A Study on Combustion of Direct Injection Diesel Engine with 150MPa Injection Pressure

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ABSTRACT

The characteristics of diesel combustion with high pressure fuel injection were investigated, using a naturally aspirated single cylinder engine and high pressure injection equipments which can produce over 250MPa injection pressure. The performance test results showed that high injection pressure in combination with smaller nozzle hole diameter provided the best performance and exhaust emission level. Also the effect of swirl was investigated. At a 150MPa injection pressure, a quiescent zero swirl showed the best performance and emission.

Observation and analysis of combustion were performed using high speed shadow-graph at a definite NOx emission, while varying the parameters of both injection pressure and swirl ratio. The result showed that in the case of high pressure injection, ignition and combustion situations are fairly different from the ordinary case of below 100MPa injection pressure.

INTRODUCTION

In order to cope with the growing demand for reducing particulate and NOx emissions from diesel engines, without worsening fuel economy, a better understanding of diesel combustion is becoming a more important subject. The key to such an understanding is a knowledge of the injection system parameters, especially injection pressure, and their interaction with in-cylinder air motion.

Recently high pressure injection is regarded as the most promising method by many researchers. The authors already presented that a high pressure injection is very effective for minimizing fuel droplet size, increasing air entrainment and improving particulate and NOx emissions from diesel engine(1)(2)* , and also at an injection pressure below 100MPa, the swirl still has an important role in the fuel-air mixture formation processes(3). In this report the authors examined further the effect of increasing the injection pressure over 150MPa on engine performance and combustion characteristics, using newly developed high pressure injection equipments HPIE and AHPI. The study includes performance and emission evaluation, indicator diagram analysis, and observation of combustion phenomena using high speed shadow-graphs.

EXPERIMENTAL ENGINE AND TEST CONDITIONS

The engine used in the experiment was a single cylinder naturally aspirated engine. Specifications of this engine are shown in Table 1. The base line of this combustion system has a jerk pump with pipe, 4 hole nozzle, square combustion chamber, and high swirl inlet port.

Two types of high pressure injection equipment, HPIE and AHPI are used in the tests. Details of these equipments are explained in the previous paper(2). Injection rate patterns of these equipments have some differeces. HPIE has a "gradual rise and sharp cut" injection rate, while AHPI has a "square" injection rate. Conventional jerk pump is also used as a base line.

All of the data obtained are at 1000r.p.m. engine speed, excess air ratio of 1.25.

Table 1 - Engine specifications

Type of Engine	Single Cylinder 4-stroke Cycle Direct Injection Naturally Aspirated
Bore × Stroke	135 mm × 140 mm
Displacement	2004 cm^3
Compression Ratio	16.5
Cylinder Head	2 Inlet Valves 2 Exhaust Valves
Nozzle Position	Center of Combustion Chamber and Installed vertically

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* Numbers in parentheses designate references at end of paper.

RESULTS OF ENGINE PERFORMANCE TEST

Effect of Injection Pressure

In the same NOx emission condition(1200ppm), the combustion characteristics by increasing injection pressure with the base line combustion system (square combustion chamber, swirl ratio 2.6, injection nozzle 0.38mm×4) were obtained using HPIE as shown in Fig.1. By increasing injection pressure, smoke decreased significantly and also the combustion period decreased. Fig.2 shows a comparison of rate of heat release curve between jerk pump as base line (Type A) and HPIE with 150MPa injection pressure (Type B). It can be seen that the ratio of premixed combustion (the 1st peak of rate of heat release) increases with the increase of injection pressure.

Effect of Nozzle Hole Diameter

In order to reduce premixed combustion, it is effective to apply a smaller nozzle hole area. So the effect of nozzle hole diameter was examined with 6 hole nozzle configurations, same injection pressure(150MPa) and timing conditions using AHPI. A quiescent(0.4 swirl ratio and ϕ 98mm shallow-dish chamber) combustion system was used in this test. As shown in Fig.3, smoke decreases by decreasing nozzle hole diameter, while the best fuel economy was obtained with a 0.17mm nozzle hole diameter at the same injection timing conditions. It is because by decreasing nozzle hole dia., injection duration and combustion period tend to increase as shown in the figure.

Effect of Swirl

Effect of swirl was investigated with 0.17 mm ×6 hole nozzle and ϕ 98mm shallow-dish chamber combustion configurations, using HPIE. As shown in Fig.4, the low swirl case (Type C) showed the best performance and emission at the same injection timing conditions. By increasing the swirl ratio, both NOx and smoke emission increase and fuel economy worsens. The heat release curve of Type C is compared with the high swirl case (Type D) as in Fig.5. The high swirl showed an increased heat release at the early stage of diffusion burning, and a shorter combustion period, which is generally considered preferable for improving smoke and fuel economy.

