

# Role of Flows and Turbulent Mixing in Combustion and Pollutant Formation in Diesel Engines

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## ABSTRACT

This paper describes the diesel combustion and its pollutant formation paying special emphasis on fuel spray, gas motions, and turbulent mixing in direct-injection diesel engines. First, enhancement of the degree of air utilization by macroscopic distribution of fuel is discussed from various aspects; they include effects of spray-wall interaction, swirl, squish, and combustion-chamber geometry. A further discussion is made on the spatial heterogeneity and its decrease over time with turbulent mixing. It is pointed out that an enhanced turbulence intensity likely promotes mixing between fuel-rich and fuel-lean fluid lumps and also assists burning of soot particles once formed. Finally, some factors affecting turbulent mixing and their effects on the particulate formation are discussed in detail.

## INTRODUCTION

Diesel engines, having an inherently high thermal efficiency, are widely used as a prime mover in many fields including transportation, industry, and agriculture. However, exhaust emissions of oxides of nitrogen, particulate matter, and unburned hydrocarbons are far greater than for other prime movers. In order that diesel engines may be more widely accepted in the future, greater effort for reducing such pollutants without a decrease in the thermal efficiency will be required than has been made up until now.

The major problems are concerned with the combustion process. However, diesel combustion is very complicated since many processes are linked to each other; they are physical processes such as fuel atomization, gasification, gas motion and mixing, and chemical processes such as thermal cracking, preflame reactions, oxidation and heat liberation. Nevertheless, much effort has been made to reveal what takes place in the combustion chamber, using e. g. high-speed photography, high-speed gas-sampling techniques, and various laser diagnosing methods as well as modeling methods for obtaining a perspective of the phenomenon, including numerical fluid dynamic simulations.

The present paper reviews the combustion research along the same lines, paying special attention to the role of gas motions and their effects on combustion and pollutant formation, based mainly on the present author's work.

The key for attaining a more complete combustion and a higher thermal efficiency in diesel engines is clearly enhancement of air utilization and speeding-up of combustion. Basically, these are governed by two significant factors; one is macroscopic distribution of fuel over the entire space of combustion, and the other is the turbulent mixing of fuel and air at the microscopic level.

To achieve a better fuel distribution at the macroscopic level, or in other words, to reduce maldistribution of fuel, the fuel must be penetrated into every part of the combustion chamber simultaneously with full use of gas motions of every kind. They are much dependent on spray characteristics, such as atomization, penetration, spray impingement on wall, and gas motions such as swirl, squish, jetting flow and so forth.

Mixing of fuel with air at the microscopic level is also considered to be crucial not only for speeding up combustion but also for pollutant formation. During the course of combustion, many fluid lumps, having different fuel concentrations of burned gas, would coexist, making contact with each other owing to a higher level of turbulence intensity. Such a heterogeneity might characterize the diesel combustion, and as time goes on the degree of heterogeneity decreases and a heat would be evolved. The exhaust smoke may be accounted for as a portion that has not undergone such a turbulent mixing during the expansion stroke.

## AIR UTILIZATION AND INFLUENCING FACTORS

We shall first discuss the matters related to the macroscopic process in direct-injection diesel engines that use a central injection with a multi-hole nozzle. Among a variety of combustion chamber geometries, a shallow-dish type deserves attention because it can be discussed without having to go into great details about flow dynamics.

Figure 1 is an example of high-speed photographs(1) showing spray-wall interactions and subsequent flame development. They were taken on an engine with a specially designed cylinder-head that simulates only one sector of the injector and the cavity. These photographs show that the spray jet hits the rim of the cavity and is deflected there. The flame formed first is air-borne momentarily covering the whole cavity space, but the flame overhanging from the cylinder head soon supervenes and gains in strength. This is due to the fuel that is conveyed to the wall of the cylinder head after

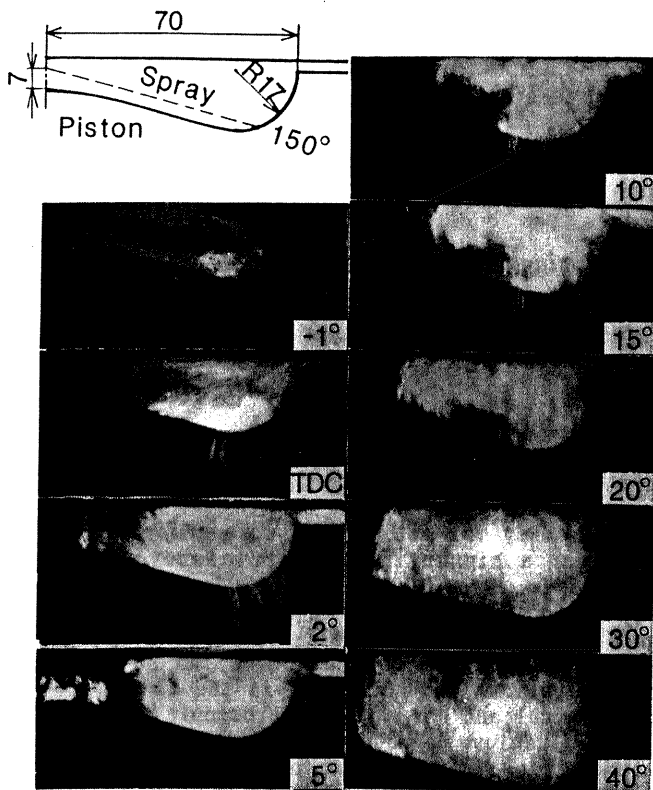


Fig. 1 High-speed photographs of spray and flame in a shallow-dish chamber (side view) [equivalent cylinder bore 180mm, stroke 100 mm, engine speed 740rpm, nozzle diameter 0.30mm]

the impingement. Such a fuel impingement of the fuel spray on the wall is usual in high-speed diesel engines, and it favors not only fuel distribution but also air entrainment into the reflected jet(2).

The spray characteristics are very important because it determines the subsequent progress of burning. A number of studies have looked at atomization, break-up length, penetration, spray angle, air entrainment, and spray-wall interactions, some of which have used non-obtruding diagnosing methods. With regard to the jet impinging on the wall, it has been shown that an impinging jet spray entrains more fresh air into the spray area than a free spray jet(3).

In the high-speed photos above, we should pay particular attention to the fact that a part of fuel is conveyed into the clearance space between the cylinder head and the piston top. Until later crank angles the flame lasts there, which finally becomes a soot cloud. Such a situation clearly corresponds to what we may call over-penetration. To remove this situation, the spray penetration should be suppressed to a certain degree, and the most usual way to achieve that is to reduce the diameter of the nozzle orifice. This requires an increase in the spray number for a fixed total nozzle flow area. Another method is to use an induction swirl which deflects the spray tip, enabling to reduce the radial penetration of the spray.

Figure 2 gives the result of a performance test on a shallow-dish chamber(4) showing the relative brake mean effective pressure and the exhaust discoloration against spray number at a constant fuelling rate and at the same total flow area of

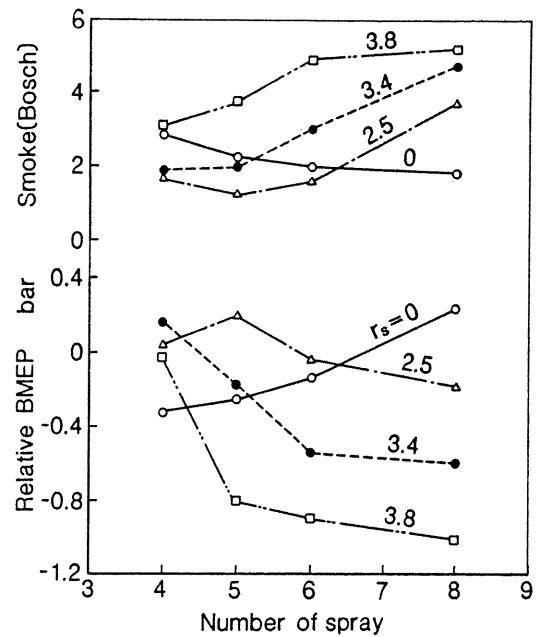


Fig. 2 Effect of spray number and swirl ratio on power output and exhaust smoke [cylinder bore 110 mm, stroke 120mm, amount of injected fuel 37.2mg/stroke, total area of nozzle holes 0.19 mm<sup>2</sup>, shallow-dish chamber]

the injector nozzle, for different swirl ratios  $r_s$ . In a no-swirl case at the swirl ratio  $r_s = 0$ , the power output increases and smoke level decreases as the spray number is increased, whereas in the swirled cases the best performance is obtained at a smaller spray number. The best performance attainable does not differ much between swirled and quiescent cases, although the swirled case is preferred in the case when the peak cylinder pressure and the noise level should be kept at lower levels. A significantly poor performance is experienced at a larger spray number with a higher swirl ratio. This situation of over-swirling stems from a poorer distribution of fuel in the outer zone. Such a situation may lead to a state of thermal pinch, as shown in Fig. 3(a); a hotter burned gas inside tends to separate with a colder air having larger density in the outer layer. A thermal pinch has the opposite effect of the thermal mixing advocated by Pischinger(5). In the case where a burning zone is located in the outer periphery in a highly swirling field, as shown in Fig. 3(b), the hot burned gas at a lower density is rolled up in the center of the swirl, enabling a better burning. This is especially so in the M-combustion system.

Deep-bowl chambers use much higher levels of gas motions than shallow-dish chambers. The reduced diameter of the piston bowl will bring about a squish motion at the end of compression. In the swirled case, the reduction in the radius of rotation will lead to an increase in the swirl angular velocity since the most air becomes confined in the piston bowl at the end of compression. Recently, it was found from numerical fluid dynamic simulation that in the swirled case, the air entering into the bowl is strongly bent at the corner of the bowl, thereby making the fuel-air mixture move downward in the bowl, unlike the toroidal motion in the case without swirl.

Such gas motions significantly affect mixture formation and the subsequent flame spread. In Fig.

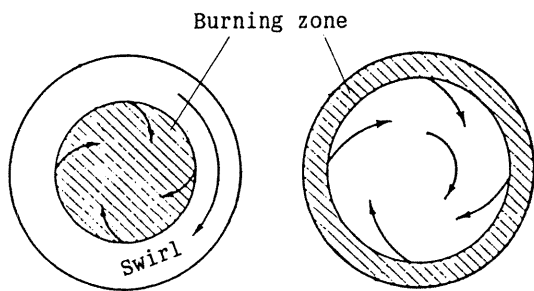


Fig. 3 Thermal pinch (left) and thermal mixing (right)

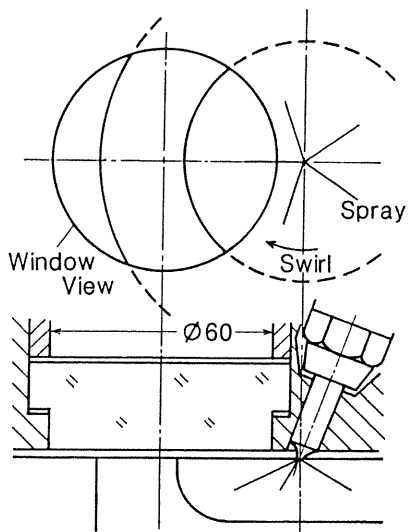


Fig. 5 Cylinder head for photographing combustion

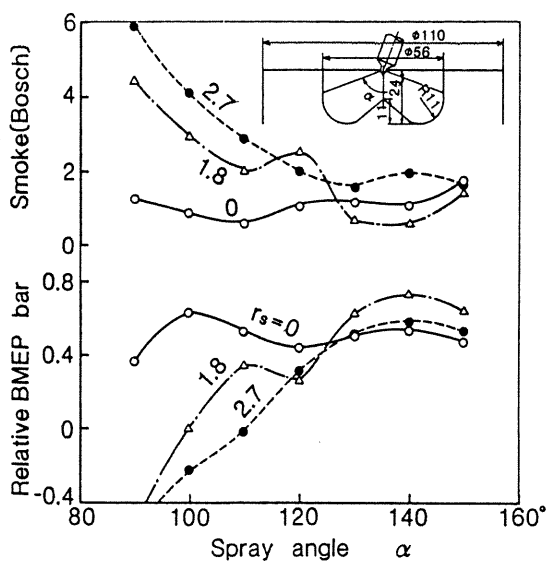


Fig. 6 Effect of spray angle and swirl ratio on power output and exhaust smoke [five 0.22mm-hole nozzle, deep-bowl chamber]

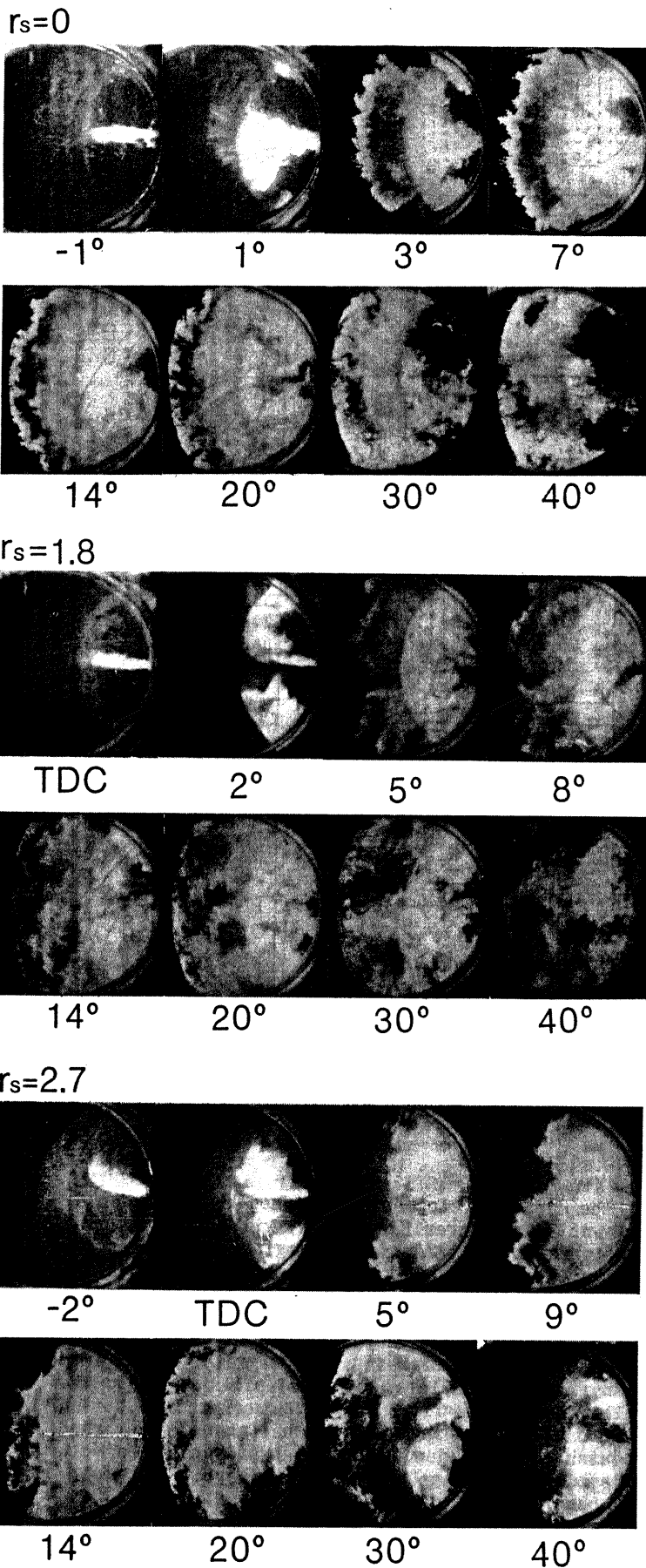


Fig. 4 High-speed photographs of spray and flame in a deep-bowl chamber (top view) [five 0.22mm-hole nozzle]

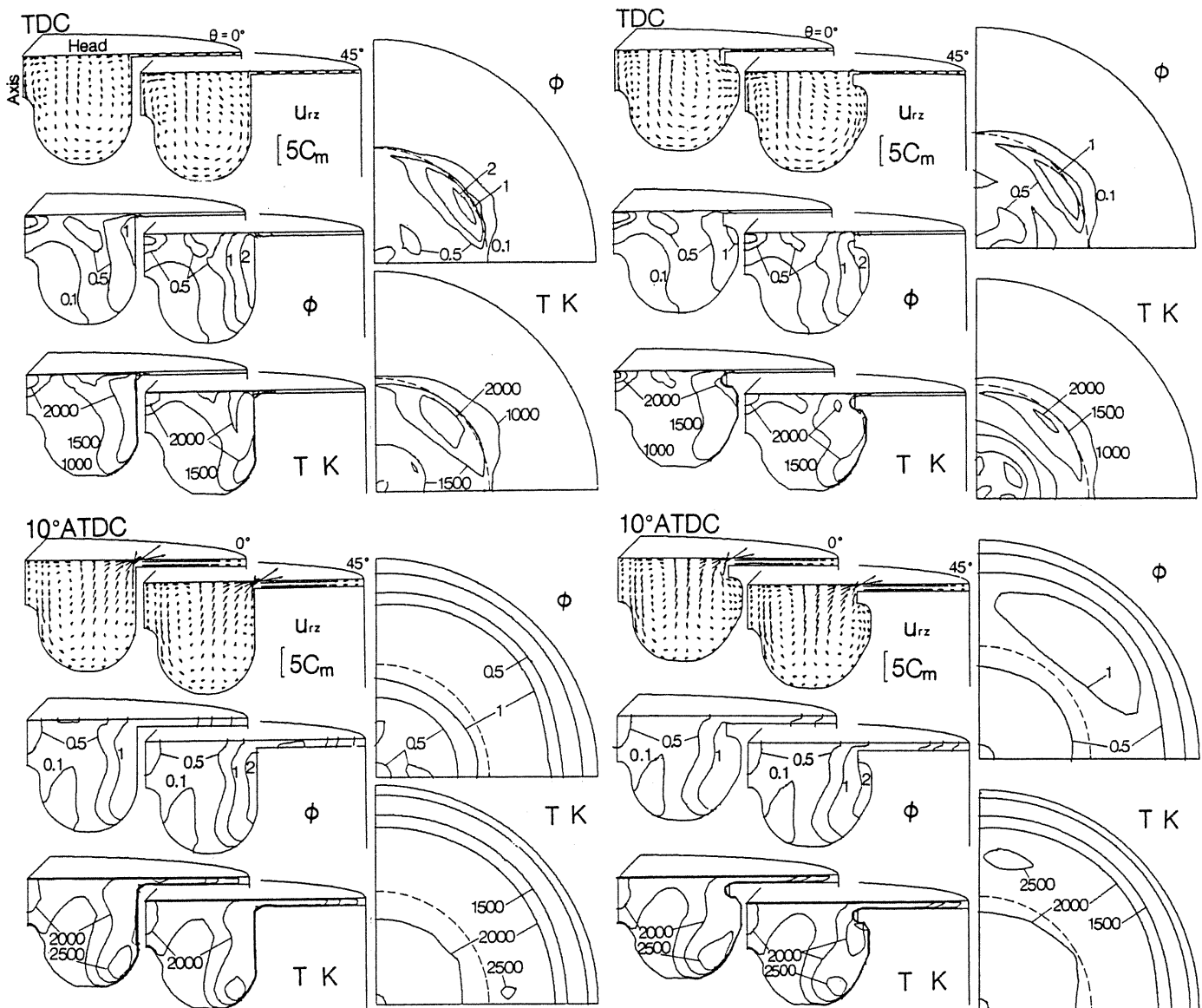


Fig. 7 Predicted distributions of resultant velocity on cylinder axial-radial plane  $u_{rz}$ , local equivalence ratio  $\phi$  and temperature  $T$  in ordinary (left) and reentrant (right) bowls [ $C_m$  is mean piston speed, swirl ratio 2.1]

4 the flame development in a deep-bowl chamber is shown for different cylinder swirl ratios. These photographs were taken on an engine having an optical access as shown in Fig. 5. Except for the case of  $r_s = 2.7$ , luminous flames are formed from the wall on which the spray hits and spreads in the direction of swirl flow. The cavity soon becomes full of flames, the flames being disintegrated later into many random flamelets. A flame spills off into the clearance space first by a fast expansion during the initial burning period and later by reversed squish associated with the downward motion of the piston. In the case of  $r_s = 2.7$ , the spray is strongly bent and no longer hit the side wall.

Such observations led to the characteristics that may be summarized as follows: Similar to shallow-dish chambers, a swirl considerably suppresses radial extension of the fuel. This keeps the fuel from excessive penetration into the clearance space, favorably preventing the mixture from suffering from an imperfect combustion. However, an

excessive swirl intensity again brings about a thermal pinch, hindering air utilization in the outer space. Such a situation may be avoided if a bigger nozzle orifice is used. The direction of fuel injection also has a significant effect on the spatial distribution of the mixture. Figure 6 shows the effect of spray angle and of swirl ratio on the engine performance. At  $r_s = 0$ , the best performance is obtained at a spray angle of  $100^\circ$  at which the geometrical spray path is the longest. At this spray direction, an increase in the swirl ratio brings about a rapid deterioration in the power output and exhaust smoke. This may well be attributed to the localized flaming zone in the middle of the cavity. In the swirled case, the best performance can be obtained at bigger spray angles ranging from  $140^\circ$  to  $160^\circ$ , and the best performance is better than that in the no swirl case.

A retarded injection should be employed to lower the concentration of exhaust oxides of nitrogen. This inherently leads to a higher concentra-

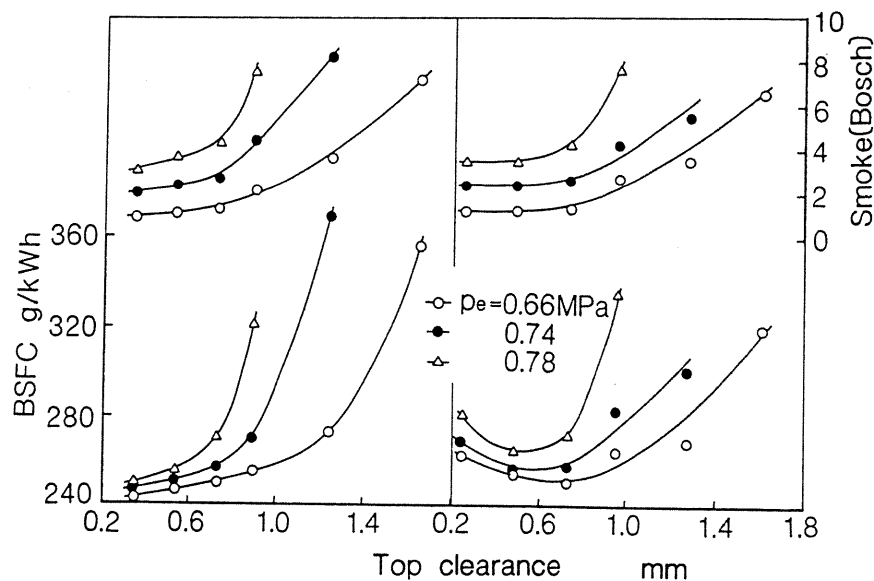


Fig. 8 Effect of top clearance on engine performance and exhaust smoke [ $p_e$  is brake mean effective pressure, cylinder bore 92mm, stroke 90mm, engine speed 2,000rpm]

tion of particulate matter including soot. In the author's view, the increased soot concentration at retarded injection is concerned with an excessive penetration of a rich mixture into the clearance space. Recently, the present author and co-workers made a high-speed gas-sampling study(7) and observed the following points: In the upper part of the piston bowl, the local average fuel-air equivalence ratio is close to unity, but the gas contains a lot of unburned hydrocarbons having a higher boiling point, and the retardation of the fuel injection results in an increase in the unmixed and unburned gas in the clearance. In the case of a reentrant bowl, that is said to have a higher capability for reducing soot even at a later injection, retarding the injection does not bring about a poor mixture during the spilling off into the clearance.

In Fig. 7 some results of numerical fluid dynamic simulation(8) are shown to explain the difference between reentrant and ordinary bowls. In the ordinary bowl, a rich mixture having a higher local equivalence-ratio is penetrated directly into the clearance, while in the reentrant bowl a clockwise vortex is formed just below the the reentrant projection, thereby suppressing the outflow of a rich mixture into the clearance space. Since excessive spill-off of a rich mixture is one of the sources of soot, the reentrant bowl may reduce the soot concentration even at a retarded injection.

Pertaining to such an outflow of rich mixture into the clearance, the effect of top clearance may be important. In Fig. 8 some results obtained at different distances of top clearance on a single-cylinder engine are shown(9), for the cases of a fixed cavity volume (left) and of a fixed compression ratio (right). In either case, a big clearance leads to a poorer combustion. Numerical fluid dynamic simulations suggested that a favorable combustion obtainable at a smaller clearance is attributed partly to the suppressed ejection of a rich mixture into the clearance and partly to the increased intensity of turbulence.

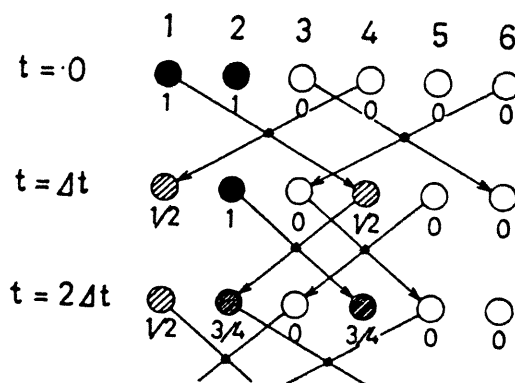


Fig. 9 Mixing process in collision-redispersion model

#### HETEROGENEITY AND THE DECREASE BY TURBULENT MIXING

As has been stated earlier, many fluid lumps having different fuel concentrations are formed and fill up the entire combustion space during the main part of burning after ignition starts. A number of large and small turbulent eddies appear to be distributed almost uniformly; such small fluid lumps having different fuel concentrations are stirred and mixed at a molecular level thereby undergoing chemical reactions. The burning rate during diffusive combustion is basically controlled by turbulent mixing, and the initial segregation tends to decrease to form a mixture of less heterogeneity as time goes on.

Such a situation can be described by a stochastic model for diesel combustion, which Ikegami and Shioji proposed in previous papers(10)(11). This model is based on the concept that combustion proceeds with turbulent mixing which causes the heterogeneity of temperature and concentration to tend to a uniform state. The progress of turbulent

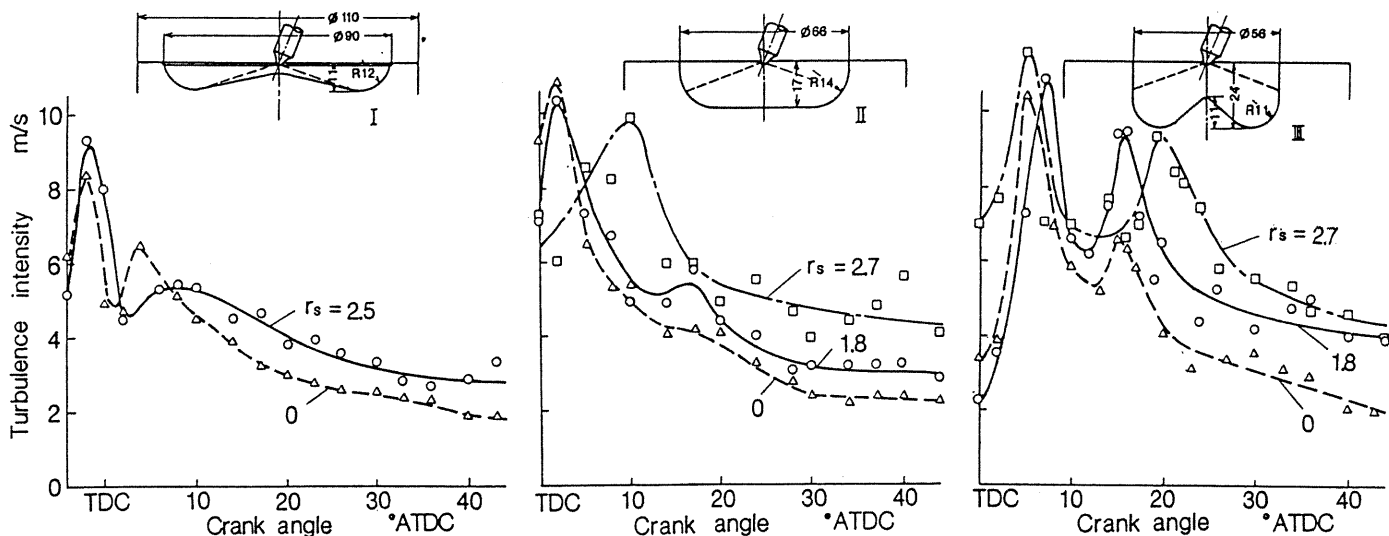


Fig. 10 Changes of turbulence intensity for different combustion chambers under various swirl ratios  $r_s$  obtained by image processing technique

mixing is expressed by Curl's collision-redis-persion model which assumes that random collision of two fluid particles having different thermodynamic states takes place at an equal probability in terms of time, thereby producing two identical particles having the same state of arithmetic mean properties as the colliding pair.

To have a better comprehension of this process, let us assume only six fluid particles, two being pure fuel (fuel mass fraction  $y=1$ ) and the remainder being pure air ( $y=0$ ) in the beginning. For instance, if random collisions take place twice during the time interval  $\Delta t$ , then the new state will be that as shown in the second line in Fig. 9. In such a way the state changes toward less heterogeneous state as time goes on, finally reaching a uniform state of an average mass fraction of  $y=1/3$ . In actual circumstances, there are many fluid particles and hence an instantaneous state should be dictated by using the probability density function. The number of collisions for one particle during a unit of time is called the collision frequency, or mixing frequency, which reflects the characteristics of turbulence. This progress may determine not only the rate of heat release but also pollutant formations that are strongly dependent both on fuel concentration and on temperature. The concept of  $k-\epsilon$  model has been introduced to describe kinetics of turbulence in the actual model calculations.

Significant points made by studies along these lines and the related ones can be summarized as follows:

(1) In the spray process, the heterogeneity significantly reflects the turbulence characteristics in the spray jet. A higher intensity and a smaller scale of turbulence make the mixture less heterogeneous.

(2) In the middle stages of combustion, turbulence intensity decreases from time to time. This is attributed to natural dissipation and the decrease in density with the piston motion, thus the mixing of fluid particles is delayed in the later stages. In Fig. 10 the measured changes of the turbulence intensity with crank angle are shown (12) for different swirl ratios and for different chamber geometries. These results were obtained from high-speed

flame photographs using an image processing technique. We may note that the turbulence decays with crank angle, that the deep-bowl chamber gives a higher level of turbulence, and that the higher the swirl ratio the higher the turbulence intensity becomes.

(3) The concentration of the exhaust nitric oxide NO is greatly affected by the temperature heterogeneity until  $20^\circ\text{ATDC}$ . Figure 11 shows the predicted fuel-air equivalence ratio pdf (probability density function), temperature pdf, and NO pdf during combustion for three different injection timings. It is noted that in the earlier stages of burning, equivalence ratio  $\phi$  and temperature  $T$  distribute quite widely. The highest end of the temperature distribution is responsible for NO formation.  $\phi$  and  $T$  become less spread as the crank angle advances. The mass-average temperature, as shown in broken line, decreases with crank angle owing to bulk quenching caused by the piston motion. NO is no longer formed at crank angles beyond  $20^\circ\text{ATDC}$ . Thus it may be stated that the temperature heterogeneity is responsible for the NO formation, and the decrease in the heterogeneity and the bulk quench finally terminate the formation. A lower NO concentration at the retarded injection timing may also be interpreted by the fact that only a small amount of fluid particles can reach a high temperature, as can be seen from Fig. 11. Figure 12 shows the predicted and the measured exhaust  $\text{NO}_x$  concentrations and the peak cylinder pressures  $p_{\text{max}}$  for three injection timings at a fixed swirl ratio of 2.7 (13). It may be seen from the figure that the present model predicts  $\text{NO}_x$  well, although there is a certain discrepancy at a higher equivalence ratio, owing to neglect of macroscopic maldistribution of fuel.

A similar situation is likely as regards soot formation that might be more dependent on the concentration heterogeneity in later stages of burning. This would be especially the case when maldistribution of fuel prevails. In later burning stages, the concentration heterogeneity tends to be frozen partly because the turbulence decays and partly because the turbulence scale increases. Unfortunately, the stochastic model is not capable

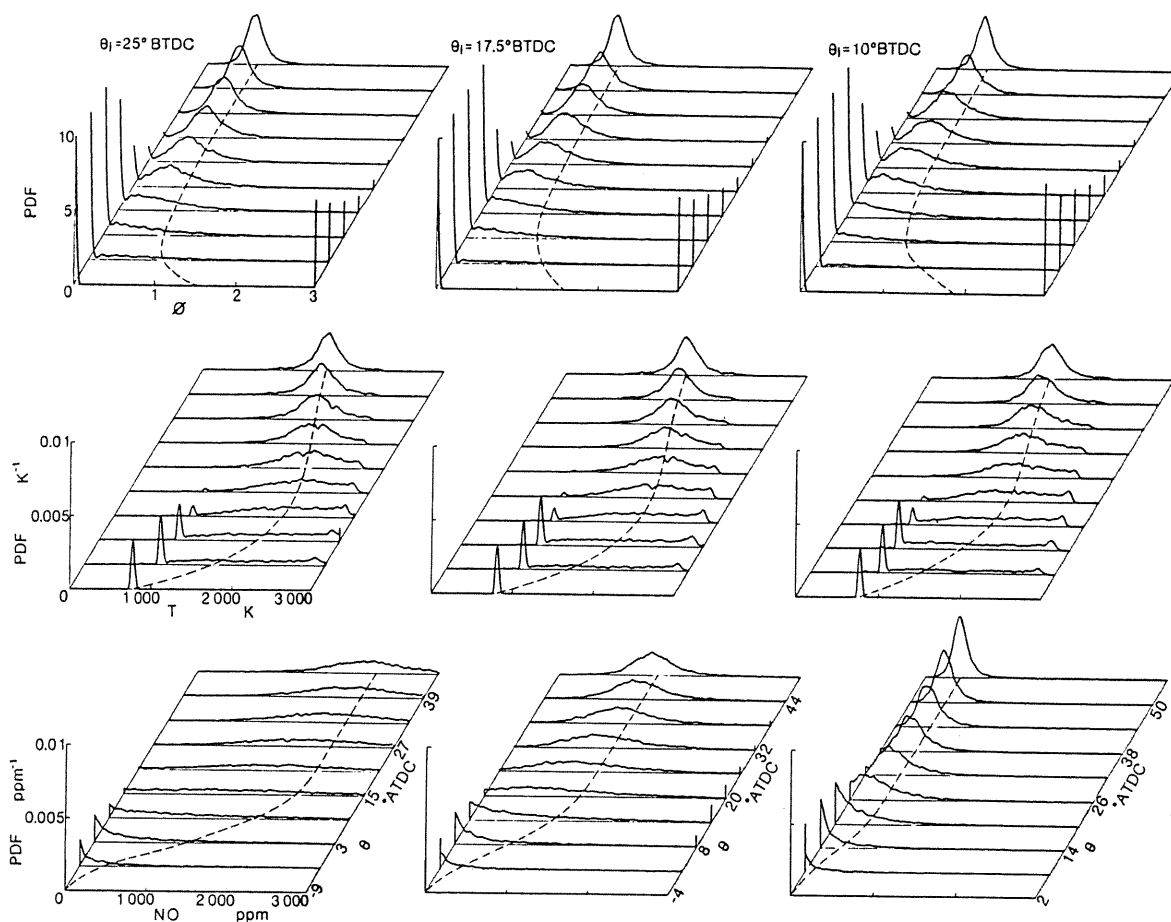


Fig. 11 Predicted pdf of equivalence ratio  $\phi$ , temperature  $T$ , and NO concentration, against crank angle  $\theta$  for different injection timing  $\theta_j$

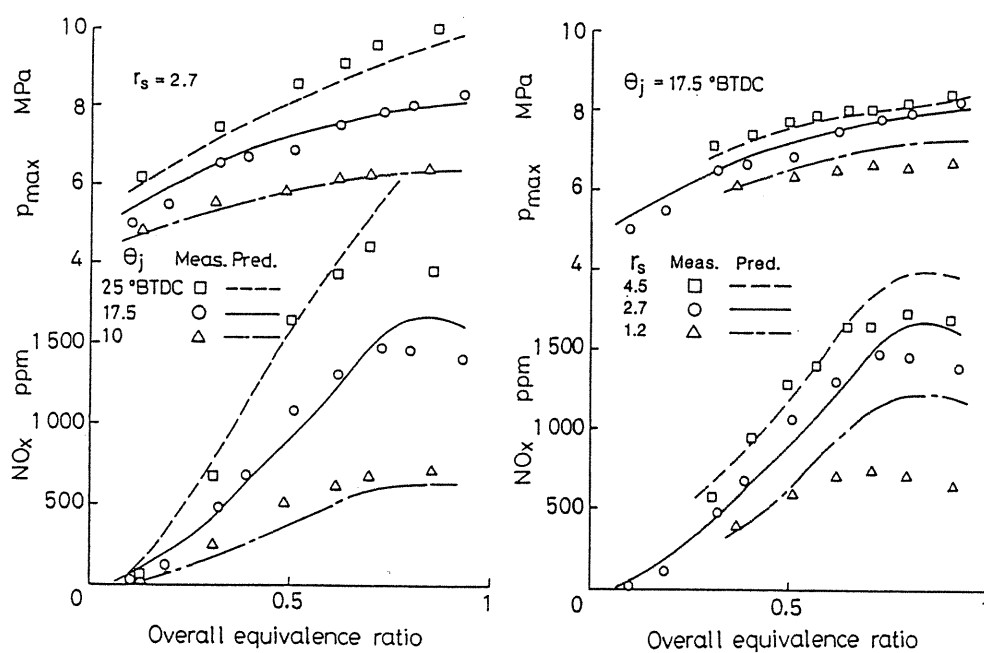


Fig. 12 Predicted and measured exhaust concentrations of oxides of nitrogen  $NO_x$  and peak cylinder pressure  $P_{max}$

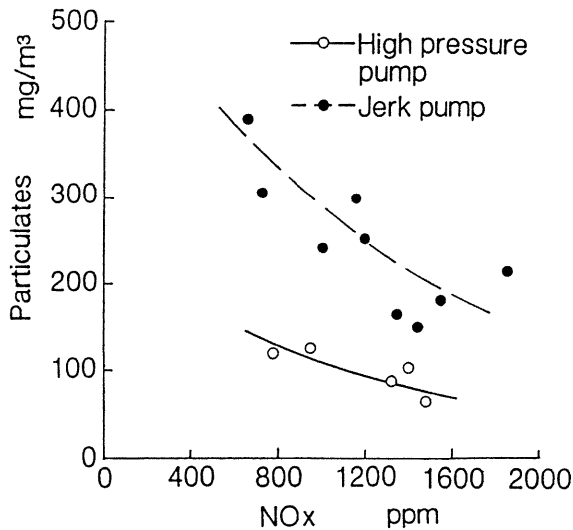


Fig. 13 Relationship between particulates and nitric oxide for different injection rates [a fixed equivalence ratio at 0.65]

of quantitatively predicting soot formed at present, but it would be possible once kinetics of turbulence and of soot formation and burn-up are established. Nevertheless, we can at least forecast the following point from the knowledge of the stochastic model: In the early stages when highly heterogeneous nature prevails, numerous soot particles may be formed on the fuel-rich side of the hot interface in fuel-rich zones. The subsequent homogenization process by strong turbulence promotes reburning of the soot particles once formed. If the turbulence decays too quickly, reburning is hindered, causing much unoxidized residual soot in the exhaust. In other words, it is necessary to keep the turbulence intensity high enough to ensure soot burning. This problem will be discussed again later from a more practical point of view.

#### REDUCTION OF PARTICULATES

Based on what have been mentioned in the preceding chapters, we now discuss the problem of reducing diesel pollutants that is of great concern at present. As has widely been accepted, reducing oxides of nitrogen increases particulate matter and this hampers their simultaneous reduction a great deal.

The most probable method to arrive at the goal is considered to use highly pressurized injection. Injection at a peak pressure around 100 MPa or more entails a smaller nozzle orifice diameter to obtain a proper degree of penetration; otherwise it may fail to keep the noise emission and white and/or blue smoke within allowable levels. There are two other advantages at the use of a smaller nozzle orifice; one being finer fuel drops that may be formed in the spray, and the other being smaller turbulence scale in the spray jet. Both promote not only fuel gasification but also turbulent mixing of fuel with the entrained air. Furthermore, the increased injection pressure gives finer fuel drops (14) and hence a greater amount of air is entrained into the spray, thereby preparing a leaner mixture in the spray(15). A higher injection pressure gives a higher spray momentum inducing turbulence at a

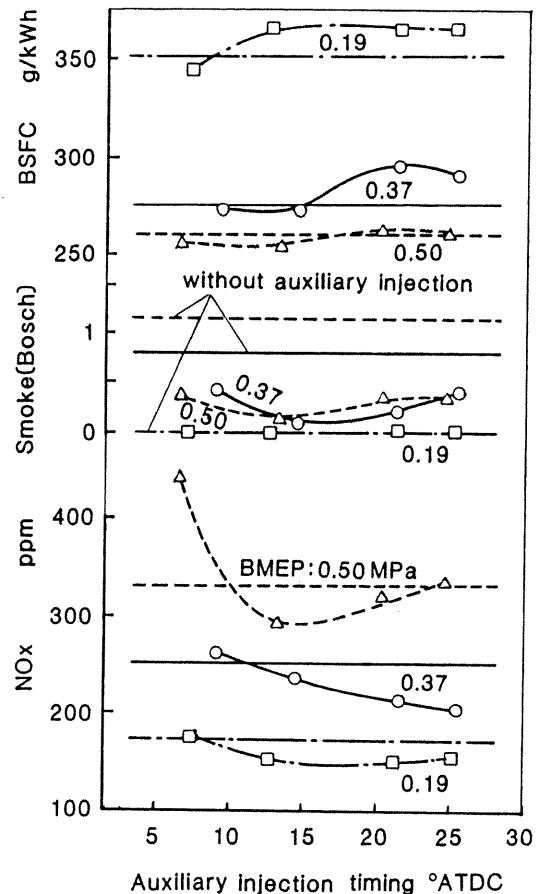


Fig. 14 Effects of auxiliary injection timing on engine performance and exhaust emissions [main injection timing 9° BTDC, engine speed 1,200 rpm]

higher intensity. All these factors will favor to decrease the concentration heterogeneity a great deal during injection.

In Fig. 13 the effect of the pressurized injection is shown(16). Tests were made on a single cylinder engine with deep-bowl chamber using two different injection systems; one is the ordinary system that uses an ordinary jerk pump and a four-0.31mm hole nozzle at a peak injection pressure at 40 MPa, and the other is a high pressure system using a four-0.20mm hole nozzle. The results clearly suggest the pressurized injection may halve the particulate emission for the same level of the oxides of nitrogen. There are many other reports that confirmed the capability of the pressurized injection for reducing particulate matter. Recently, studies were performed to look at combustion at injection at a peak pressure ranging from 150 MPa to 300 MPa(17). The results showed that at such a high injection pressure, flames are less luminous and extinct much earlier than in the ordinary case.

As has been described earlier, it is necessary to keep the turbulence intensity high enough to ensure soot burning. In the case of the highly pressurized injection, the concentration heterogeneity is destroyed faster until the end of injection than in the ordinary case, because of the elevated turbulence intensity with finer eddies. This must have brought about less luminous flames and a lower smoke level.



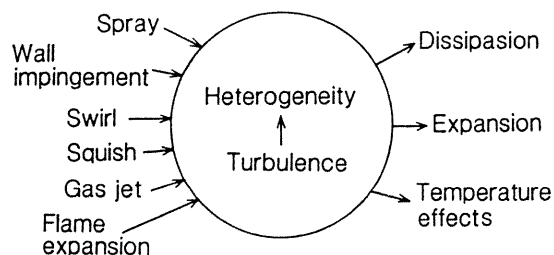


Fig. 15 Factors affecting turbulence during combustion

There are other strategies that may enhance or preserve the turbulence for quickly decreasing the heterogeneity during expansion stroke to ensure soot burning. A typical example would be the Isuzu square-bowled combustion chamber(18), which is capable of preserving a higher level of turbulence intensity until later crank angle degrees. Small turbulent eddies might be induced due to flow separation when a swirling fluid passes each corner of the bowl. In fact, this type of combustion chamber is said to work at a later injection without a large sacrifice in the exhaust smoke. Other examples are the auxiliary cell(19) and air cell(20) each installed on direct-injection diesel engines. These combustion systems are being studied by some researchers, with the intention of applying a strong disturbance to the burning gas in the middle and later stages of burning. Figure 14 shows an example of such tests made on a combustion system consisting of a deep-bowl chamber and an auxiliary cell into which an amount of fuel is injected. Over the whole range of auxiliary injection timing, the smoke level is much lower than without auxiliary injection. At the same time, the brake specific fuel consumption (BSFC) is little changed, and oxides of nitrogen are reduced in a range of auxiliary injection timing.

The above examples probably support the present view that the enhancement of the turbulence enables promoting combustion of soot particles. There are many factors that may be considered to either enhance or weaken the turbulence. Such factors are summarized in Fig. 15. Unfortunately, kinetics of turbulence are not fully established at present especially for the combusting case, and hence we must discuss them only qualitatively. Some sort of thermal effects might exert an influence on turbulence; laminarization is one of such effects and is caused by a decrease in kinematic viscosity at elevated temperatures is one of the factors to emaciate turbulence, while flame-generated turbulence an enhancing factor. These effects would require further clarification.

Lastly, the deep-bowl combustion chambers are retouched on in the context of turbulent mixing. It has been reported from the study using a laser anemometry that at the end of compression the rate of turbulence dissipation is expected to be higher in a reentrant deep-bowl than in a non-reentrant bowl(21). It has been predicted from a computer simulation that the turbulence intensity in the clearance space is much higher than elsewhere during combustion when the reversed squish becomes strong. These results suggest the important role played by turbulence in favoring turbulent mixing.

## CONCLUDING REMARKS

This paper surveyed diesel combustion from the standpoint of gas motions and turbulent mixing, based mainly on studies performed by the present author and his co-workers. It has been pointed out that the spray and gas motions should be best adapted to ensure a better air utilization. Some points that deserve attention are as follows; the spray impinged on the wall creates flames with a circulatory motion, which may appreciably improve the air utilization. However, over-penetration brings about local accumulation of fuel thereby leading to a smokier combustion. To give a proper penetration of the spray to the wall, one can use either the nozzle having a reduced orifice diameter or a swirl motion. The swirl may favorably suppress radial penetration of the spray, but one with an excessive intensity results in a thermal pinch causing a poorer air utilization. In the case of deep-bowl chamber, an excessive outflow of a fuel-rich mixture should be avoided to prevent incomplete combustion. It has been also emphasized that the heterogeneity of the mixture changes from time to time undergoing a strong turbulent motions. Accordingly, the turbulent mixing process is one of the essential factors that govern the progress of combustion and also the pollutant formation. Enhanced turbulence likely promotes mixing between fuel-rich and lean fluid particles thereby accelerating combustion. It is also plausible that nitric oxide and soot are formed in the interface between hot zones and fuel-rich zones, and that soot burning might be governed by turbulent motions in the middle and later stages of combustion.

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