Droplet Velocity Measurements in a Four-Cylinder Optical DI Diesel Engine

C.Arcoumanis, Z.Hu and J.H.Whitelaw

Department of Mechanical Engineering
Imperial College of Science, Technology and Medicine
Exhibition Road, London SW7 2BX
England

ABSTRACT

Laser Doppler velocimetry (LDV) measurements of the fuel spray droplet velocities were obtained in the combustion chamber of an optical direct-injection 1.9L diesel engine in the period between start of injection and auto-ignition at idle and low-load conditions and compared with spray images obtained with an image-intensified CCD camera. In agreement with previous results under atmospheric conditions, the leading droplets in the spray exhibited lower velocities than those of the following faster-moving droplets which decelerated rapidly after the end of injection. In addition, the LDV results indicated that the mean droplet velocities were of the order of 25m/s, close to the wall, which is in good agreement with the spray tip penetration velocities deduced from visualisation images. Comparison with previous results obtained with the same injection system but under atmospheric conditions, revealed that the maximum droplet velocities in the engine are 7 times lower than the corresponding velocities in the atmospheric test rig and about 3 time higher than the local swirl velocities in the piston-bowl at the time of injection at an engine speed of 1000rpm. At the same time, near the piston wall, the measured droplet rms velocities were about 10m/s in the main body of the sprays, which is 4.5 times higher than the corresponding rms velocities of the air flow in the motored engine at the crankangle corresponding to the start of injection. These results confirm that in small direct-injection diesel engines, at least under low-load conditions, the spray imposes its own velocity field on the air flow which implies that the pre-injection mean flow and turbulence are of secondary importance in terms of the local mixing during the injection period.

1 INTRODUCTION

Direct-injection (DI) diesel engines offer superior fuel economy to gasoline and indirect injection diesel engines, which has been the driving force for the increasing proportion of small high-speed DI diesel engines in the passenger car market in Europe over the past few years. However, further reduction of NO_x and particulate emissions is required to meet the increasingly more stringent environmental regulations. This represents a serious technical challenge to diesel engines which run overall lean and depend on exhaust gas recirculation, high injection pressure and oxidation catalysts to reduce their NO_x and particulate emissions.

The combustion and related pollutant formation processes in DI diesel engines are controlled by the fuel/air mixing process. Since the available time and chamber space for fuel/air mixing are significantly reduced in small, high-speed DI diesel engines compared to conventional larger and lower speed DI engines, the key to improve the performance of such engines is good atomization of the injected fuel sprays and utilisation of the interaction between the sprays and the in-cylinder flow for effective fuel/air mixing. Therefore, knowledge of the transient characteristics of the spray development is essential for futher refinement and improvement of combustion in DI diesel engines.

The major problem in optimising combustion is the lack of experimental techniques capable of probing the hostile environment of DI diesel engines. To overcome the difficulties associated with the study of diesel sprays under normal operating conditions, detailed investigations have been made in constant-volume combustion chambers at realistic pressures and temperatures, e.g. [1-4] and at atmospheric conditions, e.g. [5,6]. These studies have provided the essential physical insight and data for the formulation and validation of computer spray models, but quantitative differences are expected between diesel engine sprays and those under simulated conditions. Direct measurements of the spray characteristics under engine operating conditions are, therefore, urgently needed in order to quantify these differences and provide additional data to the already available spray tip penetration and spray angle for validating spray models.

This paper presents fuel droplet velocity measurements in the cylinder of a small, high-speed DI diesel engine, using a purpose-built back-scatter laser Doppler velocimeter (LDV) and spray development images using an image-intensified CCD camera. The transient characteristics of the sprays are examined along the spray geometric axes for two engine operating conditions, and the results are compared with earlier measurements obtained under atmospheric conditions [7].

