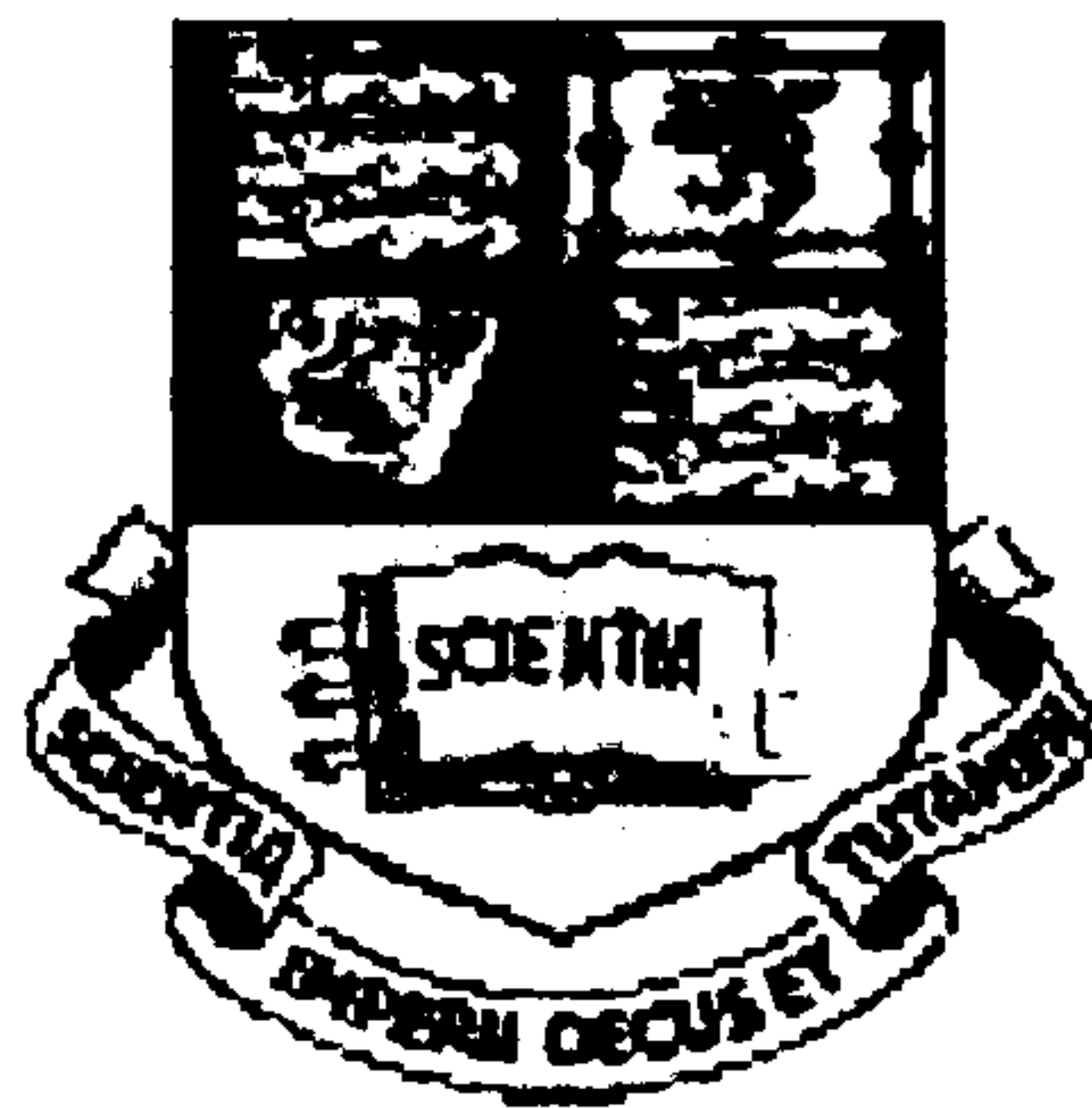


INTERNAL FLOW AND SPRAY CHARACTERISTICS OF A SWIRL PRESSURE ATOMISER FOR GASOLINE ENGINES

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ABSTRACT

This study focuses on the internal nozzle flow and spray generated by swirl pressure atomisers for direct injection gasoline engines and it tackles one of the most challenging problems in fuel injection systems which is the link between the nozzle-hole flow conditions and the spray characteristics.

Inside the nozzle, the liquid film has been visualised and its thickness has been found to remain almost constant during the injection duration with no apparent effect from the injection pressure, although it increases with nozzle length. Further downstream of the nozzle the film diverges until it breaks-down into ligaments and droplets which in turn diverge until the nozzle spray angle is reached.

Two sub-models have been introduced based on the flow observations near the nozzle exit: the first predicts the film and droplet divergence outside the nozzle exit and links three angles; the nozzle flow angle, the film divergent angle and the nozzle spray angle, while the second is based on the linear instability analysis to predict the highly unstable wave propagating on the film surface and to estimate the mean droplet diameter after film disintegration. These two sub-models have been implemented in an existing fuel injection model and are expected to contribute to development of advanced spray models.

Finally, the spray characteristics have been revealed using direct imaging and phase Doppler anemometry (PDA) system under atmospheric conditions and at high pressure/temperature environment inside a constant-volume chamber. While spray visualisation revealed the presence of different stages during the spray development, the PDA system provided the temporal and spatial distribution of the spray mean droplet diameter and velocity. The spray structure generated by this type of injector was found to be very sensitive to the ambient conditions as well as nozzle geometry. Overall, the data provided in this thesis are complete enough to validate relevant computer models that include the flow inside the nozzle and near its exit, as well as the spray characteristics of swirl pressure atomisers.

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CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1.1 INTRODUCTION

With ever growing concern for environmental protection, automobile tailpipe emission regulations continue to be pushed to lower levels in many world markets such as the European Union, Japan, and North America. CARB (California Air Resources Board) and EPA (Environmental Protection Agency) of the USA have continued establishing tighter standards, which are becoming more difficult to meet. Recently many passenger car manufacturers have become interested in the development of direct injection spark ignition (DISI) engines as a promising alternative to conventional gasoline engines and as a means of meeting the present gaseous and particulate emissions legislation and future CO₂ standards. Road transport accounts for 22 % of the total man-made emissions in EU, with 12% accounted just for passenger car traffic, as has been reported by the European Automobile Manufacturers Association (1998). Carbon dioxide, in particular, has been implicated in global warming. While air pollution from road transport in the form of carbon monoxide, hydrocarbons, nitrogen oxides, sulphur oxides and particulates presents a hazard to human health, they also contribute to eutrophication and acidification of soil and water.

The competition among the automobile manufacturers to improve engine performance with minimum emissions has driven intensive research on DISI engine development. This competition has intensified when Mitsubishi Corp. succeeded in putting into mass production the first car with a 1.8-litre DISI engine in August 1996. This car was brought into the Japanese market with claims of about 35% less fuel consumption at low loads and 10 % higher power output at all speeds relative to the conventional port fuel injection (PFI) engine. This low fuel consumption has been achieved by operating the engine over at least two injection regimes: low loads and high

loads injection. During the low load regime, the fuel is injected late during the compression stroke; a very lean overall mixture is produced which is possible through stratification of the charge. This concept is based on ignition of a stoichiometric or rich mixture present in a limited region around the spark plug at the time of ignition, with the flame propagating towards a much leaner mixture or excess air present in the rest of the combustion chamber. This concept allows better fuel economy and, consequently, reduced CO₂ emissions. At higher operating loads, power is controlled by the quantity of fuel injected, therefore the inducted air is not significantly throttled, thus minimizing the pumping work, while the charge temperature decreases as soon as the fuel is injected into the cylinder air and, consequently, the charge density also increases thus allowing more air and fuel to be introduced into the chamber for producing more power. This unthrottled operation is normally used in diesel engines but applying this technique to DISI engines is not straightforward. In diesel engines, autoignition may occur at any location inside the combustion chamber. However in DISI engines the ignition or flame kernel initiation is fixed in the combustion chamber by the spark plug location and therefore, fuel distribution has to be very well controlled in order to achieve smooth flame propagation with stable combustion independent of whether the operation mode is homogeneous or stratified. The fuel injection process is also more critical in DISI engines than port fuel injection due to the shorter times involved and the need for precise fuel distribution for every injection regime.

Accurate control of the injection performance parameters (timing, delivery, instantaneous flow rate, pressure, spray configuration, etc.) offers the most effective means to influence mixture preparation and combustion kinetics in order to achieve both clean burning and high thermodynamic efficiency. Modern fuel-injection equipment is providing design engineers with improved control of timing and delivery. This progress has resulted in tremendous improvement in engine control. Unfortunately, the success of DISI engines is based on the trial and error approach, and direct evaluation of the actual fuel-injection spray in the engine is still difficult. New and more powerful means to study and evaluate the characteristics of sprays from fuel injectors are needed to achieve the higher level of knowledge and understanding that will allow further progress in the mixing and combustion processes. Physical understanding of the mixture preparation, flow interaction and atomisation process are clearly incomplete. Moreover, the mathematical formulation of the spray characteristics and combustion process is not well established. The new concept of direct injection gasoline engine is

based on directly injecting the fuel into the cylinder. Therefore, the spray characteristics and the mixture formation are of primary importance and they are at the heart of the DISI engine development. Despite the most recent advances, the essential challenge for all new generation direct injection systems is the delivery of the fuel economy benefits under the constraints of the current and future emissions standards in both Europe and USA. This will require further development in the simultaneous control of HC and NO_x emissions under lean stratified conditions.

The development of the spray from a DISI injector can be divided into three discrete stages; in the first stage the liquid emerges from the nozzle until it starts to atomise. This stage depends not only on the fuel flow conditions and its properties at the nozzle exit but also on the airflow pressure and temperature at the nozzle exit. The second stage is the atomisation process during which the continuous flow of liquid is converted into ligaments that, in turn, are converted into droplets. The parameters that affect the atomisation process could be divided into external and internal parameters. External parameters are those controlling the interaction between the liquid jet with the surrounding air. The internal parameters control the liquid flow before being atomised and that, in turn, is controlled by the first stage. These two early stages depend mainly on injector design factors, the fuel properties, the fuel conditions as well as air physical characteristics. The third stage of spray development occurs during the spray penetration process, which is dominated by the interaction of the fuel droplets with the surrounding airflow field. The spray during this last stage is defined by its global and local characteristics. These characteristics, either define the overall spray structure, the macrostructure, that includes spray dispersion, penetration, cone angle and pattern, or qualify the internal spray structure, the microstructure, which comprises the mean droplet diameter and velocity, the droplet size and velocity distributions and the liquid and vapour concentrations of the fuel within the spray.

1.2 KEY POTENTIAL: DISI VERSUS PFI ENGINES

Besides the benefits of fuel saving by operating the engine under stratified combustion, a major advantage of DISI engines is their operation under unthrottled conditions which is known as the most efficient combustion process since the pumping losses can be avoided (Fraidl et al., 1996). The quick start and the fast transient response of DISI engines have encouraged the automobile companies to invest money

and effort in order to achieve the optimum engine design for better performance. The operation of DISI engines produces less HC emissions during cold start since the need for rich mixture is avoided as explained by Anderson et al. (1996). In DISI engines NO_x emissions can be reduced due to the decrease in the charge temperature and the possibility of its operation with high rates of EGR (Kume et al., 1996).

Yang and Anderson (1998) have discussed the improvement in both volumetric efficiency and knocking tendency with DISI engines when compared to PFI engines. They pointed out to two different basic phenomena. First, the fuel evaporation heat source in DISI engines is different from PFI engines. For DISI engines, the latent heat required for fuel evaporation is taken mainly from the air inside the chamber causing a reduction in the fuel-air mixture temperature, which can improve the volumetric efficiency and the knocking tendency relative to PFI engines where fuel evaporation takes place mainly on the hot surfaces of the port and valves. Second, the fuel injection timing affects the heat transfer rate from the combustion chamber surface to the charge. Earlier fuel injection cools the gas, causing an increase in heat transfer from the wall to the gases, however the final charge temperature at the time of ignition is reduced. For the late injection case, the heat transfer rate from the wall to the air before fuel injection is lower because the air temperature is higher. When the fuel is injected and the fuel droplets start to evaporate, the charge is cooled, thus, the gas temperature at the time of ignition is lower, resulting in a lower knocking tendency.

One of the main problems with DISI engines in stratified operation is the control of HC. This problem has recently been tackled by many researchers i.e. Iwamoto et al. (1997) and Alger (2000) by optimising the spark plug location relative to the injector and the ignition timing in addition to the spray and air flow pattern. The design of the combustion chamber may also play a role in exhaust HC emissions, especially in the case of complicated geometries.

1.3 DISI COMBUSTION SYSTEMS AND THEIR DEVELOPMENT

The first direct injection gasoline engine was introduced into the market in 1954 by Mercedes Benz which stopped its production three years later because its fuel efficiency was not very satisfactory and the car was very expensive. The history of research and development since that time has been reported by Iwamoto et al. (1997). It

seems that the idea of stratified combustion has been around for more than 70 years (Ghandi and Bracco, 1996).

