Interaction between Spray Impingement and Swirl Movement and Its Influence on Combustion in a DI Diesel Engine

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ABSTRACT

Interaction between fuel spray impingement and swirl movement and its influence on combustion characteristics of DI diesel engine were studied on an uni-flow scavenging two stroke engine with a cylinder bore of 190mm and a stroke of 350mm by experiments of both engine performance and combustion photography, and by a simple model about in-chamber gas movement related to the stratification of the charges in cylinder axial direction during combustion. In result, it was found that,spray impingement on walls of combustion chamber could lead up to locally richer fuel-air mixture and flame near the walls, and when air swirl existed, under the action of pressure gradient produced by swirling air,the mixture and flame might get movements in the radial direction due to vertical inhomogeneity of not only charge density but also circumferential velocity. With this,the different effects of air swirl on combustion in different combustion chambers, and some other characteristics, were well explained.

INTRODUCTION

Interaction between fuel spray impingement on combustion chamber wall and swirl movement is practically one of the most important phenomena in modern medium and high speed direct injection (DI) diesel engines[1]. But studies about this are quite limited because of the complexity when both spray impingement and swirl movement exist in the greatly confined combustion chamber. Many researchers concerned only the swirl movement[2]or only spray impingement[3]. The most important contribution may be the well known "thermal mixing, thermal pinch" theory applied for the "M" type combustion system[4]. But the theory concerned only the density inhomogeneity of the charge, and had the hypothesis that circumferential velocity of an element should always equal to that of the local swirl velocity wherever it went in radial direction, which was clearly disagree with the principle of angular momentum conservation. In practice, mixture in DI diesel engine could be very inhomogeneous not only in density and fuel concentration but also in velocity distributions.

The authors had presented a paper[5] at the 18th CIMAC Congress (1989, Tianjin), in which experimental studies about swirl movement and engine combustion characteristics were carried out. The performance and combustion photography results showed that in shallow bowl type chamber where spray impingement were not so important due to the large injection hole angle, effect of swirl was to disperse sprays in circumferential direction, and over-swirl meant overlap of adjacent sprays. On the other hand, in deep bowl type chamber,where spray impingement was an important factor,overlap of adjacent sprays were unavoidable and much more soot was formed during fuel injection but could be re-combusted in later time under appropriate swirl intensity.

In the present studies further experiments were carried out on the same test engine,and a simple model about in-chamber gas movement related to the stratification of charges by spray impingement during combustion was proposed. With the model, the difference in swirl effects between deep and shallow bowl type combustion chambers could be explained.

EXPERIMENTAL APPARATUS[5]

The test engine was a cross-head type, uni-flow scavenging two stroke DI diesel engine with the cylinder bore of 190mm and the stroke of 350mm. Its maximum rating is 81kW power at 510 rpm , and the maximum mean effective pressure is 0.98MPa. On this engine, visualization of in-cylinder combustion is possible too, through the bottom of combustion chamber if the piston and piston rod are replaced by a special prolonged piston,on top of which a transparent glass window is set up, as shown in Fig.1. Test conditions are shown in Table 1. Both operation test (engine performance) and visualization test had same amount of fuel injection per cycle, although in the latter engine speed (rpm) was a little lower for safety reasons and scavenging air temperature and pressure were set higher for normal ignition delay in spite of the lower engine temperatures such as chamber wall, cylinder and so on. Fuel injection pressure was kept unchanged by using different fuel pump plungers. The fuel was marine diesel oil, whose properties are shown in Table 2. Combustion chambers used in both operation and visualization tests are shown in Fig.2.

Scavenging swirl was adjusted by a so called variable swirler which was installed around the scavenging ports in the liner, as shown on the