2 EXPERIMENTAL SYSTEM

2.1 Volkswagen DI Diesel Engine

The research engine was a Volkswagen (VW) high-speed, four-cylinder 1.9L direct-injection diesel engine with an optional scroll supercharger (G40) and intercooler, a helical intake port, a re-entrant combustion chamber and a slightly off-centre multi-hole injection nozzle. The geometric and operating conditions are the same as given in [7], while a schematic of the optical engine and LDV system are shown in Fig.1. The transparent version of the standard engine was also manufactured by VW and allowed optical access into the combustion chamber for all four cylinders through mirrors positioned within the extended-pistons at 45° to the cylinder axis and 25mm thick quartz windows in the bowl base. In the present study, the in-cylinder flow and spray characteristics were examined only for the first cylinder through a mirror which was supported independently from the engine test bed in order to isolate the engine vibrations and prevent broadening errors in the velocity measurements. Since both the piston and the cylinder liner were elongated by the same amount, the optical engine retained the high compression ratio of 19.5, which represents only a small reduction from the 20.5 of the production engine due to the flat piston window. The injector was orientated such that the axes of the sprays were uniformly distributed in the piston-bowl; the angle between the spray axis and the cylinder head was nominally 15° resulting in a spray cone angle of 150°.

Two Kistler pressure transducers were installed in the cylinder and in the high-pressure fuel line near the injector to allow measurement of the cylinder and fuel line pressures, respectively, and a Hall needle lift sensor was incorporated in the injector to provide information about the start of injection. Timing information was provided by a shaft-encoder mounted on the camshaft, with a resolution of 0.36°CA.

2.2 Laser Doppler Velocimeter

A back-scatter LDV system was employed to measure the fuel spray droplet and in-cylinder air flow velocities. It comprised an Ar-ion laser operating at 514.5nm with 1.5W maximum power output, a diffraction grating for beam splitting and for providing frequency shifts of up to 15MHz, a purpose-built photomultiplier and a frequency counter (TSI 1990C) for processing the Doppler signals.

The main difficulty for back-scatter LDV measurements close to walls is to distinguish the light scattered by seeding particles within the control volume from the wall reflections of the two laser beams as discussed in [8]. In the present application, the control volume had to be placed around 4mm from the cylinder head in order to intercept the fuel droplets. A 300mm front lens was used to give a larger beam angle but further reduction of its focal length was not feasible because of the geometric restrictions in the engine. The key component in the present LDV system was a purpose-built 5X magnification telescopic lens set which provided the essential spatial separation between the images of the control volume and the beam reflections at the pinhole plane of the photomultiplier. A 150μm pinhole was used to stop the wall reflections reaching the photocathode as well as to reduce the effective measurement volume to 495μm (length) x 30μm (diameter). Scattered light from the measurement volume was detected by the photomultiplier tube equipped with a pre-amplifier in order to improve the signal-to-noise ratio. A schematic of the LDV system is given in Fig.1; this arrangement has allowed droplet velocity measurements to be obtained as close as 4mm from the cylinder head through the moving piston window with an uncertainty of ±5%.

2.3 Spray Images

Single shot images of the spray development were obtained from separate engine cycles with an intensified CCD camera (Proxicam HF1) through the piston-window and the mirror in the extended piston while the spray up to the point of auto-ignition was illuminated by a spark-light source. More details about the imaging system and associated software are given in [9]. Here, only a few spray images are presented to allow comparison of the spray tip velocity with the droplet velocities obtained with the back-scatter LDV system.

3 RESULTS AND DISCUSSION

The radial mean and rms velocities of the fuel droplets were measured at four locations A-D along the geometric axis of spray #4, as shown in Fig.2, at 1000rpm/idle (Case A) which corresponds to 5mg per injection (or 1.2mm^3 per hole per injection) and an injection frequency of 8.3Hz. Measurements at locations C and D were also obtained at 2000rpm/2bar BMEP condition (Case B), corresponding to 8mg per injection (or 1.9mm^3 per hole per injection) and an injection frequency of 16.7Hz. Location D is only about 3mm from the spray impingement point on the piston wall, hence, the swirl velocities at this point were also measured in the motored engine in order to quantify the spray/swirl interaction just before impingement on the wall. The injector tip has been used here as the origin of the coordinate system and was also used as the reference point for positioning the LDV measurement volume; it is estimated that the uncertainty in positioning the LDV volume was less than 0.5mm. It should be noted here that none of the five nozzle holes coincides with the origin of the coordinate system, but they have nearly the same radial coordinates of 1mm; therefore, all the measurement points are actually closer radially to their nozzle holes by 1mm than those suggested by their corresponding radial positions. In all cases, about 600-800 velocity samples were ensemble-averaged over a window of 0.72°CA which resulted in a statistical uncertainty in the ensemble-averaged mean and rms velocities of less than 3% and 6%, respectively.