To obtain a stratified charge the fuel must be injected shortly before the time of ignition and to be guided as a mixture towards the spark plug. Ideally, the mixture around the spark plug should be stoichiometric or slightly rich, while the rest of the combustion chamber should be filled with pure air diluted with recycled exhaust gases. The mixture preparation depends on the spray characteristics, in-cylinder flow field, spray/wall interactions, fuel properties, gas temperature and pressure as been investigated by several relevant studies by Fridl et al. (1996), Kano et al. (1998), Kume et al. (1996) and others.

Combustion characteristics at full load do not differ from homogeneous in working SI engines as quoted by Kech et al. (1998); however, during the stratified charge operation these characteristics change significantly. After combustion starts around the spark plug, the flame develops rapidly in the rich mixture region but then its speed decreases as the mixture becomes leaner away from the spark plug and close to the chamber wall.

Different combustion systems have been summarised by Lake et al. (1998) and Zhao et al. (1999), which can be divided into two main configurations: close-spacing and wide-spacing. These systems are illustrated in Figure 1-1 and described thereafter:

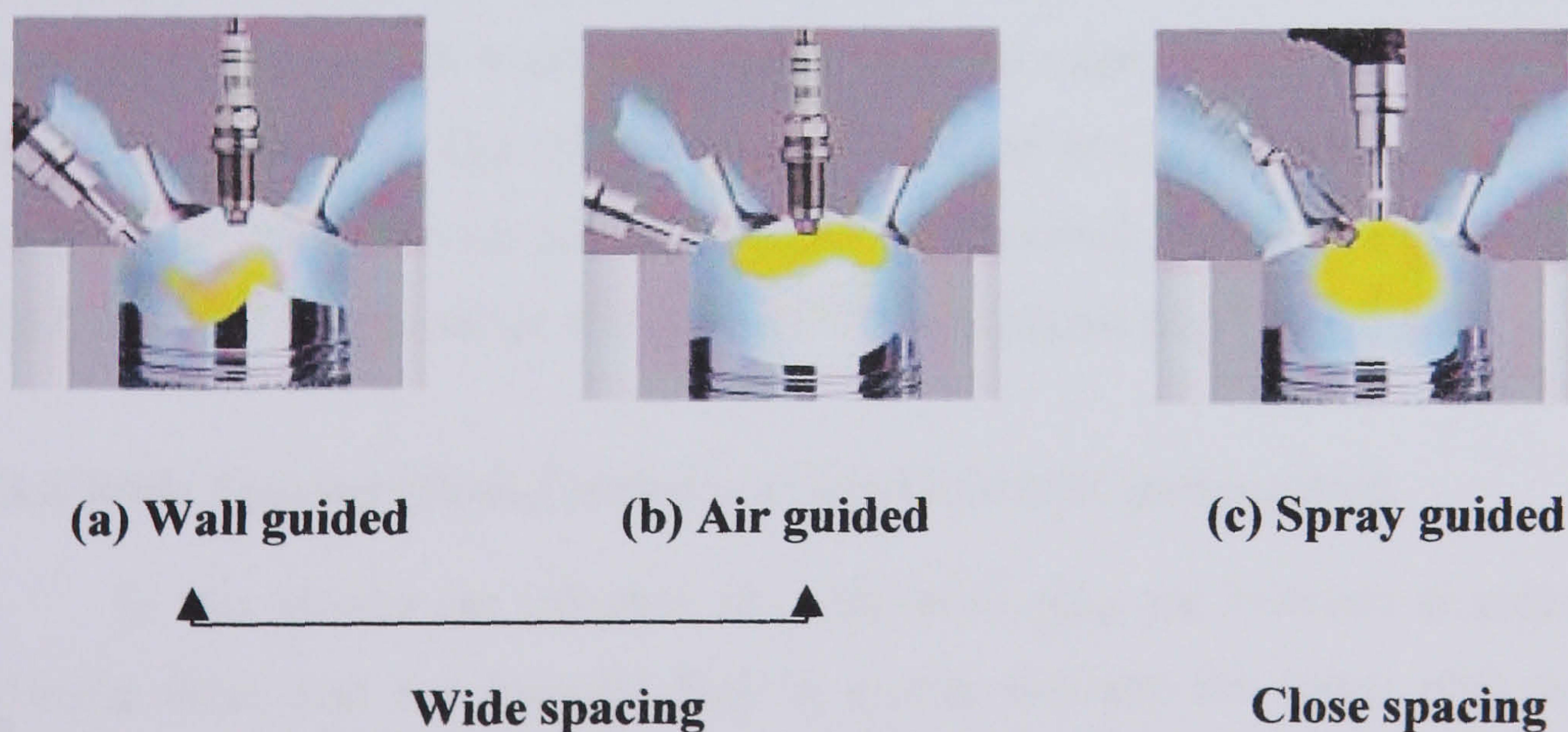


Figure 1-1 Wide and close spacing combustion systems in DISI engines

1.3.1 Close Spacing (Spray-Guided) Configuration

In this system, the spark plug is located close to the spray periphery, as shown in Figure (1-1c). This type of system leads to a spatially limited stratification, which allows a wide range of operation down to the idling loads with a global air/fuel ratio above 100 (Preussner et al., 1998). The appropriate air/fuel ratio at different engine loads is realised by the penetration depth of the fuel spray, mainly determined by the spray momentum. One of the major problems facing this configuration, is liquid fuel impingement on the spark plug due to the short distance between the injector and the spark plug. Fuel impingement on to the spark plug leads to shortening the life time of the spark plug's electrode material. Some work has been published recently on how to modify the spark plug in order to resist carbon fouling by Hanashi et al. (2001) and Miyashita et al. (2001). Also, there is a short time available for fuel to evaporate and mix with air. This system is also strongly dependent on the tolerances of the installation geometry and the spray characteristics. Therefore, its successful implementation imposes very tight tolerances for the serial production of fuel injectors, the ignition characteristics and the air motion control in order to achieve stable and repeatable spray and combustion (Pontoppidan et al., 2001).

Hitachi Ltd., as discussed by Ohsuga et al. (1997), employed a flat piston and a central swirl injector at the cylinder top next to the spark plug, while air swirl was employed to keep the fuel in the centre of the combustion chamber. Low spray velocity, small droplets and high swirl were found to be necessary conditions to achieve stable stratified combustion. The Orbital Engine Company has implemented the spray-guided system using their own air assisted injector as described in a series of papers (Houston and Cathcart, 1998; Cathcart and Railton, 2001; Cathcart and Tubb, 2002).

1.3.2 Wide Spacing (Wall-Guided and Air-Guided) Configuration

In this system the injectors and the spark plug are installed at some distance between them and the injected fuel is driven towards the spark plug through its interaction with the piston cavity wall and the tumbling air, as shown in Figure (1-1a). The wide spacing configuration allows both air guided or wall guided systems or a combination of the two. In the wall-guided combustion system, the in-cylinder flow (tumble and/or swirl) acts as a secondary guide (Whitaker et al., 1998). The fuel spray targets the piston bowl in the required area at the appropriate crank angle and in such a

way that the spray momentum carries the greatest proportion of the fuel in the direction towards the spark plug. Therefore, impingement on the cavity of the purpose-designed piston is the main guide that drives the spray into the spark plug location. In this system, mixture formation time is relatively long resulting in larger fuel-rich areas, which restrict the stratification potential and the maximum limit of the overall air/fuel ratio. On the other hand, this system is less sensitive to spray and installation tolerances.

This approach was firstly introduced by Mitsubishi Motor Corp. and described by Kum et al. (1996) and Iwamoto et al. (1997). In their system upright-straight intake ports were used to create intense reverse tumble together with a curved top piston, guiding the spray towards the spark plug within a very limited region. Also a high pressure common rail with a swirl injector were used to achieve small droplet sizes, which can easily follow the air flow and vaporise quickly. Fuel consumption was reduced by 35 % when the engine was running at 40 km/hr and a 10 % increase in power output could be observed at load operation. Another DISI engine (D4), manufactured by Toyota Motor Corp., and based on the same concept of Mitsubishi Corp. was introduced soon after and has been described by Harada et al. (1997). The main features of this engine are the piston with an involute shaped concave combustion chamber and the straight helical intake port with a swirl control valve. This design drives the spray towards the spark plug and controls the in-cylinder gas motion as well as the mixture location. A high-pressure swirl injector with an injection pressure of 120 bar was used. Ricardo engineers (Whitaker et al., 1998) have introduced four configurations for the injector and the inlet port: the side port side injector with swirl motion, the side port side injector with tumble motion, the side port central injector and the top port side injector: each type has its advantageous and limitations. The Nissan Motor Company introduced a prototype direct-injection gasoline engine with a round piston bowl, a 70 degree hollow cone spray and a swirl air control valve (Takagie et al., 1998). They showed similar results to the Mitsubishi system. Another system was introduced by FEV Motortechnik Co. in Germany and described by Baumgarten et al. (2000). In this system a tumble motion was introduced and the engine was operated in a homogeneous mode immediately after cold start and with the stratified lean operation starting as soon as a sufficient activity of the NO_x storage catalyst was achieved. This system showed a 13 % reduction in fuel consumption compared to its corresponding PFI engine whilst meeting the EU4 emission regulations.

In the air-guided combustion system, the air flow mean pattern is the main transport mechanism of the evaporated fuel to the spark plug location, as shown in Figure (1-1b). The piston bowl shape is designed to conserve the tumbling air momentum during the compression stroke in order to use it in transporting the evaporated fuel spray towards the spark plug. Therefore, a well-defined in-cylinder flow and spray are pre-requisites to ensure the adequate interaction between the injected spray and the in-cylinder flow over a wide range of engine operating conditions. The penetration velocity of the fuel spray or the mixture plume should be of the same order of magnitude as the in-cylinder flow velocity to enable adequate convective transport of the mixture by the flow field (Fraidl et al., 1996).

1.4 DISI ENGINE-OUT EMISSIONS

1.4.1 Direct Injection Engines Emissions

The pollutants formed during the combustion of sprays of liquid fuels are particulate matter (PM) like smoke, carbon and unburned fuel droplets, and gaseous pollutants like carbon monoxide, nitrogen oxides (NO_x), sulphur oxides (SO_x) and unburned hydrocarbons (UHC). The formation and control of these pollutants has been discussed in details by Williams (1990), who showed how droplet combustion can affect the pollutant emitted, as shown in Figure (1-2). This figure shows the importance of achieving a uniform droplet size distribution in the spray and good mixing with air to prevent formation of smoke and unburned hydrocarbons. The CO and NO_x formations depend on the gas temperature and the level of excess air. It has been shown that finer sprays produce less particulates and NO_x than sprays containing considerable quantities of large droplets.

It has been reported by Alger et al. (2000) and Matthews et al. (1999) that DISI engines have much higher engine-out hydrocarbon emissions (HC) than the port-injected ones. Sandquist et al. (1998) found that the mixture preparation time from the start of injection to ignition is very influential for both hydrocarbons and NO_x emissions. The mixture preparation process involves spray break-up into droplets, droplet evaporation and finally mixing with air.

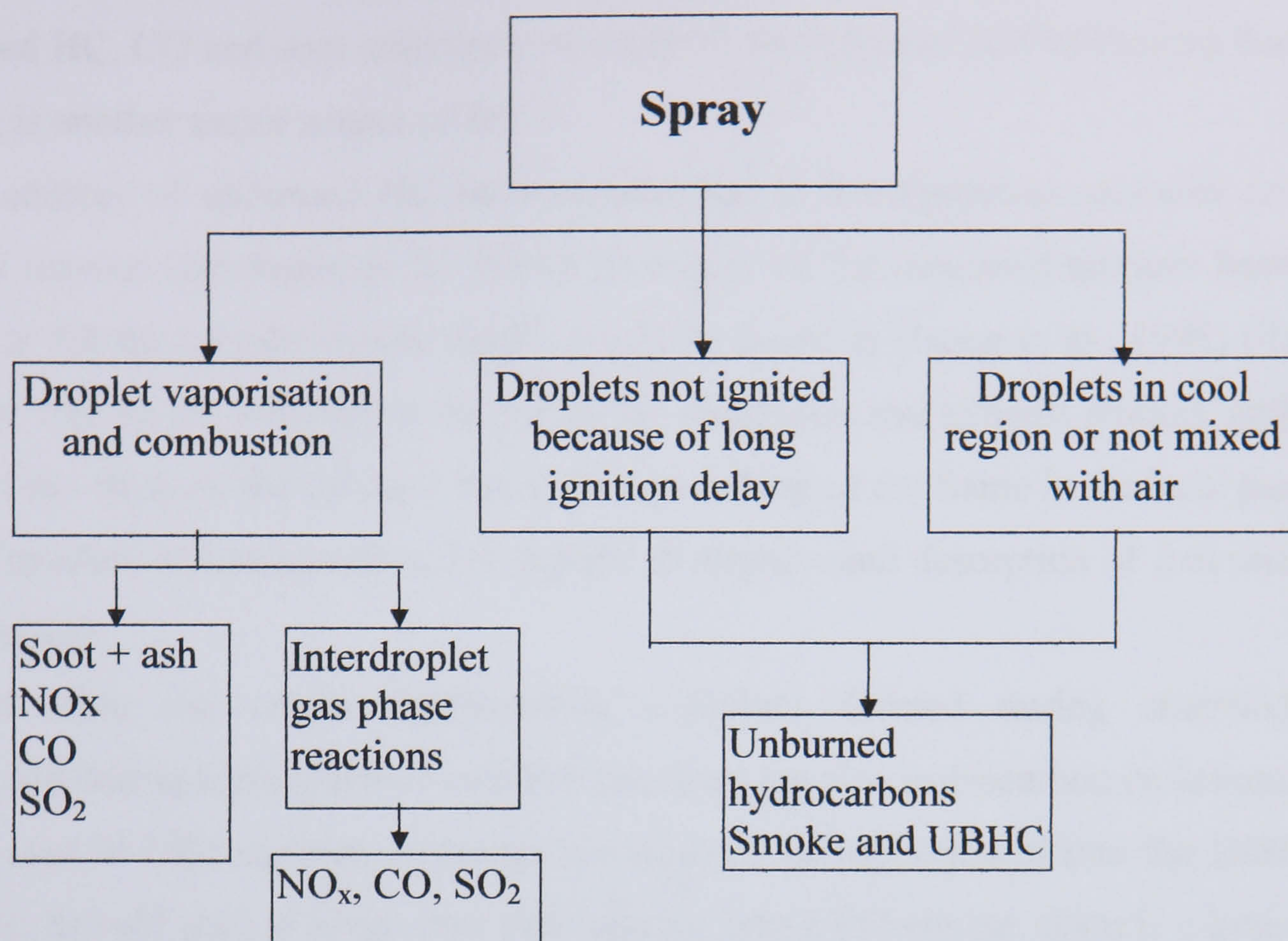


Figure 1-2 Schematic representation of the formation of pollutants during spray combustion