OBSERVATION OF COMBUSTION PROCESS

Observation was conducted from the lower side of the piston with quartz glass window. Four types of combustion systems(Type A,B,C,D) are compared as shown in Table 2. Owing to the restrictions from installing glass inside the piston, a ϕ 80mm cavity was used instead of a square cavity, for Type A and B. Also a ϕ 90mm cavity was used instead of a ϕ 98mm shallow-dish

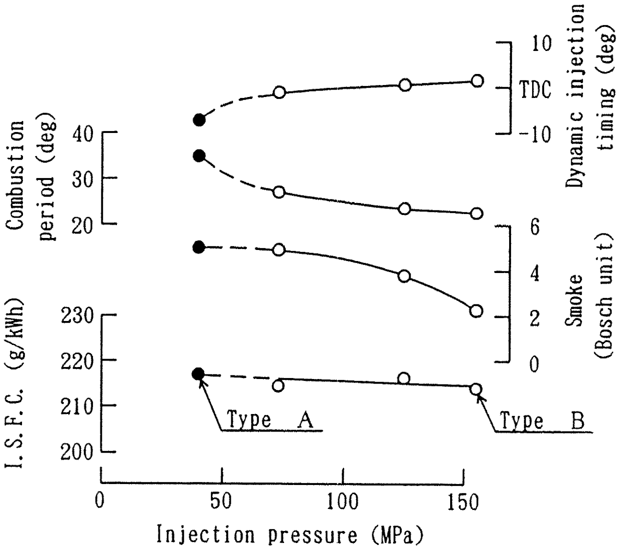


Fig.1 Effect of injection pressure on exhaust smoke and combustion characteristics