3.1 In-cylinder Air Flow

The in-cylinder flow in this engine has been extensively investigated using laser Doppler velocimetry under motored conditions at an engine speed of 1000rpm [7]. These results confirmed that at the end of the compression stroke (340°-360°CA) the swirling flow present in the piston-bowl exhibits a structure resembling closely solid-body rotation with a swirl ratio of 5.5 at TDC, and a squish flow towards the centre of the piston-bowl with velocity magnitudes of less than 0.5 times the mean piston speed (V_p = 3.18m/s at 1000rpm).

Since all the measurement locations in the previous study [7] were more than 7mm from the cylinder head and below the geometric axes of the sprays, the swirl velocities were measured in the present study in the motored engine at an engine speed of 1000rpm at location D (see Fig.3) to quantify the cross stream flow conditions for spray #4 just before its impingement onto the piston wall. The results are presented in Fig.3 and are normalised by the mean piston speed, V_p = 3.18m/s. The mean swirl velocity at this location increases from about 2.1V_p at 320°CA to 2.8V_p at 350°CA, reflecting that the air flow is undergoing a spinning-up process due to the additional momentum brought into the piston-bowl by the squish flow near TDC and the reduction in the bowl entry diameter. The peak at 350°CA is followed by a decay of the swirl velocity around TDC due to the increasing frictional losses on the combustion chamber wall as a result of the higher surface-to-volume ratio. Due to the swirl-squish interaction near TDC, the corresponding rms of the swirl velocity increases by about 30% from a value of 0.5V_p at 320°CA to 0.7V_p at TDC of compression. As will be discussed later, the magnitude of the swirl velocities is much lower than the spray droplet velocities just before impingement on the piston wall.

3.2 Spray Droplet Velocities

Measurements of the droplet velocities for spray #4 at various locations along its geometric axis are presented in Fig.4, and exhibit similar velocity distributions during injection. The mean radial droplet velocities are around 24m/s in the main body of the spray prior to their sharp decrease in the spray tail. The maximum radial velocity remains nearly the same as the measurement volume moves from r = 7.4 to 14mm, but is about 20% lower than the corresponding droplet velocity measured in spray #1 (not shown here). This is in agreement with the shadowgraphy studies of [7,9] where the spray tip penetration was quantified in the same engine under identical operating conditions (typical images of the sprays are presented in Fig.5), which clearly showed that the sprays are not uniform and that spray #4 has slower tip penetration than spray #1.

Location D (r = 14mm, z = 5.5mm) in the present study is the closest position (about 3mm) from the piston wall. At the time of impingement (~ 359°CA), the droplets at location D have mean radial velocities of about 23m/s with corresponding rms levels of about 10m/s; these levels of droplet velocities obtained with LDV are similar to the spray tip velocities deduced from the spray images shown in Fig.5, and are in general about 3 times higher than those of the cross-stream swirl flow, which were shown in Fig.3 to be 8 and 2.2m/s for mean and rms velocities, respectively. These results provide quantitative evidence of the dominant influence of the spray on the pre-injection flow field. Such high droplet velocities near the impingement point also imply that the spray/wall interaction plays an important role in the mixture formation and distribution in small direct-injection diesel engines, even at low loads. The related physical processes, such as spray splash, secondary atomization and formation of liquid film on the piston wall, need to be further examined in order to quantify their impact on the fuel/air mixing. Due to the difficulty to perform such studies in a production engine, an atmospheric test rig was set-up which allowed quantitative information to be obtained about the flow and heat transfer characteristics of the impinging spray as a function of the wall temperature and injection conditions; more details are given in [10].

