Fuel Injection Pressure and Nozzle Orifice Diameter in Direct-Injection Diesel Engines

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ABSTRACT

A high-pressure injection is effective in reducing diesel smoke. However, only a little is known as to how to select the best combination of spray number and nozzle orifice diameter at high injection-pressure. To clarify this, effects of the orifice diameter on the spray characteristics at an elevated injection pressure are assessed in detail based on existing spray theories. The results suggest that an increase in injection pressure with a reduction of nozzle-orifice diameter reduces the fuel-air ratio and enhances the microscopic mixing, thereby preparing a homogeneous mixture within the spray. In addition, it was found that a nozzle having a smaller orifice diameter should be combined with a lower swirl intensity and a higher spray number at high injection-pressure. These were ascertained by engine bench tests and observations using high-speed photography and two-dimensional soot imaging in diesel-flame by means of a laser-light sheet method.

INTRODUCTION

In a direct-injection diesel engine, an elevated injection pressure with an ordinary large orifice diameter causes an explosive combustion simultaneous with an increase in NOx. Thus, a combination with a small nozzle orifice diameter should be employed for an injection pressure at greater than 150 MPa using a low swirl shallow-dish type combustion chamber.[1] On the other hand, another report suggests that a reduction in the nozzle orifice diameter is essential rather than an increased injection pressure for keeping the smoke level lower in a deep-bowl chamber unlike the case of a shallow-dish chamber.[2] However, these reasons have not been made clear.

The present paper discusses factors that may affect spray and combustion at elevated injection pressure for the case of direct injection. To this end, the sensitivity of injection pressure on the spray characteristics are derived for an assigned relationship between the nozzle orifice diameter and the injection pressure. The spray characteristics are first described based on the momentum spray theory and other spray models. The results of analysis are compared with those of engine bench tests, and observations of diesel flames by direct high-speed photography and the two-dimensional soot imaging in flames by a laser-light sheet method.

THEORETICAL

Description of spray characteristics

The present theoretical studies on spray characteristics were carried out based on the momentum spray theory [3] and the stochastic spray model.[4] The latter enables us to describe the diesel combustion process from the viewpoint of turbulent mixing.

Figure 1 illustrates one spray with number N. Let the pressure difference between nozzle-sack and surrounding air be Δp, the fuel density ρ_F, the ambient air density ρ_A, and the spray corn angle θ, each being assumed to be constant during injection. We find the following relations.

First, on the assumption that the injection quantity of Q_F is injected at a constant rate during the injection time t_e, the injection velocity u_F and t_e,

$$u_F=\sqrt{2\Delta p/\rho_F} \tag{1}$$

$$t_e=4Q_F/(N\pi d_N^2 u_F) \tag{2}$$

The fuel flow rate J_F and the entrained air flow rate J_A are expressed as

$$J_F=(\pi/4)N d_N^2 \rho_F u_F \tag{3}$$

$$J_A \propto J_F\sqrt{u_F\, t/d_N} \tag{4}$$

where t denotes the time from the start of injection. It should be noted that J_A/J_F increases with a reduction of d_N and an increase in u_F. For this reason, a smaller nozzle diameter gives

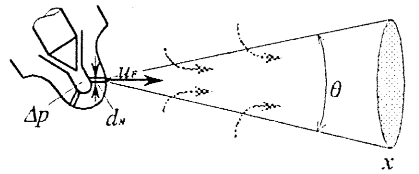

Fig.1 Illustration of a fuel spray injected from one of some orifices

a leaner mixture in the spray. The distance of spray tip from the nozzle, or penetration of spray, x, decreases with a decrease in d_N, i.e.,

$$x \propto \sqrt{u_F t d_N} \quad (5)$$

The Sauter mean diameter D_S is said to be directly proportional to d_N, whereas another report states that D_S is proportional to the −2/3 power of Δp at low injection pressure and tends to be constant at high injection pressure.[5] Consequently, D_S might be expressed as follows.

$$D_S = d_N f(\Delta p, \cdots) \quad (6)$$

From this equation, it might safely be stated that an increase in Δp and a reduction of d_N will increase the rate of atomization and evaporation of spray droplets.