The hydrocarbons emitted during stratified combustion are related to either overmixing or undermixing according to Sandquist et al. (2000). Overmixing occurs when the time of mixing is long enough to cause the outer boundaries of the fuel/air cloud to become leaner than the lean combustion limit which implies that a strong air/fuel ratio is recommended at the boundary, as suggested by Cathcart and Railton (2001). Generally the increase in tumble or swirl strength inside the combustion chamber leads to better fuel mixing which reduces hydrocarbon emissions. The swirl component of the flow field is generally preserved for the entire period of the compression stroke but is transformed into turbulence early in the expansion stroke to assist in achieving good mixing between fuel and air. On the other hand, the tumbling component of the flow field is rapidly transformed into turbulence near top dead centre of compression, significantly enhancing the fuel-air mixing process. However, excessive turbulence levels generated from swirl and tumble air motion towards the end of compression, together with an over lean mixture, leads to a decrease in flame speed at the outer boundaries of the mixture which may cause flame quenching, and increase of the HC emissions. During undermixing, on the other hand, too little time is available for air entrainment and mixing. Consequently, the flame encounters a locally rich

mixture where fuel droplets may also be present which leads to incomplete combustion with increased HC, CO and soot emissions. A study by Stanglmaier (1999) showed that wall wetting is another major source of HC.

The sources of unburned HC from combustion of homogeneous mixtures are due to many reasons (Sandquist et al., 2000); (i) escape of the unburned mixture from combustion to fill the crevices (more details could be found in Huang et al., 1998) (ii) absorption of fuel by the lubricating oil during the expansion and exhaust strokes, (iii) quenching of the flame at the cylinder walls, (iv) quenching of the flame in the bulk gas due to local mixture inhomogeneties, (v) deposit absorption and desorption of fuel and (vi) valve leakage.

Apart from the engine hydrocarbon emissions formed during stratified combustion and during homogeneous combustion, there are also hydrocarbon emissions during cold start in DISI engines. Although the quantity of fuel injected into the DISI engine during its cold start is lower than that injected into a PFI engine, there is a large degree of cylinder wetting due to spray impingement resulting in engine-out HC emissions. It has been reported that about 60 percent of the engine-out emissions from automobiles occur during the cold start period before the catalyst becomes active. About two minutes are required for the catalysts to be heated to the light-off temperature by the exhaust gas. Two at least approaches are used as quoted by Miller et al. (1998) ; the first is to use an auxillary heater and the second is to trap the emitted hydrocarbons and store them temporarily in an adsorber. Lee et al. (2000) found that the best cold start performance in DISI single-cylinder engines, in terms of the accumulated hydrocarbons, is achieved by introducing tumble and then swirl followed by the baseline system. Also they stressed the importance of choosing the optimum ignition timing for the lowest accumulated hydrocarbons. Koga et al. (2001) showed that, if the fuel is heated to 80 °C before injection, the fuel volume associated with cylinder wetting decreases by 50 %.

Choosing the optimum injection timing and in cylinder flow can contribute to reduction of the HC emissions since this determines how much surface wetting occurs and also influences the amount of mixing and the equivalence ratio in the spark gap, as reported by Agler et al. (2000). Johnson et al. (2000) and Stovell et al. (2000) have also found that fuel properties have a significant effect on engine-out emissions and the catalyst efficiency in a DISI 1998 Toyota Corona passenger car.

In earlier research in DISI engines, a high percentage of NO_x and hydrocarbons were emitted until Mitsubishi (Iwamoto et al., 1997) succeeded in raising the exhaust gas recirculation (EGR) levels up to 35 % with a developed lean- NO_x catalyst. Kume et al. (1996) demonstrated that mixing EGR with the vaporized fuel inside the combustion chamber reduces the temperature of the combustion zone by about 200K, causing a reduction in NO_x emissions. However, although the mean temperature may be low, the high local temperature could lead to the production of nitrogen radicals which diffuse into the lean region and find an easy way to react with oxygen forming NO_x . The same approach of increasing EGR for NO_x reduction has been followed by Toyota (Harada et al., 1997) with almost similar reduction in both fuel consumption and engine-out emissions. In spite of the decrease in temperature of the combustion zone when increasing the volume of EGR, Sasaki et al. (1998) showed that it also leads to a significant increase in the exhaust gas temperature, which is extremely beneficial to the activation of the catalyst. However, excessive EGR results in deterioration of the fuel economy and HC emissions and leads to engine misfire and slower flame propagation due to the reduced oxygen concentration.

1.4.2 Passenger Car Emissions Legislation

The pollutants emitted from a DISI Toyota vehicle, the current emission standards and the European regulations for the years 2000 and 2005, as described by Schindler (1998), are shown in Table 1-1. This table confirms that high NO_x levels are emitted from DISI engines without a catalyst but with EGR. When equipped with both a catalyst and EGR, DISI engines can meet the current and even future emission regulations. Reducing the CO_2 emitted from the engines, as now required in many countries, necessitates improvements in fuel consumption which can be easily achieved by DISI engines. Reduction of CO_2 represents the main target of both environmentalists and car manufacturers. In July 1998 the European Commission and the European automotive industry represented by the European Automobile Manufacturers Association (ACEA) reached a ground breaking agreement on the reduction of CO_2 emissions from passenger cars. In this agreement ACEA has committed itself to achieve an average CO_2 emissions figure of 140 g/km by 2008 for all its new cars sold in the EU. Although the target is ambitious and needs a large number of improvement measures, there is strong potential and commitment towards achieving this target; Table 1-2 shows the

limitations of CO₂ emissions for a new car registered or to be registered in the EU as reported by ACEA.

Table 1-1 DISI engine emissions and future emission regulations

	HC (g/km)	CO (g/km)	NO _x (g/km)
DI engine as presently developed	0.1	0.05	0.1
DI engine with EGR but without catalyst	2.75	3.56	0.6
DI engine raw NO _x emissions	---	----	1.85
Current emission standards (Japan)	0.25	2.1	0.25
Euro 2000	0.2	2.3	0.15
California stricter LEV (Low Emission Vehicle) standards	0.056	2.6	0.19
Expected Euro 2005	0.1	1.0	0.08

Table 1-2 CO₂ emission regulations (ACEA)

Year	1995	1999	2000	2001	2008
CO ₂ (g/km)	185	174	169	164	140

In order to achieve these restricted limits, researchers have attempted to modify the fuel used in DISI engines or replace it by renewable fuels such as LPG, CNG or DME. Unfortunately these alternative types of fuel need special attention for the safety of handling and storage, in addition to the extra costs of additional equipment that may be required and the various safety measures, as quoted by Arcoumanis (2000). However, the study performed by Stan et al. (2001) showed that there is potential for further reduction in CO₂ when using methanol with a well controlled amplitude and duration injection pressure pulse system.

1.5 FUEL INJECTION SYSTEM AND SPRAY REQUIREMENTS

The injection system of DISI engines includes all the parts required to transfer the fuel from the fuel tank to the combustion chamber. The injection system design is controlled by the injection system requirements and the spray characteristics.

1.5.1 Injection System Requirements

The fuel injection system of DISI engines is the key component in preparing a well atomised spray that matches the specific in-cylinder flow field in order to provide the desired mixture cloud at the spark plug over the entire operating range of the engine. The fuel injection system must have the capability of providing an accurate quantity of fuel during both late injection for stratified charge combustion at part loads, and during early injection for homogeneous charge combustion at full load. The main requirements of a DISI injection system have been discussed by Zhao et al. (1999). Thus, DISI injection system is designed not only to deliver a precisely metered fuel quantity but also a highly repeatable spray structure to avoid engine misfire. Well-atomised spray from the injector is essential to achieve fast evaporation and good mixing with air and, as a result, reduce the hydrocarbon emissions. The ability to provide fuel within a short time pulse is very important for light load operation where fast opening of the needle is required and improvement in the dynamic range of the injector is important. The fast opening of the needle has to be associated with a well-designed pressure damping rail system to avoid pressure surge in the injection system. Secondary injection or needle bounce has to be avoided. High injection pressure is required to achieve better atomisation and reduce the time of injection for the quantity of fuel needed for each injection, as reported by Davy et al. (2000). It is also an aim to limit the pressure inside the injection system to minimize any future maintenance demands and the possibility of leakage. There might also be a need for split injection during full load operation so that a portion of the fuel is injected during the intake stroke and the rest during compression. This technique has been proposed by Yang and Anderson (1998) to reduce the tendency of knocking and, at the same time, benefit from the increased volumetric efficiency. They also reported that 2-3 % improvement in IMEP has been achieved.

1.5.2 DISI Injectors

The high-pressure single hole, single fluid, swirl injector using an inwardly opening needle is by far the most common DISI injector type, and there is still research in progress by injector manufacturers to improve its nozzle design and spray performance.

Most current DISI injectors are inwards opening, designed to apply a strong rotational momentum to the fuel in the injector nozzle, in addition to the axial injection

velocity. The liquid first flows through a number of tangential slots or holes into a swirl chamber to emerge a single discharge hole as an annular cylindrical sheet that spreads radially outwards to form a hollow-cone spray, as confirmed by Hoffman et al. (1997). The experimental investigations of Dodge et al. (1996), Zhao et al. (1996) and Kume (1996), showed that common injection systems with electromagnetically-activated needle type swirl injectors satisfy the spray requirements adequately. They showed that the swirl injector produces a conical spray with a wide dispersion at atmospheric back pressures and small droplets of about 15 μm in diameter with low penetration which are the requirements of the early spray injection stage. Moreover, at higher air density, as in the case of injection during compression, the spray angle decreases due to the higher aerodynamic effect leading to a compact spray. Some of the manufactured engines (Toyota and Nissan), as quoted by Harada et al. (1997) and Okada et al. (1998), use an offset spray injector where the spray is angled or offset from the injector centre.

The air-assisted atomizer has also been under investigation by many authors (Meyer et al., 1997) for quite sometime. The air-assisted injection system consists of two main components, a fuel metering injector and an air or charge injector which jointly deliver a mixture of metered fuel and air into the combustion chamber at relatively low pressure (typically 6.5 bar). The major advantage of this type of injector is its low fuel injection pressure and the fact that it provides more flexibility to meet the combustion mixture requirements as discussed by Houston and Cathcart (1998). However, Wigley et al. (1999) showed that a relatively longer time is needed to inject the same quantity of fuel when compared to the single fluid swirl atomizer due to its lower injection velocity. The air-assisted injection system introduced by Orbital Engine Co. and described by Cathcart and Tubb (2002) includes two injection pumps to match the demands of both part-load and full-load engine operation, besides the compressor needed for the compressed air which needs special attention to prevent any extra noise (Bowden et al., 2000). These extra components increase the system cost and complicate the running maintenance while needing more control. The spray structure exiting from an air assisted fuel injection nozzle has been studied by for example, Meyer et al. (1997), Ekenberg and Johansson (2000) and Houston et al. (1998).

Recently other injector types have been tested for DISI engines. The spray structure from an outward swirl injector has been investigated by Delphi and described by Xu and Markle (1998). They claim that the outward opening is able to avoid the

poorly atomised pre-spray structure. Moreover it offers design flexibility since the initial liquid film thickness is directly controlled by the injector needle stroke rather than by the angular velocity of the swirling fuel. They have also indicated that the initial film thickness is smaller during the needle opening and closing periods due to the reduced pressure at the swirler as a consequence, the atomisation improving during needle's opening and closing. Another advantage of this injector is the flushing out of any deposits accumulated on the tip of the injector. The main disadvantage of this injector is the spray-to-spray variation where it has been reported that the angle changes from one injection to another and, therefore, the repeatability of the spray is inferior to that of the inward swirl injector; thus is very important since lack of repeatability leads to engine misfire.