The droplet velocities were also measured at locations C (r = 11.2mm, z = 5.5mm) and D (r = 14mm, z = 5.5mm) for Case B. The maximum droplet velocities were also observed immediately after the arrival of the spray tip at the measurement volume, with values of about 26m/s, which are 7% higher than the corresponding velocities in Case A. This is attributed to the line pressure being higher both at the start of injection and during the injection period for Case B.

The droplet velocities of the spray generated by an identical injection system, but under atmospheric conditions for Cases A and B, were also measured along the direction of the spray axis. Although the orientation of the velocity component differs by 15° from the velocity component measured in the engine (the effect is expected to be small since cos15° = 0.97), the velocity profiles shown in Fig.6a&b for the two cases have clearly similar shapes under atmospheric and engine conditions. However, the droplets in the atmospheric spray at a distance r = 10mm have maximum mean velocities of 170m/s for Case A, and

210m/s for Case B which are about 7 times higher than those obtained in the engine. Since maximum droplet velocities occur in general just behind the spray tip, it is reasonable to expect that the spray tip velocities should experience a similar level of difference under atmospheric and engine conditions. Using the correlation for spray tip penetration proposed in [11], a reduction of 7.3 times in the spray tip velocity was estimated in close agreement with the level of decrease observed in the measured maximum mean droplet velocities in the engine relative to those under atmospheric conditions.

4 CONCLUSIONS

Measurements of spray droplet velocities were obtained by a back-scatter LDV system in a transparent DI diesel engine, during the period between start of injection and auto-ignition, and compared with spray images obtained with a CCD camera and previous spray measurements obtained under atmospheric conditions. The results revealed that :

1) The maximum droplet velocities in the engine cylinder were about 7 times lower than the corresponding velocities in the atmospheric spray. This level of reduction is similar to that estimated using available correlations for the spray tip penetration.
2) The mean radial droplet velocities near the piston wall at the time of spray impingement were about 23 and 26m/s for idle and low-load conditions while the rms velocities were about 10m/s in the main body of the sprays. These velocity values when compared with the local swirl velocities provide evidence of the dominant influence of the spray on the air velocity field during the period of injection.

5 ACKNOWLEDGEMENTS

Research financed by the Commission of the European Communities within the frame of the Joule Programme, by the Swedish National Board for Industrial and Technical Development NUTEK, by the Austrian Government, and by the Joint Research Committee of European automobile manufacturers (Fiat, Peugeot SA, Renault, Rover, Volkswagen and Volvo) within the IDEA EFFECT Programme.