Second, we look at turbulence and turbulent mixing within the spray. Both are significant in controlling the heterogeneity of the fuel concentration in a spray. According to a previous study, the power of turbulence generation P_T in a spray[4] is described as follows.

$$P_T = (\eta_T/2) J_F u_F^2 / M \quad (7)$$

where M denotes the instantaneous mass of the mixture in the spray and η_T the conversion efficiency of jet power to turbulence power. The assumption that $J_F \ll J_A$ may hold except for the very initial stage of injection. Hence, we may have

$$M = \int (J_F + J_A) dt \approx \int J_A dt \propto J_F \sqrt{u_F t^3 / d_N} \quad (8)$$

if η_T is assumed to be constant. This follows

$$P_T \propto \sqrt{u_F^3 d_N / t^3} \quad (9)$$

It is noted that P_T is proportional to the 3/2 power of u_F and to the square root of d_N. The integral scale of turbulence L_T is inversely proportional to d_N,[6] i.e.,

$$L_T = d_N \sqrt{\rho_F / \rho_A} \quad (10)$$

For an uniform and isotropic field of turbulence, the dissipation rate of turbulence is in equilibrium with the generation rate at a steady state. This leads to

$$P_T = u'^3 / L_T \quad (11)$$

where u' denotes the root-mean-square velocity of turbulence. The dissipation rate of turbulence ω is

$$\omega = u' / L_T \propto \sqrt{u_F / d_N} \quad (12)$$

ω might be regarded as being microscopic mixing rate for other scalar quantities, such as species concentrations and specific enthalpy. Hence, the heterogeneity of a mixture would decay quickly with increasing ω. In other words, an increase in u_F at reduced d_N would result in a more homogenized mixture having a more uniform temperature.

The effect of air swirl on a spray is dictated by a spray theory which takes air swirl into account.[7] In the case when the deflection of the spray due to swirl is small, the spray-tip trajectory might be determined in first approximation by the ratio of momentum on spray axis, I_F, to that in the direction perpendicular to the axis, I_S. Each is expressed as

$$I_S = \rho_A \int \omega_S r d_N u_F dr = \rho_A \omega_S d_N u_F r^2 / 2 \quad (13)$$

$$I_F = \rho_F (\pi d_N^2 / 4) u_F^2 \quad (14)$$

where ω_S and r denote the swirl angular velocity and the distance from nozzle, respectively.

$$I_S / I_F \propto 2 \omega_S r^2 / (\pi d_N u_F) \quad (15)$$

I_S / I_F increases with a reduction of d_N. To keep the profile of spray unchanged at a reduced nozzle orifice diameter, ω_S should be decreased in proportion to d_N.

Effects of injection pressure and nozzle orifice diameter on spray characteristics

The above relationships will indeed give how d_N and u_F affect spray characteristics, but it is almost beyond expectation to give a perspective of finding out the optimum nozzle orifice diameter. For this reason, the sensitivity of injection pressure on each factor has been calculated based on the above relationships for a given relationship between injection pressure difference Δp and nozzle orifice diameter d_N as follows.

$$d_N \Delta p^{\alpha} = \text{const.} \quad (16)$$

where the index α represents the degree of reduction in d_N for the case when Δp is increased. We may determine the index β in the following formula for each quantity characteristic to spray, Ψ, as a function index α.

$$\Psi \propto \Delta p^{\beta} \quad (17)$$

β is the sensitivity of injection pressure on Ψ for a given α. Figures 2 and 3 show the results obtained from the above-mentioned relationships. $\alpha = 0$ is the case when Δp is increased at a fixed d_N, $\alpha = 0.25$ the case when the nozzle orifice area is changed inversely proportional to the fuel flow rate, and $\alpha = 0.5$ the case when the nozzle orifice area is altered inversely proportional to Δp. For example, we note that when Δp is doubled at $\alpha = 0$, t_e falls by 30%, whereas J_F and ω rise by 41% and 19%, respectively. This explains why an explosive combustion takes place at high pressure injection without a reduction in d_N.

The fact that the Sauter mean diameter D_S increases with α suggests that the spray becomes closer to a gas jet at a higher injection pressure with a smaller nozzle orifice.

When $\alpha > 0.5$, x decreases and I_S / I_F increase with α. A decreased penetration in the radial direction of a combustion chamber may injure air utilization due to thermal pinch[8] by swirl. To avoid the thermal pinch, the swirl intensity should be reduced when a smaller nozzle orifice is used. In addition, the spray number should be increased to reduce t_e.