A high-pressure slit injector has been tested by many authors from Toyota, e.g. (Kanda et al., 2000; Okamoto et al., 2001 and Nakashima et al., 2001). This injector produces a thin shaped fan spray that could be accommodated within the piston whose cavity shape can direct the spray towards the spark plug without any need of extra airflow. It has been claimed that this type of injector improves the stability of mixture concentration around the spark plug and minimizes any combustion fluctuations. Toyota research team considered it as a new concept as it allows stratified charge to be used at higher load and at higher engine speeds than that with the traditional swirl type injector as been investigated by Takeda et al. (2000).

More recently, multi-hole injectors have started becoming the subject of an investigation; Ortmann et al. (2001) have shown that the spray angle from this type of injector is very stable with chamber pressure. Significant improvements in fuel consumption and emissions have also been reported. An early comparison in implementing the multi-hole injector versus swirl injector in DISI engine has been investigated by Tomoda et al. (1997). They showed that the multi hole injector has a wider distribution of droplet sizes and, to achieve the same mean diameter a higher injection pressure is required.

1.5.3 Spray Requirements from Swirl Injectors

Swirl injectors are recognised to be well adapted for DISI engines because they generally produce fine sprays with a moderate injection pressure. A considerable number of publications have recently been appeared in the open literature on spray

characterisation of swirl type injectors. These studies aim to identify the optimum spray structure for each combustion chamber configuration and operating conditions in order to minimise the engine-out emissions and achieve stable combustion with maximum overall air/fuel ratio in the case of stratified combustion.

For the wall/air guided spray configuration, which so far employs the pressure swirl injector, a compact and stable spray during late injection is essential to contain all the spray inside the piston cavity. Small droplet sizes have also to be produced to ensure quick evaporation and mixture preparation before it arrives at the spark plug (Fraidl et al., 1996). Optimum timing of the spark after fuel injection is also very crucial to avoid overmixing or undermixing of the fuel and air and, therefore, reduce the engine-out hydrocarbons as quoted by Sandquist et al. (2000) and Kano et al. (1998). At higher loads, in the early injection stage, a well-dispersed fuel spray is desirable to achieve a homogeneous charge even for the largest fuel quantities. Moreover, spray penetration should necessarily be short in order to prevent the impingement of the fuel spray on the piston surface and cylinder liner which leads to increased soot and hydrocarbon emissions as well as dilution of the lubricating oil, as described by Dodge (1996). A narrow distribution of droplet sizes is essential to ensure fast evaporation and good mixing. Dodge (1996) has shown that a Sauter mean diameter of roughly 15 μm is required to achieve reasonably high evaporation rate and to provide minimum impact of the liquid spray on the surface of the piston. It is a major requirement to achieve a well controlled spatial and temporal distribution of fuel liquid and vapour inside the combustion chamber. A repeatable spray (minimum spray-to-spray variations) is also essential to avoid engine misfire and torque fluctuations during engine operation as quoted by Kano et al. (1998).

Spray has to be optimised with respect to the air flow pattern inside the chamber in order to achieve stable combustion with minimum emissions. Not only small droplet sizes are required to achieve fast evaporation but also air motion around the droplets is very important. The entire evaporation of the droplets is a diffusion controlled process, and steady evaporation reduces the droplet temperature until the vapour around the liquid droplet reaches the equilibrium (Ortman, 2001). The resulting saturation stops the evaporation process due to the absence of diffusion gradient. In order to maintain fuel evaporation a continuous supply of clean air around the fuel droplets is necessary. Therefore, the characteristics of the spray under quiescent air flow conditions have to be

well known before air is introduced and then the effect of air motion on the spray microstructure has to be quantified for each spray type.

1.6 FACTORS AFFECTING SPRAY CHARACTERISTICS OF SWIRL INJECTOR

To achieve the required spray characteristics, fuel injection equipment (FIE) has to be well designed to provide the accurate amount of fuel to the injector under the necessary conditions. The spray characteristics generated by the FIE depends on what is happening within three regions during the fuel development. The first region is located inside the injector itself; the geometry and the flow path inside the injector as well as the needle opening profile are amongst the main factors affecting spray structure. Also the nozzle surface finish has been considered as another factor affecting the stabilization of the film and the subsequent atomisation process (Tomoda et al., 1997). The second region is located very close to the nozzle exit, within 1mm. Along this short distance, the liquid film penetrates following a certain profile depending on its initial momentum and the mutual interaction between the liquid film and surrounding air inside the chamber, before starting to disintegrate to form droplets with different velocities and sizes. The third region is located further downstream, after film atomisation occurs. Within this last stage, the spray characteristics depend mainly on the mutual interaction between the liquid droplets and the air flow pattern as well as the thermodynamic conditions inside the combustion chamber. It also depends on the interaction between the droplets themselves and fuel properties. Each stage described above before has been studied separately in this thesis and the literature review of work related to these stages is described in the following subsections. In the wall guided combustion system, the spray wall impingement process has also to be considered in characterising the spray development.

1.6.1 Flow and Film Formation Inside the Swirl Injector

In the swirl injector, the liquid is injected through a number of tangential passages into a conical section from which it emerges with both axial and tangential velocity components to form a thin hollow cylindrical film at the nozzle exit with an air core in the centre.

The thickness of the liquid film created inside the nozzle from a simplex swirl atomizer has been calculated first by Giffen and Muraszew (1953) and reported by Lefebvre (1989). In their analysis, both film thickness and discharge coefficient are determined as a function of the main geometry of the swirler and nozzle exit by assuming non-viscous flow. This approach has been applied to the current DISI injector and is described in Appendix 1-A. This analysis shows no dependence on the fuel physical properties or the operating conditions and calculates a single value for the discharge coefficient. The effect of other physical parameters on the discharge coefficient seems to be very weak as revealed from the experimental work conducted by Jones (1982). Dumouchel et al. (1993) also showed that the velocity profile at the nozzle exit of a swirl injector is more influenced by the internal geometry of the swirl atomizer. They concluded that, by increasing the nozzle length, the velocity profile at the nozzle exit changes causing a reduction in the spray flow angle. These results in swirl injectors have encouraged early researchers in DISI swirl injectors to make use of the equation derived by Lefebvre (1989) to estimate the film thickness at the nozzle exit.

Han et al. (1997a) and Fans et al. (1997) were amongst the first to use this analysis and apply it to DISI swirl injectors to estimate the initial film thickness and then to their spray model after introducing a correction factor to match the computer model results with the experimental spray measurements. In their analysis, the atomisation process was neglected and the initial droplet diameter after atomisation was assumed equal to the liquid film thickness. More recently Cousin and Nuglisch (1999 and 2001) used a similar analysis with a correction parameter in their model to fit the experimental results. Datta and Som (2000) have predicted numerically for a simplex swirl atomizer the air core diameter, the coefficient of discharge and the spray angle under different nozzle design parameters. The diameter of the central air core that stabilises inside the nozzle has been predicted from a hydrodynamic situation that yields the minimum resistance to the liquid flow. Their results show that the discharge coefficient decreases while spray angle and air core diameter increase with flow rate at low values of the liquid flow rate ($<0.06\text{cm}^3/\text{ms}$). However, for flow rates higher than 0.06 the result shows no effect. Ren et al. (1997) established a 3-D model inside their swirl DISI injector assuming the hole is full of liquid and then calculated the film thickness from the location where the net mass balance of the incoming and exiting from the hole flow becomes zero. The results predicted a larger spray angle, which has

been estimated from the swirl and axial velocity at the nozzle exit, whose value remained constant and independent of the injection pressure, in agreement with Lefebvre (1989) and Nouri et al. (1999) and with the images taken in this study; a similar approach has been considered by Cousin et al. (1998). Arcoumanis and Gavaises (2000) and Gavaises et al. (2002) compared the results from a 3-D single phase model with a 2-phase 2-D model considering the volume of fluid method (VOF) and showed that the results from the single phase model fail to predict reasonably the flow process taking place inside the injector hole. The results from Kubo et al. (2001) using the same numerical technique (VOF) showed reasonable agreement with the experiments if the flow angle is considered equal to the spray angle; the deviation was evaluated to be within 8%. Le Coz and Hermant (1999) succeeded in photographing the film inside an extended transparent nozzle. Their work is similar to the work presented in this study, however the details of their injector geometry were not known for validating their model, moreover the calculated film thickness exhibited fluctuations which are most probably due to the limited number of images used for averaging and the method used for detecting the film thickness.

1.6.2 Film Development and Atomisation Process Near the Nozzle Exit

1.6.2.1 Film penetration outside the injector

A liquid jet is formed inside the swirl injector when the needle starts to open. This jet starts to open up as the centrifugal forces increase due to the increase in the swirl velocity associated with the opening of the needle and the reduced pressure drop. When the valve becomes fully open, the centrifugal force reaches a value that overcomes the liquid internal pressure and surface tension and a liquid film is created. The film is then ejected from the nozzle into the surrounding gas forming a tulip-shaped or a conical-shaped structure depending on Weber number (Ramamurthi and Tharakan, 1998). During the needle opening the injected velocity is relatively low due to the pressure drop above the needle seat, which encourages a tulip shape structure to be formed at some stage during the needle opening, provided there is no pressure fluctuations or flow disturbance. It is believed that the high pressure fluctuations during the needle opening period obstruct the liquid film formation inside the nozzle and in turn, prevent the

formation of the conical shape spray structure and could be responsible for the large droplets present in the pre-spray.

The early study on annular jets at low velocity has been performed by Baird and Davidson (1962), who considered the balance of the forces acting normally on the moving liquid sheet in order to represent mathematically the profile of the annular jet. The gravity force was neglected and the radius of axial curvature is assumed to be much larger than the jet circumference radius. Their results showed that the transition from tulip-shaped to conical-shaped structure depends on the value of the Weber number

($We = \frac{\rho_l U_j^2 h}{\sigma}$); where ρ_l , U_j , σ and h are the liquid density, jet velocity, liquid surface

tension and half the sheet thickness respectively. More recently, Hassan et al. (1997) have analytically solved the shape of a vertical annular liquid jet without the above assumptions for a non-swirling annular jet flow. Their analysis predicted the overall shapes of a converging and a diverging jet in a good agreement with their experimental results. For low velocity, their analysis was more accurate than that obtained by Baird and Davidson (1962). Another experimental and analytical study has been performed by Sivakumar and Raghunandan (1997) who used an annular swirler injector with a specific injection angle of 35 degrees. Their analysis and experiments revealed that the jet exit angle increases linearly with the liquid flow rate in the tulip regime, but in the divergent sheet regime the variation is very small. The images of the film at low velocity also showed that a multiple tulip could be formed if a swirl motion is created. Swirled liquid sheets have been analysed by Ramamurthi and Tharakam (1998) who applied the same theory considering as additional the centrifugal force. The results have been compared with the spray images obtained for different liquid Reynolds numbers and Weber numbers. Their results showed that the transition from a tulip shape to a conical divergent shape occurs when the liquid Weber number exceeds 150. By changing the surface roughness of the nozzle they observed that the surface texture of the film outside the nozzle becomes frothy, however the geometry of the spray remains the same and the transition from a tulip shape to a divergent conical shape occurs at the same Weber number. Based on these observations, the present study applied the same theory to predict the film profile exiting the pressure swirl atomiser with higher injection velocity, which has been compared with the magnified images taken close to the nozzle exit.

1.6.2.2 Atomisation and film disintegration process

The term atomisation expresses the physical process by which the continuous liquid jet gradually disintegrates into a large number of small droplets. The fundamental principle of the disintegration of a liquid, ejected from a swirl injector, depends on increasing the surface area of the sheet until it becomes unstable and breaks down into droplets. This method is usually employed to produce small droplets at lower working pressures by spreading the liquid into a thin sheet (Lefebvre et al., 1989).

Two types of atomisation could be distinguished; the first is the primary atomisation process that is responsible for the disintegration of a jet or a sheet into ligaments or droplets. The second is the secondary atomisation process where droplets of a size larger than a certain critical size continue to disintegrate into smaller droplets due to the aerodynamic effects. Primary atomisation is one aim of this study, since it is the main process affecting the spray structure and the initial droplet size distribution, which is required as input to any spray model.

Proper understanding of the primary atomisation process will lead to improvements in the injector design in order to meet the spray requirements and achieve a better spray/flow interaction. The primary atomisation is the link between the continuous liquid flow exiting from the nozzle and the atomised fuel spray. The flow inside a DISI injection system and inside its nozzle has been modelled by Arcoumanis et al. (1999) and Kubo et al. (2001) to identify the flow conditions at the nozzle exit. In order to calculate the spray characteristics, spray models need the initial droplet size and velocity distributions as inputs, which are either based on empirical equations or to fit the experimental data. Therefore, atomisation can be considered as the missing link between liquid film development inside and outside the nozzle and the spray model.

