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The Effect of Combustion Swirl on the Combustion in a Diesel Engine*

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In order to investigate the effect of the timing of swirl creation on the combustion process in a Diesel engine, varying amounts of fuel were injected from two sets of fuel-injection equipments into the main and auxiliary combustion chambers respectively at different timings. A swirl was created in the main chamber by an outflow of gas from the auxiliary chamber.

Swirl creation prior to the fuel injection into the main chamber brings a small improvement on the combustion state, but is accompanied by an unfavorably rapid and high pressure rise.

Swirl creation after ignition is more effective in shortening the after-burning of fuel, as ascertained by the high-speed photography. When fuel is injected into the auxiliary chamber after TDC, the noiseless and smokeless combustion with fairly low peak pressure is materialized without the expense of fuel consumption.

1. Purpose of this study

In all cases of the combustion process in high-speed Diesel engines, it is a well-known fact that an air-swirl in the combustion chamber plays an essential function in mixture formation within the short period of time available, and in the subsequent combustion. There have already been many investigations done on the relationships between the intensities of swirl and the combustion; for example, H. R. Ricardo's experiment⁽¹⁾, in which the effect of the intensity of swirl was examined systematically with a sleeve-valve four-cycle engine.

It has been commonly supposed that a substantial difference exists in the process of mixture formation and combustion, when fuel is injected prior to the swirl creation, or following it. However, few investigations of this problem have been done.

The utilization of the air-swirl in auxiliary combustion chamber type Diesel engines may be divided into the following three classes:

(a) Swirl creation prior to the fuel injection; e. g., the combustion in a swirl combustion chamber.

(b) Swirl creation almost simultaneous with the fuel injection; e. g., the combustion in the main combustion chamber in a typical pre-combustion type engine.

(c) Swirl creation after the fuel injection; e. g., the combustion in the main chamber in a Lanova air-cell type engine⁽²⁾, or in the MWM balanced-pressure pre-combustion type engine⁽³⁾.

These three types are thought to have, of course, different characteristics of combustion, which depend principally on the relative timing of the fuel injection and the swirl creation, the intensity of the swirl, and the relative position of the fuel injection into the swirl of air. Furthermore, it may be inferred that a suitable combustion for the above conditions exists for each system of combustion chamber.

2. Experimental apparatus

A four-cycle water-cooled single cylinder Diesel engine (which was reconstructed of the ST-95 type of the Yanmer Diesel Engine Co.) was used throughout this experiment.

Fig.1 shows the cross-sectional view of the combustion chamber. Varying amounts of fuel were injected from the two sets of injection device (Bosch-type; their plunger diameters were 6.5 mm and 7 mm respectively) into the main and auxiliary chambers at different timings. The outflow of gas

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from the auxiliary chamber, which was created by the pressure rise in the auxiliary chamber due to the combustion within, created a swirl in the main combustion chamber located below the exhaust valve. In this construction the timing and intensity of the swirl could be altered at will by varying the timing and amount of the fuel injection into the auxiliary chamber. Further, the engine may be operated as a pre-combustion type or an air-cell type by bringing one or the other of the fuel injection devices into action. The above mentioned dimensions of the auxiliary chamber are not fully suitable for either operating system, but are

ascertained to lie midway between the optimum dimensions for each system. The fuel used was always heavy oil A (specific weight 0.85, cetan number 45).

The pressure diagrams in the main and auxiliary chambers were taken with pressure pick-ups of the strain-gage type with natural frequency of 35 kc/sec, being recorded on an oscillogram of electromagnetic type (with B-type vibrator having a constant sensitivity up to 3.5 kc/sec) after amplification by a transistor circuit. Furthermore, the body-noise of the engine was recorded on the same oscillogram-film simultaneously with pressure diagrams. The discoloration of exhaust smoke was measured with a smoke meter of the Saurer filter-method type.