Also, it should be noted that the index for the ratio of air and fuel flow rates, J_A / J_F, increases with α. This suggests that the mixture formed in a spray becomes more diluted as Δp increases and as d_N is reduced.

Indices for P_T and ω are always positive over a wide range of α. This means that an elevated injection pressure with a smaller nozzle orifice is effective in attaining a homogeneous mixture. An increase in ω is helpful in attaining a less heterogeneous mixture, reducing local fuel-rich and high temperature zones. This is one of the significant reasons for a lower soot yield at a higher ω. This might be effective as well in terminating the reactions of oxides of nitrogen because of early disappearance of high temperatures.

DISCUSSION

Combustion and spray characteristics for various cases of α

In this section, we discuss various cases of α to find out how the nozzle orifice diameter acts on spray characteristics. α may be divided into three cases as follows.

Case of very large α This is the case when the nozzle orifice diameter, d_N, is excessively small in comparison with the increase in injection pressure. For a sample case when a conventional injection system has d_N = 0.3 mm and Δp = 50 MPa at α = 1.5, d_N must be reduced to 0.05 mm for an elevated injection pressure at 150 MPa. If such a condition is realized, there is a possibility of attaining a very homogeneous and lean combustion that may lead to lower NOx and smoke emissions. However, it is necessary to increase the spray number to keep the injection period short enough to suppress degradation of combustion. Also, the spray penetration might be not enough and the spray-tip be easily deflected by the cross-wind due to swirl. For this reason, air utilization in the periphery of the combustion chamber would be sacrificed at an ordinary swirl intensity. Therefore, swirl intensity should be much weaker. Injection from the side wall of a combustion chamber should be employed, unlike ordinary central injection, to maintain satisfactory air utilization.

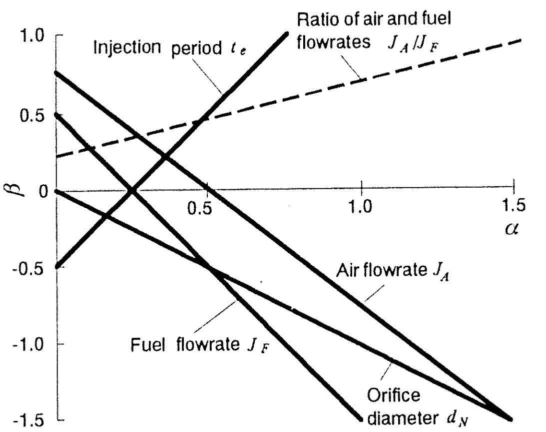

Fig.2 Relationship between β and α for spray characteristics

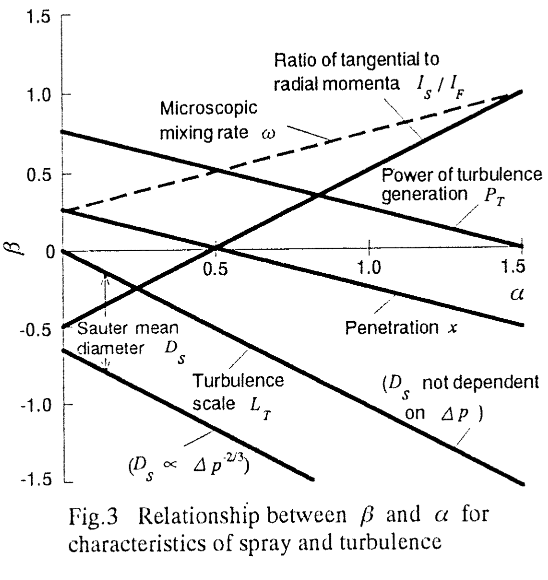

Fig.3 Relationship between β and α for characteristics of spray and turbulence

Case of α around 0.5 This case might be of practical use in ordinary central injection for direct-injection. A typical example is that reported by New A.C.E.[1]. For an increase in the injection pressure from 40 to 150 MPa, the nozzle orifice diameter is reduced from 0.38 mm to 0.17 mm, swirl ratio is reduced from 2.0 to 0.4 and spray number is altered from 4 to 6. In this case, a ratio of injection pressure rise is 4.14, and hence α is 0.57. According to Fig.2, this value of α gives the index of t_e to be 0.64. This means that the injection period is 2.48 times that at the use of the original pump for the same spray number. The spray number should be increased to keep the injection period. In fact, the injection period is decreased by 19 % from that of the original jerk pump for increased spray number. Such a longer injection period may lower the NOx concentration. At α = 0.57, I_S/I_F is increased by 10% from the original case. Consequently, the swirl intensity must be reduced, so as to compensate this increase. However, a further reduction in the swirl intensity is needed because the spray number is increased in this case.

