Combustion of a Diesel Engine with the M-combustion System*

(Effects of Swirl Ratio and Surface Temperature of Combustion Chamber)

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This paper gives the experimental results obtained with a two-stroke cycle single-cylinder research engine using an electrically heated combustion chamber to simulate the combustion chamber of a low-heat-rejection(LHR) DI diesel engine with the M-combustion system. In this report, the experiment was performed to investigate the influences of combustion chamber surface temperature and swirl ratio on the combustion of a LHR DI diesel engine with the M-combustion system. It is found that the combustion of fuel deposited on the surface is greatly affected by the combustion chamber surface temperature, and the diesel combustion is controlled by the swirl and the surface temperature of the combustion chamber.

Key Words: Internal Combustion Engine, DI Diesel Engine, M-combustion System, Chamber Surface Temperature, Model Combustion Chamber, Swirl Ratio

1. Introduction

The M-combustion system is characterized by the mixture formation process, in which most of the injected fuel is deposited as a film on the wall of the combustion chamber, and then evaporates and burns. Because of these features, the M-combustion system would be suited well not only to heavy duty DI diesel engines but also to small sized ones^{(1)~(3)}. For this reason, as the combustion chamber decreases with the size of the DI diesel engine, the amount of fuel impinging and being deposited on the wall should increase even in conventional DI diesel engines. Thus it is thought that the small-sized DI diesel engine, with the advantages of the M-combustion system such as lower NO_x emission and combustion noise, could be realized by utilizing combustion control to prevent its disadvantages, such as smoke, hydrocarbon emission

and odour, and that better fuel consumption than in other combustion systems currently employed in small diesel engines could be expected.

It is considered that combustion chamber surface temperature and swirl greatly affects the combustion of the M-combustion system. It has already been confirmed that, for combustion in the M-combustion system, the combustion chamber surface temperature is an important factor affecting diesel combustion and that combustion is greatly improved by increasing the combustion chamber surface temperature^{(4)~(6)}. Although it is thought that the airflows in the combustion chamber of a diesel engine, such as swirl and squish, affect fuel evaporation, mixture formation, combustion, engine performance, fuel consumption and exhaust emissions not only in the conventional DI diesel engines but also in the M-combustion system, it is not clear how the swirl affects the combustion in the M-combustion system^{(3)~(6)}. Furthermore it is considered that the air motion must be actively utilized to promote the combustion in the M-combustion system, as in the conventional DI diesel engines.

The purpose of this work is to investigate experimentally the effects of swirl and combustion chamber

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surface temperature on the combustion of a diesel engine with the M-combustion system. In the experiment, electrically heated model combustion chambers were used to simulate the bowl-in-piston-type combustion chamber of a low-heat-rejection diesel engine (hereafter called "LHR diesel engine") and a small-sized DI diesel engine with the M-combustion system. Combustion characteristics and its processes were investigated by analyzing an indicator diagram^{(5),(6)}.

2. Experimental Apparatus

2.1 Model combustion chamber

Figure 1 shows the construction of the model combustion chamber. The structure of the combustion chamber on the whole is same as the swirl chamber diesel engine. But because the swirl chamber is spherical and the passage area is larger than that of conventional swirl chamber diesel engines, part of the swirl chamber of the model combustion chamber simulate the bowl-in-piston-type combustion chamber of a small-sized DI diesel engine with a strong induction swirl. In this model combustion chamber, swirl strength was changed by altering part of the passage.

Figure 2 shows the three parts of the passage and Table 1 shows the main specifications of the model combustion chambers including the altered part of the passage. The combustion chamber with an R_p value (=passage area/cylinder bore area) of 2.0% was

called the A-type, $R_p=3.1\%$ the B-type and 4.9% the C-type, respectively. The calculated swirl ratio, ignoring the pressure difference between the swirl chamber and cylinder⁽⁷⁾, was about 30 for the A-type, about 17 for the B-type and about 8 for the C-type, respectively; the ratio was about 3.8:2.1:1. It is thought that swirl affects the processes of mixture formation and the condition of the fuel on the wall. However their effects were not analyzed and only the final results of the combustion processes appear in this study. The combustion chamber surface temperature could be controlled up to about 820 K from 600 K by the two electric heaters installed in the top(130 W) of and inside(200 W) of the combustion chamber.

