Physics behind Diesel Sprays

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Abstract

Diesel spray formation that is a dominant phenomenological event for performance of diesel engine and its combustion emission, has been received much attention not only from engineering but also from scientific field. Cavitation in an injector, breakup process of a high-speed fuel jet, fuel spray development in a combustion chamber, and combustion process following these atomization processes contribute to the scientific and engineering progresses in hydrodynamics. Diesel spray cavitation suggested the physical importance of turbulence caused by cavitation bubble disruption. Liquid surface stability problem in a breakup process has leading many liquid disintegration concepts and their numerical simulation methods. Measurement of a diesel spray promoted new scientific achievement of laser diagnostics. Engineering achievement of these items usually couples with physical considerations of diesel spray. Then finding of unknown physics behind diesel spray is the essential of next engineering approach of diesel spray. This paper highlighted physical approaches concerning spray behaviour such as liquid breakup, spray penetration, spray volume, velocity distribution, and air entrainment.

Introduction

At February 17, 1894, Rudolf Diesel and Hans Linder had succeeded the first self-running of compression ignition engine (diesel's engine) [1]. They used gasoline fuel and air-blast injection system to supply fuel into a cylinder as a diesel spray. Airless injection was called solid fuel injection, and its injection mechanism was developed by James McKechnie of Vickers in 1910 [2]. His poppet type injection system (1914) named "Vickers hydraulically actuated nozzle needle valve", became the technology origin of updated diesel injection system. From an early age of diesel engine development, distress problems of fuel injection were how to attain good atomization, fuel mass metering and timing control. Importance of these required performance items on diesel spray has been unchanged still now.

Solid fuel spray so called diesel spray was characteristic naming to an intermittent fuel spray formed in high ambient pressure surroundings and having high momentum and high turbulence, which were resulting in a good spray-air mixture in the combustion chamber. Figure 1 is an illustration of diesel spray behaviour and its combustion in a DI diesel engine. With an injector set on the cylinder head, diesel fuel was injected into a combustion space that was formed with piston cavity and upper portion of the cylinder. As shown in the figure, suitable mixture for well-controlled low emission combustion greatly depended on its formation process.

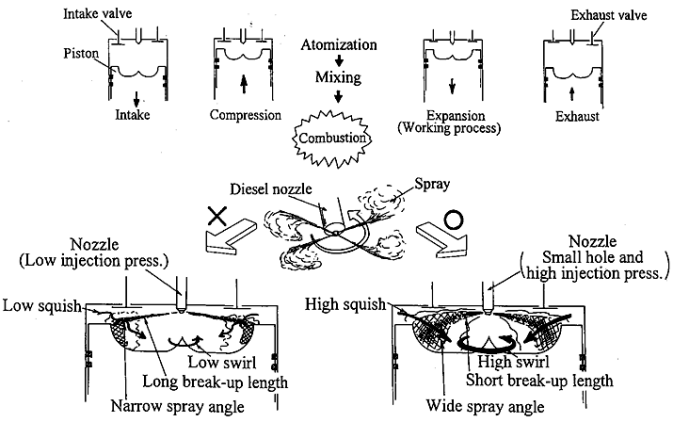

Figure 1 Diesel spray combustion in DI diesel engine

The first process of diesel spray formation was atomization. Liquid fuel should be rapidly atomized into a spray. Short breakup length of fuel jet and wide spray angle were suitable to promote well-distributed spray. Promotion of quick vaporization of fuel preferred to fine droplet spray. The second process of diesel spray formation was a mixing between fuel and air. Swirl flow induced by an intake air affected the bulk movement of spray and controlled the air utility in the combustion chamber. Squish flow induced by the piston motion agitated the mixing at around the lip of the cavity. Turbulence induced by bulk flows such as swirl and squish also promoted local mixing inside the spray.

The engineering achievements of diesel spray had been coupled with many hydrodynamics considerations such as pulsating pipe flow of high-pressure fuel [3], turbulent jet [4] [5] [6], liquid surface stability [7], and so on. For example, finding of cavitation phenomena in a nozzle entrance [8] contributed great advance on understanding of breakup mechanism of high-speed liquid jet. Then findings of unknown physics behind diesel

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spray become the essential scientific research work of next engineering approach of diesel spray. In this review, physical approaches concerning spray behaviour such as liquid breakup, spray penetration, spray volume, velocity distribution, and air entrainment are highlighted. Further, some of new approaches for future research works were proposed.

Diesel Spray

To understand the spray characteristics, we have to list up characteristic parameters of a diesel spray and to investigate relationships between these parameters and engine performance. The parameters shown in Fig.2, were well known as macroscopic parameters of diesel spray [9].

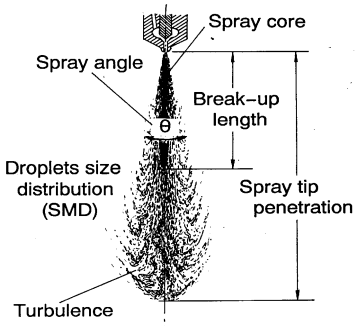


Figure 2 Characteristic parameters of diesel spray [9]

(1) Spray angle: The larger the spray angle, the wider the spatial distribution was.
(2) Breakup length: The original meaning of it was a length of liquid column in the spray or a length of liquid core that did not disintegrate. Sometimes it meant the extremely dense spray region near the injector. From a viewpoint of fluid dynamics, it corresponded to the potential core of a jet.
(3) Core of spray: It was a high dense spray region in the centre portion of diesel spray. It included un-breakup portion of liquid fuel.
(4) Spray penetration: It meant a macroscopic development of diesel spray.

A spray of large angle usually meant the spray having a short breakup length and a short core of spray. A spray having a long breakup length corresponded to a narrow unsteady spray and it resulted in wall adhesion of fuel when it impinged to a wall of combustion cavity. As shown in Fig.1, a spray with long breakup length resulted in high HC and PM emissions. When the mixture formation progressed, spray volume increased with an entrained air increase. This mixing behaviour well reflected to the penetration of a spray because air entrainment and spray velocity had strong relationship as explained later. Then the spray penetration evaluated easily by photographic observation was one of the most important characteristics of a diesel spray.

Not only macroscopic parameters but also microscopic parameters listed below were important for diesel spray combustion. There were a huge number of research works on mean diameter and size distribution of diesel spray. However, there was no definitive theoretical background of droplet size of diesel spray.

(5) Size distribution of spray: The smaller the size of fuel droplet, the faster the evaporation was. It meant fine droplet spray could easily make vaporizing fuel. However, the fine droplet spray could not maintain its momentum and it could not accelerate fuel-air mixing after losing the momentum. Many mathematical expressions of droplet size distribution were proposed and introduced in the literature [10].

(6) Mean diameter of spray: The Sauter mean diameter X_{SMD} is the most popular mean diameter. The definition is;

$$X_{SMD}=\frac{\sum_i n_i X_i^3}{\sum_i n_i X_i^2} \qquad (1)$$

where n_i is numbers of X_i diameter droplet. The Sauter mean diameter is a representative diameter corresponding to equivalent surface of spray and could show the average evaporation characteristics of a spray. Typical empirical equations of the Sauter mean diameter of diesel spray was as follows [11].

$$\frac{X_{SMD}}{D_n}=\mathrm{MAX}\left[\frac{X_{SMD}^{LS}}{D_n},\frac{X_{SMD}^{HS}}{D_n}\right] \qquad (2)$$

$$\frac{X_{SMD}^{LS}}{D_n}=4.12\cdot Re^{0.12}\cdot We^{-0.75}\cdot\left(\frac{\mu_l}{\mu_a}\right)^{0.54}\cdot\left(\frac{\rho_l}{\rho_a}\right)^{0.18} \qquad (3)$$

$$\frac{X_{SMD}^{HS}}{D_n}=0.38\cdot Re^{0.25}\cdot We^{-0.32}\cdot\left(\frac{\mu_l}{\mu_a}\right)^{0.37}\cdot\left(\frac{\rho_l}{\rho_a}\right)^{-0.47} \qquad (4)$$

Where, MAX[A,B] means the larger value of the two. μ is viscosity of liquid (l) and air(a), and ρ is density. Positive index of Re (Reynolds number) and negative index of We (Weber number) meant that both of shear force in a nozzle and jet stability were dominant factors on X_{SMD}. However, applicable range of nozzle diameter D_n was unclear and it was one of fundamental issues of liquid atomization. X_{SMD} of a traditional type diesel spray was ranging into 25μm-35μm, and that of 10μm-20μm was attained with a high-pressure injection.

(7) Spatial distribution of spray: This is an important factor to control combustion characteristics of diesel spray. Many photographic observations and numerical simulations have been carried out until now. However, no direct and definitive evaluation method for the spatial distribution of spray was established. Spray angle, spray penetration, spray volume, spatial number concentration of spray droplets and a combination among these parameters were often used for characterization of spatial spray distributions.

(8) Turbulence: Spatial and time scale of turbulence and their intensities were important to promote an internal mixing process of bulk spray that was consisted of fuel droplets, fuel vapour, and entrained air. Further, an injection-to-injection variation (diesel spray fluctuation) was also important for the engine performance.

Demand for Diesel Spray and Task of Diesel Spray Injection

Diesel combustion model shown in Fig.3 was proposed by J.E.Dec [12]. In the model, NO and soot formation mechanisms that were most distress problems of diesel combustion were discussed based on the motion of diesel spray. No-flame region near the injector [13] was a unique characteristic of diesel spray and it sometimes called lift-off length of diesel spray. B.Lewis and G. von Elbe [14] introduced lift-off behaviour of premixed gaseous jet, but lift-off of diesel flame was somewhat different from the original meaning of lift-off. Lift-off of diesel flame related to the breakup and core length of diesel spray. Combustion air was supposed to be mixed in this region. Flesh oxygen entrainment was the result of air entrainment to the spray. Rich fuel/air mixture zone was the core of diesel spray. Soot was formed, and was remained in an insufficient oxygen combustion zone where air entrainment was restricted. NO was formed at high temperature zone where the spray tip movement was stagnated. Figure 4 is a more detailed and advanced model proposed by Aizawa et al. [15]. It based on photographic observation of diesel spray injected by a common rail injection system with high pressure. Main differences between these two models were location of air entrainment and inside spray motion. In this model, air entrainment into the main body of spray was emphasized. In other words, difference of models came from different concepts of diesel spray turbulence. The latter one uniquely modelled internal structure related to air entrainment caused by turbulent mixing motion in the spray.