3. Effect of combustion swirl

Effect of the timing of fuel injection

In this system of dual injection explained above, it will be easily supposed that the swirl creation in the main combustion chamber is affected by four factors, namely, the quantities and timings of both main and auxiliary injections. Now, as the first experiment, the quantities of fuel injected into both chambers were kept nearly equal, and the timing of the main injection was chosen at three fixed timings. Fig. 2 shows the effect of the timing of the fuel injection into the auxiliary chamber on the specific fuel consumption when running at a constant speed and load. It is noted here that the most suitable timing of the auxiliary injection for the minimum fuel consumption corresponds nearly to the timing of the main injection in each case. The injection timing throughout this experiment was measured from

the movement of a needle valve of the injector. From the experimentation, it is apparent that the swirl created immediately after the main injection is very effective in improving the combustion; this swirl in the main chamber is created, as is explained later, in a certain period after the auxiliary injection. Another decrease in fuel consumption is, however, attained when the auxiliary injection is done near TDC. This fact makes the effectiveness of the swirl as a factor influencing the fuel consumption appear all the more remarkable.

Effect of the ratio of fuel injected into the auxiliary chamber

With the increase of fuel quantity injected into the auxiliary chamber, the intensity of the swirl produced in the main chamber becomes naturally higher, while

Cylinder bore/stroke 95 mm/115 mm
Compression ratio 15:1
Ratio of the auxiliary combustion chamber to the total compression volume 25 %
Cross-sectional area of connecting passage 0.25 % of piston area

Fig. 1 Section through the combustion chamber of the experimental engine

Timing of main injection	Injector used for auxiliary injection
—●— -10.5°	DN8S1
—○— -7.0°	DN8S1
—×— -3.0°	DN8S1
—□— -10.5°	DN4S1

Engine speed: 1250 r.p.m., BMEP: 5.5 kg/cm²
Injector used for main injection: 0.25mm ϕ -2 holes
(see Fig. 5-N)

Fig. 2 Effect of the timing of fuel injection into the auxiliary chamber

additional heat loss and throttling loss at the connecting passage are inherently unavoidable. Thus, the adequate ratio of the quantity of auxiliary injection to the total should be decided from the view point of the balance between the improvement on combustion and the above losses incurred by strengthening the swirl. On the other hand, the timing of the swirl creation in the main chamber, which depends mainly on the timing of the auxiliary injection, affects to a considerable extent the effect of swirl on the combustion process. Furthermore, the thermodynamic loss of the cycle of the engine increases, as an inevitable consequence, with the retardation of the timing of fuel injection, this loss becoming especially great in the case of a larger ratio of the quantity of auxiliary injection to the total. Fig. 3 illustrates the comparative specific fuel consumption and exhaust gas temperature curves when running at a constant speed and load but at various ratios of fuel delivery into the auxiliary chamber. In all cases, the injector DN8S1 was used for the auxiliary injection, and the beginning of the main injection was kept constant (10.5° before TDC). It will be shown that even a small quantity of fuel injected into the auxiliary chamber is effective in improving the combustion, and that a noticeable decrease in the specific fuel consumption can be obtained over the wide range of the timing of the auxiliary injection (from 20° before TDC to 10° after TDC); variation of the auxiliary injection timing is permissible since the cycle of the engine is determined

primarily by the timing of the main injection. With an increase in the quantity of fuel delivered into the auxiliary chamber, the allowable range of the timing of the auxiliary injection becomes gradually more narrow. However the minimum specific fuel consumption can be decreased to 216 ~ 219 g/BHP/hr, if the auxiliary injection takes place at a proper timing.

Fig. 4 shows the effect of the timing of the main injection at a fixed timing of the auxiliary injection and at several ratios of fuel distribution. In this case, there is also a similar tendency as observed in the preceding experiment; a remarked improvement on the combustion can be obtained by only a small amount of auxiliary injection. The allowable timing range of the main injection depends on the ratio of the quantity of auxiliary injection to the total.

From the above experimentations on the dual injection, it may be concluded that the effect of

Auxiliary injection: DN8S1, -10°
Main injection: $0.25\text{mm}\phi$ -2 holes (see Fig. 5-N)
Engine speed: 1250 r.p.m., BMEP: 4.98 kg/cm^2
Fig. 4 Effect of the timing of main injection

Main injection: $0.25\text{mm}\phi$ -2 holes (see Fig. 5-N), -10.5° , Auxiliary injection: DN8S1
Engine speed: 1250 r.p.m., BMEP: 4.98 kg/cm^2

Fig. 3 Effect of the ratio and timing of auxiliary injection

the quantity of the auxiliary injection depends to a considerable extent on its timing. There is little difference in the minimum specific fuel consumption obtainable in the range of the ratio 20~75 %, if the auxiliary fuel is injected at the suitable timing. Accordingly, in this case, since the thermal efficiency remains equal in spite of the increase in the auxiliary fuel, there will be evidence to show a considerable improvement in the combustion compensating for the increase in heat loss and throttling loss at the connecting passage. For example, in the case where 75 % of the total fuel is injected at TDC into the auxiliary chamber, the exhaust gas temperature is considerably higher than that at the other delivery ratio showing an equal fuel consumption as shown in Fig. 3.