A round jet break-up has been photographically investigated by Ohnesorge (1936) who established the well known four distinct regimes on a log-log plot representing Ohnesorge number ($Oh=We^{0.5}/Re$) versus Reynolds number (Re). The first regime is the Rayleigh capillary mechanism region at lower values of Oh and Re ; in this region the aerodynamic interaction is not significant. The second regime is the first wind-induced region where non-axisymmetric sinusoidal oscillations occur resulting in droplet sizes comparable to the jet radius. The second wind-induced region results in small droplets, while the atomisation region with even higher value of Oh and Re results in the smallest droplets. As the Weber number or Ohnesorge number increases

further, the break-up length tends to decrease. Thus the atomisation regime is the domain where break-up occurs at the nozzle exit.

Similarly for liquid sheets Dombrowski and Fraser (1954) have performed an extensive photographic investigation into understanding the disintegration of the liquid sheet produced from a single hole fan-spray nozzle and a spinning disk. Three modes of sheet disintegration have been identified; rim, perforated and wave. In the rim mode the surface tension causes the edge of the liquid sheet to contract into a thick rim which then breaks-up in the same way as a free jet. This mode is most prominent when the viscosity and surface tension are both high. In the perforated sheet disintegration mode, holes appear in the sheet and are delineated by rims formed from the liquid that was initially included inside. These holes grow rapidly in size until the rim-adjacent holes coalesce to produce ligaments of irregular shape that finally break-up into drops of varying size. In the wave disintegration mode, the sheet disintegrates through the generation of a wavy motion on its surface and in the absence of perforation, whereby areas of the sheet, corresponding to half or full wavelength of oscillation are torn away before the leading edge is reached. Wave sheet disintegration due to aerodynamic interaction with the surrounding gas is more dominant with practical injectors. Hashimoto and Suzuki (1991) have studied the effect of injection velocity on the fine interfacial waves produced on the sheet. A contact needle and camera with strobe light were used to measure the sheet thickness and the interfacial wave pattern of the liquid sheet. They confirmed that both the upper and lower interfaces of the liquid sheet periodically vibrate with the same phase. Five patterns have also been identified, depending on the value of $R_e = \frac{u_j h}{\nu}$ and the dimensionless surface tension, S , which is

the reciprocal of the Weber number $W_e = \frac{\rho_l u_j^2 h}{\sigma}$ where ρ , σ , ν , h , u_j are the fuel

density, the fuel surface tension, the fuel kinematic viscosity, the film thickness and the jet velocity of the liquid film respectively. These patterns can be described as follows:

- The first pattern (smooth) is the flow where the interface of the liquid sheet is smooth and no waves appear.
- The second pattern is the two-dimensional wavy flow where roughly two-dimensional and finely striped waves begin to appear on the smooth interface in the vicinity of the nozzle exit.

- The third pattern is the evolved two-dimensional wavy flow where the two-dimensional waves grow and a more obvious striped pattern can be observed.
- The fourth pattern is the pebble wavy flow where the partially evolved waves grow and deform into dimples with a finely stripped pattern which the interface becomes rough and presents a pebbly pattern at positions far from the nozzle.
- The fifth pattern (S-wave) is the sand-paper-like wavy flow in which the interface of the liquid sheet in the downstream area is quite rough although the evolved two-dimensional wave and the pebble waves are still recognizable near the nozzle.

By measuring the light intensity gradient of the inner and outer surfaces of the liquid sheet, it was found that both surfaces periodically vibrate with almost the same phase and amplitude in the vicinity of the nozzle exit (antisymmetric wave). They also found that the small fluctuations grow to become the interfacial wave downstream of the nozzle and this growth increases with increasing Reynolds number (Re). The amplitude of the interfacial waves was found to be very small when compared to the sheet thickness. In an effort to know the source of this interfacial waves, Hashimoto and Suzuki (1991) also solved numerically the internal sheet flow and compared the results with experiments. They found that, the sheet surface wave was closely related to the internal instability of the liquid sheet.

The atomisation process is governed by complicated physical mechanisms and different types of instabilities which make it difficult to be modelled to such a degree that Bayvel (1993) claimed that it is hard to expect to see significant progress in the development of the theory of atomisation in the future and until today not much progress has been achieved in this field. Although direct numerical simulation can be applied, it may be inaccurate because the initial conditions can not be exactly identified. Moreover, it is difficult to be practically applied due to the very high cost of the computations. These difficulties are due to the dramatic scale difference between the wavelength in the sheet surface ($\approx 10 \mu\text{m}$), the engine bore and the stroke ($\approx 10 \text{ cm}$).

The atomisation process is governed by both internal and external parameters. The internal parameters include the flow conditions before the fuel is atomised and that depends on the turbulence intensity produced within the nozzle, the nozzle surface roughness, the physical properties of the fuel and the injection pressure which all affect the behaviour of fuel break-up. The external parameters include the outside air pattern in the combustion chamber, the air pressure, the temperature and the physical

properties. These external parameters are highly transient specially when combustion starts. Although the effect of the internal parameters on the flow conditions at the nozzle exit can mathematically be modelled, it is quite difficult to accurately validate this type of model because of the small scale of turbulence and surface disturbances that occur in the flow at the nozzle exit. The other way of representing the atomisation process is to perform a parametric study to find out a correlation that will include both the internal and external parameters.

In spite of the high interest in spray-producing injectors, the design of these injectors remains largely empirical due to the lack of knowledge of the basic atomisation mechanisms. Understanding of these mechanisms is incomplete because of the difficulties in experimentally observing the flow in the very limited space close to the nozzle, and in modelling the emerging turbulent two-phase flow and the different types of instabilities in this highly transient flow.

1.6.2.3 Mechanisms of atomisation

Over the last century, various theories have been proposed to explain the liquid jet atomisation process which could be applicable to sheet atomisation. These theories introduce several mechanisms for atomisation such as the aerodynamic interaction between the liquid jet and the ambient gas medium, the liquid turbulence developed within the nozzle passage, liquid boundary layer acceleration, liquid cavitation phenomena occurring at the nozzle hole, liquid supply pressure oscillations and flash boiling. Reitz and Bracco (1982) found that none of the theories alone could satisfactorily explain their experimental results obtained in a liquid jet.

For the aerodynamic interaction between the liquid and gas media, a small perturbation arises from the interaction between the sheet and the surrounding gas. The winner of the competition between the capillary and aerodynamic forces determines whether a disturbance may grow or decay. The increasing perturbation leads to the disintegration of the liquid sheet into fragments. These fragments rapidly contract under the effect of surface tension into unstable ligaments which finally, are broken into a multitude of droplets. Experimental observations have indicated that the unstable waves during the inception stage of the interfacial waves are almost two-dimensional, and the wave amplitude is very small compared to the sheet thickness. Theoretically, Squire's theorem has dictated that for any three-dimensional disturbance, there exists a

corresponding two-dimensional disturbance which is unstable for plane parallel flow (Li, 1996).

Turbulence may also play a role in destabilising the jet at higher speeds. It has been shown that there is a tendency to associate the turbulent flow with a Reynolds number higher than the critical one, which is around 2320 for a jet flow. If the flow at the nozzle exit is fully turbulent, the radial velocity component can soon lead to disruption of the surface film. No aerodynamic forces are needed for break up so that break-up can proceed even in a vacuum (Lefebvre, 1989). The effect of the liquid boundary layer acceleration on atomisation depends on the state of the boundary layer at the nozzle exit. Some researchers have suggested that the boundary layer turbulence plays a critical role in the atomisation process.

At higher jet velocities, nozzles with turbulent boundary layers provide poorer atomisation than laminar jets. In addition, the theoretical work of Leib and Goldstein (1986) suggests that turbulent velocity profiles are actually more stable than the more fully developed profiles associated with laminar flows. Hilbing and Heister (1998) have studied the effect of the boundary layer development on the atomisation process of a liquid jet by solving the boundary layer equation by CFD in the viscous region and the boundary-element method at the edge of the boundary layer. In this way, the entire jet was modelled in two modes: one at the gas/liquid or liquid/wall interface and the other at the edge of the liquid boundary layer. The obtained results by the model indicated that atomisation mechanisms that rely on boundary layer relaxation or velocity rearrangement cannot generate the instability required to atomise the jet. These results have been supported by the experiments done by Reitz and Bracco (1982) who also noticed that there is a swelling at the exit plane whose size depends on the boundary layer thickness at the exit plane. This swelling provides a large amplitude disturbance for the initiation of the aerodynamic instabilities.

Cavitation can generally be produced due to the local low pressure inside the injector which leads to the formation of fuel vapour film and/or bubbles. Some of these bubbles collapse at the nozzle exit resulting in enhancement of the atomisation process. The development of cavitation inside multi-hole diesel injectors has been studied by Arcoumanis et al. (1998, 1999, 2000) and Roth et al. (2002). The effect of liquid pressure oscillations on the atomisation process depends on the amplitude and frequency of these oscillations. These pressure fluctuations at the nozzle exit may have some subsidiary influence on atomisation in DISI swirl injectors. These may be

attributed to the small time scale of the atomisation process relative to these fluctuations. However, investigation of the effect of pressure fluctuations on atomisation has not been presented up to now and this is part of this study.

Since the aerodynamic instability is responsible for the jet break-up at lower relative velocities, it may be inferred that it is still playing a very important role in the atomisation regime (Reitz and Bracco, 1982). The linear instability effect on a liquid film also showed very similar results with the direct numerical simulation data obtained by Ashgriz and Mashayek (1995) for low Reynolds numbers. At higher Reynolds numbers, the linear instability analysis overpredicts the growth rate, although the trend is rather similar.

1.6.2.4 Types of aerodynamic instabilities

Aerodynamic instability of the liquid sheet causes waves on its surface whose amplitude increases with time and also along the liquid film. For simplicity these waves are always studied either temporally or spatially. In the temporary instability of a jet, it is assumed that the disturbed amplitude grows up everywhere along the sheet, even in the immediate neighbourhood of the atomiser. Keller (1998) noticed in their study of a circular liquid jet that the temporal instability cannot be actually applied to the swirl atomizers and that the wave sets in after the sheet is formed, and develops downstream along the sheet. However, Li (1996) found that the Gaster's (1962) relationship can be used to relate the temporal and spatial growth rate for plane sheets at high velocity. Their results showed that, at large Weber numbers, the same dimensionless cut-off wave numbers and cut-off frequencies are obtained for both types of instability. Thus, the spatial stability results have one to one correspondence with the temporal stability analysis because the group velocity is equal to one in the case of large Weber numbers.

1.6.2.5 Linear instability analysis (LIA)

LIA is based on the hydrodynamics theory in which infinitesimally small perturbations are superimposed upon a stationary flow field. These perturbations are time dependent, and their amplitudes may exhibit three stages; decrease, remain constant, or increase with time. For the first two cases, the flow is stable because perturbations do not give rise to atomisation, whereas the last case is unstable as it leads to atomisation. This does not mean that every unstable wave will cause atomisation because some of these waves, after some time, may reach a certain finite value at which

the LIA theory is not applicable. Then, the flow field is disturbed by this wave and it may adopt another stationary state without the occurrence of atomisation. According to most of the analyses using linear instability, whether on jet, sheet or hollow cylindrical surface, two modes of instabilities could be distinguished: these two modes are; the axisymmetric (varicos-dilational) waves or the antisymmetric (sinuous). According to LIA, the amplitude of unstable waves δ will grow up exponentially with time according to:

$$\delta = \delta_o \exp(\omega t + ikz + inz)$$

where δ_o is the initial amplitude and ω is called the growth rate, k is the axial wave number, n is the circumferential wave number, t is the time and z is the axial distance,. Accordingly, many waves with different wave number k , which is related to wavelength λ through $\lambda=2\pi/k$, will grow up on the liquid surface with different growth rates. The wave with the maximum growth rate is considered responsible for the break-up. The wavelength of the wave with the maximum growth rate is called the optimum wavelength and the maximum growth rate is called the optimum growth. The minimum wavelength at which waves start to grow on the surface is called the cut-off wavelength and its corresponding wave number is called the cut off wave number.

Linear instability analysis applied on flat liquid sheet and jet. The earliest mathematical investigation on primary break-up has been performed by Rayleigh (1892) who analysed the instability of a non viscous incompressible liquid round jet under laminar flow conditions neglecting the air effect. His analysis showed that all disturbances with wavelength greater than its circumference will grow up. This growth is exponential and takes the form of $e^{\omega t}$, where ω is the growth rate. Furthermore, his results showed that one class of disturbances, with certain wavelength λ_{opt} , grows fastest and eventually controls the break up. The wavelength corresponding to maximum growth is called the optimum wavelength λ_{opt} which is related to the jet diameter $\lambda_{opt} = 4.51d_j$ by considering a cylinder of length $4.51 d_j$ and diameter d_j to be converted into a sphere of diameter D where $D = 1.89 d_j$. Thus, according to Rayleigh mechanism of break up, the average drop size of a laminar jet is nearly twice the diameter of the undisturbed jet. Different models have been introduced using the capillary instability applied on a liquid jet which have been reviewed by Ashgairz and

Mashayek (1995). The analysis of a liquid round jet is considered outside the scope of this thesis.