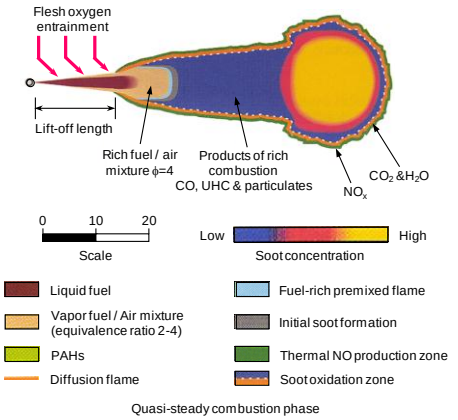
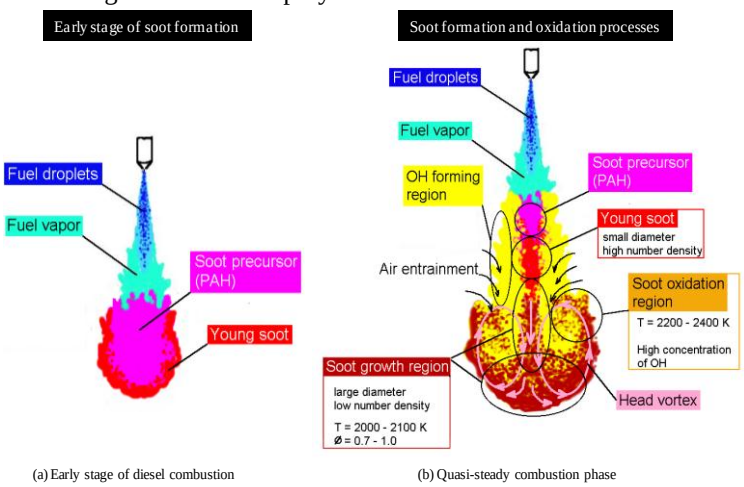


Figure 3 Diesel spray combustion model [12]



(a) Early stage of diesel combustion (b) Quasi-steady combustion phase

Figure 4 Diesel spray combustion and soot formation model [15]

Figure 5 shows the injection timings of diesel spray for various purposes and heat release related with each injection. Early and pilot injections were effective for preparation of combustible mixture for ignition. Self and spontaneous ignition was the start of combustion heat release. Main combustion usually divided into following three stages.

1st stage: Initial combustion, premixed combustion of accumulated mixture before ignition.

2nd stage: Main combustion, high turbulence diffusion combustion of simultaneously injected fuel.

3rd stage: Afterglow combustion, low turbulence diffusion combustion of remained fuel.

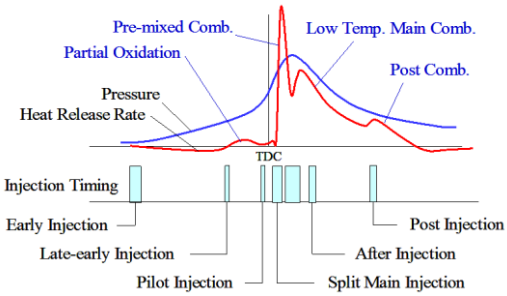


Figure 5 Injection timing of diesel spray for various purposes

Combustion models introduced above were corresponded to the 2nd stage of main combustion. Sprit main injection was designed to attain high turbulent mixing of diesel spray in this stage. After-grow combustion for oxidation of soot was sometimes promoted by after-injection. Purpose of post injection was exhaust gas temperature elevation for after-treatment system.

Table 1 is a list of future engine hardware for NOx and other emission control.[16][17] As for the diesel fuel injection, injection pressure has increased from 180MPa to 220MPa, and ultra-high pressure injection such as 300MPa is now considered. Downsizing and high boost engine are effective reduction ways of low fuel consumption. Consequently, in-cylinder pressure becomes increasing. In an ultra-high boost engine, compressed air density at injection timing might increase beyond 40kg/m^3. Then the circumstances around diesel fuel injection have been changing drastically.

Table 1 Engine hardware [16][17]

Engine-out NOx levels	Euro VI 04.g/kW-hr	~US2007 1.6g/kW-hr	Euro VI -90% deNOx ~US2010 -94% deNOx 3.2g/kW-hr
Fuel injection pressure, bar	3000 bar	2200 bar	1800 bar
Peak cylinder pressure	230 bar	180 bar	150 bar
EGR at full load	45%	27%	15%
Charge cooling relative to rated engine power	90%	50%	30%

General demand for a diesel spray was adequate mixture supply to a space of compressed hot surroundings in a diesel engine cylinder for controlled combustion. Then, the task of diesel fuel injection was as follows.

(1) Fuel mass and combustion timing control
(2) Homogeneous/heterogeneous mixture formation
(3) Fuel preparation in a combustion chamber space

As for the first item, it was involved in practical engine designing as a kind of combustion control methods. As concerns of homogeneous/heterogeneous mixture formation with well-atomized diesel spray, many research works and discussions have been performed for improvement of diesel spray combustion.

When we consider the essential task of diesel spray injection, **"fuel transportation to a desired space at a desired timing"** is the most important task as shown in Fig.6. This concept was partially introduced into an optimum matching of combustion chamber and diesel spray. However, we need more deep understandings for mixture preparation in a combustion space and task of fuel injection. Fuel injection should be considered as one of the methods to supply an adequate fuel mixture in a desired space and at a desired timing. When an alternative and innovative method to supply fuel mixture directly into a desired space is developed, no discussion about diesel fuel injection and atomization processes is needed.

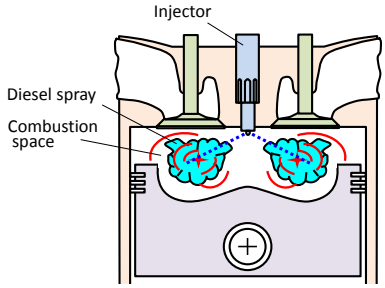


Figure 6 Task of diesel spray injection

Nozzle Flow and Cavitation

A typical layout of common rail injection system with electrically controlled needle valve is shown in Fig.7. Needle valve was driven by solenoid or piezo-electric actuator so that multi-time injections in one combustion cycle were possible. Liquid fuel was injected with a pressure of around 200MPa into a combustion chamber, and a fine and high momentum diesel spray was formed.

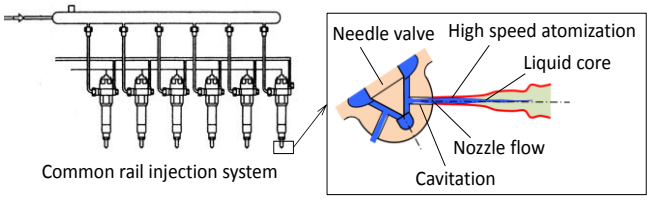


Figure 7 Nozzle flow and liquid jet atomization

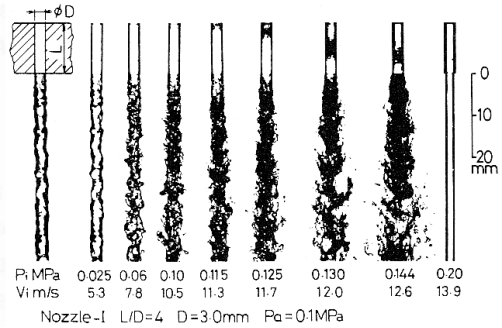


Figure 8 Nozzle cavitation and liquid jet [18]

Liquid surface deformation caused by Rayleigh instability and shear force caused by high speed movement were main driving forces of atomization for laminar liquid jet. More important factor for atomization of high-speed liquid jet was reported with Hiroyasu et al.[18] They used a transparent scale-up nozzle and observed atomization phenomena related with cavitation. Their result shown in Fig.8 suggested that smooth jet coupled with no cavitation was hardly atomized. Figure 9 was their summary of cavitation effect [8] on high-speed liquid jet. When the velocity increased beyond a certain limit, cavitation fixed at the entrance appeared as shown in (d). Disruption of the cavity generated intense turbulence that could perturb the jet surface close to the exit. This situation was generally observed in an atomization process of a high-speed liquid jet.

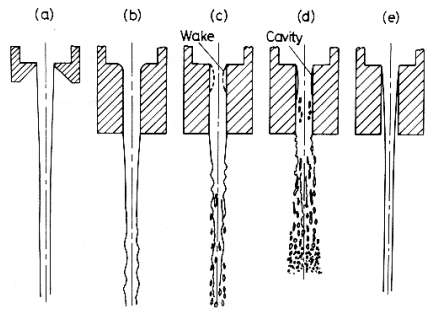


Figure 9 Model of cavitation and liquid jet [18]

According to the fluid dynamics analysis, cavitation number K_c shown by Eq.5 was a characteristic index of cavitation.

$$K_c = \frac{P_1 - P_v}{\frac{1}{2}\rho V_1^2} \qquad (5)$$

Generally, K_c<1 meant an onset of cavitation. Definitions of P_1, P_v and V_1 are indicated in the internal flow model shown in Fig. 10. Using this model, cavitation number coupling with nozzle friction f and back pressure P_a was obtained as follows.

$$K_c = \left(\frac{D_c}{D}\right)^4 \left\{\frac{P_a - P_v}{\frac{1}{2}\rho V_i^2} + f\frac{(L-L_c)}{D} + 1\right\} - 1 \qquad (6)$$

$$f = \frac{0.316}{Re^{0.25}} \qquad (7)$$

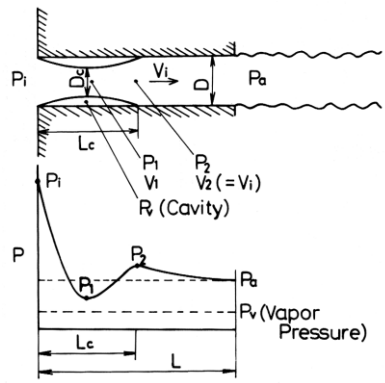


Figure 10 Cavitation model in a nozzle [8]

Where, f meant a friction factor based by Blasius's equation. The higher the back pressure (ambient pressure Pa) and the longer the L/D of nozzle, cavitation number increased more. It meant a suppression of cavitation.