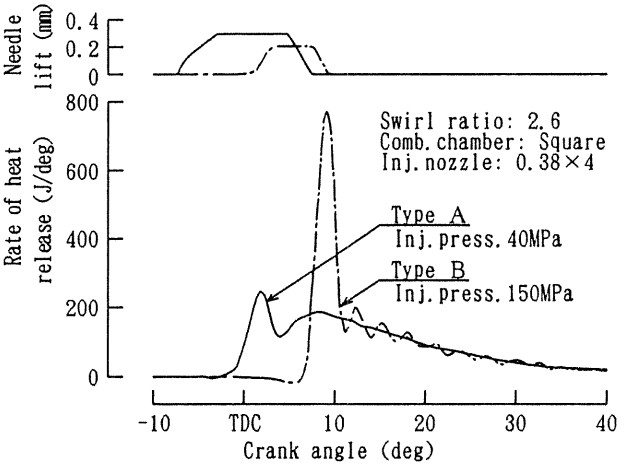


Fig.2 Combustion characteristic curve at different injection pressures

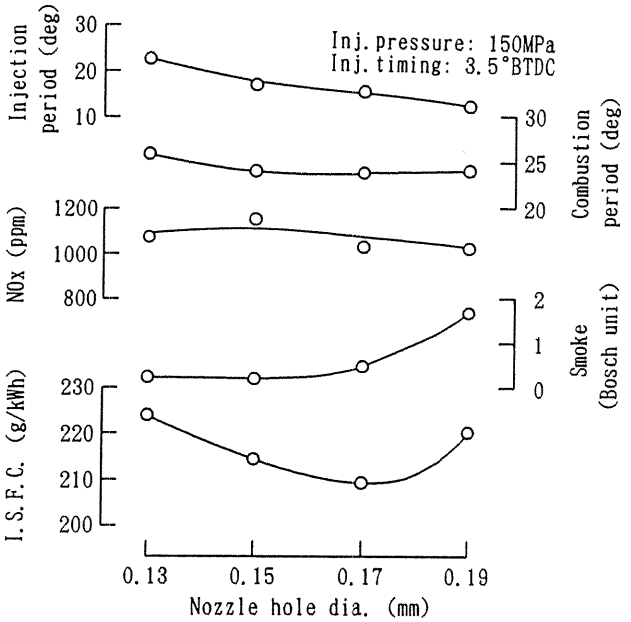


Fig.3 Effect of nozzle hole diameter on fuel economy and exhaust smoke

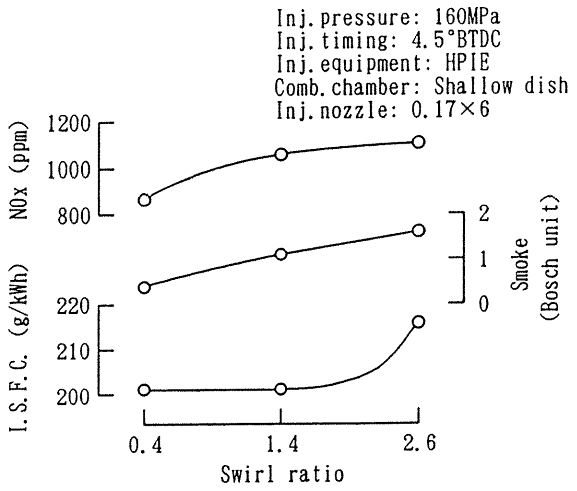


Fig.4 Effect of swirl on fuel economy and exhaust smoke

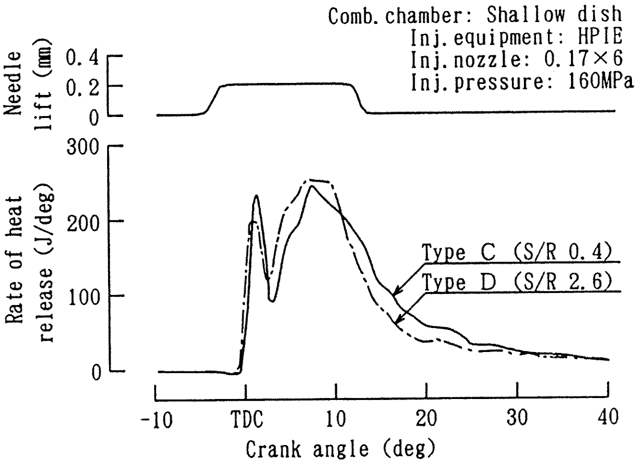


Fig.5 Combustion characteristic curve at different swirl ratios

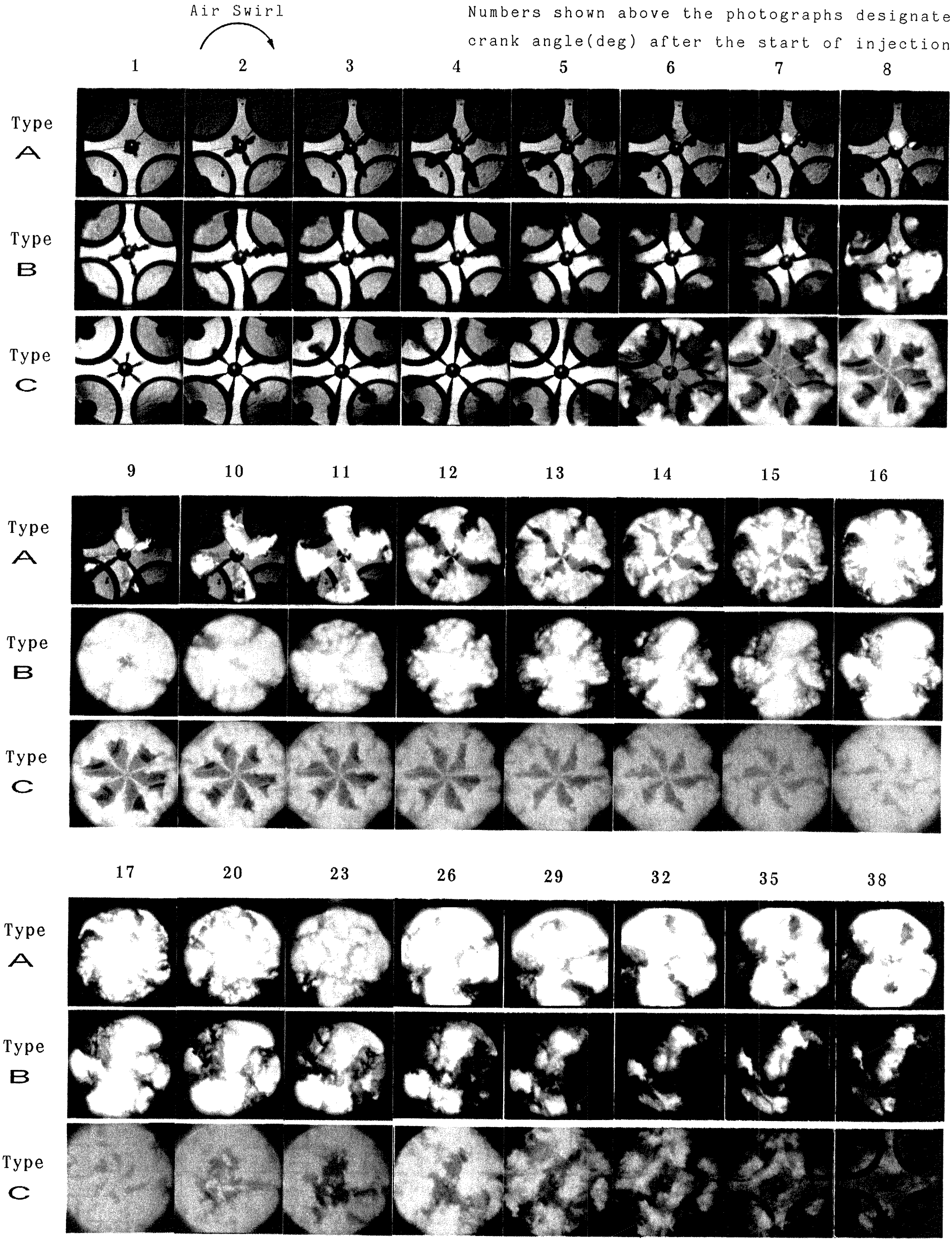
cavity, for Type C and D. The high speed camera was driven at a rate of 6000 frames per second, which corresponds to 1 deg.C.A./1 frame.

Both Type A(40MPa injection pressure) and Type B(150MPa injection pressure) combustion photographs are compared in Fig.6. When comparing photographs at the early stage of injection, Type B gives an increased penetration. In Type A, the first ignition flames are observed in the neighbourhood of the nozzle tip, and tend to expand to the downstream of each spray cone. At the middle stage of combustion, burning mainly occurs at the spray and wall jet region. While in Type B, the first ignition flames are observed at the wall jet region, and expand quickly and cover the whole observation area. It seems that the spray impinges on the cavity wall, changes direction along the cavity wall as a wall jet, and then impinges on the wall jet of the neighbouring spray, and turns back to the center of the cavity. This well developed comparatively lean fuel air mixture burns quickly, and at the middle stage of combustion, burning mainly occurs at the spray and wall jet region as observed in Type A. It is noticed that, even at the end of combustion, there still remain lumps of fuel rich mixtures burning slowly in both cases. These lumps have their origin in each fuel spray and its wall jet. This suggests that entrainment of fresh air into the spray is not enough in both cases.

Combustion photographs with high pressure, small nozzle hole, low swirl and shallow-dish cavity combinations (Type C) are also compared in Fig.6. Photographs of Type C show different characteristics. When comparing photographs at the early stage of injection, in Type C the first ignition flames are observed at the wall jet region, and tend to expand slowly to the center of the cavity. At the middle stage of combustion,

Table 2 Combustion system specifications for high speed photograph

Type	A	B	C	D
	Low Inj. Press. Large Nozzle Hole Dia. High Swirl	High Inj. Press. Large Nozzle Hole Dia. High Swirl	High Inj. Press. Small Nozzle Hole Dia. Low Swirl	High Inj. Press. Small Nozzle Hole Dia. High Swirl
Injection Pump System	Jerk	HPIE	←	←
Peak Injection Pressure (MPa)	40	150	←	←
Injection Nozzle	0.38 mm×4	←	0.17 mm×6	←
Combustion Chamber Cavity dia. (mm)	80	←	90	←
Intake Port Swirl Ratio	2.6	←	0.9	2.6



Numbers shown above the photographs designate crank angle(deg) after the start of injection

Fig 6 Comparison of combustion photographs between Type A,B and C an unburned fresh air area still remains at the cavity center. Burning mainly occurs at the wall jet region. But the dark color area at the wall jet region which is observed in Type A and B, and may correspond to the generated soot, is not observed in Type C. In the middle stage of combustion, the spray shape diminishes, and also the luminous flame which may correspond to the lumps of fuel rich mixtures, diminishes very quickly. These phenomena suggest that, in Type C, a well developed lean and homogeneous fuel air mixture is already formed in the spray, before the spray reaches the cavity wall. And also by this, the spray already loses its momentum by entraining so much air before reaching the cavity wall.