The capillary instability of a thin inviscid liquid film, was first studied by Squire (1953), who divided the oscillations, which may happen on the liquid film surface into two modes. These modes are symmetric where the middle plane is undisturbed and antisymmetric where the displacements of the corresponding points on the free surfaces are equal in magnitude and of the same direction. His analysis revealed that the degree of instability for the axisymmetric disturbances is very much less than for the antisymmetric disturbances. He also showed that one of the necessary conditions for instability of the liquid film is the liquid Weber number $We_l = \frac{\rho_l u_j^2 h}{\sigma}$, which is the ratio between the inertia force to the surface tension force to become higher than one. This condition can be defined in terms of the air Weber number, which is the ratio between the aerodynamic forces to the capillary forces, by multiplying both sides of the equation by $\frac{\rho_a}{\rho_l}$; the condition of instability thus becomes $We_a = \frac{\rho_a u_j^2 h}{\sigma} > \frac{\rho_a}{\rho_l}$; where ρ_a , ρ_l , u_j , h , σ are the air density, liquid film density, liquid film velocity, half the film thickness and the film surface tension respectively.

The analysis of Squire (1953) shows that the wavelength of the most unstable wave is proportional to the surface tension of the liquid and inversely proportional to the density of ambient air and the injection pressure which in turn is proportional to the square of the injection velocity. Photographs of the liquid sheet verified this model and the trend for each parameter; however, the calculated wavelength was about half the measured values in some cases. Moreover, the analysis was based on the assumption that half of the injection pressure is lost in the nozzle and that it is kept constant.

Another experimental and theoretical study has been performed by York et al (1953) and quoted by Lefebvre (1989) at nearly the same time as Squire (1953). The aim of their analysis was to study the mechanism of sheet disintegration exposed to air on both sides. Their analysis showed the amplitude of the wave δ along the disturbed liquid film to increase exponentially with time. They claimed that the sheet disintegrates when the amplitude of the disturbance reaches a value equal to the sheet thickness. By using this criterion of sheet disintegration, both the break up length and the break up time were calculated assuming that the initial disturbance δ_0 is known.

Hagerty and Shea (1955) as quoted by Lefebvre (1989) extended the Squire analysis (1953) for a flat sheet to include both the axisymmetric (varicos or dilational) and antisymmetric (sinuous) waves and reported that the antisymmetric wave is always greater than the axisymmetric wave and, consequently, predominant. They also measured the growth rate of the wave by photographing and compared the value with the maximum growth rate calculated from the analysis; quite good agreement has been found. However, their experimental points were not enough to cover the entire theoretical curve.

For the purpose of studying the effect of sheet thickness attenuation on film disintegration, the analysis of Squire (1953) has been applied by Dombrowski et al. (1962) on a flat sheet produced from a single hole spray nozzle where its thickness decreases. The equations of the maximum growth rate and the corresponding wave number have been simplified for three ranges depending on the sheet thickness ($2h$) and the optimum wave number k . They found that, for small density ratio and $kh > 0.63$ which is the case applied to a DISI injector film, both the wavelength and the growth rate are independent of the sheet thickness, thus allowing the analysis to be applied to the whole entire sheet. They also found that for kh less than 0.75, the maximum growth rate of the sinuous wave is always greater than the dilational wave while for kh greater than 0.75, the two growth rates become identical.

Lefebvre (1989) studied the influence of the initial liquid film thickness ($2h$) on mean droplet size and droplet size distribution. Measuring the sheet thickness by a direct needle contact device and the droplet sizes using the light scattering technique, it was found that the Sauter mean diameters for both water and kerosene are proportional to $(2h)^{0.38}$. He mentioned that, as the velocity increases, the liquid sheet disintegrates earlier into fragments and forms nearer to the nozzle tip before rapidly converting into ligaments by the effect of surface tension. These ligaments tend to be thinner and shorter and disintegrate into smaller drops. Moreover, the effect of increasing the viscosity was found by photographing to result in damping of the oscillations taking place on the sheet surface.

Weihs (1978) attempted to analyse the effect of radial spreading of the sheet and the resulting change in thickness, on the sheet stability. He formulated an equation for the antisymmetric wave which was identical to that proposed by Squire (1953), in the case of an inviscid liquid. He found that for liquids with low viscosity, such as water, the change in wave number even at high velocity (100 m/s) does not exceed 20% when

the sheet thickness changes by a factor of 100, confirming that the assumption of inviscid flow is satisfactory. However, for viscous liquids ($1\text{cm}^2/\text{s}$) the wave number of maximum instability is much longer than that predicted for an inviscid sheet. Similar results have been reported from the analysis made by Dombrowski and John (1963).

Li and Tankin (1991) analysed a moving thin viscous liquid sheet in an inviscid gas medium. The results revealed that two modes of instability exist for viscous liquids; aerodynamic mode and viscosity mode in contrast to the inviscid liquid sheets whose only mode of instability is the aerodynamic one. They found that the viscosity of the liquid sheet enhances the instability at low gas Weber numbers below (ρ_g/ρ_l) because the aerodynamic instability disappears. On the other hand, at high Weber number, the aerodynamic instability dominates. It is also found that the axisymmetric disturbances control the instability process for small Weber numbers, while antisymmetric disturbances dominate for large Weber numbers. The same results were obtained by Shen and Li (1996) for an annular viscous liquid jet.

A more recent study has been introduced by Dumouchel et al. (1995), in which a simplified mathematical formula is derived based on the work of Dombrowski and John (1963). This study considered a small interface displacement compared to the wavelength and the sheet thickness with the aim of clarifying the effect of the viscous force on sheet disintegration, relative to the aerodynamics and surface tension forces. The results showed that, as the viscosity increases, the maximum growth rate decreases as well as its corresponding wave number, hence it has a stabilizing effect. They found that the effect of viscosity on the disintegration process of a flat liquid sheet is not a matter of the value of viscosity only but it is strongly related to the values of other physical parameters characterizing the liquid system studied. They defined a parameter

$$M = \frac{\mu \rho_g^2 u_j^3 h}{\rho_l \sigma^2} \text{ where } \mu, \sigma, \rho_l \text{ and } u_j, \text{ are the liquid viscosity, surface tension, density}$$

and velocity, respectively and found that when $M < 0.01$, the effect of viscosity is negligible and the system may be regarded as non viscous. When $M > 0.01$, the mean droplet diameter is directly proportional to $\mu^{\frac{1}{6}}$. Since the viscosity decreases as the temperature increases, the mean droplet diameter decreases at high air temperatures.

To reveal the effect of air impingement (entrained air) on the liquid sheet instability, Takuoka et al. (1985) used a contact needle to the sheet, at conditions of liquid velocity 25 m/s and air velocity 100 m/s. They found that air impingement plays

the role of an agitator and forms a relatively large initial wave on the sheet. They showed that, when the amplitude of the wave is small, the growth rate increases rapidly at higher relative velocities until the amplitude reaches a certain value at which air resistance cannot be neglected and the growth rate is reduced. The waves are then deformed by air resistance and collide with the following waves, disintegrating into drops. Three regions have been distinguished along the liquid film; smooth, wavy and break-up region. However, this classification cannot be identified quantitatively since it depends on the measurement technique as well as the accuracy of the measurement.

Linear instability analysis applied on annular cylindrical liquid sheet. Meyer and Weihs (1987) claimed that the annular cylindrical film could be treated as a full liquid jet or as a two-dimensional liquid jet depending on its thickness. Their analytical investigation of the stability of a viscous annular liquid jet moving in an inviscid medium has defined a critical thickness as a function of air density, velocity, surface tension and viscosity. They demonstrated that if the film thickness is less than that critical thickness the annular liquid film could be represented by a two-dimensional liquid sheet and the most unstable wave becomes the antisymmetric waves.

Shen and Li (1996) examined an annular liquid sheet, using the linear instability analysis, subjected to air from both sides. When comparing the results for a flat sheet with those for a conical sheet, they found that the curvature effect leads in general, to an increase of the disturbance growth and may not be neglected for the break up process. Hence, an annular jet with a sufficiently small thickness tends to break-up faster than the corresponding plane liquid sheet and that is in agreement with Mayer and Weihs (1978). Their analysis showed that there is a critical wave number below which the surface tension becomes the source of instability while above it the instability is suppressed by surface tension and promoted by the aerodynamic interaction between the liquid and the gas phase. The effect of viscosity was found to be similar to that in the liquid plane sheet. In this study, the swirl motion and the effect of curvature on the optimum wavelength, relative to a flat sheet, were not considered.

On the experimental side, Kawanto et al. (1997) studied the wave motion on the gas liquid interfaces in an annular liquid sheet and its relation to the atomisation process using a high speed camera at 4000-5000 frame/s and an electro-optical displacement device connected to an FFT analyser. They produced an annular water sheet with a velocity of 0.5 m/s and a coaxial air flow from outside and inside with velocities of up

to 35 m/s from a slit of 0.2 mm. Their results showed that the annular sheet oscillates periodically and that the wave on the axial direction is a progressive wave contrary to the circumferential direction where it is a standing one. They also found that when the air velocity is relatively low, a regular pattern of sheet break-up occurs while at high air velocity the structure becomes more complicated. The comparison between the measured wavelength and the calculated one from the linear instability analysis showed good agreement at the low air velocity but less so at the high air velocity.

Ponstein (1959) has introduced the analysis of instability of a hollow cylindrical jet by applying the hydrodynamic theory. The effect of viscosity and the inertia of the surrounding and contained air has been neglected. He pointed out that atomisation occurs due to unstable perturbations although not every unstable perturbation leads to break-up. Since the linear theory of hydrodynamic stability is valid only for infinitesimally small perturbations, then after a finite interval of time the unstable perturbations may grow to a finite value for which the linear theory of hydrodynamic stability is not valid anymore. Therefore, a flow field disturbed by unstable perturbations may adopt itself to another stationary state without the occurrence of atomisation.

The application of the linear instability analysis to a swirl atomizer has been presented by Cousin et al. (1995). Their analysis was based on a cylindrical sheet rather than a conical one because of the difficulty to resolve the problem in a conical sheet and the negligible effect of the internal diameter of the cylinder on the results. They assumed that the circumferential velocity component and the sheet thickness decrease with downstream distance from the nozzle and that the internal diameter increases. The maximum growth rate was calculated at different locations downstream from the nozzle to identify the location of maximum growth rate at which break-up takes place. By assuming that the cylindrical sheet breaks up into circular ligaments and using Rayleigh's theory and the maximum entropy formalism proposed by Sellens and Brzustowski (1985) and Li and Tankin (1987), they calculated the mean droplet diameter and the droplet size distribution from the primary break-up; satisfactory agreement with the experimental results was obtained using a Malvern particle sizer. Despite this, the results presented were limited and did not show the extent to which the linear instability analysis is valid for different operating conditions and various physical properties of liquid and air. Moreover, the maximum injection pressure used in this

study was 10 bar which is much less than the normal operating pressure of DISI swirl injectors.

Another study described by Panchagnula et al. (1996) investigated the effect of air velocity inside and outside an inviscid cylindrical liquid sheet and that of the swirl motion on the maximum growth rate. They found that the air velocities affect the axial and circumferential wave number as well as the maximum growth rate. However, the presented results showed that in the case of no swirl there are still circumferential wave numbers which affect the maximum growth rate; obviously this is not physically correct. Since if there is no relative motion between the liquid sheet and the air, there is no source of instability.

Until recently the only study that used the linear instability analysis for DISI spray injectors was that of Schmidt et al. (1999) which was based on a flat sheet rather than a cylindrical one. In the last few years, however a number of studies of DISI spray models based on the linear instability analysis using either flat sheet or hollow cylindrical films were published.

1.6.2.6 Non-linear instability analysis

The non-linear term in the governing equations has an effect on the results of the film disintegration analysis. This term predicts the deviation from a sinusoidal wave distortion and the subsequent deviation from a monodispersed spray. Therefore, the satellite droplet formation could be explained by a second-order modulation amplitude perturbation analysis. Details about the implementation of the non-linear analysis are reported by Sirignano and Mehring (2000).

A non-linear analysis of the initial small disturbances at an interface separating two fluids of different velocity and density, including the surface tension effects, was investigated by Rangel and Sirignano (1988) using the vortex sheet discretization approach. They found that for density ratios less than 0.05 the surface tension force plays a decreasingly important role as the stabilising agent of these low-density ratio inertially stable waves. Therefore, continuously decreasing the density ratio causes more stable travelling waves in agreement with the linear theory prediction. Another interesting study by the same authors (1991) focused on a sheet liquid film for axisymmetric and antisymmetric waves. Their results showed that the minimum thickness of the sheet occurs each half wavelength for the sinuous waves and each full wavelength for the dilational waves. It may be worthy mentioning that the non-linear



analysis depends on the linear analysis since the output wavelength and initial amplitude should be inputs to the non-linear analysis.

Kinnersley (1976) has extended the exact solution of Crapper (1957) for capillary waves of infinite depth on a fluid, to be applicable for finite depths. He found two finite depths at which elliptic functions can be solved which were interpreted as large amplitude symmetrical and antisymmetrical waves on the fluid sheet.