The measuring points of the combustion chamber surface temperature and direction of injection are shown in Fig.3. The combustion chamber surface temperature was measured at 8 points. It was assumed that the injected fuel would be deposited on the wall around measuring points 1, 3, 4 and 5 and that the combustion chamber surface temperatures at those points would greatly affect fuel evaporation and combustion. The combustion chamber surface temperatures measured at measuring points 1, 3, 4 and 5 were not so very different⁽⁶⁾. Thus the combustion chamber surface temperature(T_s) was represented by the temperature measured at measuring point 1. Temperatures at measuring points 2, 6, 7 and 8 were slightly lower than the temperatures at other measur-

Fig. 1 The model combustion chamber

Table 1 The specifications of the model combustion chamber

Combustion Chamber	A	B	C
Type	Swirl Chamber		
Volume of Combustion Chamber cc	38.5		39.2
Compression Ratio	19.2 : 1		18.9
Compression Ratio at Scavenging Port closed	15.5 : 1		15.3
(Volume of Combustion Chamber) (Clearance Volume) %	74.6		73.3
(Passage Area) (Cylinder Bore Area) %	2.0	3.1	4.9
Swirl Ratio (Calculation)	30	17	8

Fig. 2 Part of the passage

ing points. The maximum difference between the temperatures was about 50 K. Cylinder pressure was measured at the swirl chamber with a piezoelectric indicator(KISTLER 601A).

2.2 Experimental apparatus

Figure 4 shows the experimental setup and Table 2 shows the main specifications of the test engine. The test engine had a two-stroke cycle with the model combustion chamber installed. The test engine was driven by electric motor at a constant speed, about 470rpm. The injection pump was also driven by a timing belt from crankshaft. Diesel fuel was injected into the combustion chamber for only one cycle by moving rapidly the control rack of the injection pump using a solenoid. As the solenoid was started by a trigger pulse generated at intervals of several revolutions by the electric circuit thinning out the BDC pulses and delaying it after that so as to synchronize with the injection timing, combustion occurred constantly at intervals of several revolutions during motoring. Fuel flow rate and return fuel from nozzle were measured. Charge air was supplied to the engine via the heater.

The experiment was performed under constant conditions: injection timing 4 degBTDC, air excess ratio 2.0, charge air temperature 453K at the entrance of the engine, combustion taking place every 5 revolutions. The combustion chamber surface temperature was changed. The fuel, as shown in Fig.3, was injected

Fig. 3 Measuring points of combustion chamber surface temperature and direction of injection

toward the wall and the heated area of the combustion chamber via a single-hole nozzle. The angle between the direction and the centerline was 75 deg. Because the distance from nozzle to wall was about 12mm in this injection direction, the most greater part of the injected fuel would deposit on the combustion chamber surface. The hole diameter of the nozzle was 0.4 mm and its opening pressure was 16.7 MPa. The cylinder pressure of the combustion cycle was recorded at intervals of 1 deg.CA. The combustion characteristics, the rate of heat release and the indicated thermal efficiency were calculated from an average cylinder pressure of 50 cycles.

3. Results and Discussion

3.1 Combustion characteristics

Figure 5 shows the combustion characteristics, the peak pressure P_{max} , the maximum rate of pressure $(dP/d\theta)_{max}$, and ignition delay in the case of A, B and C-type combustion chambers, namely, the changing

Table 2 The main specifications of the test engine

Type	Two-Stroke Cycle Single Cylinder
Bore	90 mm
Stroke	110 mm
Swept Volume	700 cc
Scavenging System	Loop Scavenging
Scavenging Port Timing	BBDC, ABDC 61 degCA
Exhaust Port Timing	BBDC, ABDC 71 degCA

Fig. 4 The experimental setup

swirl strength.

(1) Peak pressure: In each combustion chamber, A, B or C-type, the peak pressure increased with increasing combustion chamber surface temperature. These results show the same tendency as previously reported^{(4)~(6)}. But the peak pressure was higher with increasing swirl strength.

(2) Maximum rate of pressure rise: In the M-combustion system, the maximum rate of pressure rise decreases with increasing combustion chamber surface temperature^{(4)~(6)}. The C-type combustion chamber result ($R_p = 4.9\%$), where swirl strength was lowest, shows the same tendency. However, with stronger swirl, the degree of the decrease of the maximum rate of pressure rise with increasing combustion chamber surface temperature becomes small. Especially, in the case of the A-type combustion chamber ($R_p = 2.0\%$), where swirl was strongest, the maximum rate of pressure rise first decreased, but then began to increase with increasing combustion chamber surface temperature. But this tendency was slight. The maximum rate of pressure rise for the 3 types of combustion chamber is significantly low, about 200 kPa/deg, compared with that of a conventional DI diesel engine. This indicates that the rate of the maximum pressure rise of the M-combustion system is very low irrespective of swirl strength, and this is a great merit on the M-combustion system. Its low maximum rate of pressure rise will contribute low combustion noise.