Case of α around 0.25 At α around 0.25, it is not necessary to increase spray number, because the index of injection period is zero as shown in Fig.2. The swirl intensity should be increased to avoid the tendency towards an over-penetration. Based on experimental results, combustion in such a case will be shown below.

Table 1 shows main specifications of the test engine together with injection systems employed. The baseline jerk pump injection system has a peak injection pressure of 25 MPa and the nozzle orifice diameter of 0.29 mm. At high-pressure injection, the nozzle orifice diameter, d_N, was reduced from 0.29 mm to 0.20 mm, giving α = 0.25 at a peak injection pressure of 90 MPa. The engine was operated at an engine speed of 1800 rpm and at an equivalent ratio of 0.54. The high-pressure injection system, KD-3, has a novel principle for injection using spool acceleration and oil-hammering in a convergent injection pipeline. For details, see the previous report.[9]

Figure 4 shows the results of Bosch smoke density,

Table 1 Main specifications of a engine and injection systems

Engine type	Single-cylinder, Four-stroke, Direct-injection
Bore × Stroke	102 mm × 105 mm
Displacement	0.857 ℓ
Compression ratio	17.8
Combustion chamber	Deep-bowl (d/D = 0.56)
Injection pump & Nozzle	KD-3, DLLA150P204 Bosch PE2A, DLLA150S294

NOx concentration, exhaust temperature T_e, peak cylinder pressure p_{max}, the ignition delay period τ and the injection period θ_e for various peak injection pressures of p_{Nmax}. Figure 5 shows the courses of cylinder pressure p, the rate of heat release $\dot{q}$, rate of cylinder pressure rise $\dot{p}$, injection pressure p_N, and nozzle needle-lift h_Z for four peak injection pressures p_{Nmax}.

In Fig.4, it is noted that the ignition delay τ is shortened by 3 degrees in crank angle from that with the original jerk pump. Smoke is significantly reduced, whereas the NOx concentration increases with p_{Nmax}. To obtain the same t_e, or in other words, to attain α = 0.25 from the case of the ordinary jerk pump, p_{Nmax} was selected to be 90 MPa. At p_{Nmax} > 90 MPa, t_e is less than that of jerk pump. From Fig.5, it is seen that $\dot{q}$ at the middle stage of combustion is higher and the combustion period is shorter than at the jerk pump. These may be effective in lowering smoke density. However, it seems likely that a high $\dot{q}$ at the middle stage of combustion produces local high temperature zones, and as a result NOx concentration becomes higher.

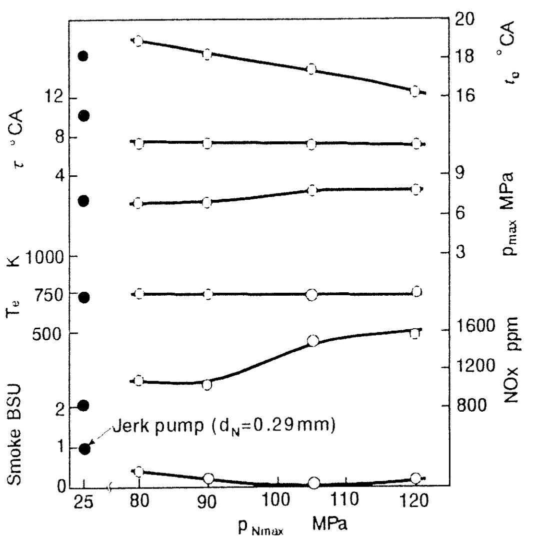

Fig.4 Effects of injection pressure, p_{Nmax}, on smoke, NOx, exhaust temperature, T_e, peak cylinder pressure, p_{max}, ignition delay, τ, and injection period, t_e (engine speed 1800 rpm, equivalence ratio 0.54, injection timing 11 ° BTDC for high pressure injection and 15° BTDC for jerk pump, and nozzle orifice diamater, d_N, 0.20 mm for high pressure injection)

At p_{Nmax} less than 90 MPa, as shown in Fig.5, the rate of heat release $\dot{q}$ is lower in the initial stage of combustion than at the jerk pump. At the same time, the peak of rate of cylinder pressure rise, $\dot{p}$, is lower. This might be caused by a decreased ignition delay and the reduced amount of premixed mixture formed. However, at p_{Nmax} greater than 90 MPa, or α < 0.25, the amount of the mixture is increased, due to the high injection rate, as shown in the β index of J_F in Fig.2. This results in a rapid pressure rise in cylinder and higher NOx concentration. Therefore, in view of reducing the NOx concentration, a nozzle orifice diameter should be selected so as to keep α = 0.25.

Dilution and homogenizing of mixture

Among factors that may affect the spray characteristics, the formation of a fuel-leaner mixture and a higher turbulence dissipation rate within the spray deserves special attention. To inquire into these factors in greater detail, the flame luminosity and soot distribution have been investigated using high-speed direct-flame photography and two-dimensional soot imaging by a laser-light sheet method, for the case when α is near at 0.25. Experiments were carried out under the

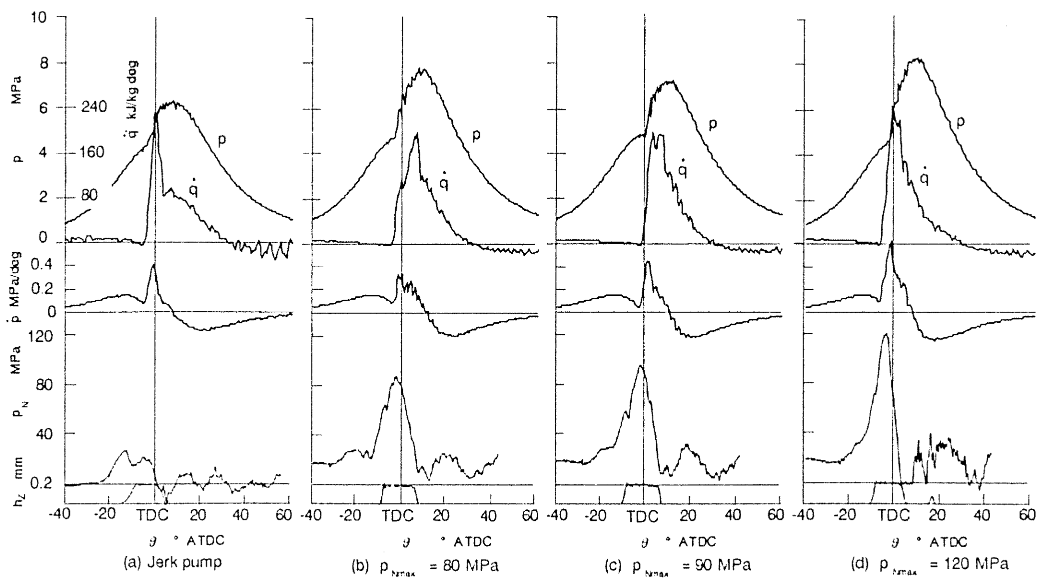

Fig. 5 Courses of cylinder pressure, p, rate of heat release, $\dot{q}$, rate of cylinder pressure rise, $\dot{p}$, injection pressure at inlet of nozzle holder, p_N, and nozzle needle lift, h_Z, for jerk pump and high pressure injection (engine speed 1800 rpm, equivalence ratio 0.54, injection nozzle orifice diamater 0.29 mm for jerk pump and 0.20 mm for high pressure injection)

following conditions; the nozzle orifice diameter was reduced from 0.22 mm to 0.16 mm, and injection pressure was elevated from 30 MPa to 75 MPa. The test engine was a two-stroke, single cylinder, direct-injection diesel engine, which was operated at an engine speed of 900 rpm and injection timing of 5 °BTDC using tridecane as the fuel. Figure 6 shows the cross-section of an optical-access direct-injection diesel engine for laser-light sheet method. Two-dimensional images of soot clouds during combustion were obtained by the Mie-scattering light-sheet technique using a pulsed YAG. Direct flame and soot clouds were photographed simultaneously by optical set-up as shown in Fig.7. For the details, see previous reports.[10][11][12]