1.6.3 Characteristics of Swirl Injector Sprays for Direct Injection Engines

Swirl injectors produce hollow cone sprays whose characteristics depend on the fuel properties and droplet velocities and diameters immediately after atomisation and later on, on the interaction between droplets and the surrounding air in the chamber as well as between the droplets themselves (droplet-to-droplet interaction). These characteristics, either define the overall spray structure also known as ‘macrostructure’ such as the dispersion, the spray penetration, the spray cone angle and the spray pattern, or quantify the internal spray structure known as ‘micro structure’ such as the mean droplet diameter and velocity, the droplet size and velocity distribution and the fuel vapour concentration and its distribution. The techniques used for identifying the spray geometry or the macroscopic structure are simpler than those used to quantify the microscopic structure of the spray. Some of the published work on the spray characteristics and relevant measurement techniques are tabulated in Table 1-3

1.6.3.1 Spray dispersion

The degree of fuel dispersion is determined by the ratio of the volume of the spray to the volume of liquid contained within it. The spray dispersion from swirl type injectors varies during spray development and also as a function of the operating conditions. For example, Zhao et al. (1999) and Shelby et al. (1998) divided the spray during its development into three phases just after the injector delay time: the liquid jet phase, the wide hollow spray-cone phase and the fully developed spray phase. During the last fully developed phase the hollow cone structure has an almost constant spray angle. A low-pressure zone is formed inside the spray cone that entrains air and small droplets into the central hollow cone, as has been discussed by Kano et al. (1998) and Hentschel et al. (1999). The fully developed spray geometry is not much affected by the injection pressure but is affected by the ambient pressure. The conical spray collapses

closer to the nozzle exit, causing a decrease in spray width, as the gas density increases according to VanDerWeg and Hochgreb (2000b). Davy et al. (2000) studied the effect of fuel composition on spray characteristics in a firing DISI optical engine. Their results showed that when a typical multi-component fuel distribution is used under warm engine conditions, during the induction process, the spray collapses due to the fast boiling (flashing) of some of the high volatility component contained within the gasoline blend.

Hoffman et al. (1997; 1998) developed a mechanical system, a spray patternator, to measure the temporal and spatial variation of the spray mass flux which is based on the collection of the fuel injected over many injections at certain locations within the spray during different time steps. Yue et al. (2001), on the other hand, introduced an optical technique to estimate the instantaneous spatial distribution of the mass flux within 1 mm from the nozzle exit using the X-ray absorption method.

The presence of large droplet number densities or liquid volume fractions makes the characterisation of the spray behaviour from PDA measurements quite difficult. The difficulty arises when liquid travels through the control volume without being measured as a result of either weak intensity of the scattered light (due to beam attenuation) or the existence of non-spherical droplets (e.g. ligaments). Therefore, the characterisation of the spray behaviour from the droplet size distribution using results from the subset of the measured liquid mass can be misleading. This problem is particularly evident when measuring the mass flux.

1.6.3.2 Spray penetration

Spray penetration has been defined by Lefebvre (1989) as the maximum distance the fluid reaches when injected into stagnant air. It is worth mentioning that the penetration of the spray is much greater than that of a single droplet. The early injected droplets, from the nozzle impart their energy to the surrounding gas which begins to move with the spray, thus offering a less resistance to the next droplets, which can consequently penetrate further. Spray penetration is mostly defined by direct imaging of the spray using either forward or backward lighting.

The spray penetration from DISI swirl injectors has been studied by many researchers, for example, Kume et al. (1996), Zhao et al. (1997), Tomada et al. (1997) and Arcoumanis et al. (1999). However, the method of determining spray penetration varies from the simple manual method to more complicated methods using imaging

analysis; there is thus no standard way to estimate the spray penetration. Caution has to be exercised in identifying spray tip penetration from images of this type of spray as the spray tip appears only during the early stage of spray development; later on, as the spray develops, the tip starts to disappear as the light reflected from the main spray overshadows that reflected from the tip of the spray. Abo-Serie et al. (1999) have suggested considering the spray tip penetration during the early stage of spray development while using the location of the maximum width as representative of the spray penetration soon after as the spray starts to form a conical shape. Their results showed that spray penetration increases with the injection pressure until the end of injection, while after the end of injection this increase is not distinct. The effect of back pressure on spray penetration has been studied by for example Kume et al. (1996) and Ipp et al. (1999), who found that there is a decrease in spray penetration when the ambient pressure increases. The effect of air temperature was reported by Ipp et al. (1999) to have only a small effect on spray penetration. Increasing the fuel temperature causes an increase in the spray penetration according to VanDerWege and Hochgreb (2000a) and Araneo et al. (2000).

1.6.3.3 Spray cone angle

Most of the work published up to day did not identify where and how the spray angle was measured. The fact is that the aerodynamic interaction of the spray liquid causes the slope of the spray envelope to change axially from one point to another. To circumvent this, the spray angle is often given as the angle formed by two straight lines drawn from the nozzle exit to cut the spray contours at some specified distance from the nozzle exit. The interest in measuring the spray angle is due to its link to liquid radial dispersion inside the chamber. While larger a spray angle is required during full load operation in order to achieve homogeneous combustion, a relatively narrow spray angle is preferable during part load operation to achieve a controlled stratified mixture around the spark plug. Itoh et al. (1999) showed that the smoke level decreases almost linearly as the spray angle increases until it reaches an asymptotic value (for their system it was 70 deg). Similar trends have been observed by Ohsuga et al. (1997) when the swirl injector was positioned in a close spacing configuration and they commented on the importance of the spray to be dispersed in the air and not to form a liquid film on the piston surface, for preventing smoke formation.

The cone angle of swirl pressure atomizers is not only affected by the swirl atomizer internal geometry but also by the operating conditions. A higher spray angle is achieved by increasing the swirl velocity relative to the axial velocity which could be achieved by increasing the inclination angle of the swirler. Cooper and Yule (2001) showed in a large-scale pressure swirl atomiser that the swirl velocity relative to the axial velocity decreases as the nozzle length increases. Accordingly, it is expected that the spray angle will decrease with increasing nozzle length. Nogi et al. (1998) also showed that a narrow spray angle is accompanied by higher spray droplet velocities that cause more fuel impingement on the cavity surface and lead to a higher smoke level; in the case of a wider spray angle, the droplets have lower momentum and therefore, may not reach the spark plug. Accordingly, spray angle has to be optimised separately for engine operation under homogeneous and stratified conditions.

Araneo et al. (2000) have shown that the spray angle decreases along the spray axis while Shelby et al. (1998) showed that the spray angle increases during spray development until it reaches its maximum value and then decreases slightly after the spray becomes fully developed, due to the difference between the air pressure outside and inside the spray. Injection pressure seems to have no effect on spray angle as concluded by Nouri et al. (1999) and Ren and Nally (1998); however, a slight decrease in spray angle with increasing injection pressure has been reported by Shelby et al. (1998). Increasing the back gas pressure leads to a decrease in spray angle due to the pressure difference inside and outside the spray as reported by Shelby et al. (1998), Nogi et al. (1998) and Ipp et al. (1999). No effect on the spray angle, however, is observed if it is measured close to the nozzle, as reported by Araneo et al. (2000) and Arcoumanis et al. (2000). Araneo et al. (2000) also showed that the spray angle increases with increasing nozzle temperature. Ambient temperature, on the other hand has only a slight effect on spray angle according to Ipp et al. (1999).

1.6.3.4 Droplet size and velocities

Droplet size and velocity measurements within the spray are very important in engine applications as they are indication of how fast the droplets penetrate and the degree of the evaporation process which determines the quality of mixing and affects the quantity of engine-out emissions. To characterise the droplets it is standard practice to use optical techniques for measuring their sizes and velocities within the spray. Optical methods can be classified into imaging and non-imaging ones. A variety of

optical methods have been employed for spray analysis. Each method has its own advantages and limitations, but they all have the important attribute of allowing size measurements to be made without disturbing the spray as when a probe is inserted into the spray. Spray droplet sizes and velocities have been quantified by many researchers, for example Kume et al. (1996), Zhao et al. (1997), VanDerWeg and Hochgreb (2000a) and (2000b), Wigley and Pitcher (2001) and many others. A review of the techniques used for spray investigations and the main findings are tabulated in Table 1-3.

1.6.3.5 Spray characteristics inside the DISI engines

The successful operation of modern DISI engines depends on optimising the combustion chamber shape with the in-cylinder flow field to achieve a well dispersed spray and complete air utilization during the homogeneous charge full load operation and to provide a well contained pocket of combustion mixture during stratified charge part-load operation. The design of the combustion chamber, the piston crown, inlet manifold and ports and inlet and exhaust valves locations are among the main factors affecting the air flow pattern inside the chamber and the fuel distribution. In the case of the wide spacing combustion system, the piston surface roughness and temperature and the injector location relative to the spark plug are additional important factors.

Many researchers have discussed the use of the swirl injector in a firing engine and their conclusions are relevant to their specific engine configuration and operating conditions for example, Stanglmaier et al. (1998) and Gold et al. (2001) showed that the intake air flow has a significant influence on deflecting the conical spray and that influence increases as the engine speed increases. Davy et al. (2000) showed also that injection timing during the early injection DISI operation affects the spray homogeneity close to the spark plug due to the flash boiling of the volatile components in the gasoline fuel. Urushihara et al. (1996) and Baumgarten et al. (2001) indicated that there is an optimum swirl/tumble intensity to achieve the required stratified or homogeneous mixture. Iwamoto et al. (1997) and Kano et al. (1998) showed that the timing of ignition is an important factor. For the system used by Mitubishi where the behaviour of the fuel spray is controlled by impingement on the piston cavity and reverse tumble, Iwamoto et al. (1997) explained that there is a time interval between the end of injection and spark plug ignition achieving stable combustion. If this interval is too short, an over rich mixture is prepared around the spark plug while when it is too long fuel diffuses making the mixture around the spray plug too lean; in both cases misfire may occurs.

1.7 PRESENT CONTRIBUTION

This study focuses on characterising the fuel injection process from DISI pressure swirl atomisers. The flow characteristics have been investigated within three regions: inside the injector nozzle, close to the nozzle exit and further downstream of the nozzle exit. The effect of the flow characteristics from one region on the next has also been discussed.

Inside a transparent nozzle, attached to the production nozzle, it was possible to visualise a liquid film soon after the needle reaching its full opening. The mean temporal and spatial variation of the film thickness have been identified taking into account the error arisen due to the change of the light pathway which is attributed to the curvature of the inter-phases and the difference in the refractive indices of the transparent nozzle, the liquid film and the air. It was also possible to identify the swirl velocity component relative to the axial velocity component at different locations inside the transparent nozzle from the marks appearing on the outer surface of the liquid film and by means of image analysis techniques. The data provided in this part are complete to validate any mathematical model aiming to predict the film thickness and the mean injection velocity from swirl type atomisers.

Close to the nozzle exit (up to 2mm), high magnification images of the fuel flow showed that the liquid fuel exits the nozzle initially parallel to the nozzle axis and then diverges radially away from the nozzle exit until it reaches an asymptotic divergent angle that is equal to the nozzle spray angle. By balancing the forces acting on the liquid film outside the nozzle it was possible to derive a total differential equation to represent the film divergence. Solving this differential equation together with the droplet penetration profile after the film breaks-up showed similar profile to that in the images and provided an estimate for the film break-up length after taking into account the images of the flow profile near the nozzle exit. Images close to the nozzle exit also showed three-dimensional waves are developing on the liquid film surface. These waves have been mathematically modelled by the linear instability analysis (LIA). The LIA model has been deduced for a swirling hollow cylindrical liquid film subjected to air flow from both sides in order to identify the highly unstable wave and to provide an estimate for the mean droplet diameter soon after the film breaks-up representing the atomisation process.

The liquid film divergent model together with the atomisation model represent the missing link between liquid film formation and spray initial conditions. This link is always a challenging problem in fuel injection systems and much of the recent research neglects this stage and focused on validating spray models with spray data, these models are rather limited and can easily fail when the design parameters or the operating conditions change.

Spray characteristics have also been identified globally by imaging and locally using 1-D PDA system under atmospheric conditions and inside a constant volume chamber under high pressure/temperature ambient conditions. Detailed temporal and spatial distributions of droplet mean diameter and velocity have been presented.

The data provided in this thesis are complete to test and develop spray models from this type of injectors. In addition to the validation of existing models, the thesis introduces two sub-models to characterise the flow located close to the nozzle exit, the first model predicts the film divergence and the second represents the atomisation process. There are currently a number of publications focusing on spray characteristics, however the complete study that includes the injection system, the film formation inside the nozzle and close to the nozzle exit and the atomisation process as well as their effect on spray structure is still under investigation and requires further research. The quantitative data obtained for the liquid flow conditions at nozzle exit and the spray characteristics in addition to the two sub-models, have already been employed to develop and validate a complete CFD model. That model includes 1-D model for the fuel injection equipment, 3-D model for the flow inside the injector up to the nozzle exit, 2-D-two-phase model for the liquid film inside the swirl injector nozzle, the developed film divergent model, the developed atomisation model and the conventional spray model by Gavaises and Arcoumanis (2001) and Gavaises et al. (2002).

The work presented in this thesis focuses on characterising the fuel injection and spray in stagnant air and does not include the operation of the injector inside an engine. However, the design process of the combustion system has to be based on a fundamental knowledge of the fuel/air mixture preparation mechanism. To consider the spray development inside the combustion chamber and the subsequent mixing, the airflow pattern has to be optimised to achieve the optimum combustible mixture for each mode of operation and this optimisation is different for each combustion system.