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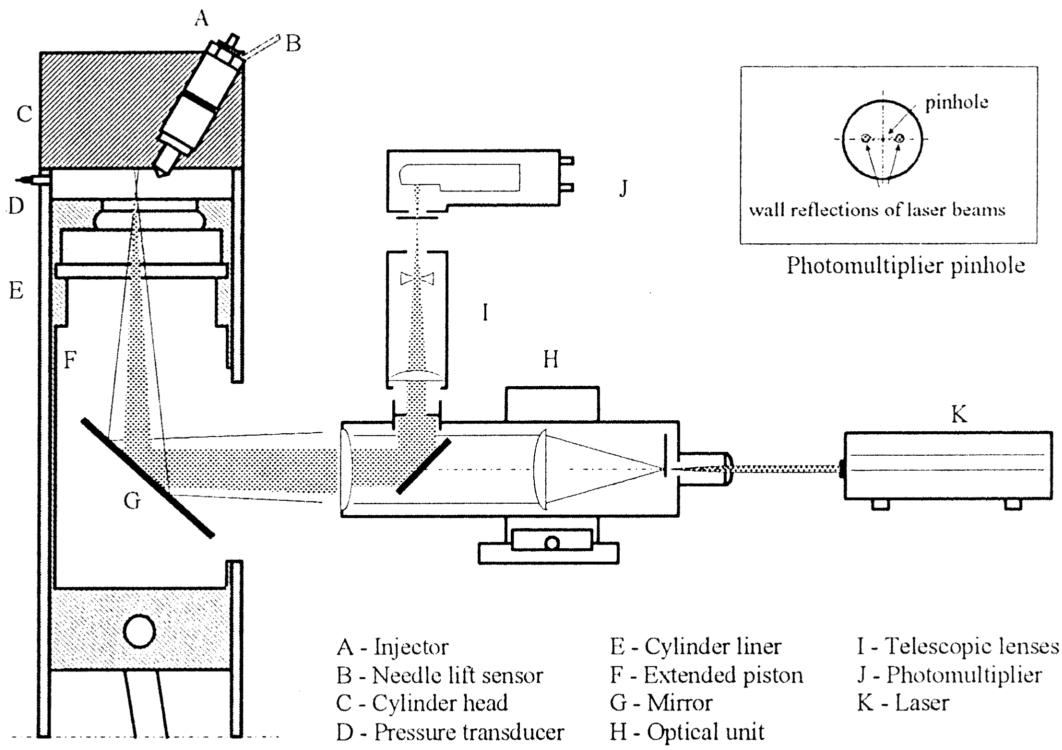
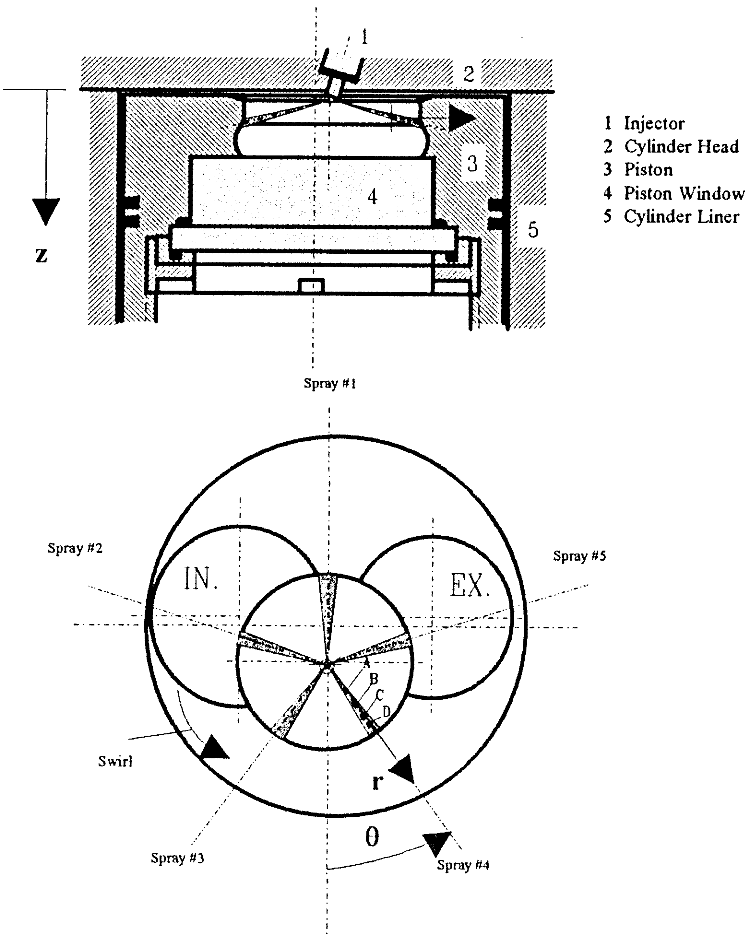


Fig. 1 Schematic of optical engine and back-scatter LDV system.



Measurement Locations	Coordinates (r, θ, z)
A	(7.4, 36, 4.0)
B	(9.2, 36, 4.5)
C	(11.2, 36, 5.0)
D	(14.0, 36, 5.5)

Fig. 2 Schematic of combustion chamber, fuel sprays and measurement positions.