Combustion zoom-up views using a 35000 frame/sec. high speed drum camera of both Type C and Type D(higher swirl than Type C) are compared in Fig.7. Photographs of these two types are rather similar. But in detailed analysis, some differences exist. When comparing photographs at the early stage of combustion, in both cases the first explosion and ignited flames are observed at the spray swirl downstream near the wall jet region, which expands the burnt mixture to the downstream of swirl. In Type C, the boundary between the mixture and fresh air is clear. While in Type D the boundary is obscure, which suggests that mixing and diffusion in this region is fairly active. At the middle stage of combustion, photographs of both types are almost similar, and at the last stage, Type D shows faster combustion end as estimated from the heat release curve shown in Fig.5. But it is noticed that the lumps of fuel rich mixture burns slowly at the end of combustion in Type D. This may suggest that the good air entrainment into the spray is somewhat been disturbed in Type D.

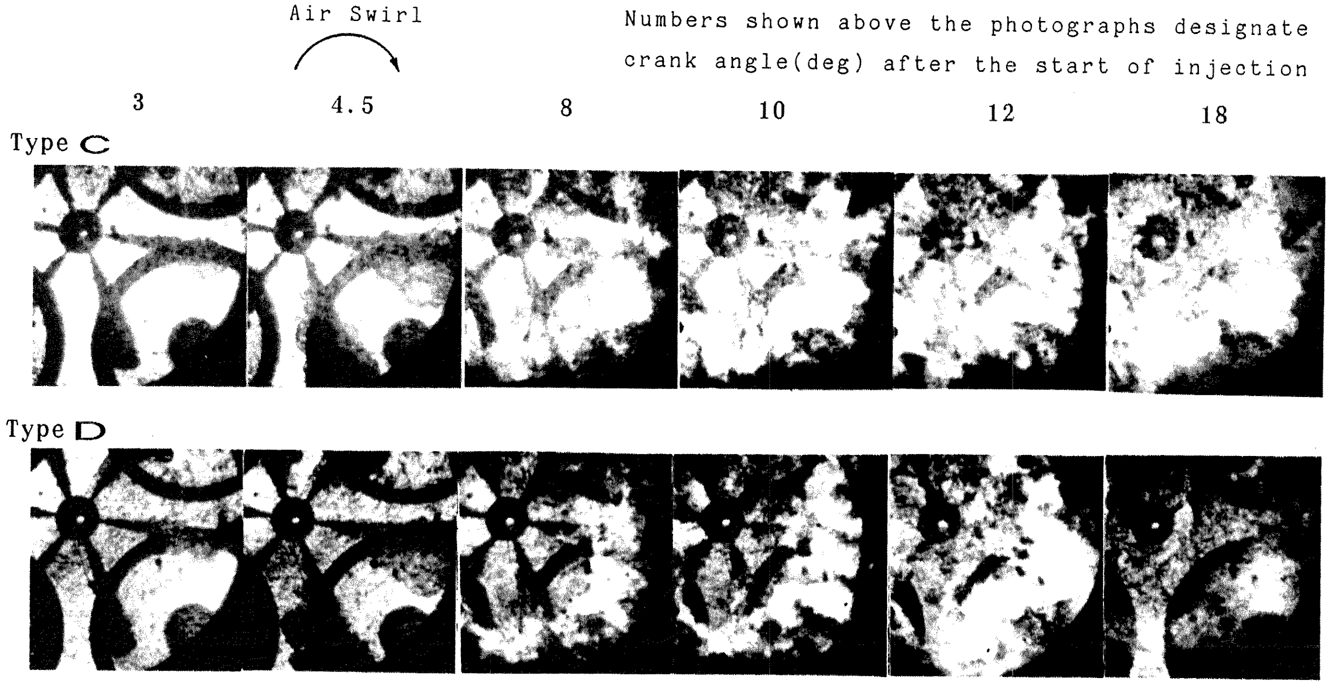
DISCUSSION

Comparing the photographs between Type A and B,C,D, it may be concluded that by increasing injection pressure, the ignition point tends to shift from the neighborhood of nozzle tip to the wall jet region. In the case of high pressure injection there exists an intense air entrain movement into the spray, and the lean fuel air mixture formed outside the spray core may be dragged into the core and brought to the wall. It seems that the differences of ignition point result from the differences of the air entrainment.