(3) Ignition delay: In the M-combustion system, with increasing combustion chamber surface temperature, evaporation of fuel deposited on the combustion chamber surface and the formation of the premixed

Fig. 5 Combustion characteristics

mixture becomes rapid. As a result, ignition delay becomes smaller with increasing combustion chamber surface temperature^{(4)~(6)}. This tendency was maintained also in the case of changing swirl strength. However, there was a tendency for the ignition delay to be not so short at a very high surface temperature, over 750 K.

3.2 Combustion processes

(1) Influences of the combustion chamber surface

Fig. 6 Combustion processes (A-Type combustion chamber)

Fig. 7 Combustion processes (B-Type combustion chamber)

Fig. 8 Combustion processes (C-Type combustion chamber)

temperature :

Figures 6, 7 and 8 show the cylinder pressure, the rate of heat release and the mass burned ratio for A, B and C-type combustion chambers, respectively. From these figures, it is commonly found that the peak pressures increased and the ignition delay became small with increasing combustion chamber surface temperature in the cylinder pressure diagrams. It is also found that diffusion burning was promoted and, as a result, the mass burned ratio increased according to the increase of the combustion chamber surface temperature. As the fuel deposited on the combustion chamber surface evaporated faster with increased combustion chamber surface temperature, mixture formation also became faster; thus the ignition delay was reduced. In the diffusion burning period, because of the faster evaporation of fuel deposited on the combustion chamber surface, which was thought to control the diffusion combustion, the rate of heat release increased. On the other hand, in the C-type combustion chamber, where swirl was weakest, the peak of premixed combustion decreased with increasing combustion chamber surface temperature. The peak of premixed combustion is closely related to the ignition delay. The formation of the premixed mixture became rapid with increasing combustion chamber surface temperature, but the quantity of the premixed mixture decreased rather than increased because of the reduced ignition delay. However, in the B- and A- type combustion chambers, having stronger swirl, the peak of premixed combustion did not change so much, or had a slight tendency to increase in spite of the reduced ignition delay. This result indicates that swirl greatly affects the formation processes of the premixed mixture and also helps to increase the quantity of the premixed mixture during the ignition delay in the M-combustion system.

In this study, because the distance from the injection nozzle hole to the combustion chamber wall was 12mm, most of the fuel injected into the combustion chamber would be deposited on the combustion chamber surface. It is considered that even over such a short distance, strong swirl tends to increase the fuel in the air and accelerate the evaporation of the fuel on the surface. It is also considered that strong swirl tends to accelerate the mixing of fuel vapor and air and increases the quantity of the premixed mixture because of the significantly accelerated fuel evaporation in the case of very high surface temperature in the combustion chamber.

(2) Influence of swirl strength: Figures 9 and 10 show the cylinder pressure, the rate of heat release and mass burned ratio for different swirl strengths in

the case of the low and high surface temperature in the combustion chamber, respectively. It is found that the stronger the swirl, the higher the peak pressure in the pressure diagram, and the peak of premixed combustion in the rate of heat release increased. This indicates that the premixed mixture increased during ignition delay. It is also found that the stronger the swirl, the earlier the appearance of the promoted diffusion burning. Regarding the rate of heat release, the depression between the premixed combustion and diffusion combustion, which have typically been clearly apparent in the combustion of the DI diesel engine, become indistinct with increasing swirl strength. It is considered that strong swirl advanced the start of diffusion combustion and, furthermore, promoted the diffusion combustion by accelerating the evaporation of the fuel deposited on the combustion chamber surface and the formation of the mixture. These effects of increasing swirl strength became remarkable with increasing combustion chamber surface temperature. The diffusion combustion was very active and finished earlier because the evaporation of the fuel deposited on the combustion chamber surface and mixture formation became fast with increasing

Fig. 9 Combustion processes ($T_s = 673$ K)

Fig. 10 Combustion processes ($T_s = 803$ K)

Fig. 11 Indicated thermal efficiency

combustion chamber surface temperature and swirl strength. Furthermore, it is considered that the effects of swirl would last throughout the combustion process and that the combustion duration would be shortened. It is also thought that strong swirl would make the fuel film spread on the wall wider and thinner⁽⁸⁾. Results showed that the fuel evaporated more quickly with increasing combustion chamber surface temperature. It is supposed that these swirl effects and increasing combustion chamber surface temperature would decrease hydrocarbon emissions.