Figure 8 shows a series of the direct-flame photographs, abbreviated to DP from here on, and laser-light sheet photographs, LS, at different crank angles for two peak injection pressures. Each set of DP and LS photos are not

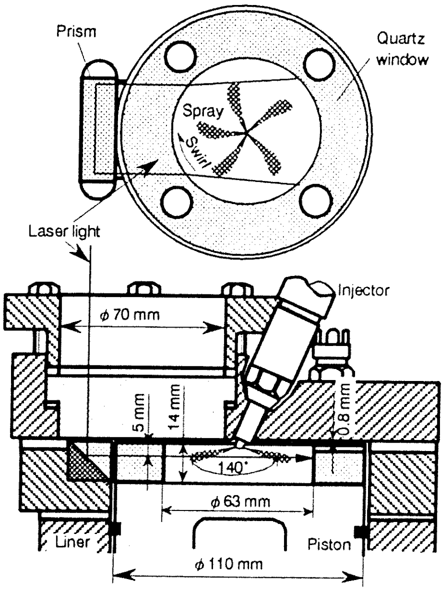

Fig. 6 Cross-section of an optical-access direct-injection diesel engine (Bore × Stroke, 110 mm × 120 mm)

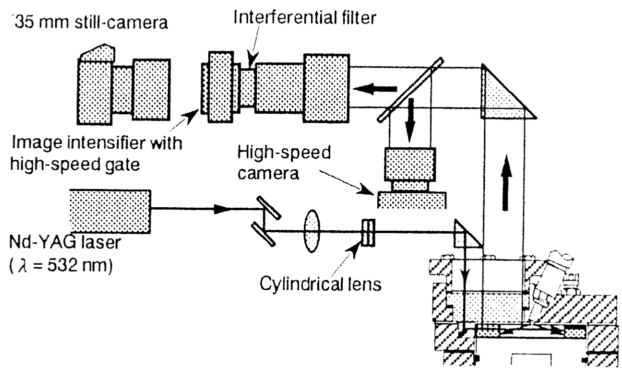

Fig. 7 Optical set-up for high-speed photography and laser-light sheet method

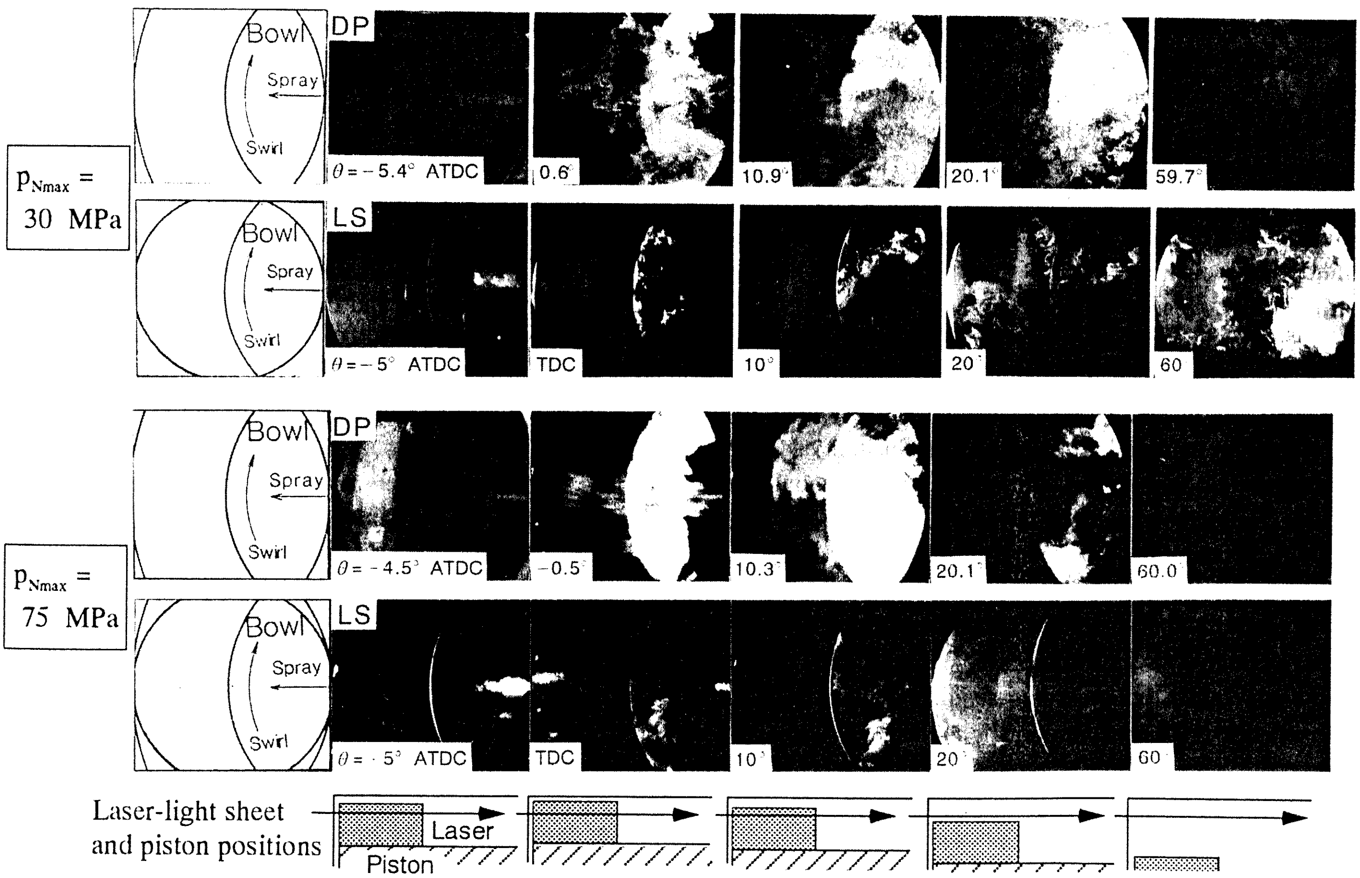

Fig. 8 Direct flame photographs, DP, and laser-light sheet photographs, LS, at different crank-angles for two injection pressures, p_{Nmax} (engine speed 900 rpm, injection timing 5 ° BTDC, injection quantity of 29 mg/st, nozzle orifice diameter 0.22 mm for p_{Nmax} = 30MPa and 0.16 mm for p_{Nmax} = 75MPa)

those obtained at the same cycle but at sequential cycles. It is found that fluctuations in flame luminosity at p_{Nmax} = 75 MPa is much less over the entire combustion space than at p_{Nmax} = 30 MPa. No luminous flame is observed at later crank angles. In addition, it is observed that the flame color is close to white at high pressure injection. Yokota et al.[13] also reported that non-luminous flame and less smoky combustion are attainable at an injection pressure of 250 MPa and at a nozzle orifice diameter of 0.15 mm.

From LS photos in Fig.8, it is found that the soot clouds no longer exist at p_{Nmax} = 75 MPa in the later crank angles. According to a previous study[11] that laser light was introduced into the top clearance, much soot-clouds were observed in the top clearance space at p_{Nmax} = 30 MPa, but there were no soot-clouds in the top clearance space at p_{Nmax} = 75 MPa. This means that the utilization of air in flame is much improved during combustion.