Air entrainment and uniform fuel air mixture formation seems to play an important role in combustion processes. High pressure in combination with smaller nozzle hole is very effective for this. As introduced by Wakuri's momentum theory, the rate of entrainment of surrounding gas Js is expressed by the following equation.

$$Js \propto V^{1/2} d_0^{-1/2} t^{3/2} Jf$$

Where, V means the spray initial speed, d_0 means nozzle hole diameter, t means the time after the start of injection, and Jf means the rate of injected fuel. Namely, by increasing injection pressure and decreasing nozzle hole diameter,better air entrainment is expected. Fig. 8 shows the mean equivalence ratio in the lateral cross section at the relevant distance along non-



Numbers shown above the photographs designate crank angle(deg) after the start of injection

Fig. 7 Comparison of combustion photographs between Type C(low swirl) and D(high swirl)

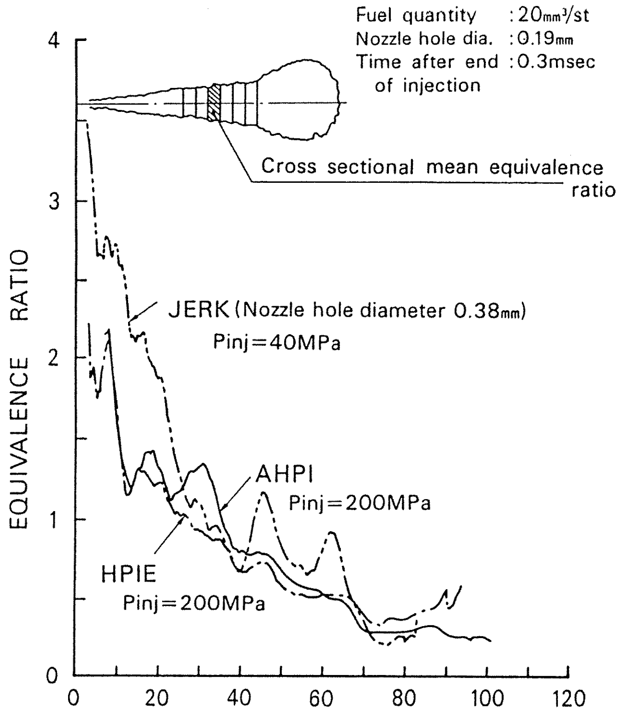


Fig.8 Effect of injection parameters on equivalence ratio inside the non-evaporating fuel spray (from Ref. 2)

evaporating fuel spray axis. These data are obtained using image analysis techniques, and assuming fuel droplet size constant. Combinations of high injection pressure and smaller nozzle hole dia. gives better air entrainment. Besides, micro turbulence generated by a smaller nozzle hole, may contribute much to the turbulent mixing speed inside the spray.

Further investigation is necessary for clarifying the mechanism of bad performance by the swirl in the case of Type D. One assumption for this is due to overlapping of sprays as proposed by several researchers(4)(5) which may occur at the wall jet region. Another possible assumption for this is due to interference between sprays which may occur at the cavity center. By the interference between sprays, fresh air surrounding the spray is replaced with the fuel-air mixture, resulting in the reduction of fresh air entrainment. In Type D the air motion around the spray should be fully turbulent, owing to the active air entrainment motion by high injection pressure, accompanied by high swirl motion. Diffusion of the mixture may occur far earlier in accordance with the complicated turbulent eddy motion, than expected by solid body rotation. It seems important not to disturb the fresh air entrainment by excess air motion, when the fuel spray itself has sufficient ability to produce a lean and homogeneous mixture.

Diesel combustion is considered, as characterized in the combustion photographs of Type C, having a premix region at the center of cavity, and the premixed mixture burns mainly downstream of spray and wall jet region through the whole injection period. The mixture thus formed seems to dominate the whole combustion process, because the lumps of fuel rich mixtures which remain at the end of combustion have their origin in each spray. It may be concluded that for the modelling of diesel combustion, it is necessary to describe both the growth of the spray (which means the rate of fresh air entrainment into the spray by spray momentum, swirl motion, and other factors) and the homogeneity or mixing process inside the spray.

CONCLUSIONS

Combustion characteristics with 150MPa injection pressure were examined using newly developed high pressure injection equipments HPIE and AHPI. The results obtained are;

1. With quiescent combustion systems, a smaller nozzle hole gives less smoke emission. By the combustion observation, generation of soot in the wall jet region is hardly recognized, which suggests that a lean and homogeneous mixture is already formed before the spray reaches the cavity wall.

But an excess reduction of hole dia. brings an increase of injection duration and combustion period, resulting in worse fuel economy.

2. Effect of swirl was also investigated, with 0.17mm×6 nozzle geometry and shallow-dish combustion chamber combination. The quiescent low swirl combustion showed the best performance and emission.

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