It was found that the stronger the swirl, the higher the mass burned ratio was immediately after ignition, independent of the combustion chamber surface temperature. This indicates that the combustion in the early and middle stages was accelerated by the strong swirl and that the M-combustion system was also influenced by swirl in the same way as conventional DI diesel engines. But it was found that the stronger the swirl, the less the final mass burned ratio. It is considered that the cooling loss increased with increasing swirl strength. It is thought that the final mass burned ratio attaining only about 0.5~0.6 in the case of lower combustion chamber surface temperature was due to the heat loss cancelling out the small heat release in the later stage of the combustion process⁽⁶⁾ because of the delay in the combustion.

3.3 Thermal efficiency

Figure 11 shows the indicated thermal efficiency. It is found that the indicated thermal efficiency was significantly improved by increasing the combustion chamber surface temperature independent of swirl strength. This is the result of a great improvement in the diffusion combustion caused by increasing the combustion chamber surface temperature, similar to the results previously reported^{(6)~(5)}. But in spite of improving the combustion and shortening the combustion duration by increasing the swirl strength, as mentioned in 3.2 (2), the indicated thermal efficiency

was decreased. This decrease in thermal efficiency must be because of increasing heat loss. It is considered that the heat flux to the wall of the combustion chamber from the combustion gas increased according to increasing swirl strength⁽⁹⁾. In this particular study, because of the relatively low engine speed, about 470 rpm, compared with the conventional DI diesel engine operation, the influence of the cooling loss would be greater.

4. Conclusions

The effects of swirl and combustion chamber surface temperature in the combustion of the M-combustion system are summarized as follows.

(1) The swirl greatly influences the combustion of the diesel engine with the M-combustion system in a similar way to the conventional DI diesel engine.

(2) By increasing the swirl strength and the combustion chamber surface temperature, the premixed combustion and diffusion combustion are both promoted and the combustion duration is shortened.

(3) Through promoting combustion by increasing the swirl strength and combustion chamber surface temperature, the peak pressure and the maximum rate of pressure rise are slightly increased. But the maximum rate of pressure rise is very low compared with conventional DI diesel engines. This is one of the great merits of the M-combustion system.

In this report, a study was conducted on the influences of swirl strength and the combustion chamber surface temperature. However, further investigations are required to determine other properties such as the spray direction for the swirl, the influences of swirl on the fuel behavior on the wall and, exhaust emissions. The authors would like to thank Mr. T. Kato, and Mr. T. Kawase, who assisted in the experiments during this study.

References

- (1) Neitz, A. and D'Alfonso, N., The M.A.N. Combustion System with Controlled Direct Injection for Passenger Car Diesel Engines, SAE Paper, No. 810479(1981).
- (2) Martin, J. K. and Ahmad, T., Heat-Release Characteristics of an Open-Chamber Diesel Engine Employing a Wall-Wetting Combustion System, SAE Paper, No. 860420(1986).
- (3) Teshirogi, N., Koyama, Y., Sasaki, H. and Uno, S., Heat-Insulated Diesel Engine with an M-combustion System (Measurement of the Combustion Chamber Surface Temperature of a Heat-Insulated Piston and Its Effects on Combustion), Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 53, No. 494, B(1987), p. 3130.
- (4) Teshirogi, N. and Oohashi, K., Control of Diesel

- Combustion by Regulation of Wall Temperature of Swirl Chamber, Trans. Jpn. Soc. Mech. Eng.,(in Japanese), Vol. 41, No.344(1975), p.1263.
- (5) Sasaki, H. and Teshirogi, N., The Combustion of a Diesel Engine with the M-combustion System (Effect of the Combustion Chamber Surface Temperature), Trans. Jpn. Soc. Mech. Eng.,(in Japanese), Vol. 52, No. 484, B(1986), p.4094.
 - (6) Sasaki, H. and Teshirogi, N., The Combustion of a Diesel Engine with the M-combustion System (Effect of Injection Direction and the Combustion Chamber Surface Temperature), p.3124. on Reference (3).
 - (7) Sitkei, G., Kraftstoffaufbereitung und Verbrennung bei Dieselmotoren, (Translated into Japanese by Tubouchi, T. and Kato, K.), (1966), p.197.
 - (8) Werlberger, P. and Cartellieri, W. P., Fuel Injection and Combustion Phenomena in a High Speed DI Diesel Engine Observed by Means of Endoscopic High Speed Photography, SAE Paper, No. 870097(1987).
 - (9) Harigaya, Y., Ooyagi, S., Wakatsuki, H. and Yoshida, S., Prepr. of 5th Joint Symposium on Internal Combustion Engines (in Japanese), (1985), p.157.
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