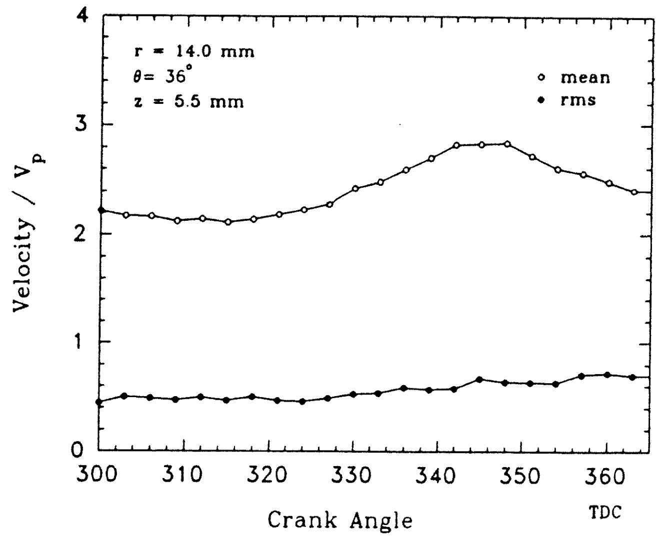


Fig.3 Evolution of swirl velocities of the in-cylinder flow near the spray impingement point (spray #4), V_p = 3.18m/s @ 1000rpm.

