratio of the speed of equivalent solid body rotation, which had an angular momentum equal to the actual swirl in the cylinder, to the engine speed. Spatial mean intensity of turbulence, $(u')_v$, was determined from the distribution of turbulence intensity.

4. The Effect of Swirl on the Combustion

4.1 The effect of swirl ratio on the combustion

Propane-air mixtures were ignited by a spark for the several cases of compression ratio, engine speed and swirl ratio. The equivalence ratio was equal to one. Figure 7 shows examples of pressure-volume diagrams under the conditions that the compression ratio is 3.2, engine speed is 600 rpm, ignition timing is

Fig. 7 Pressure-volume diagram

20 degrees before TDC, and values of swirl ratio near TDC of the first, second and third compression strokes (N1, N2, N3) were 2.21, 1.52 and 1.17, respectively.

As shown in Figs. 8 and 9, mass fraction burned, x , and burn rate, $dx/d\theta$, were determined from the pressure diagram. These were obtained from a similar method of Lavoie et al.⁽¹⁴⁾, and the heat loss was estimated by using a heat transfer coefficient presented by Eichelberg⁽¹⁵⁾.

It can be clearly understood from these results that the swirl flow has an important effect on engine combustion. With the increase of swirl ratio, the burn duration decreases and the maximum value of burn rate increases.

4.2 Burning velocity of a turbulent flame in a cylinder

Turbulent burning velocity, S_T (i.e., the entrainment velocity of unburned gas against the flame front), is obtained by considering the mass balance of unburned gas in the turbulent flame;

$$S_T = \dot{M}_b / (\rho_u A_T) + \dot{V}_{Bu} / A_T, \quad (5)$$

where ρ_u =unburned gas density, $\dot{M}_b$ =rate of mass burned in the total combustion chamber, A_T =envelope area of the turbulent flame, V_{Bu} =volume of unburned gas in the burning zone of the turbulent flame and $\dot{V}_{Bu}=dV_{Bu}/dt$. The volume of V_{Bu} is determined as follows:

$$V_{Bu} = V_T - V_L \quad (6)$$

where V_T =the volume of the turbulent flame and V_L =the volume of the laminar flame of which the mass fraction burned is equal to the value in the case of a turbulent flame. In laminar flame propagation in a constant-volume vessel, the volume fraction burned, y , is expressed as a function of the mass fraction burned, x , as follows⁽¹⁶⁾:

$$y = 1 - (1 - x)(P_{01}/P)^{1/\kappa_u} \quad (7)$$

where P =the pressure in the cylinder, κ_u =the ratio of the specific heats for unburned gas and P_{01} =the pres-

sure under the initial condition. The volume of the combustion chamber changes during the combustion process. The relation between x and y at any crank angle is obtained from Eq.(7). The initial condition (subscript 01) of the mixture is assumed to be changed adiabatically;

$$P_{01} = P_i (V_i/V_{01})^{1/\kappa_u} \quad (8)$$

where subscript i indicates the ignition timing and V_{01} =the volume of the combustion chamber at the time at which the value of y is determined. Consequently, the volume of laminar flame is obtained as follows:

$$V_L = y V_{01} \quad (9)$$

The volume V_T and the envelope area A_T of the turbulent flame are determined from the measuring results of flame configuration.

4.3 Burning zone thickness and configuration of turbulent flame in a cylinder

Turbulent flames are thought to have both burned and unburned gas clouds in a certain width of the turbulent burning zone. Assuming that the volume of burned gas in the burning zone of a turbulent flame is equal to the unburned volume, the burning zone thickness, δ_T , is expressed as

$$\delta_T = V_B/A_e = 2V_{Bu}/A_e, \quad (10)$$

where V_B =the volume of the burning zone and A_e =the surface area of the equivalent flame to the volume V_L .

The flame arrival time was measured with a small ion probe at many points in the combustion chamber, and the average shape of turbulent flame in the cylinder was determined as shown in Fig. 10. In this figure, the Z -axis is the cylinder axis, the r -axis is the radius, the origin is the spark point and t denotes the elapsed time after the spark.

Fig. 8 Mass fraction burned

Fig. 9 Burn rate

Fig. 10 Configuration of the flame front

Fig. 11 Effect of $(u')_v$ on the turbulent burning velocity, S_T

4.4 Effect of turbulence intensity on the burning velocity and the burning zone thickness of the turbulent flame

The values of A_T , A_e , $\dot{V}_{bu}$, $\dot{M}_b$ and ρ_u were deter-

mined from measurements of the flame configuration and the cylinder pressure. The turbulent burning velocity, S_T , and the burning zone thickness, δ_T , were obtained as shown in Figs. 11 and 12. The values of S_T and δ_T increase with the increase of the turbulence intensity, $(u')_v$. The turbulence intensity of unburned gas during the flame propagation was estimated from the value measured under the noncombustion condition as follows⁽⁷⁾:

$$(u')_v = (u'_0)_v (\rho_u / \rho_{u0})^{1/3} \quad (11)$$

where subscript o denotes the noncombustion condition. This relation is derived from the assumption of the conservation of angular momentum of the individual eddies of turbulence. The relation between S_T and $(u')_v$ which are normalized by the laminar burning velocity, S_L , is shown in Fig. 13. The curve in Fig. 13 is expressed as follows:

$$S_T/S_L = 1 + 5.6((u')_v/S_L)^{1.8} \quad (12)$$

where S_L was calculated from the equation proposed by Lavoie⁽¹⁷⁾, and Eq. (12) is useful under the condition of turbulence intensity being greater than about 0.3 m/s, because the flame structure in the very weak turbulence (<0.3 m/s) is different from that in the actual engine⁽¹⁸⁾.

As shown in Fig. 14, a linear correlation between S_T and δ_T exists except where the turbulence is very

Fig. 12 Effect of $(u')_v$ on the burning zone thickness, δ_τ

weak. The amount of unburned gas entrained into the flame front increases with increasing the turbulence intensity. However, the mean reaction rate in the burning zone does not increase as much as the increase of the entrained unburned gas. Consequently, the thickness of the burning zone increases with increasing the turbulent burning velocity.

5. Conclusions

The effect of turbulence on the combustion of a stoichiometric propane-air mixture was investigated using a single-event spark-ignition engine with a pancake type combustion chamber. The main results are as follows.

- (1) Swirl flow in the cylinder was examined for the noncombustion case using three kinds of methods, i. e., smoke-wire method, hot-wire anemometry and laser Doppler anemometry. These results coincided with each other, and the certainty in velocity measurements was confirmed.
- (2) Turbulent burning velocity was determined in the spark-ignition engine cylinder. The effect of turbulence intensity on the turbulent burning velocity is expressed by Eq.(12) under the condition of turbulence intensity being greater than about 0.3 m/s. In this case, a spatial mean intensity of turbulence with a frequency component higher than 100 Hz is used.
- (3) There is a linear correlation between the turbulent burning velocity and the burning zone thickness, except where the turbulence is very weak.

Fig. 13 Relation between S_T/S_L and $(u')_v/S_L$ Fig. 14 Relation between δ_τ and S_T

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