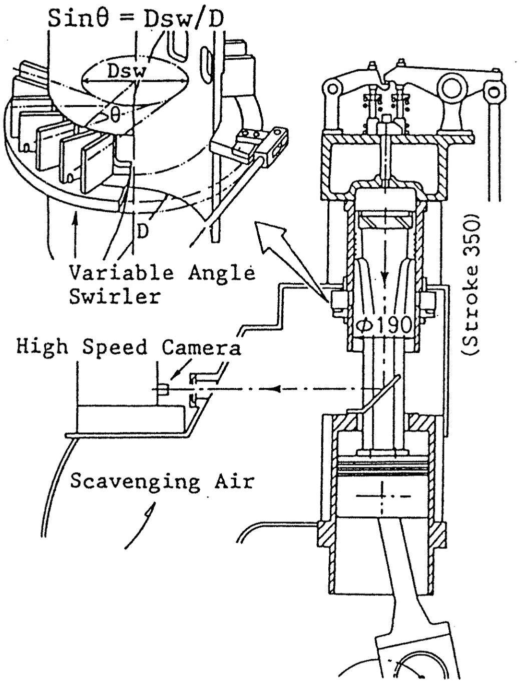

Fig.1 Cross section of test engine

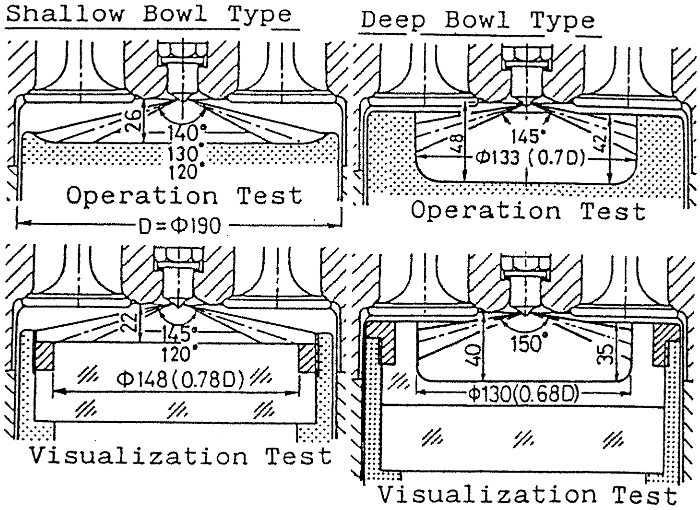

Fig.2 Cross sections of combustion chambers

Table 1 Test conditions

	Operation Test	Visualization Test
Output/Pme	70.6kW/0.875MPa	Amount of fuel injected per cycle is equivalent to the operation test
Engine Speed Compression Ratio Scav. Press./Temp. Excess Air Ratio	488rpm 12 0.178MPa/303K 1.6~1.7	430~440rpm 13 0.181MPa/333K 1.4

Table 2 Fuel properties

	Marine Diesel Oil
Density (15°C)kg/m³	843
Kinematic Viscosity (50°C)mm²/s	2.5
Flash Point °C	72
Wt% C	86.1
Wt% H	13.0
Wt% S	0.84
Residual Carbon Wt%	0.07
Lower Calorific Value KJ/kg	42500

upper left of Fig.1. Parameter Sinθ (θ is the scavenging inflow angle) was used to present swirl intensity. Fig.3 shows the mean swirl velocity near top dead center (TDC) measured from the movement of flame and soot taken on the high speed color film. As can be seen, the swirl velocity distribution was quite flat, and the bigger Sinθ is, the higher the velocity. Also, swirl in deep bowl chamber was much higher than that in shallow bowl one.

EXPERIMENTAL RESULTS

Fig.4 shows some of the high speed combustion photographs in deep bowl type combustion chamber. Clearly because of spray impingement onto the side wall of the bowl, spray dispersion in circumferential direction was very fast, and adjacent sprays met each other as early as 5 degree ATDC. Fuel injection was from 5 degree BTDC to 13 degree ATDC, so large amount of rich mixture and flame were located near the wall, forming a