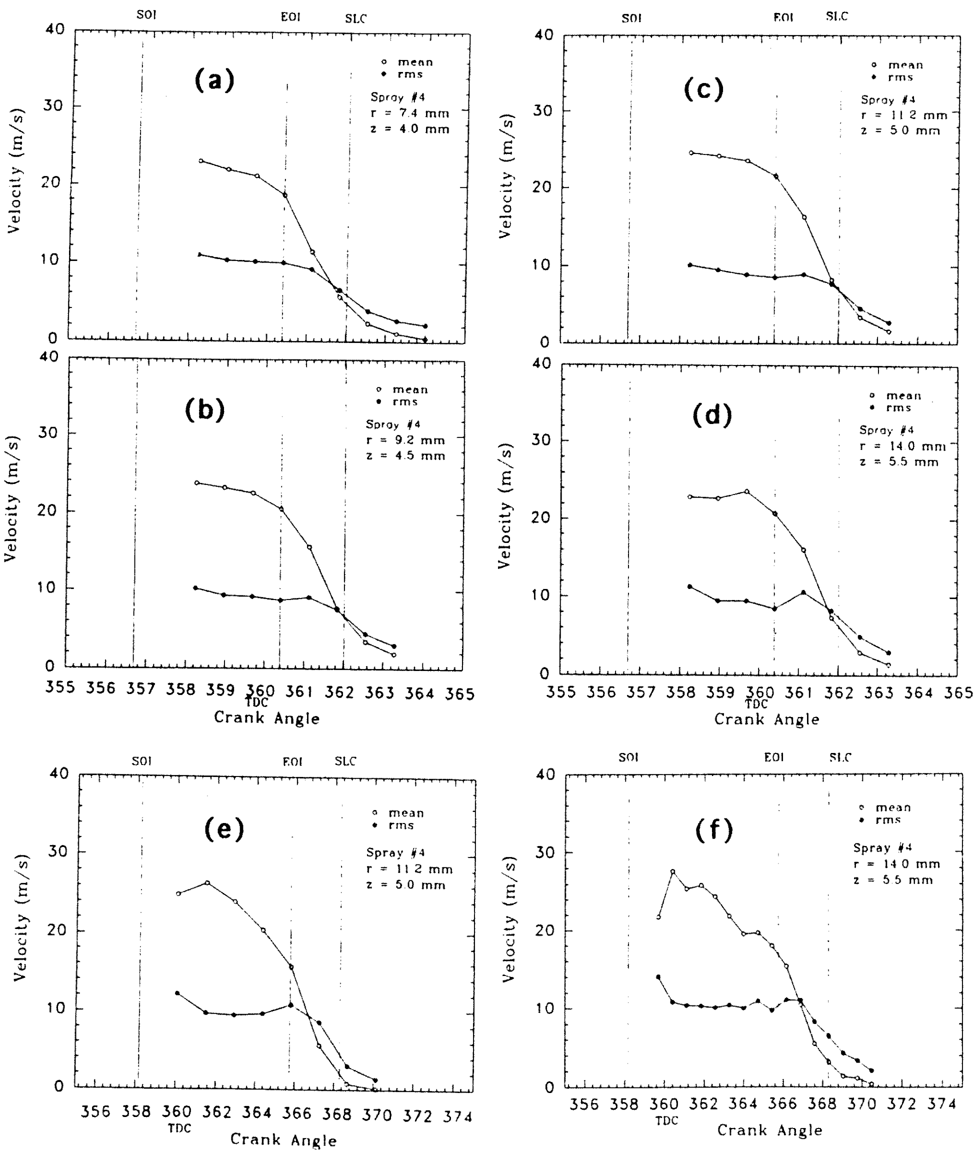


Fig.4 Variation of mean and rms radial droplet velocities with crankangle along the geometric axis of the spray #4
(a)-(d) Case A: 1000rpm/idle, (e)-(f)Case B: 2000rpm/2 bar BMEP

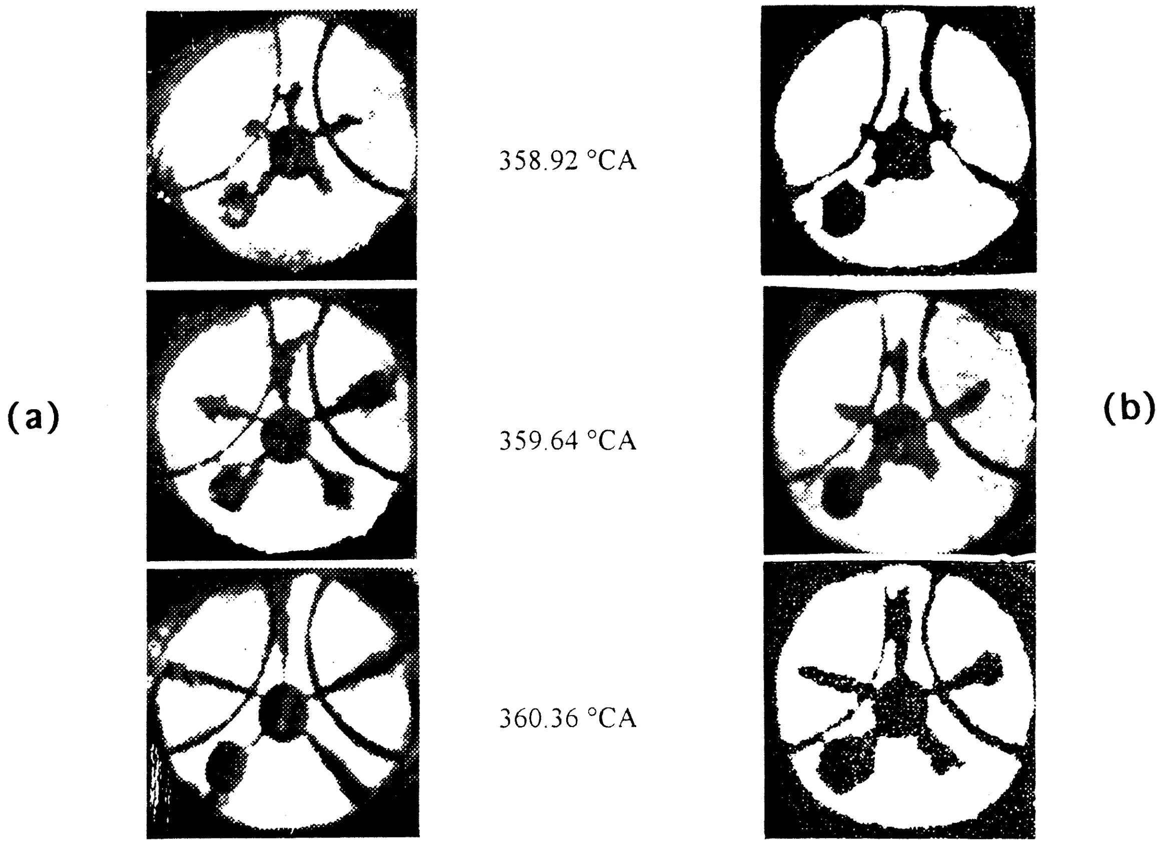


Fig.5 Spray images at various crankangles
a) Case A: 1000rpm/idle, b) Case B: 2000rpm/2 bar BMEP