1.8 THESIS OUTLINE

The thesis contains five chapters including this introductory chapter. In chapter 2 the experimental set up, including the specifications of the mechanical equipment used and the optoelectronic instrumentations are described. The details of the timing system and the software developed to drive the injector are also described. Following a detailed description of the test rig, images of the liquid film inside an extended transparent nozzle are presented. The mathematical method used to identify the film edge and the film thickness is then explained. Finally the effect of injection pressure on the film thickness along the nozzle is estimated. The experiment carried out to identify the error in the estimation of the film thickness due to refractive index mismatching between the transparent nozzle and the fuel is described and the error has been calculated. Pressure fluctuations measurements inside the common rail pipe and very close to the nozzle exit are also presented.

Chapter 3 presents the main theoretical part of the thesis which includes two sub-models the first estimates the film divergent profile prior to atomisation and the second considers the atomisation process using the linear instability analysis. The assumptions and the derivation of the final differential equation that controls the liquid film divergence outside the nozzle from the balance of the forces affecting the film during its penetration are presented. A parametric study on this film divergence model is then considered and described. Having identified the film divergent profile, the break-up length is then calculated by taking into account the measured spray angle. The linear instability model is then introduced after discussing the assumptions used and the derivation of the characteristic equation. The criterion of the sheet break-up and the assumptions made are also presented. The effect of different parameters on wave formation and ligament sizes has then been examined and presented.

In chapter 4, the experimental results of spray imaging at atmospheric conditions and in a constant volume chamber are introduced. The analysis for the images to characterise the overall spray structure, with special attention to the early stage of spray development, is described. The main parameters of the phase Doppler Anemometer, the most common technique for measuring droplet size and velocity, are provided. The droplet size and velocity of the droplets within the spray are then measured and their mean temporal and spatial variations are calculated and presented for different operating conditions.

Chapter 5 summarizes the main conclusions of this work and provides some recommendations for future research into swirl type injectors for direct injection gasoline engines.

Table 1-3 Spray characteristics of pressure swirl atomisers

Author(s)	Experimental Set-up + Technique	Main Findings
Kume et al. (1996)	- swirl type injectors - Mitsubishi upright straight intake port and spherical compact piston cavity - visualisation - laser sheet of light - PDA	- axial and swirl velocity are approximately of the same order for optimized swirl injector. - intense air flow and turbulence at the centre of the hollow cone are observed - at high ambient pressure the spray takes the shape of a compact solid jet with decreased penetration
Zhao et al. (1997)	3 different injectors with different spray angle - fuel used is Vicor-B - Atmospheric conditions 2-D laser Mie scattering - Nd-YAG laser - PDA	- with increasing injection pressure penetration increases; however at the end of injection this increase is not distinct - two counter rotating recirculation zones were observed at the head of the spray towards the end fuel injection - the circumferential distribution of the spray has irregular shape for a large spray cone angle injector compared to the small angle one. - complex relationship between the injection pressure and droplet sizes and velocities. - at the spray centre, small droplet sizes with high velocities are observed.
Tomoda et al. (1997)	- Piezoelectric actuator (PZT) hole nozzle and swirl nozzle - spray visualization - Laser diffraction method	- spray penetration of the swirl injector and hole nozzle fit well the momentum theory - swirl injector has a quite narrow droplet size distribution compared to the hole nozzle type - for evaporating the droplets produced from the swirl injector, partial heat transfer from the piston is required while in the hole nozzle heat transfer from the piston is mainly dependent. - misfire happens if the pressure decreases to 80 bar because of the deterioration of the evaporation process. - swirl injector is more dispersible and preferable to produce stratified mixture, however precise quality control of nozzle tip processing is required
Shelby et al. (1998)	- square cross section optical engine (CR=8) - compressed nitrogen and hydraulic accumulator to generate up to 80 bar injection pressure - Chrysler Corp 50 ° and Zexel Corp 20° and 60° swirl injectors are used - Planer laser induced fluorescence (PLIF) - CCD camera with 50 ns exposure time	- spray phases could be divided into four phases : - delay phase ~ 360 μs - a solid jet phase (about 150 μs for 50 bar) - a wide hollow cone phase - a fully developed spray - increasing the injection pressure leads to slightly decrease in the spray angle - increasing the ambient pressure leads to a decrease of the spray angle due to the increase of the pressure difference inside and outside the spray - atomisation is much poor at low injection pressure

Author(s)	Experimental Set-up + Technique	Main Findings
Hochgreb and VanDerWege (1998)	• square cross section optical engine(CR=8) • early injection only (homogeneous mixture) • 5 MPa injection Pressure • 60° swirl injectors are used • Iso-octane fuel • PLIF (dopant: Acetone and 3 pentanone)	◆ PLIF and scattering images showed that by increasing the nozzle temperature and decreasing the ambient pressure, flash vaporisation of the volatile component in the fuel changes the hollow cone spray structure into a solid cone. ◆ image intensity showed that about 40 % reduction in the droplet diameter is achieved due to flash vaporisation. ◆ significant change in the spray structure is noticed when the temperature is 20 ° above the bubble temperature of the mixture.
VanDerWege and Hochgreb (1998)	• square cross section optical engine(CR=8) • early injection only • two swirl injectors • 5 MPa injection Pressure • Iso-octane, Acetone , 2-butanone and 3- pentanone • PLIF • CCD camera with 50 ns exposure time • planer Mie scattering	◆ different injectors may require different amounts of superheat to exhibit flash boiling because boiling depends on nucleation process ◆ calculations of bubble point and single component tests suggest that a superheat of 20 K is required for flash boiling to be noticeable. ◆ collapsing of the hollow cone spray, at higher temperature, may occur due to flash boiling at the interior portion of the injected fuel
VanDerWege and Hochgreb (2000a)	• The same engine as above • Multi component fuel • PLIF • PDA	◆ effect of fuel temperature and fuel volatility on spray structure could be divided into three regimes ◆ (1) slow evaporation when the fuel temperature is lower than the mixture boiling temperature ◆ (2) non-disruptive evaporation during which most of the volatile component exists at the axis of the spray ◆ (3) disruptive evaporation or flash boiling during which the initial spray behaviour is affected by the vaporization of the light component causing the spray to become compact ◆ increasing the ambient density causes a decrease in both spray penetration and width ◆ increasing fuel temperature causes an increase in spray penetration but a decrease in spray width with an increase in the quantity of vapour located in the spray centre

Author(s)	Experimental Set-up + Technique	Main Findings
VanDerWege and Hochgreb (2000b)	The same as above	- Droplet mean diameter increases as the air density increase. - The correlation proposed by Lefebvre (1989) for Sauter mean diameter has been modified to take into account the chamber pressure and temperature - Spatial and temporal variation in mean droplet diameter were found to be relatively small compared to the variation due to the change in operating conditions. - PDA measurements of dense spray are reliable at the edge of the spray - Under conditions of high potential for vaporization, an additional droplet diameter reduction has been observed and this believed to be due to flash boiling of the more volatile components in the fuel
Araneo et al. (2000)	- constant volume chamber - injector nozzle is heated and the fuel inside is assumed to be in thermal equilibrium with the nozzle surface - TV camera is used for spray visualisation - PDA was utilized to measure droplet velocities and diameters. - PDA measurements achieved at 10, 15 and 30 mm and 3000 measurements were acquired $P_{\text{ambient}}=0.4$ to 8 bar $T_{\text{nozzle}}=20$ to 120 oC $P_{\text{injection}}=70$ bar	- the cone angle decreases along the spray axis - as the nozzle temperature increases, a wider nozzle spray angle is observed but the spray undergoes a rapid decay along the spray axis (spray collapses faster) - the chamber pressure does not affect on the nozzle spray angle. - axial velocity decreases towards the edge of the spray - at 30mm away from the nozzle exit, by increasing the nozzle temperature, the mean droplet velocity increases however the maximum mean velocity remains the same - at 15 mm away from the nozzle, with increasing the nozzle temperature to 120 °C the mean droplet diameter becomes half the value of the 20 ° C injector temperature
Kano et al. (1998) and Kano et al. (1999)	4-valve 1998 cc engine - C.R.=10 - injection Press. =12 MPa - wall guided spray - swirl injector - helical and straight intake port with swirl control valve - compact CCD video camera located in the place of the spark plug 6 sampling valves and gas analyser with fast response time	- there is an optimum time delay between injection and ignition which achieves A/F from 10 to 20 at the spark plug location. A drop in IMEP is observed if this delay increases or decreases. - swirl intensity should be optimised otherwise, a dispersion of the mixture from the spark plug occurs - the droplet sizes should be smaller than 15 μm to follow the air motion - if the spray parameters are optimised to minimise mixture fluctuations cycle to cycle fluctuations becomes too small to cause torque fluctuations.

Author(s)	Experimental Set-up + Technique	Main Findings
Ipp et al. (1999)	- constant volume chamber - swirl injector 90 ° - max. air Temperature and pressure are 800K and 5 MPa - constant injection pressure using a diaphragm system - Iso-octane fuel is used with seeding of benzene and treithylamine2-D Mie scattering and LIF - Nd:YAG laser produces 200 µm laser sheet thickness - two CCD cameras.	- increasing the ambient pressure causes the liquid and vapour spray to become compact - increasing the ambient pressure leads to a decrease in both the spray penetration and cone angle. - increasing the ambient temperature has slightly effect on spray penetration and cone angle. - the fuel vapour phase follows the liquid fuel phase throughout the injection process.
Gold et al. (2001)	- top entry optical engine - LDA - PDA - LIF	<u>During the early injection mode</u> - different injection timing results in different mixture formation processes. - as the time between injection and ignition increases the combustion stability improves - fuel vapour is carried out from the piston bowl by the reverse tumble air motion during the early injection.
Gold et al. (1999)	- optical engine - swirl type injector - atmospheric and high pressure chamber - LDV 2-D PDA system - PIV	- leading edge of the spray contains larger droplets. - small droplets are located in the central region of the fully developed spray. - examining six injectors showed similar penetration trends; higher penetration rate during the first millisecond than thereafter. - the wide low penetration umbrella structure of the spray improves air entrainment at high fuelling. - larger droplets can be found on the outer edge of the spray envelop.
Ismailov et al. (1998)	- swirl injector - Pinj=70 bar - LDA - PDA	- instantaneous flow rate has been measured using LDA. - close to the nozzle exit <2mm , velocity at the centre of the spray is less than at the edge. - D₃₂ at 45 mm from the nozzle varies in a range of 20 microns.

Author(s)	Experimental Set-up + Technique	Main Findings
Wigley and Pitcher (2001)	- Mitsubishi DISI swirl injector - Injection duration = 4.2ms - Injection frequency of 10Hz - PDA	- droplet sizes have been classified into two classes : the first are the droplets whose diameter are less than 20 microns while the second class are from 20-40 microns. - the dynamics of the smaller droplet class can be considered indicative of the air flow generated by the injected spray. This class also has significant population densities in the axial and extreme peripheral regions of the spray. - the larger size class has considerably lower sample densities and they are indicative of the hollow cone nature during the fully developed stage.
Davy et al. (2000)	- fuel used was Unleaded gasoline, Iso-octane and a mixture of Iso-octane, Iso-pentane and n-nonane. - swirl injector of 55 deg. Spray angle and spark plug are located at the cylinder centre (close-spacing) - Mie scattering - PLIF	- When using the unleaded gasoline the spray structure changes from the hollow cone to a filled column under full load engine conditions which has been attributed to the flash boiling of the high volatility components. - The collapse of the spray at higher ambient temperature has been attributed to flash boiling as well as to the low-pressure region inside the spray.
Iwamoto et al. (1997)	- wide spacing 4 cylinder Mitsububishi DISI engine - tumble and piston impingement - swirl type injector P_{inj}=50bar	- swirl velocity of the fuel adds a swirl component to the in-cylinder air flow. - DISI engines have higher vibration than PFI because of the higher compression pressure. - there is optimum timing of ignition after the end of injection to achieve stable stratified combustion and avoid misfire. - soot was found due to the presence of fuel layer on the piston surface and lowering the tip penetration was recommended to avoid this layer. - decreasing the injection pressure and enhancing the swirl velocity decreases the spray penetration.

Author(s)	Experimental Set-up + Technique	Main Findings
Tomonori, et al (1996)	• 4-valve optical engine with control on inlet swirl and tumble intensity • LDV • LIF	♦ the necessary conditions for axial mixture stratification in the cylinder is to limit the tumble ratio. ♦ large swirl is not always effective for stratification. ♦ tumble air motion enhances the homogeneity of the mixture and reduces the NO_x during the stratified combustion in a wide range of fuel injection timing
Stanglmaier et al. (1998)	• Zexel swirl injector 60 ° • Optical single cylinder engine • flat piston • injection at 90 ° ATDC (early Injection) • LDV measurements at 38 mm from the cylinder head • laser sheet using Nd-YAG thickness=3mm • CCD camera • LDV system for cylinder flow measurements	♦ the intake air flow deflects the fuel spray and deforms the cone shape. This deflection increases with increasing the engine speed. ♦ the formation of the spray reverse vortex is inhibited with increasing the engine speed. ