Fuel Spray Motion in Side Injection Combustion System for Diesel Engines

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ABSTRACT

The experimental and theoretical analysis of fuel spray motion in the side injection combustion system which is intended to control the fuel spray motion by the air swirl for dynamic fuel-air mixture formation were shown in this paper. The experiments were performed by high speed photographic observation of the fuel spray motion with air swirl in the constant volume test rig. The theoretical analysis of the fuel spray motion to determine with what characteristics it moves and spreads to form the fuel-air mixture within the combustion chamber involves calculations, using CFD multidimensional analysis, of fuel spray velocity, spray trajectory, and excess air ratio in the fuel spray. From all the foregoing studies the important results concerning with the characteristics of the fuel spray motion and dispersion in the side injection combustion system was obtained and usefulness of CFD multidimensional analysis to improve the spray dispersion in the combustion chamber was shown.

INTRODUCTION

Confronted by the soaring price of marine fuel oils and the pressing need for energy conservation marine diesel engine manufacturers have been striving vigorously for devising fuel-efficient engines. Prerequisite to reducing the fuel consumption is curbing the consumption of combustion air in the cylinder, which means that fuel must be burned completely even with low excess air ratio. To achieve the high performance and low specific fuel consumption, the side injection combustion system is employed by large-bore 2 stroke-cycle marine diesel engines.(1)

In this paper, the authors show the studies of fuel-air mixing process in the side injection system, which were performed by experimental observation by photography of the fuel spray motion in the constant volume test rig composed of an actual engine cylinder cover, and thoretical analysis of fuel spray motions using CFD multidimensional analysis.

SIDE INJECTION COMBUSTION SYSTEM

Combustion in the diesel engine, though defined basically as fuel spray combustion, differs from ordinary fuel sprays undergoing combustion, in that air available for use in burning fuel spray is only that present in the enclosed combustion chamber-space and, idealy, the limited quantity of air should be fully utilized for the fuel to burn out completely after being injected.

The fuel spray trajectory and development within the combustion chamber, therefore, are an important consideration,(2) and there are two ways to be taken to deal with this problem in connection with fairly large size engines, one being the center injection combustion system and the other the side injection combustion system which are respectively no named depending on the location of the fuel valve position.

The side injection combustion system which is shown in Fig. 1 is intended, beside dispersing the fuel spray geometrically in the combustion chamber, to control the fuel spray motion by taking advantage of the strong air swirl, for dynamic fuel-air mixture formation. Therefore, with this system, in addition to the question of geometrical dispersion of fuel spray, how best to coordinate the fuel spray motion and air swirl becomes an even more important design consideration.

In this study, taking into account not only the relatively simple question of geometrical fuel spray dispersion but also the problem of how best disperse the fuel spray by taking advantage of dynamic movement of air swirl, the analysis of the fuel spray motion within the combustion chamber

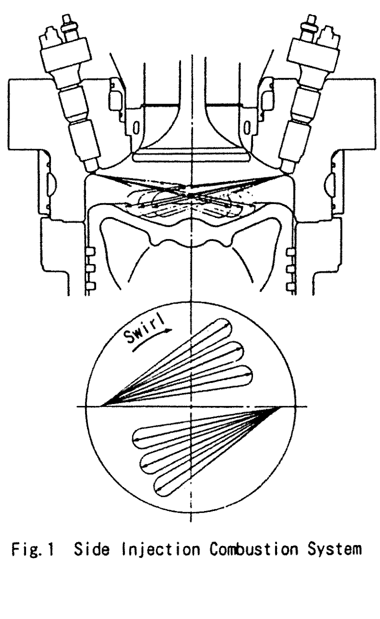


Fig. 1 Side Injection Combustion System

was performed by calculation using CFD and also using data obtained through a series of tests which were conducted for observation of movement of sprayed fuel within the constant volume test rig composed of an actual large-bore engine cylinder cover.

OBSERVATION OF THE FUEL SPRAY TRAJECTORY

Experimental Apparatus

Fig. 2 shows the experimental apparatus with the constant volume test rig using the actual large-bore engine cylinder cover with inner diameter 450mm which is covered on the under side and exhaust valve side with the thick acrylite plate and is filled with high pressure Nitrogen gas, into which the fuel in injected.

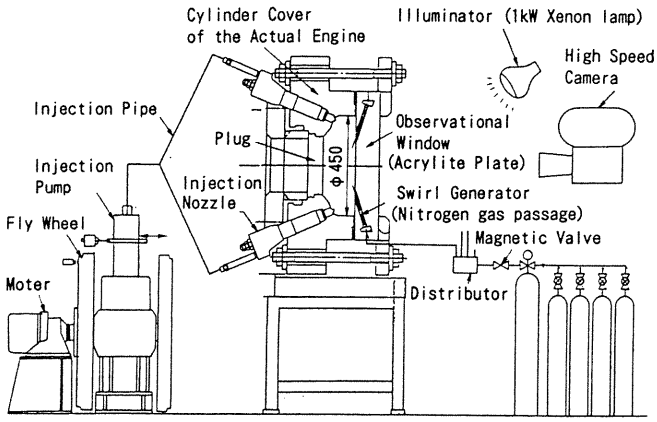


Fig. 2 Experimental Apparatus

The swirling motion in the test rig is formed during Nitrogen gas is flowing into the chamber through the 4 passages those are bored in the acrylite cover plate in the direction to give a tangential component of the gas flow. The velocity of swirling motion becomes lower after the magnetic valve closing as shown in Fig. 3. So, the strength of the swirling gas motion in the test rig chamber at the fuel injection timing is controlled by means of the delay time of the fuel injection timing after the magnetic valve closing.

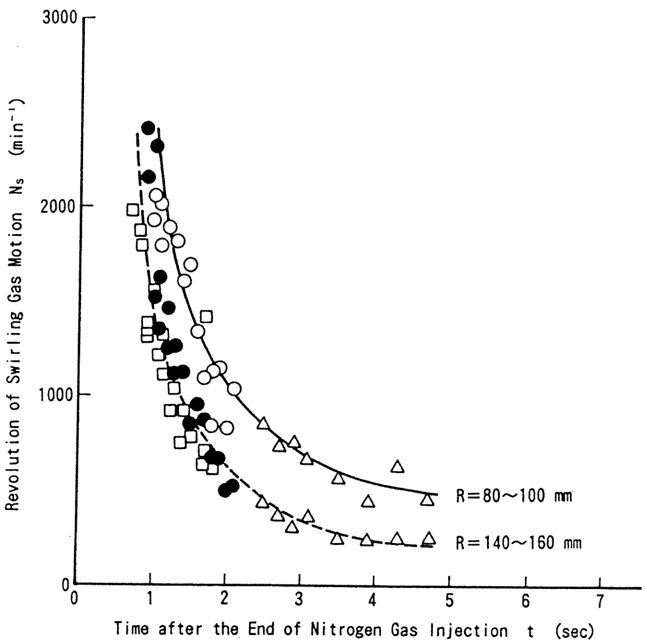


Fig. 3 Time History of The Swirling Motion in The Test Rig

Gas density ρ_g in the combustion chamber of the actual large-bore 2 stroke-cycle diesel engine under the condition of BMEP-1.7MPa (scavenging pressure Ps=0.27MPa, effective compression ratio ε=14) is 42.8kg/m^3. This data is used as a reference of gas density in the test rig chamber in this experiment, which is controlled by changing the pressure of Nitrogen gas at the room temperature (288K).

Fuel injection pump whose scale is same as that of an actual large-bore 2 stroke-cycle diesel engine is Jerk type, of which plunger diameter dp is 53mm, and the plunger speed is set to get the same fuel injection rate as that of the actual engine. The maximum fuel injection pressure Pfmax is 103MPa, with this fuel injection system.

The combustion chamber of an actual engine has a shallow cavity at the top of the piston as shown in Fig. 1. However this test rig has a transparent cover plate corresponding the flat-shaped piston top.

The fuel spray motion is photographed by a high speed camera from underside of the cylinder cover with lighting by the 1kW Xenon lamp. The fuel injection pressure, the needle lift of the injector, the chamber pressure and time from the start of experiment are measured during the high speed photography of the fuel spray motion. Main specifications of experimental apparatus are shown in Table 1 and experimental conditions are shown in Table 2.

Table 1 Specification of Experimental Apparatus

Chamber	Diameter	φ 450
	Hight	149 mm (max)
	Limit Pressure	9 MPa
Fuel Injection System	Cam Velocity	0.97 m/sec/77min^{-1}
	Plunger Diameter	φ 53
	Injection Pipe	φ 6 × 2400 ℓ + φ 4 × 660 ℓ × 2
	Injection Nozzle Area	1.767 mm^2 (φ 0.75 × 4)

Table 2 Experimental Condition

Gas Dencity	42.5 kg/m^3 (Pg=3.5MPa, Tg=288K)
Swirl Velocity	14 , 6 , 0 m/sec (max.)
Fuel Injection Quantity	19.5 cc/st
Fuel Injection Pressure	103 MPa (max.)
Fuel Injection Nozzle	φ 0.75×4N×-1° , 9° , 35° , 45° (C42) φ 0.75×4N×-1° , 9° , 40° , 50° (C49) φ 0.75×4N×-6° , 4° , 35° , 45° (C55) φ 0.75×4N× 4° , 14° , 40° , 50° (C50) φ 0.75×4N×-1° , 9° , 35° , 50° (C57)

Experimental Result

Fig. 4 shows the example of the frame-by-frame photograph of high speed film on the fuel spray in the swirling high density Nitrogen gas within the chamber of the test rig. Just after the start of fuel injection, the fuel spray develops along the geometric direction of injection because the spray penetration is strong near the fuel injection nozzle. However, with the side injection combustion system, especially the inner fuel sprays of which geometric direction is toward the central part of the chamber are bended violently to the next outer fuel sprays by receiv-

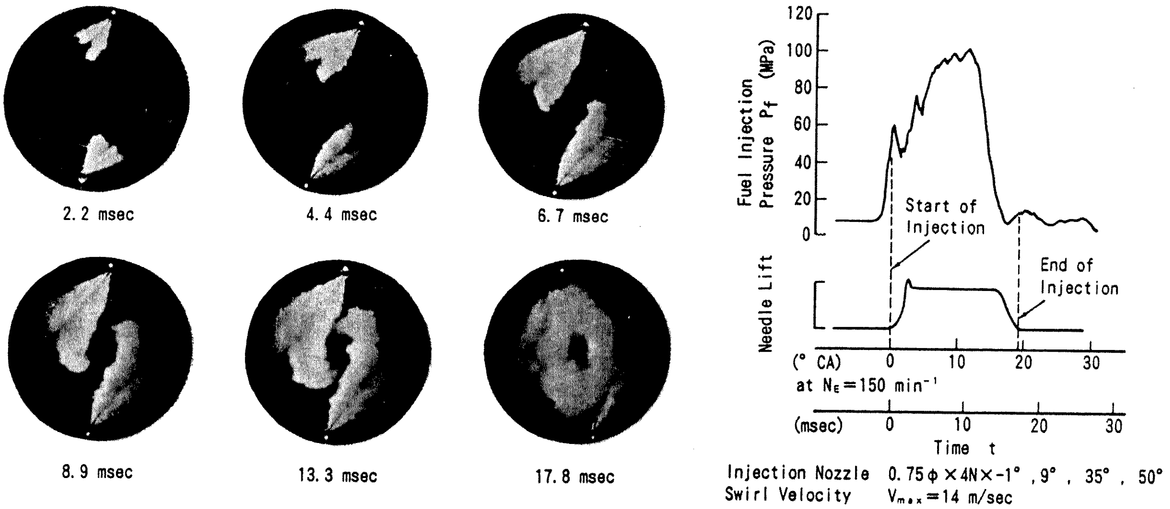


Fig.4 Fuel Spray Motion in The Side Injection Combustion System with The Swirling Gas Motion

ing a strong effect of the swirling gas motion, because the fuel valves are located at the outer side of combustion chamber where the swirling gas motion is stronger. On the other hand the fuel sprays injected toward the outer part near the side wall of the chamber develop almost along the geometrical direction of injection supported by the swirling motion of the gas within the outer part of the chamber. The fully developed fuel sprays that have entrained sufficient air lose their penetration and move with the swirling gas in the chamber. With the side injection combustion system the strong swirling gas motion is necessary for changing the direction of fuel spray motion to reduce the worse effect of the spray impingement on the combustion chamber wall because the force of spray motion during the injection is so strong.

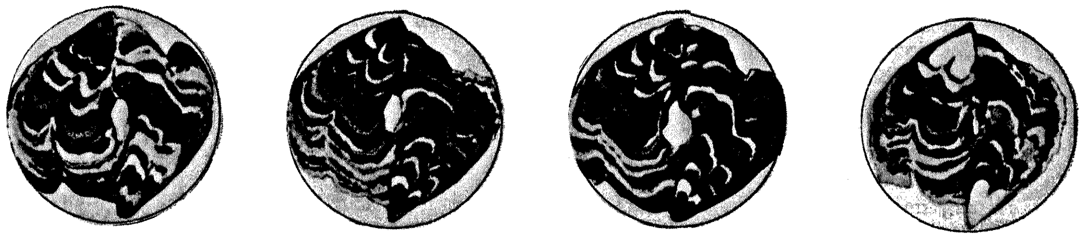
Effect of the injector configuration. The fuel spray trajectories of four kinds of injectors those configurations are shown in the Table 1 are observed. Each injector has four nozzle holes of diameter 0.75mm. Horizontal angles of the No.C42 injector nozzle holes between the injection direction and the axis to the center of the chamber are -1°, 9°, 35° and 45° respectively. Those angles of No.C49 injector nozzle holes are -1°, 9°, 40° and 50° respectively and 2 sprays injected outwards with this injector are brought closer to the side wall of the chamber. Those angles of No.C55 injector nozzle holes are -6°, 4°, 35° and 45° respectively and 2 sprays injected inwards with this injector are brought far to the next outer 2 sprays. Those angles of No.C50 injector nozzle holes are 4°, 14°, 40° and 50° respectively and the 2 sprays injected inwards with this injector are brought close to the next outer 2 sprays.

Fig. 5 shows the effect of injector nozzle configuration on fuel spray development in the chamber. It can be seen that the better spray dispersion is obtained with No.C42 injector nozzle. On the other hand with No.C49 injector nozzle 2 sprays injected outwards reach the chamber side wall sooner and at a high impinging speed commensurate with the force of spray motion. And with No.C50 injector nozzle 2 sprays injected inwards are brought outward and the air within the central part of the chamber is utilized less effectively. Furthermore, with No.C55 injector nozzle the fuel is injected too much inwards and the air within the outer part of the combustion chamber is utilized less effectively, in contrast to the case with No.C50 injector nozzle.

Effect of the air swirl. Fig. 6 shows the effect of air swirl strength on fuel spray development with No.C57 injector of which horizontal angles of nozzle holes between the injection direction and the axis to the center of the chamber are -1°, 9°, 35° and 50° respectively. With weak air swirl the fuel sprays almost along the geometrical direction of injection without changing the spray destination and sprays injected outwards reach the chamber wall extremely soon and at a high impinging speed, and besides injected fuel gathers to the geometrical direction of injection and consequently, the dispersion of fuel spray deteriorates. Whereas with strong swirling gas motion the fuel spray, with its motion suitably controlled, undergoes a high degree of dispersion. With the lower gas density, the fuel spray angle is narrower and the force of spray motion becomes so strong that the higher swirling gas velocity is required to control the fuel spray motion. With the side injection combustion system, if the fuel valve is designed with worse care for fuel spray formation, the development of spray combustion flame will be disturbed by flame quenching by the combustion chamber wall as well as highly heat loss from the combustion chamber wall where fuel spray impinges will be caused. So, obviously from the experimental results of fuel spray motion observation with this system it is important that the air swirl strength is put to the best use and the fuel valve is designed with special care for optimum fuel spray formation to control the fuel spray so as to avoid the undesirable effect of combustion chamber wall while fully utilizing the combustion space (i.e. air).

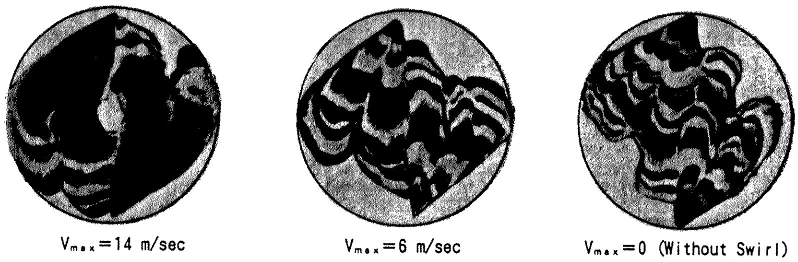
NUMERICAL ANALYSIS OF FUEL SPRAY MOTION

Concerning the characteristics of the fuel spray motion in the side injection combustion system with the air swirl, the numerical analysis of the fuel spray motion in combustion chamber was performed calculating the relative correlation of the motion of fuel droplets with the air swirl and the result was compared with the experimental results.



Φ0.75×4N×-1°, 9°, 35°, 45° (C42) Φ0.75×4N×-1°, 9°, 40°, 50° (C49) Φ0.75×4N× 4°, 14°, 40°, 50° (C50) Φ0.75×4N×-6°, 4°, 35°, 45° (C55)

Fig.5 Effect of The Nozzle Configuration on The Fuel Spray Motion



V_{max}=14 m/sec V_{max}=6 m/sec V_{max}=0 (Without Swirl)

Fig.6 Effect of The Swirling Intensity on The Fuel Spray Motion

Calculation Method

Basic equation. The basic equations governing the gas flow in a cylinder are expressed in cylindrical coordinate by the conservation form of mass, momentumn, energy and concentration of vaporized fuel gas.

$$\frac{\partial \mathbf{F}}{\partial t}+\frac{\partial \mathbf{F} V_r r}{r\partial r}+\frac{\partial \mathbf{F} V_\theta}{r\partial \theta}+\frac{\partial \mathbf{F} V_z}{\partial z}=\mathbf{p}+\mathbf{q}+\mathbf{r} \quad (1)$$

$$\mathbf{F}=\begin{pmatrix}\rho\\ \rho V_r\\ \rho V_\theta\\ \rho V_z\\ \rho E\\ \rho C\end{pmatrix} \quad (2)$$

$$\mathbf{p}=\begin{pmatrix}0\\ -\frac{\partial p}{\partial r}+\rho V_\theta^2\\ -\frac{\partial p}{r\partial \theta}-\frac{\rho V_r V_\theta}{r}\\ -\frac{\partial p}{\partial z}\\ -\frac{\partial p V_r r}{r\partial r}-\frac{\partial p V_\theta}{r\partial \theta}-\frac{\partial p V_z}{\partial z}\\ 0\end{pmatrix} \quad (3)$$

In viscous term, the effect of subgrid scale turbulence is modeled[3] as shown below.

$$\mu_e = \mu+\mu_t \quad (4)$$

$$D_e = D+\mu_t/S_c \quad (5)$$

$$\mu_t = (C.\ \Delta)^2\ \rho \sqrt{2\Phi} \quad (6)$$

Another basic equation of state is expressed as follows assuming perfect gas.

$$p = \rho RT \quad (7)$$

These equations are descretized by using the modified FLIC Method.[6][7]

Spray model. The liquid fuel spray injected in a cylinder is treated as assembling parcels composed with many circular droplets whose conditions diameter, velocity and temperature are equal.[4][5] The atomization of fuel at the exit of a fuel nozzle is not analyzed but the boundary condition of droplets here are assumed as known. The changes of the position, mass and the temperature of a droplet representing a parcel is analyzed in Lagrangian form considering the interaction between the droplet and gas.

The mass and heat transfer to the gas phase caused by evaporation of droplets, reaction force of aerodynamic drag and the heat transfer at the surface of a droplet are contributed to the gas phase analysis as the source term in Eq. (1). But the collision between droplets and breakup are not considered. The governing equation of the trajectory and velocity of a parcel are given by

$$\frac{dx}{dt} = u \quad (8)$$

$$\frac{du}{dt} = \frac{3}{4}C_D\frac{\rho_g}{\rho_l}\frac{1}{d}|v-u|(v-u)+R \quad (9)$$

The heat and mass transfer processes of a parcel are described by the following equations.

$$\frac{d(mC_pT)_l}{dt} = \pi d^2 h(T-T_l)+H\frac{dm_l}{dt} \quad (10)$$

$$h = \lambda Nu/d \quad (11)$$

$$Nu = 2+0.6\cdot Re^{1/2}\cdot Pr^{1/3} \quad (12)$$

$$Re = |u-v|\cdot d/\nu_v \quad (13)$$

$$\frac{dm_\ell}{dt} = \pi d p_\ell D Sh(1-m_v) \tag{14}$$

$$Sh = 2+0.6\cdot R_e^{1/2}\cdot S_c^{1/3} \tag{15}$$

For the initial condition, the distribution is given for the swirl velocity as shown in Fig. 7 and the constant values for other variables as shown in Table 3.

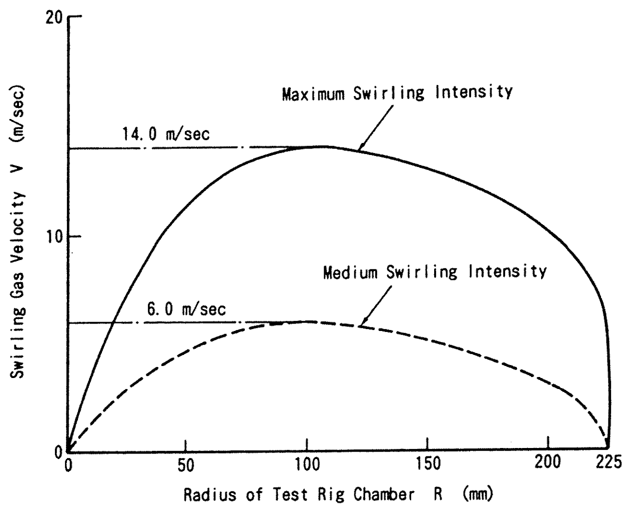


Fig. 7 Swirling Gas Velocity Distribution for The Calculation

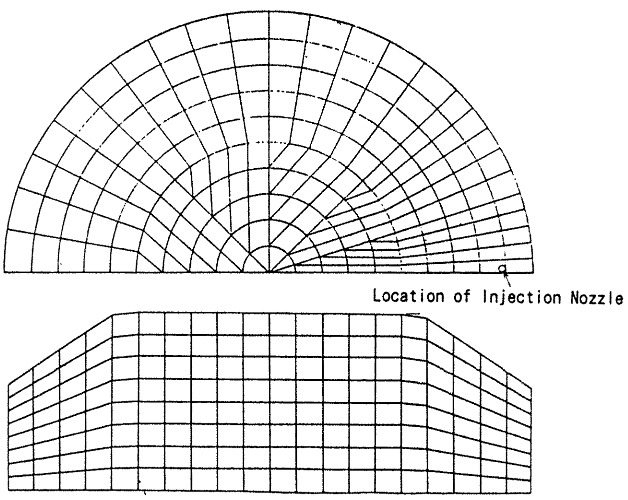


Fig. 8 Mesh Division for Calculation

Table 3 Calculational Condition

Fuel Injection Pressure	Based on Experimental Result
Fuel Spray Angle	14°
Fuel Droplets Diameter	50 μ
Gas Density	42.5 kg/m³ (P_g=3.5MPa, T_g=288K)
Swirling Gas Velocity	14, 6, 0 m/sec (Fig. 7)
Mesh Division	Horizontal 200, Vertical 8 } Total 1600 (Fig. 8)

<u>Comparison of Fuel Spray Development by Calculation and Experiment</u>

The trajectories established by tracing fuel spray motion produced through No.C57 injector nozzle with high swiring gas motion and those determined by the theoretical analysis which was performed using CFD as stated earlier are shown in Fig. 9 for comparison. Although the spread of fuel spray determined by test not necessarily corresponds to the theoretical spread of fuel spray due to the lack of consideration of break up, collision and coalescence between fuel droplets, the spray trajectory and the spray dispersion involving the vaporized fuel determined by calculation and those traced by test are in good agreement, witnessing to the propriety of the spray model used in calculation.

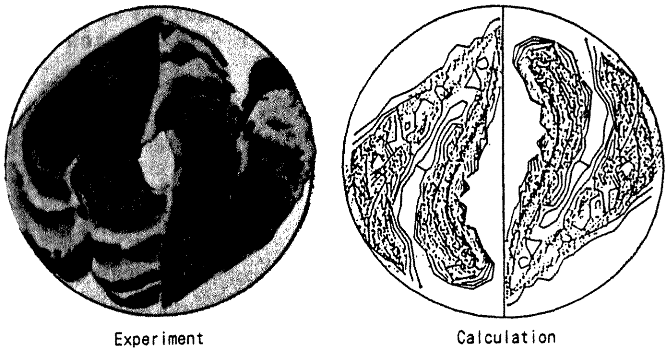


Fig. 9 Comparison of Calculation Results with Experimental Results (No. C57 Injection Nozzle)

Fig. 10 shows the penetration length of fuel sprays through No.C57 injector nozzle with high swirling gas motion. The penetration length of fuel spray by calculation is coincide will that determined by experiment.

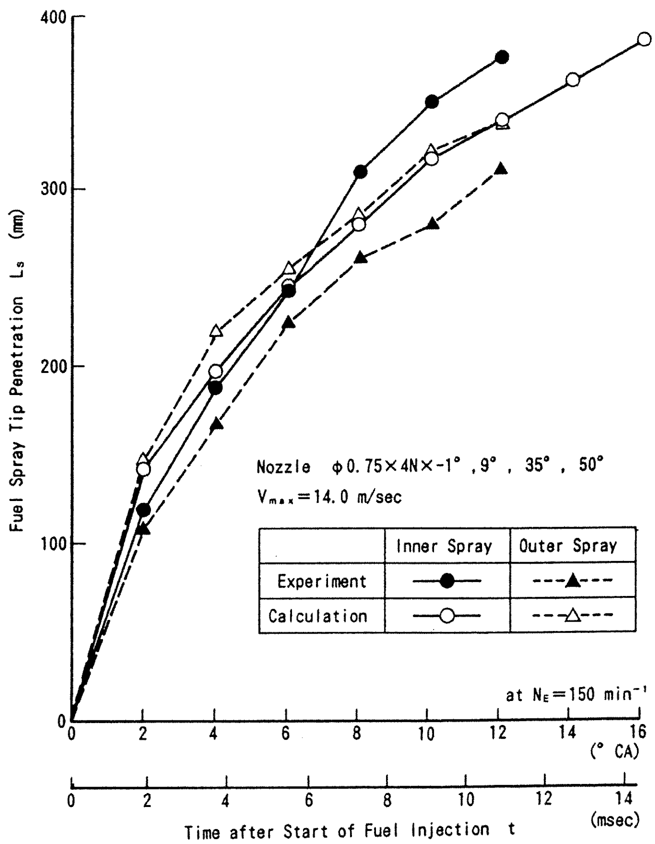
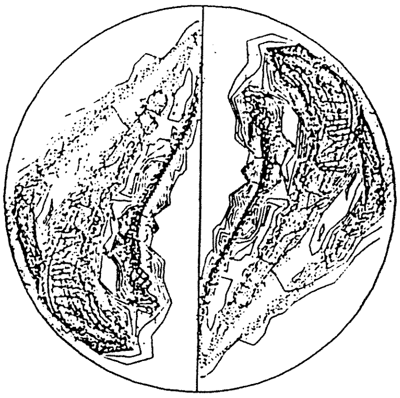


Fig. 10 Comparison of Calculated Spray Penetration Length with that of Experiment

From the above results, it is clear that this theoretical calculation based on CFD can predict the motion of the fuel spray in the swirling gas within the combustion chamber very well, and is powerful tool to improve the fuel dispersion in the combustion chamber with the side injection combustion system.

The calculation result of an example to get a high degree of dispersion of fuel in the combustion chamber is shown in Fig. 11. With this injector the horizontal injection angles of nozzle holes between the injection direction and the axis to the center of the chamber are -5°, 20°, 30° and 50°.

As shown in Fig. 11 the fuel spray, with its motion suitably controlled, a high degree of dispersion corresponding with air distribution in the combustion chamber.



Nozzle $\phi 0.75 \times 4N \times -5°, 20°, 30°, 50°$
Swirl Velocity $V_{max} = 14.0$ m/sec

Fig.11 Calculation Results of An Example to Improve The Fuel Spray Dispersion

CONCLUSION

From all the foregoing experimental and theoretical studies performed this time the following conclusions were obtained.

(1) With the side injection combustion system especially the fuel sprays injected inwards are bended strongly by the swirling gas motion to be changed from straight motion along the geometrical injection direction to relational motion. So it is absolutely necessary to give adequate design considerations not only to the geometrical dispersion of fuel spray but also to the dynamic dispersion of spray utilizing the air swirl.

(2) The theoretical calculation based on CFD multidimensional analysis can predict the fuel spray motion of the side injection combustion system and dynamic fuel dispersion within the combustion chamber. It is the useful tool to determine the dimensions of the combustion system.

NOMENCLATURE

C_D = drag coefficient of droplet
C = specific heat, J/(kg·K)
D = diffusion coefficient of vaporized gas, m^2/s
d = diameter of droplet, m
E = energy, J
H = latent heat of vaporization, J/kg
h = heat transfer coefficient, W/(m·K)
m = mass of droplet, kg
Nu = Nusselt number
Pr = Prandtl number
p = pressure, Pa
R = gas constant, J/(kg·K)
Re = Reynolds number
q = viscous term caused by interactions with fuel droplets
R = source term caused by gravity force
r = source term caused by interactions with fuel droplets
S = surface, m^2
Sc = Schmidt number
Sh = Sherwood number
T = temperature, K
t = time, s
u = velocity of parcel, m/s
V = volume, m^3
v = velocity of gas, m/s
x = position of parcel, m
λ = heat conductivity in gas, W/(m·K)
μ = viscosity of gas, Pa·s
ν = kinematic viscosity, m^2/s
ρ = density, kg/m^3
Φ = discipation function
Δ = subgrid length scale, m

Subscripts
e = effective
g = gas phase
ℓ = liquid phase
j = number
ℓ = direction component
m = direction component
n = direction component
p = constant pressure
t = total
v = kinematic
v = constant volume

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