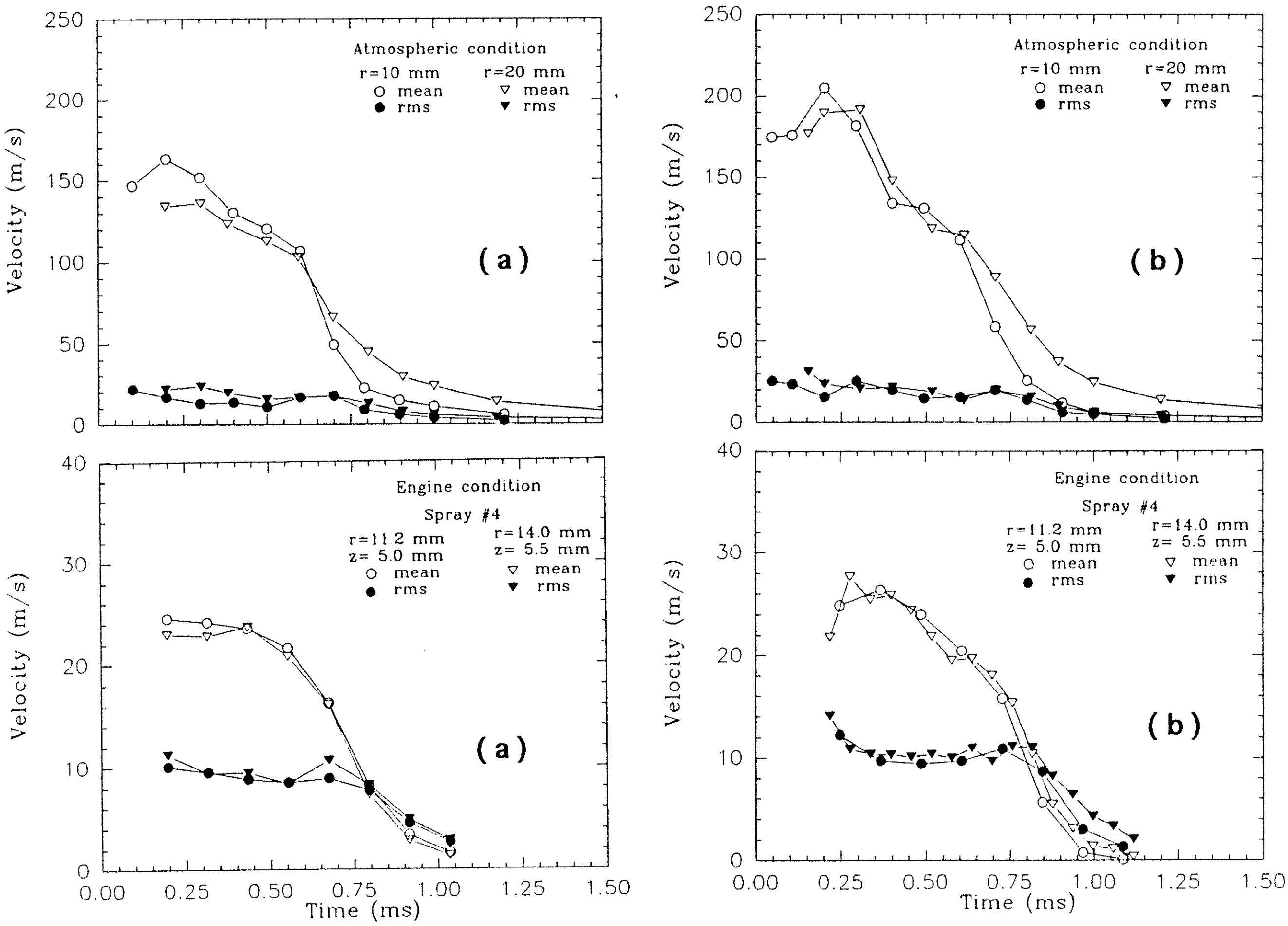


Fig.6 Comparison of the temporal variation of droplet velocities under atmospheric and engine conditions
a) Case A: 1000rpm/idle, b) Case B: 2000rpm/2 bar BMEP