Relative directions of the fuel injection and the air-swirl

In order to examine the influence of the direction of the fuel injection on the air-swirl, the direction of main injection was altered by utilizing different types of injector, as is shown in Fig. 5. Fig. 5 compares the specific fuel consumption and the exhaust smoke discoloration at a constant timing of the main injection and a fixed quantity of fuel being injected into each chamber. In the case when the injection is directed upstream in the air-swirl (III), the improvement in combustion is not realized and the exhaust smoke discoloration becomes higher, especially with simultaneous injections into both chambers. In the case of injecting downstream (I), the specific fuel consumption reaches a minimum with the simultaneous injections

into both chambers, but increases gradually with the retardation of the auxiliary injection. In the case of the injection aiming at the center of the main combustion chamber (II), the fuel consumption attains an optimum at the almost simultaneous timing of both injections. In the case where the two jets of fuel aim at both sides of the center of the chamber (IV), the exhaust is least smoky and the fuel consumption curve has two minimum

Main injection : 16.4 mg/stroke, -10.5°
 Auxiliary injection : 15.6 mg/stroke, DN8S1
 Engine speed : 1250 r.p.m.

Fig. 5 Influence of the relative directions of fuel injection and air-swirl

Curves indicate from the above: Pressure in the auxiliary chamber (N), Pressure in the main chamber (H), rate of pressure rise in the main chamber (H'), timing of auxiliary injection, and timing of main injection and crank position

Fig. 6 Examples of indicator diagrams

points, i.e., at 10° before TDC and 6° after TDC.

The exhaust smoke discoloration, as a whole, is reduced with the retardation of the timing of the auxiliary injection. This observation shows that the complete combustion of fuel can be realized by creating an effective swirl after the ignition occurs in the main chamber. For this purpose the auxiliary injection should be retarded to such a degree that the thermodynamic loss of the cycle of the engine is allowed within an appreciable amount; therefore the improvement on the specific fuel consumption is not always accompanied by an improvement on the combustion state.

4. Observation of the combustion process by means of measuring the pressure patterns and taking high-speed photographs

Typical examples of indicator diagrams are collated in Fig. 6, in which oscillograms Nos. 1~5 illustrate a comparison of the pressure patterns observed when running at a fixed timing of the main injection but varied timing of the auxiliary injection. No. 6 is the pressure pattern with the single injection into the main chamber (direct-injection type). Table 1 shows the summary of the operating conditions of the above oscillograms.

Fig. 7 shows the characteristics of combustion measured from the oscillograms, together with the results of the performance test, when varying the timing of the auxiliary injection. In this case, the quantities of fuel injected into both chambers were maintained constant, and the timing of the main injection was fixed at 10° before TDC. At an excessively advanced timing of the auxiliary injection, the ignition is initiated in the auxiliary chamber, while the maximum peak pressure in the main chamber becomes higher than that in the auxiliary chamber. With the retardation of the injection timing, the timing of the ignition and the peak pressures in the two chambers approach same values; the latter attains a minimum at 13° before TDC when the ignition occurs simultaneously in

the two chambers. With further retardation, the ignition starts in the main chamber earlier, and the peak pressure in the main chamber is lower than in the auxiliary chamber. The variation of the rate of pressure rise in the main chamber shows the same tendency, reaching a minimum at the above timing. These phenomena are considered to be caused by the decrease in the intensity of the gas outflow from the auxiliary chamber owing to the simultaneous combustion in the two chambers. As the timing of the auxiliary injection is retarded further, the peak pressures in both chambers as well as the rate of pressure rise increase once to some extent. After TDC, however, these values decrease suddenly and thereafter reach constant

Main injection: $0.25\text{ mm}\phi$ -2 holes, 16.2 mg/stroke , -10°
 Auxiliary injection: DN8SI, 15.4 mg/stroke
 Engine speed: 1250 r.p.m.

Fig. 7 Characteristics of combustion at varying timings of auxiliary injection

Table 1 Operating conditions of the oscillograms shown in Fig. 6

Oscillogram No.	Main injection			Auxiliary injection			Engine speed r.p.m.	Output BHP	Remark
	Injector mm	Timing deg	Quantity mg/st.	Injector	Timing deg	Quantity mg/st.			
1	2-0.25 ϕ	-10	16.2	DN8SI	-20	15.4	1275	4.8	Dual injection
2	"	"	"	"	-10	"	"	5.5	
3	"	"	"	"	-6.5	"	1280	5.3	
4	"	"	"	"	0	"	1260	"	
5	"	"	"	"	3	"	1240	"	
6	"	"	35.2	—	—	—	1295	5.6	Dir. inj.

magnitudes; $P_{\max}=39$ at, $(dp/d\theta)_{\max}=1.4$ at/deg. On the oscillograms Nos. 4 and 5, it is worthy to note that the pressure patterns in both chambers have two stages of pressure rise, i. e., the curve of the rate of pressure rise has two peak points.

By comparing the combustion state in the case of the later timing of the auxiliary injection (for instance, 6° after TDC) with that at 10° before TDC, it will be clear at a glance that a perceptible improvement is obtained without any expense of fuel consumption; i. e., the rate of pressure rise decreases to $1/8$ of that in the latter case, the discoloration of exhaust smoke decreases to $1/3$ and, furthermore, the decrease in the peak pressure in the main chamber is 15 kg/cm^2 . On the other hand, a strong swirl accompanied by the early auxiliary injection accelerates excessively the mixture formation during the period of the ignition delay in the main chamber, resulting in increases in the peak pressure, the rate of pressure rise and, moreover, the exhaust smoke.

The curve of "combustion noise" in Fig. 7 indicates the maximum amplitude of audible body-noise of the engine, recorded on the oscillograms together with the pressure patterns (Fig. 6). The curve in the range of injection timing earlier than TDC resembles that of the rate of pressure rise, but it shows a periodic rolling in the later range. This phenomenon is considered to be closely related to the distance between the two peak points on the curve of pressure rise in Fig. 6, Nos. 3, 4 and 5.

If the combustion state and the heat loss are quite independent of the timing of fuel injection, the specific fuel consumption and exhaust gas temperature should increase as the timing is retarded. However, the specific fuel consumption continues to remain low even in the timing later than 3° before TDC and decreases once at 6° after TDC, notwithstanding the gradual rise in the exhaust gas temperature. All these observations prove that the later auxiliary injection brings about a noticeable improvement on the combustion, decreasing the discoloration of exhaust smoke. After that, however, the specific fuel consumption increases gradually, for the loss owing to the imperfect expansion in the cycle can no longer be compensated for by this improvement.

Another investigation was made by means of high-speed photography, taken through the glass window mounted on the main combustion chamber. The cylinder head of a loop-scavenging two-cycle engine was reconstructed, as is shown in Fig. 8. Fig. 9 shows some examples of the high-speed pictures.

Film 1 shows the combustion process with the single injection into the main chamber. From the observation of the flame pattern in this case, it is noted that a very small amount of fuel is supplied into the auxiliary chamber by an air-stream during the compression stroke, and that this plays no appreciable part in mixture formation and combustion in the main chamber, for the weak outflow of gas starts barely from 25° after TDC. Film 2 shows the combustion process with the dual injection; equal amounts of fuel are injected simultaneously through the auxiliary and main injectors. The outflow of the flame from the auxiliary chamber occurs somewhat after the ignition in the main chamber (about 5° after TDC), and creates a high rate of swirling motion in the main chamber. The time available for the disappearance of the flame in the main chamber is shortened appreciably compared with that in film 1, in spite of the two-fold quantity of injected fuel. In film 3, in which the main fuel injection was done through a single orifice injector, the outflow of hot gases from the auxiliary chamber prior to the ignition in the main chamber may be seen. The other observations were made on the combustion process in the case of the auxiliary injection done at the later timing, from which it was found out that the flame pattern maintained partially the

Ratio of auxiliary chamber to total compression volume: 25 %
 Area ratio of connecting passage: 0.25 % of piston area
 Compression ratio: 18 : 1

Fig. 8 Sectional view of the apparatus for high-speed photography

features of the fuel spray, until the outflow of gas from the auxiliary chamber began.