Direct evidence of diesel nozzle cavitation was captured with Bosch researchers and C.Tropea group. [19][20] Figure 11 is their transparent diesel nozzle. Figure 12 is a wall cavitation captured by shadowgraphy, laser light sheet, and fluorescent particle image velocimetry with YAG laser. Figure 13 shows possible forms of cavitation in sac volume and nozzle. A lot of cavitation studies on expanded scale model nozzles and numerical calculations on a flow field including cavitation have been carried out. These research works have made great contribution on diesel nozzle development. However, there is a little knowledge about the relationship between cavitation and atomization.

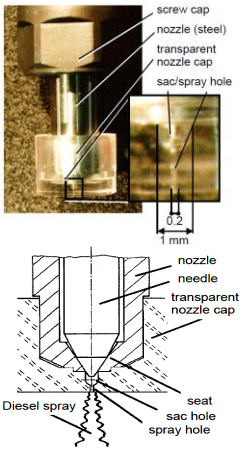


Figure 11 Diesel nozzle with transparent cap [19]

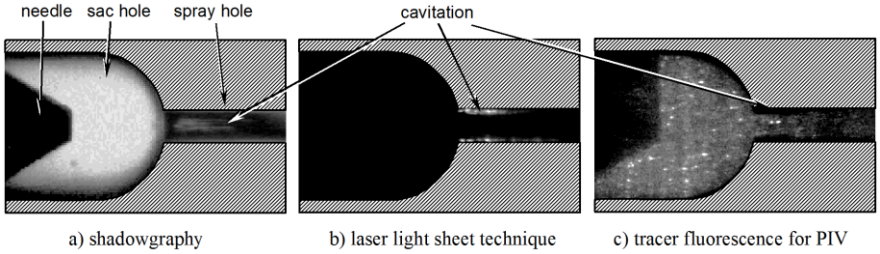


Figure 12 Internal nozzle flow with cavitation films; P_I=10MPa, Needle lift=60μm [19][20]

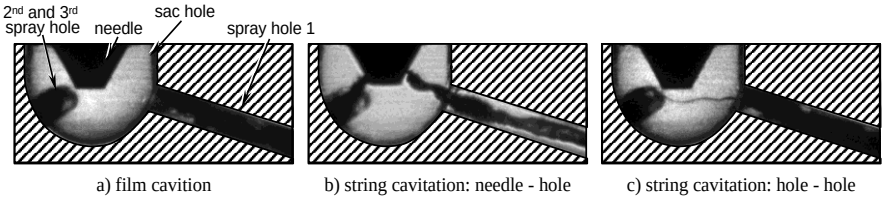


Figure 13 Possible forms of cavitation [20]

Unknown physics behind diesel nozzle cavitation are listed up as follows.

(1) Cavitation bubble and vapour pressure of fuel: There are many kinds of diesel fuel. Most of them are mixture of various liquids having different vapour pressures. Then intense of cavitation might be affected the fuel properties.

(2) Turbulence caused by cavitation bubble disruption: It might be a real origin of atomization. However, origin of atomization is still unknown physics.

(3) Physicochemical effect of cavitation: Cavitation consumed some kinetic energy and might change physicochemical properties of fuel. Owing to a sudden shrink of cavitation bubble at disruption, pressure and temperature in the bubble increase and exited molecules induced some fluorescence light emissions. It might result in some chemical effects on fuel molecules.

(4) Cavitation control: Suppression or enhancement of diesel nozzle cavitation means a new management of diesel fuel atomization.

(5) Cavitation number: It is a good index of cavitation, but we need other index to explain a relationship between cavitation and atomization [8][21].

Breakup Behaviour of Liquid Jet and Its Modelling

1) Breakup model of liquid jet

Since the diesel fuel jet near an injector exist is covered with too high dense spray and the jet velocity is also too high so that observation of breakup phenomena is impossible, the detailed mechanism of diesel fuel breakup is still unclear matter for diesel spray researches.

However to meet the demand of numerical simulation of diesel combustion, many kinds of breakup model had been proposed. Target of diesel spray modelling focussed on the breakup region with liquid core and dense spray surrounding it. Figure 14 is a basic breakup scheme for modelling and Fig.15 is breakup regimes of droplet.[22]

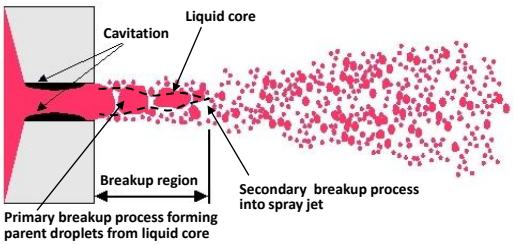


Figure 14 Breakup model of diesel fuel jet (modified from ref. [22])

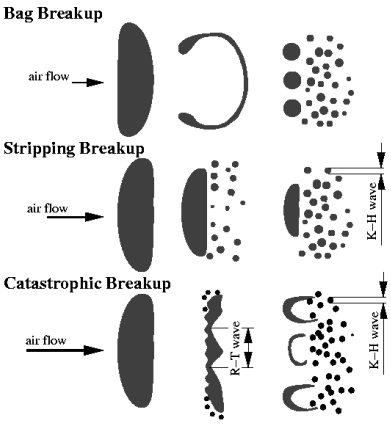


Figure 15 Schematics of droplet breakup mechanisms [22]

Wave Breakup (WB) model proposed by Reitz et al. [23][24][25] was based on a wavy-like surface disturbance. Taylor Analogy Breakup (TAB) model was introduced by O'Rourke [26] and developed by Tanner et al. (Enhanced Taylor Analogy Breakup, ETAB [27][28]). Those were combination models of liquid jet breakup forming large droplets which size was similar to the jet diameter and so called parent droplets, and secondary breakup based on Weber number instability caused by drag force. As shown in Fig.14, droplet breakup was controlled by a Weber number of a droplet and surrounding gas stream. It was classified as follows; Bag breakup: 6<We<80, Stripping (shear) breakup: 80<We<350, and Catastrophic (surface wave) breakup: We >350. These models and detail mathematical descriptions were developed coupling with numerical simulation codes such as "KIVA" and "FLUENT".

As the standpoint of phenomenological breakup behaviour, relationship between cavitation in the injector nozzle and primary breakup was obvious but no adequate modelling proposed. Then, physical consideration for complete modelling needs more detail observation of a liquid core hidden behind dense spray surroundings.

When a diesel spray was injected in a hot ambient gas and impinging to a combustion cavity wall, rebound of droplets, liquid film formation from adhered droplets and re-atomization of fuel film took place on the wall with spray evaporation. Spray behaviour on the wall was also modelled with a droplet movement controlled by the Weber number of each impingement droplet. Detail models of spray on a hot surface and re-atomization process were proposed by J.Senda et al. [29][30][31]. Their impingement diesel spray model with evaporation is shown in Fig.16. They classified the droplet behaviour using Weber number as shown in Fig.17. When the weber number was less than 80, it was supposed that droplets were stuck on the wall and fuel film evaporation occurred. When the Weber number of the droplet was larger than 80, rebound of droplet or impingement breakup were to be the main feature of diesel spray impingement. However, owing to the lack of real information of the spray behaviour, reliability of the model could not be checked, except the macro-scale movement of impingement spray. Further, it was dynamically interesting that critical Weber number (*We*=80) from "Bag Breakup" to "Stripping Breakup" in Fig.15, and from "Film Formation" to "Rebounding" in Fig.17 were the same. It meant that the stability of a droplet was the physical base of both models. Figure 18 is their general models for heat transfer, impinging breakup, and droplet dispersion.

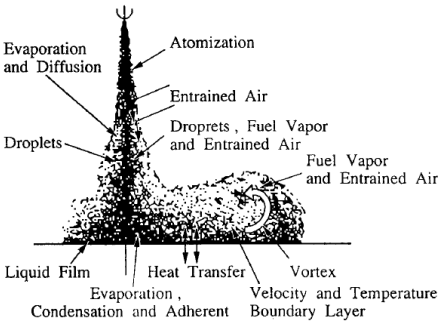


Figure 16 Model of evaporating and impinging diesel spray [30]

Breakup Form	d_{10}/d_i	V_b/V_i	Range in *We* No.
I non-breakup (into film)	—	—	$We<80$
II large breakup	0.3	0.3	$80 \le We < 600$
III small breakup	0.1	0.5	$We \ge 600$

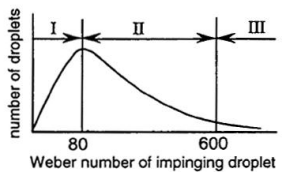


Figure 17 Fuel film breakup model for impinging diesel spray [29][31]

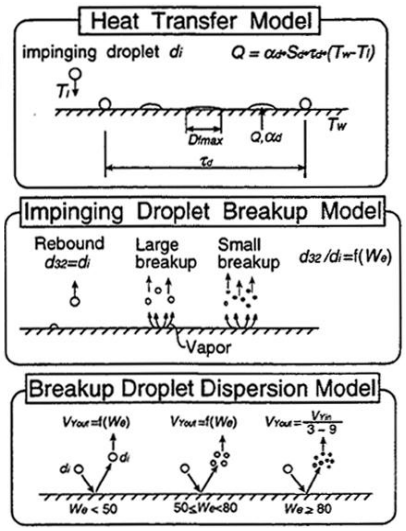


Figure 18 Diesel spray impingement model on high temperature wall [29][31]

2) Spray model

As shown in Fig.19, there were three kinds of diesel spray model for numerical simulation. These were package, parcel and two-phase flow models. Phenomenological non-dimensional simulation of diesel spray often used a package model. Package consisted of various size droplets and gaseous mixture. Package volume increased with air entrainment. All the evaporation and combustion phenomena were included in the package. On the contrary, parcel only contained various size droplets. Its movement was supposed to be the same as that of a representative droplet in the parcel. Each parcel meant source terms of mass, momentum and energy in the Navier–Stokes equations. It was adopted for many 3-D simulations of diesel spray. There were a lot of discrepancies between models and real movement of diesel spray, but it considered that priority of easy numerical simulation was higher than discrepancies. Two-phase flow model was used in a direct simulation of a diesel spray. However without many hypothetical simplifications for droplets, it might be difficult to use for diesel spray simulation.

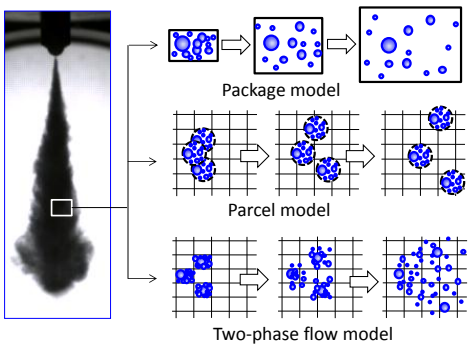


Figure 19 Diesel spray model for numerical analysis

Diesel Spray Development

1) Spray tip penetration

Since **"fuel transportation to a desired space at a desired timing"** was the main task of injection, one of the important characteristics of diesel spray was its penetration. Hiroyasu and Arai [32][33] measured diesel spray penetration. They reported the penetration data shown in Fig.20 and penetration model shown by Fig.21. Using a theoretical breakup analysis of Levich [4], they derived the following empirical equations of diesel spray.

Since velocity deceleration before breakup of liquid jet was negligible, spray tip moved with initial velocity at the nozzle exit. However, its velocity was somewhat lower than that of theoretical injection velocity because of lower velocity coefficient caused by complicated configuration of injector nozzle. Then, spray tip movement at initial development stage was expressed as follows;

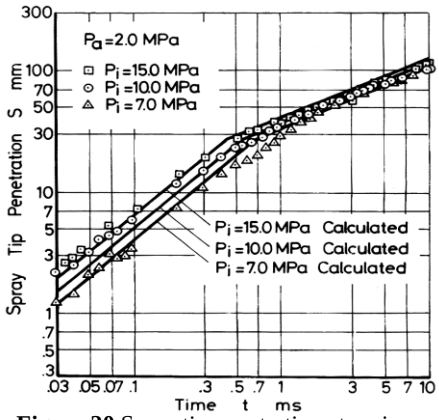


Figure 20 Spray tip penetration at various injection pressures [32]

$0 < t \leq t_b$

$$S = 0.39\sqrt{\frac{2\Delta P}{\rho_l}} \cdot t \qquad (8)$$

$$t_b = 28.7\frac{\rho_l D_n}{\sqrt{\rho_a \Delta P}} \qquad (9) \qquad\qquad L_b = 15.8\sqrt{\frac{\rho_l}{\rho_a}} \cdot D_n \qquad (10)$$

where, S is penetration length and L_b is breakup length (Fig.21). ΔP is effective injection pressure, D_n is nozzle diameter, ρ_a and ρ_l are densities of air and liquid fuel. Breakup time t_b means a corresponding time when a liquid jet breaks up to a spray. Figure 22 shows the breakup time obtained by Eq.9. It was known that ignition delay of diesel spray consisted of physical delay and chemical reaction delay, and physical delay consisted of breakup delay and evaporation delay. Owing the breakup time (breakup delay) being the same order of the total ignition delay and overlapping on other delay periods, ignition delay of diesel spray became complicated phenomena to be hard to analyse.

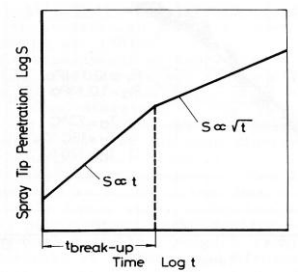


Figure 21 Schematic diagram of spray penetration [33]

After breaking up of the liquid jet, spray tip developed with the following equation [33] was derived from momentum conservation [34].

$t_b < t$

$$S = 2.95\left(\frac{\Delta P}{\rho_a}\right)^{0.25} \cdot \sqrt{D_n t} \qquad (11)$$

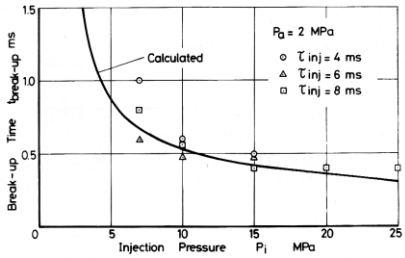


Figure 22 Breakup time [33]

These empirical equations were generally used for evaluation of diesel spray. For high-pressure injection spray with recent common rail system, tip penetration was little longer than that estimated here. It mainly caused by the estimation error of initial velocity. In other case of high-boost engine, cavitation number shown in Eq.6 increased with an increase of ambient pressure P_a and it meant that onset of cavitation tended to be supressed by back pressure of the nozzle and resulted in higher velocity owing to higher velocity coefficient of nozzle. Higher velocity resulted in longer penetration than that estimated with Eqs.8 and 11. Even though, above equations with modified coefficients were applicable for up-to-date diesel spray, and sometimes these equations were used for verification of numerical simulation works of diesel spray.

When non-dimensional scales of time and penetration length are introduced using Eqs.12 and 13, spray tip penetration could be simply expressed by Eqs.14 and 15. It meant that characteristic time and length of breakup phenomena gave similarity scales for spray tip penetrations of various kinds of sprays not only diesel sprays but also sprays in many engineering fields.

$$t^* = \frac{t}{t_b} \qquad (12) \qquad\qquad S^* = \frac{S}{L_b} \qquad (13)$$

$$0 < t^* \leq 1 \qquad S^* = t^* \qquad (14)$$

$$1 < t^* \qquad S^* = \sqrt{t^*} \qquad (15)$$

When a homothetic (similitude) development with constant spray angle and conservation of spray momentum were maintained, deceleration of spray tip velocity V_{tip} (=dS/dt) was considered to be caused by spray mass enrichment by entrained air. Then, following equations were derived;

$$\int_t \frac{\mathrm{d}M_f}{\mathrm{d}t} V_{inj}\mathrm{d}t = \int_t \left(\frac{\mathrm{d}M_f}{\mathrm{d}t} + \frac{\mathrm{d}M_a}{\mathrm{d}t}\right)\frac{\mathrm{d}S}{\mathrm{d}t}\mathrm{d}t \qquad (16) \qquad\qquad M_a = \int_{M_f}\left(\frac{V_{inj}}{\mathrm{d}S/\mathrm{d}t} - 1\right) d\,M_f \qquad (17)$$

where, M_a is entrained air. M_f is injection fuel mass and V_{inj} is injection velocity. Equation 17 shows that high injection velocity and high velocity deceleration of spray tip resulted in large air entrainment. This equation

was an estimation base of air entrainment and used as a principal concept of phenomenological diesel combustion modelling such as "HIDECS".[35][36]

2) Catch-up motion of diesel spray tip and spray tail behaviour

Even though a common rail injection system was used for diesel spray injection, fuel injection rate was fluctuating, and it was obvious at injection start and end. Further pilot injection and split injection were sometimes adopted for improvement of diesel combustion. An evidence of catch-up motion was reported by Arai et al. [37][38] Figure 23 is their optical measurement system for spray density and their results for split diesel spray. Photo-detector at the exit of injector showed splits spray injection phenomena. Spray density fluctuation corresponding to the split spray injection was clearly detected by photo-multiplier set at 10mm apart from injector exit. On the contrary, at 40mm, there was a slight trace of split injection and it means that the first and the second split sprays were mixed there.

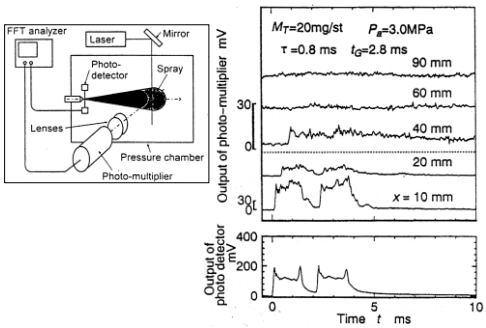


Figure 23 Optical analysis of spray density fluctuation caused by split sprays [37]

Figure 24 is an illustration of catch-up and push-away motion of diesel spray where the injection rate of second stage was higher than the first stage. Since the drag force of the second stage spray tip was smaller than the first one, the spray tip penetration of the second was faster than the first even if the injection rates of them were the same. Owing to the faster movement of the second, the spray tip of the second caught-up the first one, and both were stagnated at almost the same positions. Correlation length in the figure means the maximum length where the initial fluctuation of spray was remained. The second spray that penetrated beyond this correlation length was interacting with the first one.

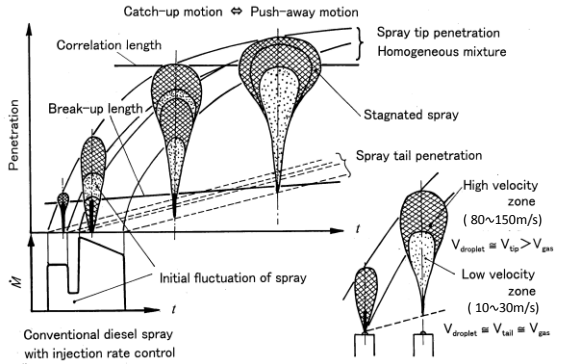


Figure 24 Tip penetration and catch-up process of diesel spray

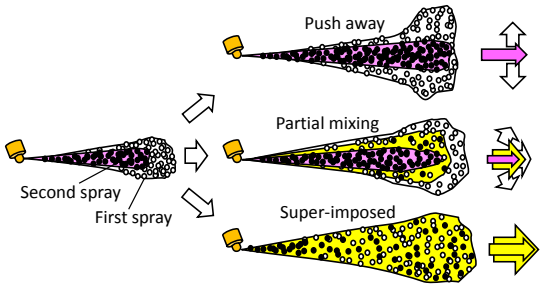


Figure 25 Illustrations of push-away, partial mixing, and super-imposed motion of diesel spray

Typical feature of spray tip movement is illustrated in Fig.25. When droplets in a spray were transported by entrained gas stream, the secondary spray tip that initially followed the first tip pushed away the first one to the lateral side. On the other hand, the second spray tip was super-imposed on the first one when droplet movement was completely separated from entrained air movement. Those two typical patterns depended on injection rate and so on.

Recently Zama et al. [64] have been performed PIV analysis on spray tip behaviour. Figure 26 is an example of velocity vector map near the tip of diesel spray. Detail condition of PIV analysis is explained in the final section of this paper. Downward direction in the figure was the penetration direction and lateral velocity near the spray periphery suggested the existence of push away motion modelled in Fig.25. At "A" location indicated by arrow was a stagnation point. Around this point, a vortex motion was clearly observed. It proved that air entrainment mechanism modelled in Fig.4 was reasonable.

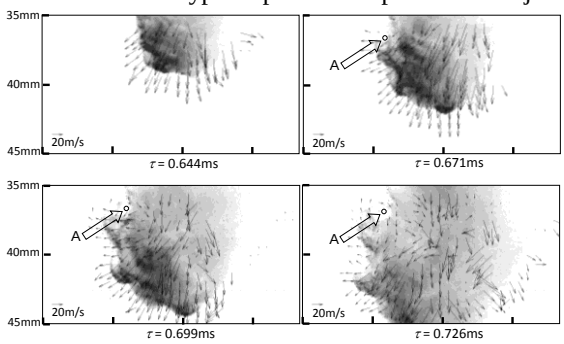


Figure 26 PIV analysis of inside velocity field near spray tip, P_{inj}=150MPa, P_a=4MPa [66]

Figure 24 also shows the behaviour of spray tail. Since no catch-up motion occurred in a zone of spray tail where the injection rate decreased, spray tail behaviour became simple. However, it resulted in another problem of spray behaviour. According to the result in Fig.23, spray density at 40mm was continuously detected beyond 8.0 milliseconds. It means that spray tail remained in the space for a long time. Low injection rate sometimes resulted in insufficient atomization so that low penetration rate and slow mixing were inevitable at the spray tail. There were a few studies concerning spray tail motion and its effect on combustion.

3) Combustion phenomena of diesel spray

A photographic example of diesel combustion is shown in Fig.27.[39] There were two possibilities for ignition location. One was the spray side periphery before impingement and the other was a location in the spreading impingement spray. During fuel injection period, there was no flame around a spray near the vicinity of injector [13]. Liquid length [40] or flame lift-off length [41][42] that was illustrated in Fig.3 was clearly observed in the photographs. Here, set-off length indicated in the photograph means this characteristic length.

The lift-off concept of flame was investigated widely as a kind of flame behaviours of combustible mixture jet. It was well explained by B. Lewis and G. von Elbe [14]. A lifted flame was observed when the issuing jet velocity exceeded flame propagation velocity, and when decayed jet velocity balanced with flame propagation velocity at a position apart from the issuing port of jet. When the lift-off of diesel spray flame was controlled with the balance of velocities of spray and upward flame propagation, it might be similar with lifted flame of premixed gaseous jet. However, diesel mixture set-off (set-up/preparation of combustible mixture and start of visible flame) that was not included in the concept of lifted flame, was the main controlling factor of no-flame length of diesel spray. Then the physics of diesel spray set-off (no-flame phenomena in lift-off region) was far different with that of the lift-off of gaseous jet flame.

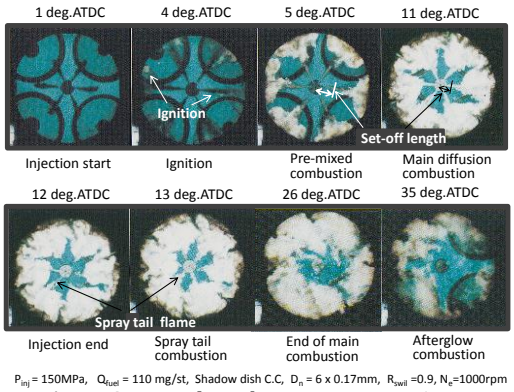


Figure 27 Diesel combustion at P_{inj}=150MPa, reconstructed from ref.[39]

Set-off (preparation of combustible mixture in hot environment) of diesel spray flame was controlled with liquid fuel evaporation, air entrainment, and ignition lag of fuel air mixture. Further physical meaning of set-off length was far different from breakup length. It was longer than the breakup length mentioned above because of no flame penetrating into un-breakup and dense un-evaporation portions of liquid jet. At the end of spray injection, penetration velocity of spray tail was slow and spray tail might stagnate near the injector. Then spray tail flame appeared near the vicinity of nozzle at 12 and 13 deg.ATDC (Fig.27). At that timing, nominal set-off length became very short.

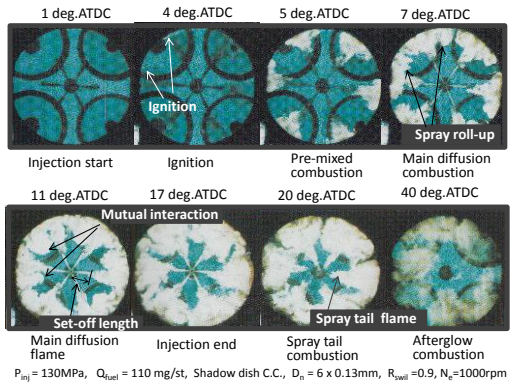
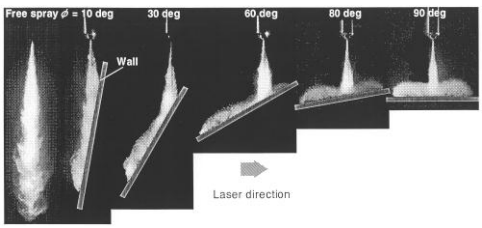


Figure 28 Diesel combustion at P_{inj}=130MPa, reconstructed from ref.[39]

In other example shown in Fig.28, first flame kernel of ignition appeared in an impingement spray. Further, roll-up motion of flame from chamber wall and mutual interaction of impingement sprays were clearly observed. Owing low injection pressure, spray tail flames remained in the space more longer period than those in Fig.27. Diesel spray combustion models shown in Figs.3 and 4 did not involve these phenomena so that practical combustion scheme might be far complicated than those models. However, detail discussion on the diesel combustion was the out of scope of this review.

4) Interaction with wall

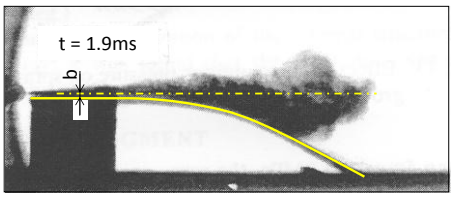
One of the unique characteristics of diesel spray was interaction with wall that was inevitable phenomena of diesel spray combustion in confined space. Figure 29 shows diesel spray impingement to a wall set normally or inclined. Development of diesel spray impinging to a slight inclined wall was not so different from free diesel spray because of less momentum loss on the wall. However, diesel spray impinging normally to a wall lost much of momentum and spread to radial direction.



P_{inj}=25MPa, D_n=0.25mm, M_f=13.7mg, τ_{inj}=1.8ms, P_a=1.5MPa

Figure 29 Wall impingement diesel spray [43]

Diesel spray impinging to a side surface of piston cavity or cylinder was a typical case of normal impingement. It was popular impingent phenomena shown Figs.27 and 28, and its model was established as shown in Fig.16. On the contrary, a slight inclined wall impingement occurred on the bottom surface of piston cavity. When a diesel spray developed along a surface, deviated development occurred sometimes owing to a Coanda effect of flow. Figure 30 is a typically evidence of it observed in a spray development on a curved surface.



P_{inj}=25MPa, D_n=0.25mm, M_f=13.7mg, τ_{inj}=1.8ms, P_a=3MPa, b=1.5mm

Figure 30 Coanda effect on diesel spray development [44]

As shown in Fig.16, fuel film might be formed by an impingement of diesel spray on a wall. Ko et al. [45][46] performed a direct trial of fuel film observation on the wall. They used 40mm width glass plate where a portion of impingement diesel spray spread over beyond the side periphery of a glass plate. Bottom view shown in Fig.31 indicated clearly the fuel film motion on it. At t =1.10ms and t=1.66ms, a trace of impinging spray or thick fuel film was clearly observed at around the impingent centre and its diameter was corresponded with a diameter of diesel spray jet before impingement. Around this thick trace, thin flared trace existed. At t=2.35ms, this flared trace was observed not only on the glass plate but also outside position of it. There was no difference between flared traces on the glass plate and outside. Since the movement of adhered fuel film was restricted by high viscosity of liquid fuel, it was doubtful that observed trace was the adhered fuel film on the glass plate. There was a possibility that fuel film or dense spray was slipping on the glass plate. These phenomena were important for evaporation phenomena of impingement fuel that had great adverse contribution on HC emission from diesel combustion.

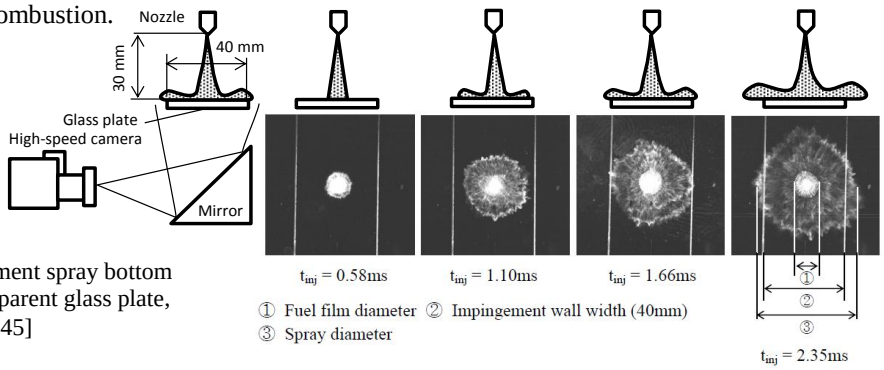


Figure 31 Impingement spray bottom views through transparent glass plate, modified from ref. [45]

In another report by Ko et al. [47], they conducted the fuel mass adhered of the wall. Figure 32 is their result indicating that around half of the injected mass was adhered on the wall. Owing to the adhering of fuel, fuel mass of post-impingement spray reduced comparing to that of pre-impingement spray. Spray volume after impingement increased owing to the entrained air increase that occurred from turbulence promotion by impingement. Increase of air entrainment is generally a reason of lean mixture feature of post-impingement spray. However, adhering fuel loss is another reason of lean mixture formation, but it has not been clarified quantitatively. Both of Figs.31 and 32 suggested that wall impingement and evaporation model shown in Fig.16 was insufficient to explain the whole phenomena of wall impingement.

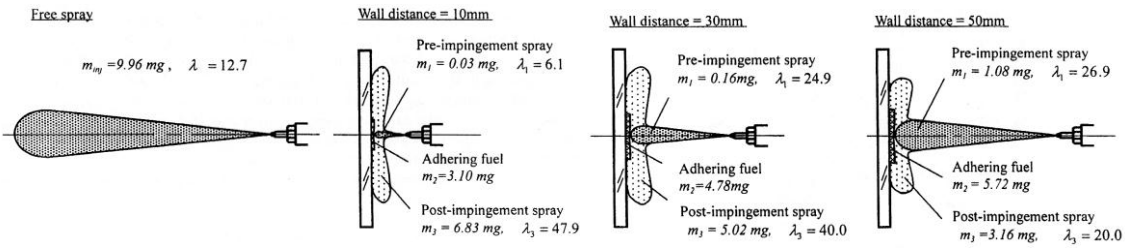


Figure 32 Distribution of injection fuel mass and air-fuel ratio on the free and the impingement sprays, at t_{meas}=1.8ms, (Injection condition; P_a=1.5MPa, t_{inj}=1.8ms, m_{inj}=9.96mg) [47]

5) Mutual interaction of sprays

Roll-up motion of normal impingement diesel spray occurred and mutual interaction of impingement spray seemed sometimes to make a main turbulent diffusion flame in the combustion chamber space as shown in Fig.28. Figure 33 shows the various cases of spray impingement and mutual interactions between two impingement diesel sprays. In case (A), mutual interaction of impingement diesel sprays that were spreading along the wall surface formed new roll-up motion of diesel spray. In case (B), carpet roll-up motion [48] occurred on a tip of radially spreading diesel spray and its mutual interaction formed new spray configuration in the space. More complicated impingement interaction existed in case (C).

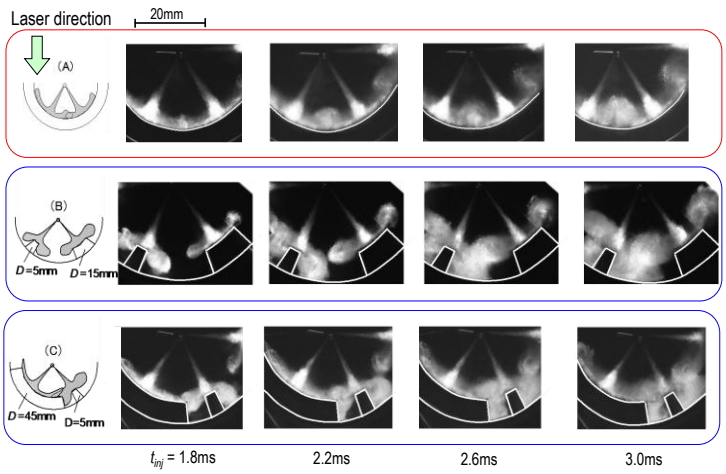


Figure 33 Roll-up motion of impingement spray and mutual interaction of sprays

Behaviour of a mutual interaction sprays had a possibility of new method for controlling spatial distribution of spray. Arai et al. [49] used two EFI nozzles to investigate the mutual interaction of sprays. They suggested that droplet-to-droplet collision in gasoline sprays rarely occurred. Chiba et al. [50][51][52] showed that mutual interaction of spray-to-spray collision in diesel sprays resulted in strong effect on the spray behaviour after collision. According the results shown in Fig.34, spray behaviour after mutual interaction between fully developed sprays was similar to that of gas jets interaction. On the contrary, when an undeveloped diesel spray (in breakup process) and fully developed diesel spray interacted together, spray behaviour after collision was much different from original spray behaviour. As shown in Figs.34 and 35, spray behaviour after collision was controlled by collision angle, relative collision position of each spray, also spray density and momentum. Recently, droplet-to-droplet collision models and numerical simulation based on the models were proposed, and outline of the mutual interaction mechanism was clarified. [53][54] However there was a few feasibility research works for application of mutual interaction sprays.

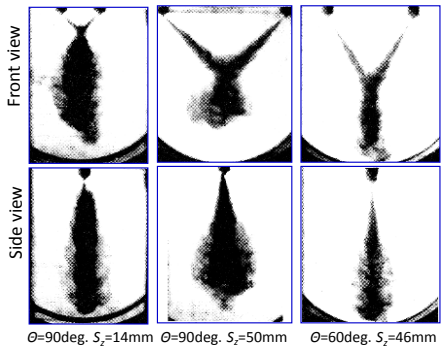


Figure 34 Behaviour of impingement sprays [50]

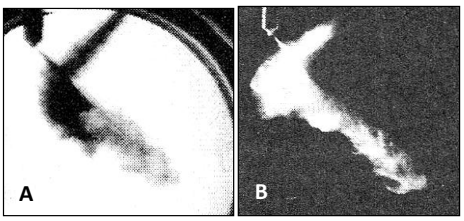


Figure 35 Jet to spray impingement (t_{ing}=0.6sec, 15mm x 45mm), A: shadowgraph, B: Tomographic view with laser sheet [50]

6) Ultra-high pressure injection

To meet the demand of clean diesel combustion, injection pressure of common rail injection system tended to increase. Shock wave generation around diesel spray was reported [55][56], but not so much attention was paid on this fact. When a diesel spray is injected by injection pressure of 300MPa into a combustion chamber, injection velocity may increase up to 750m/s and it might be far faster than the sound velocity. Hiroyasu [57] and Nishida et al.[58] reported an early study on ultra-high injection pressure diesel spray. Using Schlieren optical observation system, they reported the Mach wave around diesel spray as shown in Fig.36. It means that there might be other breakup and spray tip penetration processes, even though many research reports supported the empirical equation of Eq.11 for diesel spray of which injection pressure was beyond 200MPa.

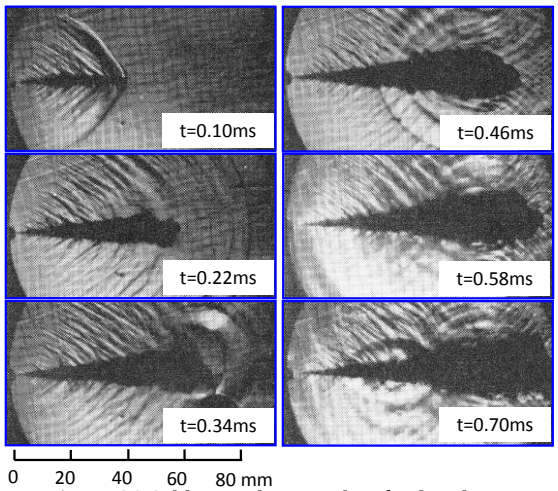


Figure 36 Schlieren photographs of a diesel spray and Mach wave around the spray, P_{inj}=275MPa, D_n=0.25mm, P_a=1.2MPa, T_a=298K [58]

Air Entrainment and Spray Angle

1) Spray boundary and air entrainment

There were a huge amount of discussions about spray tip penetration, spray angle, spray volume, and spray configuration, those were relating to the definition of spray boundary. However, there was no direct discussion about the spray boundary itself. As for a liquid jet, the boundary of jet was clearly defined as a liquid-gas interface or simply as a liquid surface. On the contrary, as shown in Fig.37, we needed a more deep consideration about the spray itself.

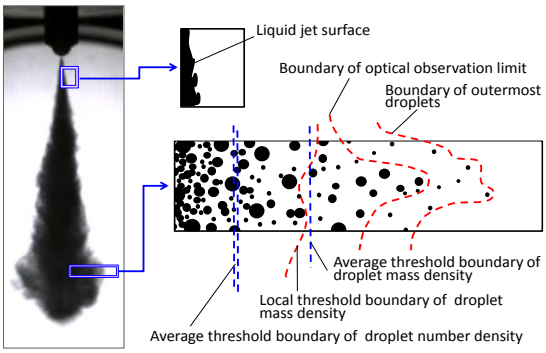


Figure 37 Boundary of spray jet

The physical meaning of spray is two phase substance consisted of liquid droplets and gaseous medium. When the photographic observation of diesel spray was performed to evaluate spray behaviour, we usually considered only droplets or droplet clusters in a spray. For example, diesel spray shown in Fig.37 should be considered as a photograph of fuel droplets clusters in a diesel spray because of no information of gaseous substance in the spray was involved in the photograph. Even though, there are many ways to determine the boundary of droplet clusters. Some of them are listed below.

(1) Boundary of outermost droplets
(2) Boundary of optical observation limit
(3) Average threshold boundary of droplet mass density
(4) Local threshold boundary of droplet mass density
(5) Average threshold boundary of droplet number density
(6) Other definitions relating to droplets cluster

When air fuel mixture was considered as the state of diesel spray, velocity of the mixture could be used for definition of diesel spray boundary. Some of the combustion characteristics of fuel air mixture were other candidates for boundary definition. Figure 38 is these illustrations and list of boundary definition is as follows.

(7) Velocity boundary such as 90% of centre velocity
(8) Stoichiometric mixture boundary
(9) Flammable limit boundary
(10) Other definitions relating to combustion characteristics

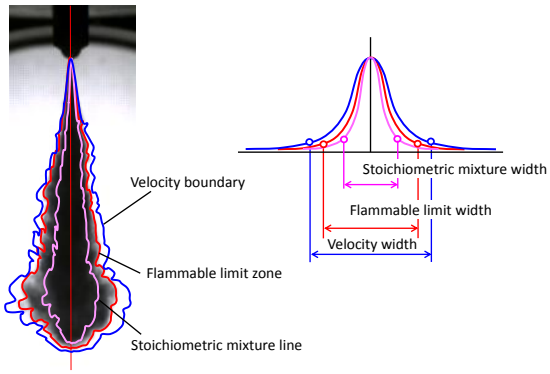


Figure 38 Boundary of fuel spray mixture

Spray surface and volume depended on the definition of spray boundary. Then as shown in Fig.39, various spray volumes such as V_1, V_2, · · · could be obtained with a definition of spray surface S_1, S_2, · · ·. Entrained air M_a meant the total mass of involved air in a spray. Its mathematical definition was the total air flowing into across a spray surface S_i and was expressed by Eq.18. Where U_a is air velocity vector and n_i is normal unit vector to the spray surface S_i.

$$M_a = \int_t \int_{S_i} \rho_a \left(\vec{U_a} \cdot \vec{n_i}\right) dS_i dt \qquad (18)$$

When a volume of spray was known, it was simply evaluated by the volume of spray.

$$M_a = \rho_a \left(V_i - V_f\right) \qquad (19)$$

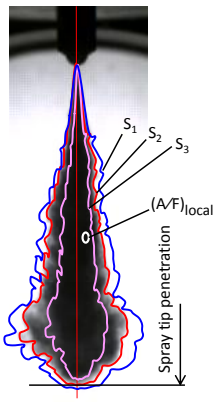


Figure 39 Spray volume

where ρ_a is density of air, V_i (=V_1, V_2, · · ·) is the volume of spray and V_f is liquid fuel volume in a spray. Local air fuel ratio A/F_{local} is defined by Eq.20. When local air fuel ratio was obtained directly, entrained air could be obtained with volume integration using Eq.21.

$$(A/F)_{local} = \frac{\rho_a \left(V_i - V_f\right)_{local}}{\rho_f \left(V_f\right)_{local}} \qquad (20)$$

$$M_a = \int_{V_i} (A/F)_{local} \cdot \rho_f \left(V_f\right)_{local} \mathrm{d}V_i \qquad (21)$$

Even though, spray volume directly relating the spray boundary was the dominant factor for evaluation of the air entrainment.

2) Spray angle

Definition of spray angle was also unclear item for the field of diesel spray research. Figure 40 is the two kinds of definition for diesel spray angle proposed by Zama et al. [59] One of the definitions was the maximum angle θ that was defined by the maximum internal angle between spray peripheries. On the other hand, in Case 2, spray angle ϕ was derived from spray width W at fixed axial location and Eq.22.

$$\phi = \tan^{-1}\left(\frac{W}{z}\right) \qquad (22)$$

Spray boundary mentioned before had great impact to define the spray angle. Here, optical observation limit was used to define the spray boundary. According to the general characteristics of injection rate change, three phases of injection rate after start of injection were observed. The first phase was a period of injection rate increase, and it was called Phase A. Phase B was a period of constant injection rate, and Phase C was a last period of injection rate decrease.

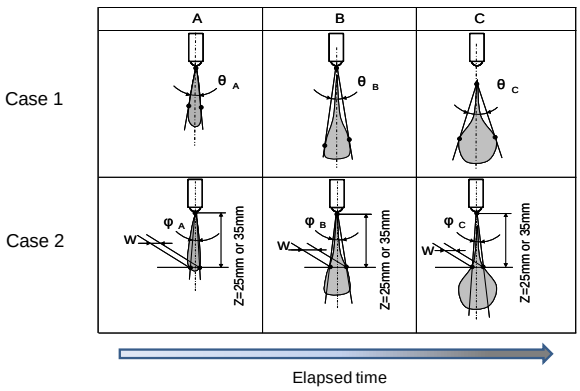


Figure 40 Definition of spray angle [59]

Figure 41 shows time history of spray angle at P_a = 1.0MPa, t_{inj} = 3.2ms and z = 25mm. θ is a spray angle measured with the definition of Case 1. W is a spray width measured at z = 25mm. ϕ is a spray angle of Case 2 and is calculated by Eq.22. Both of spray angles θ and ϕ changed with elapsed time from injection start. Owing the fluctuation of spray angles, it was difficult to determine a unique spray angle from the angle θ, but it could be defined using mean value of angle ϕ in Phase B.

Average spray angle θ for each ambient pressure condition was evaluated using a mean spray angle ϕ of Phase B. Figure 42 shows relationship between ambient pressure and average spray angle. Here, mean spray angles θ (ϕ in Fig.41) at 25mm and 35mm were plotted. Spray angle in phase B at 35mm seems to have no difference from that at 25mm. It means that mean spray angle did not depend on the axial measurement position. Therefore, mean spray angle θ (ϕ in Fig.41) was suitable for determination of an average spray angle. Ambient pressure range in Fig. 42 involved high-pressure condition similar to an ultra-high boost engine. From the results, it clearly confirmed that spray angle increased with increasing ambient pressure.

Wakuri et al. [34] derived an equation of spray angle from the theory of momentum conservation. They reported that spray angle could be described by equations shown in Eq.23 or Eq.24;

$$\theta = F\left(\frac{\rho_f}{\rho_a}, \frac{V_0\, d_0\, \rho_f}{\mu_a}\right) \tag{23}$$

$$\theta = \tan^{-1}\left\{\sqrt{c\cdot\rho_f / \rho_a} \Big/ \left(\frac{K}{\sqrt{V_0 d_0}}\right)^2\right\} \tag{24}$$

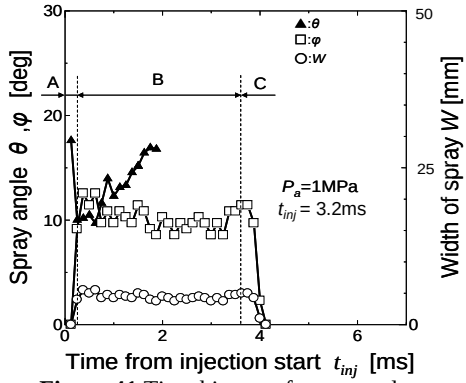


Figure 41 Time history of spray angle (P_{inj}=150MPa, P_a = 1.0MPa, t_{inj} = 3.2ms, z=25mm) [59]

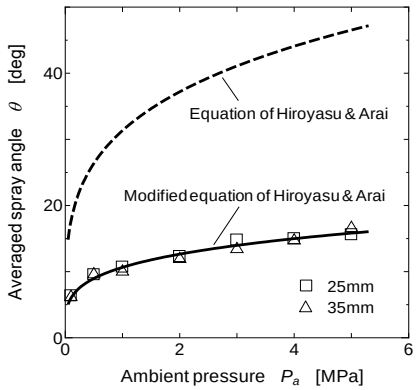


Figure 42 Averaged spray angle vs. ambient pressure (P_{inj}=150MPa, t_{inj} = 3.2ms) [59]

where, θ is spray angle of steady state diesel spray. d_0 is diameter of nozzle hole. ρ_f is density of fuel. ρ_a is ambient gas density. μ_a is coefficient of ambient gas viscosity. V_0 is velocity of fuel at outlet of nozzle. c is coefficient of contraction. K is constant related to spray penetration. For approximation of spray angle, it needs suitable value of a parameter K in Eq.24, and K should be found in experiment. Based on the function introduced by Wakuri et al. [34], Hiroyasu and Arai [60] derived an empirical formula shown in Eq.25.

$$\theta = 0.05\left(\frac{d_0^{\,2}\cdot\rho_a\cdot\Delta P}{\mu_a^{\,2}}\right)^{0.25} \tag{25}$$

Here, ΔP means P_{inj} - P_a and ρ_a is ambient gas density. In their empirical equation, spray angle was proportional to 0.25 power of ambient gas density. The empirical equation of Eq.25 was used for estimation of the spray angle. As shown in Fig. 42, it did not fit to the present experimental plots.

According to the literature of Hiroyasu and Arai [60], they used a jerk-pump-injection system. In their traditional injection system, steady injection period was shorter than that of a recent common rail injection system. And there was a possibility that maximum spray angle corresponding to the Case 1 spray angle in Phase A or in Phase C was used for derivation of the empirical equation. Sometimes, injection rate of a jerk-pump-injection was unsteady and the maximum injection rate resulted at the final stage of injection. When the push away motion shown in Fig.25 appeared in the end of jerk-pump-injection diesel spray, spray angle at this stage increased unexpectedly. In their data analysis, an opening pressure of needle valve was used as a representative injection pressure and real injection pressure was probably larger than this pressure. Moreover, Arai et al. [61] did not assess their empirical equation over 15MPa of injection pressure. Then applicable range of Eq.25 did not cover a high injection pressure condition such as 150MPa in the present injection system. Thus, coefficient of Eq.25 was changed from 0.05 to 0.017, and following equation was obtained.

$$\theta = 0.017\left(\frac{d_0^{\,2}\cdot\rho_a\cdot\Delta P}{\mu_a^{\,2}}\right)^{0.25} \tag{26}$$

Estimation of spray angle using Eq.26 fitted well to the experimental plots as shown in Fig.42. Therefore, it was found that the empirical equation proposed by Hiroyasu and Arai could be adapted to spray angle under high ambient gas density condition even though small modification was required.

Diesel Spray in High Density Surroundings

Spray density could be evaluated with various optical methods. One of them was the light extinction method.[62] Manaka et al. [63] used this method and estimated air excess ratio in a diesel spray. They evaluated the air excess ratio of diesel spray injected into high-density surroundings that corresponded with an in-cylinder pressure of ultra-high boost engine. Figure 43 is the result where leaner and more homogeneous mixture was obtained with an increase of surrounding pressure. In other words, promotion of internal mixing occurred with an increase of the surrounding pressure. They did not explain the reason why it occurred. They only suspected that shear force increase with pressure rise was the main reason of it.

Recently, Zama et al. [64] reported the diesel spray behaviour in ultra-high pressure surroundings. They investigated diesel sprays under various ambient pressure conditions by using shadow imaging method. Figure 44 shows photo-density distributions of shadow images at ambient pressures of 0.5MPa (5.8kg/m^3) and 4.0MPa (46.5kg/m^3). Spray tip and remarkable local patterns on the side-edge of the spray were indicated with circles, and those patterns were sequentially tracked. According to the indicated patterns in Fig. 44(a), movements of spray tip and spray side-edges were almost same. It seems that they moved with similar axial velocities. On the other hand, in the case of Fig. 44(b), spray tip penetration increased with elapsed time, but spray side-edges hardly moved. According to the results obtained in various gas density conditions, it was clear that local movement of side-edge spray showed far different from that of spray tip under high gas density condition. Local spray near the side-edge became stagnated with an increase of the density of ambient gas. These photographs suggested that inner structure of diesel spray might change depending on the gaseous density of surroundings. Then precise measurement and much more attention should be needed on the diesel spray in an ultra-high boost diesel engine.

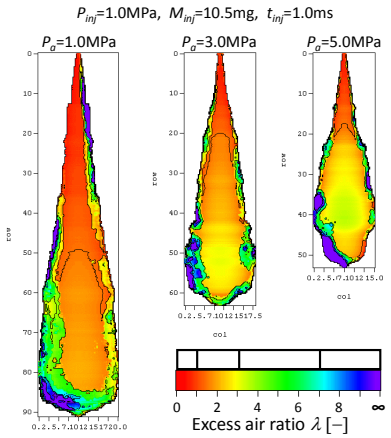
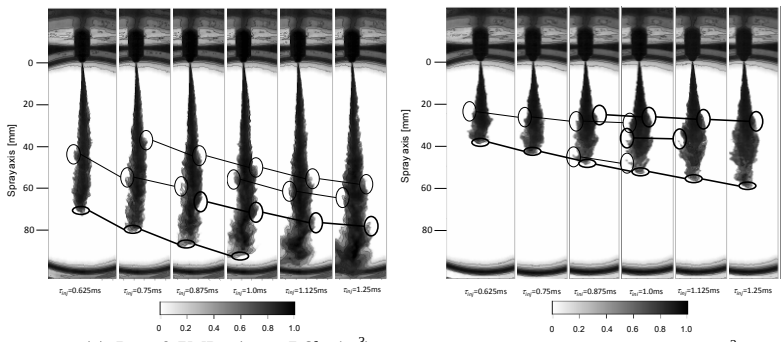


Figure 43 Effect of surrounding pressure on local air excess ratio in a diesel spray [63]



(a) P_a = 0.5MPa (ρ_a = 5.8kg/m^3), P_{inj} = 150MPa, M_{inj} = 31.5mg

(b) P_a = 4.0MPa (ρ_a = 46.5kg/m^3), P_{inj} = 150MPa, M_{inj} = 31.5mg

Figure 44 Effect of surrounding pressure on photo-density distribution of diesel spray [64]

Velocity Distribution inside a Diesel Spray

1) PIV analysis

In order to investigate diesel spray behaviour in terms of a velocity map in a spray, time-resolved PIV was applied for a measurement of velocity distribution inside a spray[65][66][67]. Table 2 is their setting condition of a digital high-speed video camera. For PIV analysis, correlative tomographic images with narrow time interval were required. Thus optimum frame rate of the digital high-speed video camera was selected by considering spray velocity that depended on injection pressure and ambient gas density. Since PIV analysis needs an ultra-high frame rating over 100,000 fps, image size and spatial resolution were restricted by the performance of the digital high-speed video camera. Thus both of image size and view area were changed with ambient gas density condition. Here, spatial resolution was 0.156mm/pixel, and view area was around 5mm×20mm at 0.5MPa, 10mm×20mm at 1.0MPa, and 20mm×40mm at 2.0 and 4.0MPa.

Table 2 Camera condition for PIV [65]

	0.5MPa	1.0MPa	2.0MPa	4.0MPa
Frame rate (f.p.s)	291,666	170,731	130,232	65,116
Exposuer time (μs)	0.5	1	3	3
Image size (pixel)	32×128	64×128	128×256	128×256

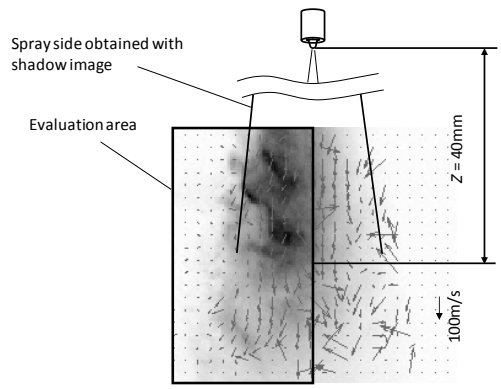


Figure 45 Evaluation area for PIV [65]

Figure 45 shows evaluation area for PIV analysis and an example of velocity vectors. According to the insufficient spatial resolution (0.156mm/pixel) of video image, each droplet in a spray could not be resolved on the image, and cluster of spray droplets was captured as photo-density distribution in a spray image. Velocity dis-

tribution of droplets cluster in a spray could be obtained from PIV analysis based on direct cross-correlation method. In PIV analysis, interrogation spot size was 12pixels×12pixels which corresponded to 1.87mm×1.87mm of real image.

2) Velocity distribution

Figure 46 shows the axial velocity distribution in diesel spray at 40mm apart from injector. Its distribution fitted well with the Gaussian distribution shown in Eq. 27.

$$U(r_s) = U_c \exp\left(-\frac{(r_s)^2}{2\sigma^2}\right) \qquad (27)$$

Where, U_c is the axial velocity of spray centre and σ is the standard deviation of distribution.

Figure 47 shows normalized axial velocity distributions where radial axis was also normalized using spray half width W_{shadow} to clear up the relative position of spray side-edge. Here, W_{shadow} corresponded to the boundary of optical observation limit mentioned previously. In the case of 0.1MPa, due to extremely high-dense and high-speed spray, no PIV analysis could be performed. Then, only velocity distribution estimated by Eq.27 is shown in the figure.

Normalized velocity at spray side-edge (r_s/W_{shadow} = 1.0) of P_a = 0.5MPa (ρ_a = 5.8kg/m^3) was 45% of spray centre velocity, and that of P_a = 4.0MPa (ρ_a = 46.5kg/m^3) was 30% of centre velocity. It indicates that relative velocity difference between centre and side-edge of spray increased with increasing the ambient pressure. It seems that velocity around side-edge of a spray decreased due to the momentum exchange between a spray and high dense ambient gas. This fact was well corresponding with stagnant behaviour of spray side-edges observed in Fig.44. Further it might suggest that turbulent diffusion mechanisms of momentum and droplet clusters were different.

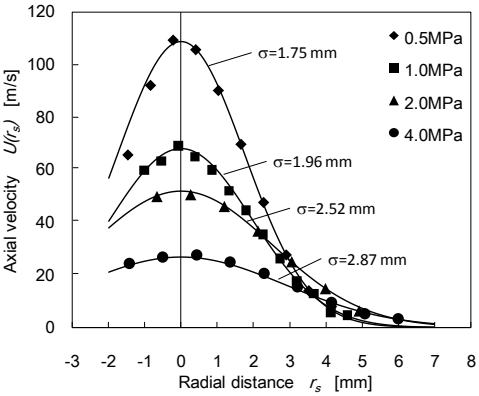


Figure 46 Axial velocity distribution fitted with an axisymmetric Gaussian function, P_{inj} = 150MPa, M_{inj} = 31.5mg [59]

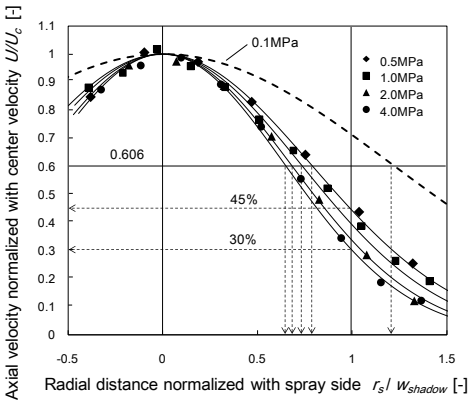


Figure 47 Axial velocity distribution normalized with centre velocity and spray width [65]

3) Mixing zone

Radial position, where velocity gradient is the highest, can be obtained from the second order differential coefficient of Eq.27 being set zero. According to Eqs.28 and 29, maximum normalized gradient of the velocity is derived when r_s is set σ, and $U(r_s)/U_c$ = 0.606 is obtained.

$$r_s = \sigma \quad \text{at} \quad \frac{d^2}{dr_s^2}\left(\frac{U(r_s)}{U_c}\right) = 0 \qquad (28)$$

$$\left.\frac{U(r_s)}{U_c}\right|_{r_s=\sigma} = 0.606 \qquad (29)$$

Considering the physical meaning of maximum velocity gradient, its radial position seems to correspond with highest shear stress location. Further, intense mixing zone might appear around this location. Then, $U(r_s)/U_c$ = 0.606 is indicated in Fig.47 as a characteristic value of the velocity distribution.

When we considered a relative radial position normalized by r_s/W_{shadow}, the radial position corresponding to $U(r_s)/U_c$ = 0.606 shifted from outside to inside of a spray with increasing the ambient gas density. It means that intensive zone of mixing shifted from outside to inside with an increase of the ambient pressure. As for predicted velocity distribution at 0.1MPa, intensive mixing zone placed at far outside of spray side-edge (r_s/W_{shadow} > 1.0). It means that the spray at 0.1MPa had weak mixing performance with ambient gas.

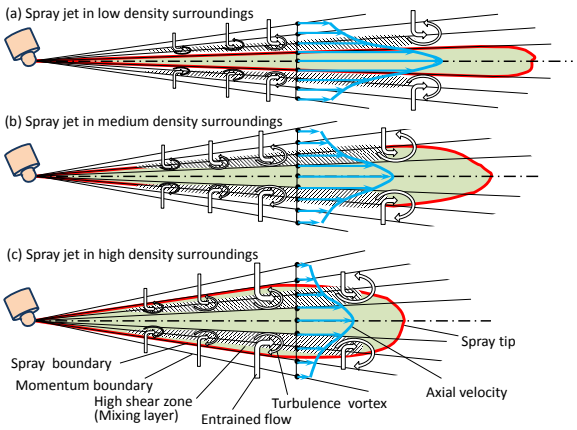


Figure 48 Diesel spray and mixing zone model [65]

From the results indicated by Fig.47, new spray concept with intense mixing zone is introduced as shown in Fig.48. Mixing zone in low ambient pressure condition locates at outside of a spray. Since the intense mixing only occurs in the surroundings, it means no promotion of mixing between spray jet and surroundings. As for

the mixture formation between fuel and air, a spray in medium gas density field might be better because mixing zone is located around side-edge of the spray. Under high ambient pressure condition, mixing zone is located inside of a spray. It might promote inside homogenization of the spray. However, it has a limited promotion effect on an air entrainment into the spray.

As shown in Fig.43, internal mixing became increase with an increase of ambient pressure. It was explained with that the mixing zone tended to shift from outside to inside of diesel spray with increasing the ambient pressure. Not only by high shear force with high-density surroundings but also by the shifting of mixing zone to inside of spray resulted in homogeneous mixture of spray injected into high ambient pressure surroundings.

Summaries

Owing to the high-speed transient phenomena of diesel spray in high temperature and pressure condition, it is far difficult to observe the real diesel spray in a diesel engine. However, diesel spray behaviour partially clarified with its fundamental studies, gave the lot of suggestions for not only improvement of diesel engine but also scientific advancements such as liquid atomization and two-phase spray flow.

Detailed information on cavitation and breakup phenomena contributed greatly to designing and manufacturing of injector nozzle. Optimum design of combustion chamber is always following with the spray characteristics such as spray penetration and spray angle. Air entrainment related to spray volume was a key process of diesel combustion and its emission. Further, cavitation and other breakup mechanisms made new insights on two-phase hydrodynamics.

There are still too many unclear phenomena to describe breakup and spray development of liquid jet injected from diesel injector. Surface deformation related with cavitation and internal turbulence in a jet is a physically unclear problem concerning to the liquid jet disintegration. Observation of liquid jet core near the exit of injector is hard because of high dense droplet clouds surrounding it. As for the internal structure of diesel spray, only a little information is available for understanding the mixture formation and combustion of impingement spray. Further, there is almost no information for future diesel spray with ultra-high pressure injection system for ultra-high boost diesel engine. Then, it needs more fundamental measurement and deep physical understanding of diesel spray for future advancement of diesel engine and atomization technology.

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