The above-mentioned results suggest that the mixture prepared in spray is more homogenized having a more excess air, and the microscopic-mixing between fuel and air progresses rapidly after ignition. Hence, homogeneous and lean combustion takes place at high pressure injection.

CONCLUSIONS

The present paper discussed factors that may affect spray and combustion at elevated injection pressures for the direct-injection diesel engine. To this end, the sensitivity of injection pressure on spray characteristics was assessed for an assigned relation between the injection pressure difference Δp and the nozzle orifice diameter d_N, i.e., $d_N \Delta p^{\alpha}$ = const. As the results, it was shown that an increase in injection pressure with a reduction of nozzle orifice diameter reduces the fuel-air ratio and enhances the microscopic mixing, thereby preparing a homogenous mixture within the spray.

According to the analysis, spray number and swirl intensity should be selected depending on α. In the case of very large α, the injection period becomes be very long and the air utilization on radial direction of a combustion chamber is decreased. For this reason, the spray number should be increased considerably without swirl. In the case of α at around 0.5, the spray number should be a little increased and the swirl intensity be reduced. In the case of α at around 0.25, the same spray number and swirl intensity with those at jerk pump are permitted. Even in this case, smoke density can be reduced significantly, but pressure rise in the cylinder is faster and the exhaust NOx concentration is higher.

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