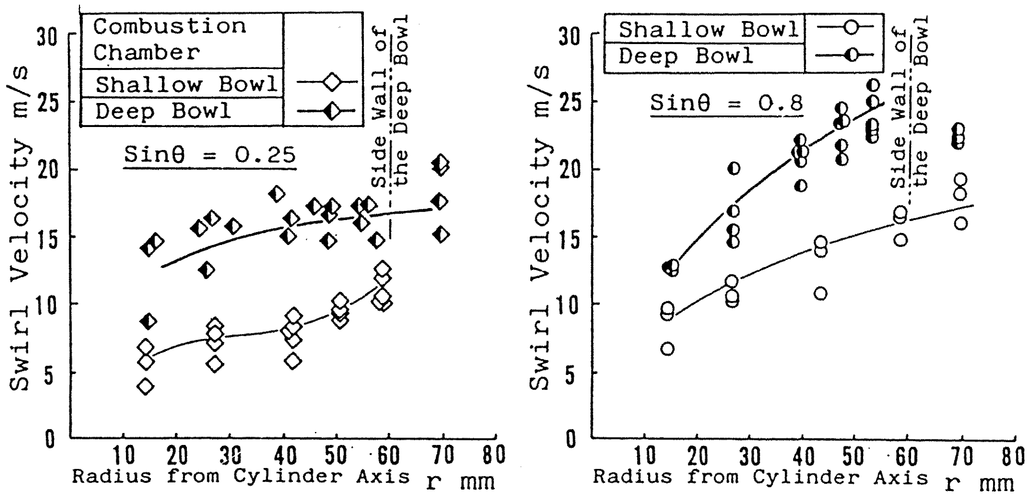

Fig.3 Swirl velocities at ATDC3°~20°

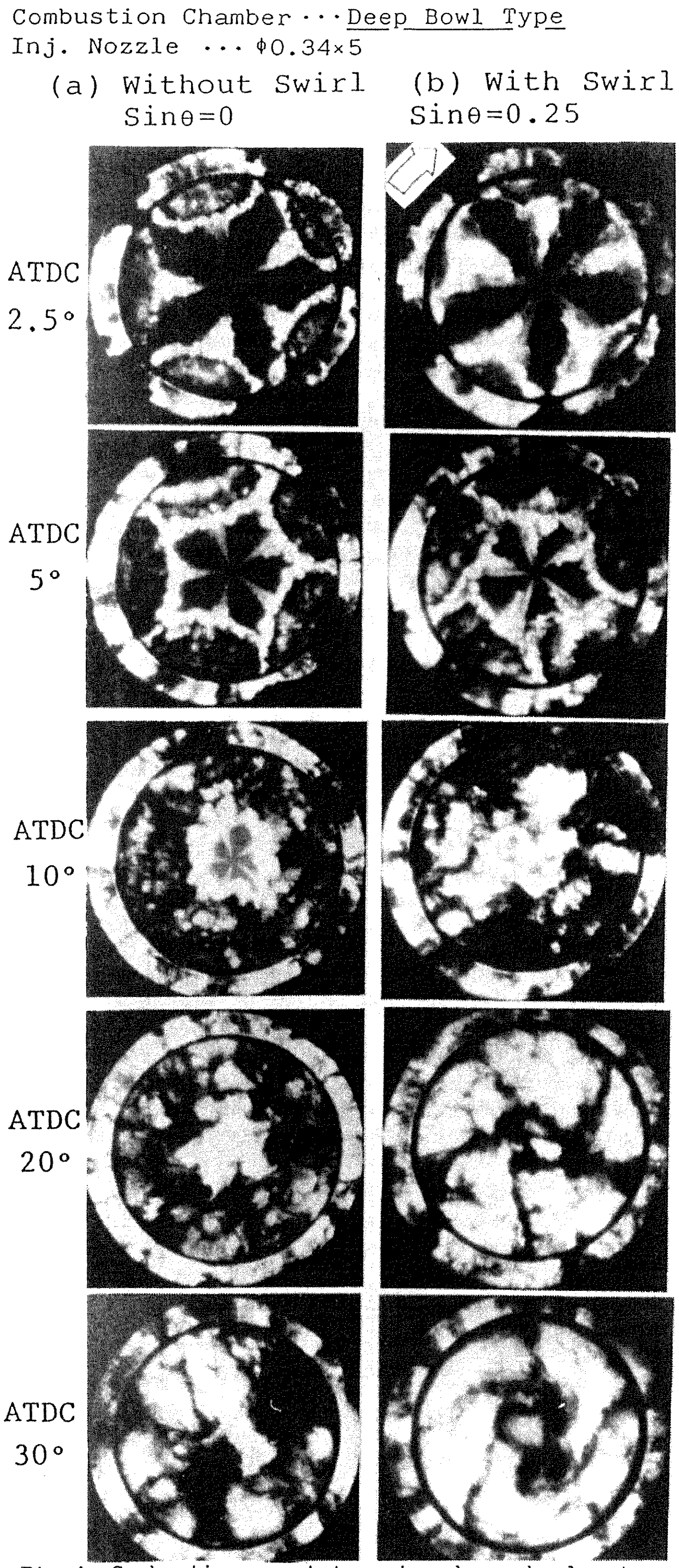

Fig.4 Combustion states in deep bowl type combustion chamber

rich mixture-flame doughnut on the bottom surface of the chamber. As can be easily imagined, stratification in cylinder axial direction was unavoidable in such case. Comparing the photographs with and without swirl, it could be found that the doughnut development and characteristics were quite different, especially in later crank angles. This will be analyzed in detail in the next sections.

Fig.5 shows some of the combustion photographs in the shallow bowl type chamber. In

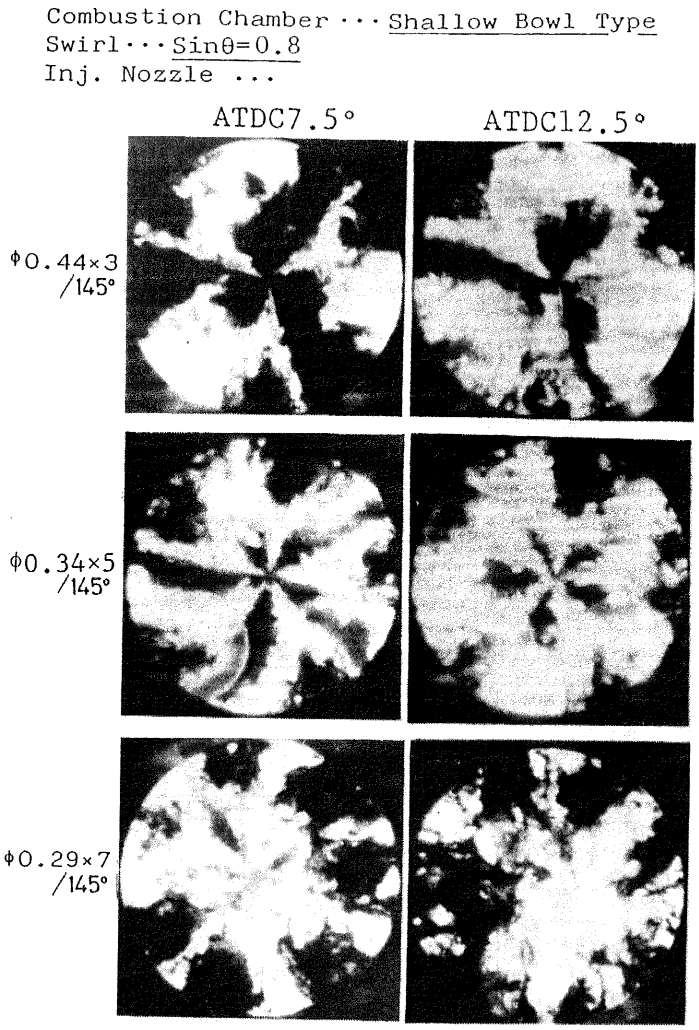

Fig.5 Combustion states in shallow bowl type combustion chamber

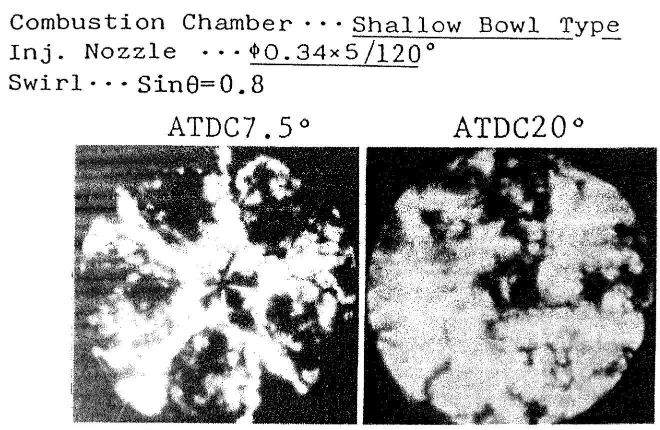

Fig.6 Combustion states with small injection nozzle hole angle

those cases, impingement of sprays was at large distance from the chamber center, dispersion in circumferential direction was mainly by swirl blowing. Adjacent sprays could not meet each other even until after fuel injection ended in the case of only 3 nozzle holes. However, in the case of 7 nozzle holes (but total hole area unchanged), adjacent sprays overlapped quite early causing much visible soot cloud. This was a typical example of over-swirl state[5].

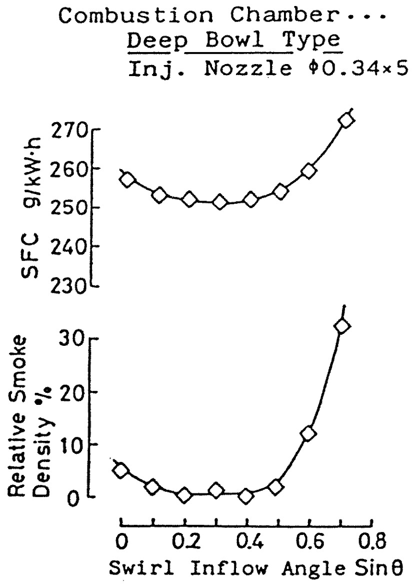

Fig.7 Operation test results(1)

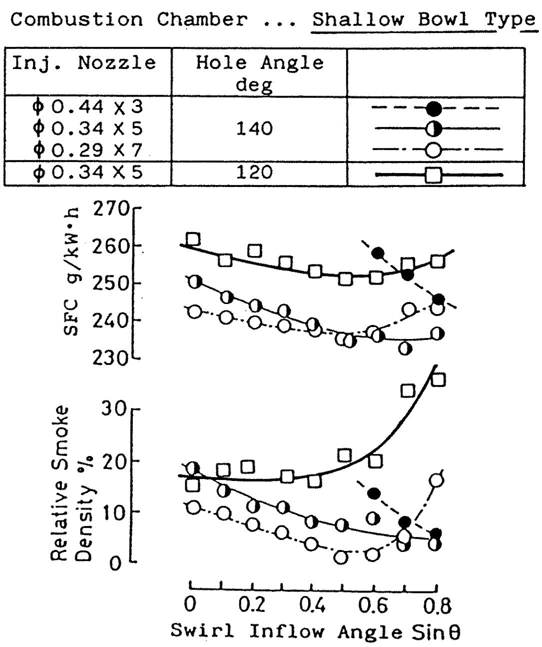

Fig.8 Operation test results(2)

Fig.6 shows combustion photographs with smaller injection hole angle in the same shallow bowl chamber. Spray impingement occurred near chamber center led to early overlaps, and the more outstanding was that fuel penetration to the large outside space was much slowed down, and much soot cloud remained unburnt.

Fig.7 and 8 were the engine performance test results. In deep bowl type chamber, $\sin\theta = 0.25$ gave the best fuel economy and exhaust smoke density. In shallow bowl chamber, the case injection hole angle of 140 degrees and hole number of 3 needed the strongest swirl, and that with 7 holes moderate swirl. However, with small injection hole angle (120 deg.), even moderate swirl led to over-swirl. The smoke density could not be improved by increasing swirl intensity. Those results agreed well with the above visualization results.

MODEL ABOUT NEAR WALL CHARGE MOVEMENT

As has been seen in the combustion photographs, when strong spray impingement takes place, stratification of the charges in cylinder axial direction will be unavoidable. Fig.9 shows a simple model describing the movement of the formed rich mixture-flame doughnut layer near the bottom wall of the bowl, which has a thickness of several millimeters[6], greatly larger than the boundary layer but much less than the chamber vertical dimension. Because of difference in velocities between sprays and air, and some other reasons such as combustion, mass density ρ and velocity V_θ (tangential) of the charge within the doughnut layer, are not necessarily equal to the mass density ρ_a and swirl velocity Vs of the other charges (air) respectively . Due to the air swirl, a static pressure gradient dp/dr will act on the layer. If we make ideal assumption, the pressure gradient will be balanced by inertial force of the layer, so equations of motion could be written as follows in cylindrical coordinates:

$$\frac{dV_\theta}{dt} + \frac{V_r V_\theta}{r} = 0 \qquad (1)$$

$$\frac{dV_r}{dt} - \frac{V_\theta^2}{r} = -\frac{1}{\rho}\frac{dP}{dr} \qquad (2)$$

$$\frac{dP}{dr} = \rho_a \frac{V_s^2}{r} \qquad (3)$$

Where t is time, r is radius and Vr is radial velocity component. Solutions of them can be easily got for unsteady movement of a mass element inside the layer:

$$V_\theta = \frac{C_1}{r} \qquad (4)$$

$$V_r^2 = C_2 - \int \frac{2}{r}\left(\frac{\rho_a}{\rho} V_s^2 - V_\theta^2\right) dr \qquad (5)$$

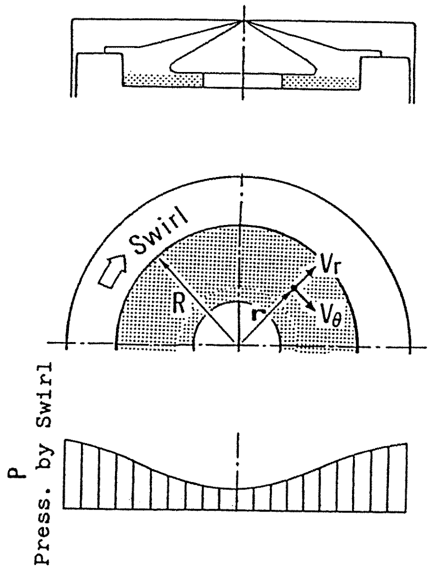

Fig.9 Model for charges in combustion chamber

where C1 and C2 are constants for one definite element dependent on its original position and velocities. Equ.(1) and (4) simply mean that the angular momentum of a mass element will be conserved, and Equ.(2) and (5) mean that it will be accelerated or decelerated in radial direction depending on the centrifugal force difference between the element and the swirling air. Fig.10 shows diagrammatically an example of the solutions (the continuous lines). The swirl velocity was approximated by a solid part plus a constant speed part. It is assumed that the original state of the element is in point ⓐ, and so in later time the state will change along the lines to ⓑ and ⓒ, and then turns to move outward, as indicated by the arrow. In a real engine, of course, the angular momentum of the element is hard to be conserved, so it would more likely follow the sketchily dotted line in the figure.

Fig.11 shows the influence of gas density ratio ρ/ρ_a. As has been known for long, the smaller the ratio is, the stronger the radial movement of the doughnut.

Fig.12 shows the influence of initial velocities. Smaller initial V_θ will give larger movement toward swirl center. Initial Vr has relatively weaker affection on the most inside position that the element can approach.

DISCUSSIONS

The movement in radial direction had been pointed out long ago from the view point of secondary flow[7] or mass density inhomogeneity[4]. However, because the viscous boundary layer can not be thick enough in a steady confined flow, to influence the combustion, it has been considered negligible. Only the density inhomogeneity has been recognized widely as an important factor. From the above photographs and model analysis, it can be said that when spray impingement and air swirl exist at the same time, the near bottom wall process is more meaningful than a general boundary layer in modern high and medium speed DI engine.

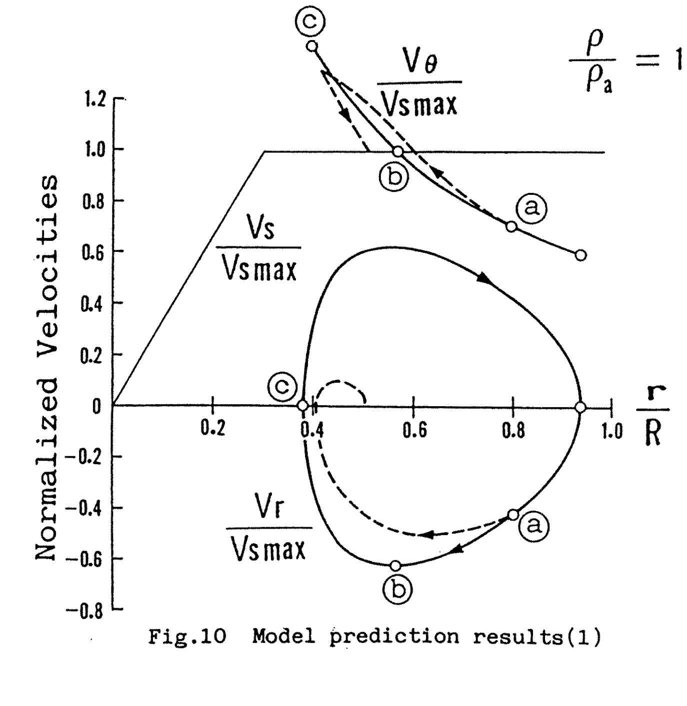

Fig.10 Model prediction results(1)

In Fig.4, gas(flame) movement in case (b) with appropriate swirl was stronger than in case (a) without swirl, particularly after fuel injection ended, and clearly re-combustion of the formed soot visible in the photographs was much better done (comparing those at 20 and 30 deg. ATDC). A motion direction change of the flame-soot cloud from inward to outward was also observed in case (b) at later crank angles, as has been predicted in Fig.10. As a result, the exhaust smoke density was lowest although the fuel economy was not as good as in shallow bowl type chamber,

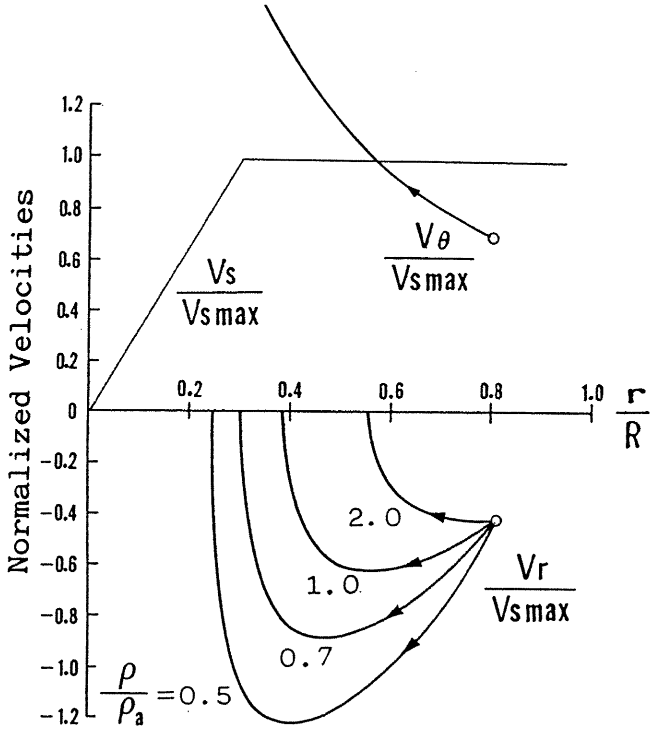

Fig.11 Model prediction results(2)

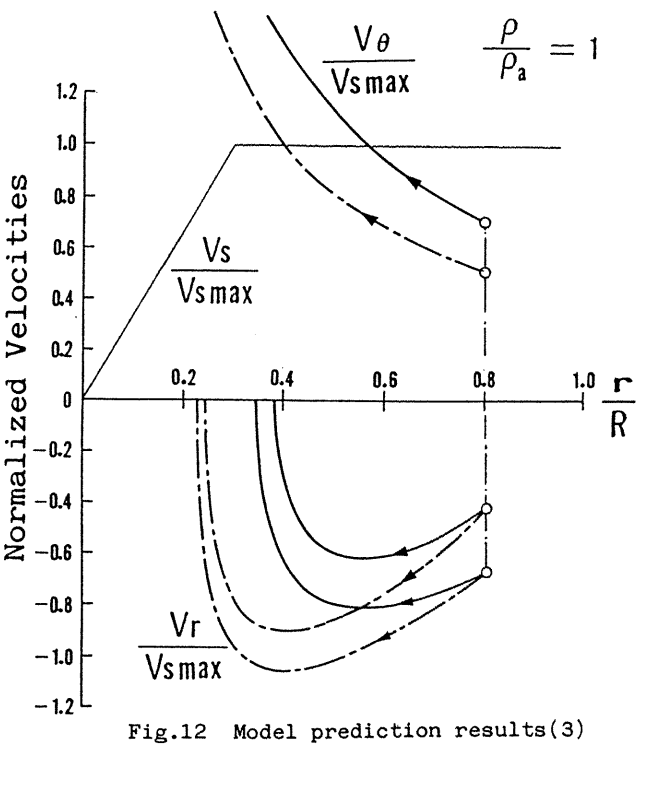

Fig.12 Model prediction results(3)

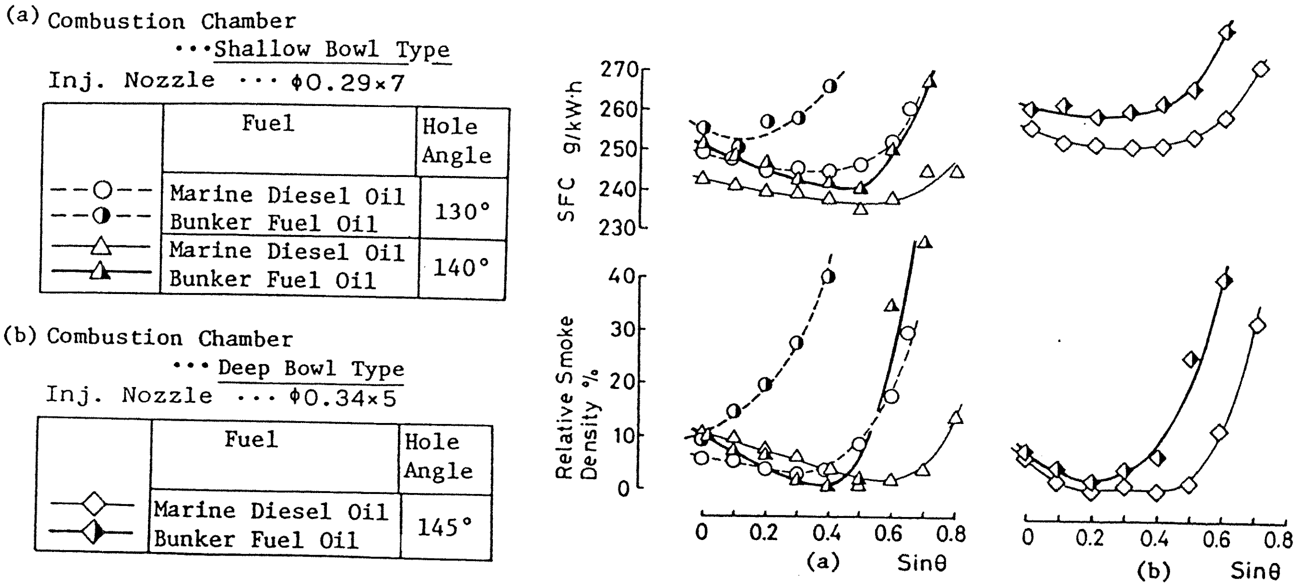

Fig.13 Change in combustion characteristics with fuel properties

due to the larger surface area of the deep bowl chamber and slow combustion of the near wall rich mixture. That is, interaction between spray impingement and air swirl in deep bowl chamber may accelerate strong radial movement and furthermore the associated vertical movement of the rich mixture and soot for further mixing and combustion, especially in later period at which swirl, squish and other perturbation likely have decreased a lot.

On the other hand, in shallow bowl chamber, sprays must be dispersed to outside space by their own momentum, and to circumferential space by swirl blowing. When spray impingement occurs at large enough distance from swirl center, the main problem will be to disperse sprays by using swirl, and over-swirl means overlap between adjacent sprays[5]. But if impingement occurs near swirl center, as had been seen in the combustion photographs in Fig.6, expansion by impingement and larger angular velocity of swirl at smaller radius, will make overlap easier, besides, a rich mixture-flame layer, though not evident as in deep bowl chamber, can also be formed near the bottom wall. From the model, air swirl will block the outward advance of the layer in radial direction, causing oxygen insufficient inside and useless outside, thus soot formed by overlap or so, can not be fully re-combusted but exhausted (see Fig.8). That is, air swirl has plus effect of blowing sprays and minus effect of blocking radial dispersion at the same time, and so the optimum swirl is a compromise of the two in such case. Considering also that better cycle efficiency can be got by fast combustion and less soot formation in early period and less after burning (soot re-combustion), larger injection nozzle angle cooperated with higher optimum swirl intensity should be used in shallow bowl type combustion chamber.

Lastly, the model can also be used to analyze the combustion characteristics of heavy low quality fuel such as bunker fuel oil[8]. Fig.13 was an example. With bunker fuel oil, optimum swirl intensity became lower, which might be partly resulted from the stronger spray impingement and so stronger mixture stratification caused by worse atomization and vaporization.

CONCLUSION

A model describing the interaction of spray impingement and swirl movement was proposed. With it, the different effects of air swirl in different combustion chambers upon combustion characteristics in DI diesel engines, and some other characteristics could be explained quite well.

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