5. Conclusions

The process of the gas outflow from the auxiliary chamber depends mostly on the pressure gradient between the two chambers; in effect, on the rate of combustion in both chambers and also on the displacement of piston. It is very difficult to accurately observe the process even in a pre-combustion chamber. Therefore, it seems to be impossible in the case of the dual injection to presume the intensity and timing of the combustion swirl created by the auxiliary injection. However, the following conclusions as to the effect of combustion swirl upon the Diesel-combustion might be derived from the investigation made above:

(1) Swirl creation prior to the fuel injection into the main combustion chamber brings about a small improvement in the combustion state and in the exhaust discoloration, but is accompanied by

an unfavourably rapid and high pressure rise, which results in louder combustion noise.

(2) Swirl creation after the ignition of fuel spray is very effective in shortening the after-burning and also in decreasing the exhaust smoke. If fuel is injected into the auxiliary chamber after TDC, the noiseless and smokeless combustion with fairly low peak pressure is realized. This improvement on the combustion state, however, does not always decrease the fuel consumption in consequence of a lower thermodynamic efficiency of the cycle.

(3) The rate of the fuel delivered into the auxiliary chamber to the total amount does not have a large influence on the minimum specific fuel consumption, if the timing of the auxiliary injection is properly chosen in each case. On the other hand, the stronger swirl caused by increasing the quantity of auxiliary injection is more effective in increasing the output of an engine.

Fig. 9 Examples of high-speed photographs

On the application of this dual injection system to the combustion in a free-piston gas-generator⁽⁴⁾, it will be expected that the exhaust gas of higher temperature can be produced without the sacrifice of higher fuel consumption by injecting a small amount of fuel into the auxiliary combustion chamber at the later timing. An additional advantage will be the realization of a smooth and smokeless combustion. This fact is very effective

in improving the thermal efficiency of the power plant combined with the gas turbine.

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The Influence of the Combustion Products on the Piston-Ring Wear of a Diesel Engine

(1st Report, Motoring)*

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A small type Diesel engine is driven by a motor with various $\mu U/W$ values, then water steam is added to the inlet air, and exhaust gases of the other combustion engine is sucked in. The weight loss of the piston-rings is measured, and the following results have been obtained.

(1) When the oil film is thin to a certain extent and at the same time, the combustion products attack on it, the wear is increased greatly; however, if one of these conditions is absent, the wear becomes very small.

(2) When the exhaust gases of the fuel with high sulphur content are sucked in, and the cooling water temperature is low, the engine is corroded very violently by the condensed water, containing the sulfuric acid.

1. Introduction

The piston-ring wear will be very small, if the products of combustion do not take part in the friction between the ring and the cylinder; that is, only the air is compressed or expanded by the motoring. Then the remarkable characteristic of the phenomena of wear in the internal combustion engine in comparison with the other machine parts is the promotion of wear by the combustion products.

Recently Japan consumes mainly the crude petroleum from the Near and Middle East. Naturally the diesel oil which does not pass the desulphur process contains much sulphur compounds. So it is important to reduce the corrosive wear

which is caused by SO_3 in the combustion products of diesel engines.

To know the effects of each product of combustion on the piston ring wear, in this study a small-type diesel engine is driven by a motor and the air only, exhaust gas only or its partial component is introduced.

2. Measurement of wear

Fig. 1 shows the experimental engine in which the temperature difference of the cooling water is made as small as possible. The engine is a $70\phi \times 100$ mm single-cylinder Yanmer K-2 type precombustion-chamber diesel engine, with maximum output 3 HP at 1300 r.p.m., and average hardness of the cylinder wall H_RB 102. It has four compression-rings with the size $B \times T = 2.5 \times 2.8$ mm, and one oil-ring 4×2.5 mm with 2.5 mm width groove; these piston-rings are made by one-casting

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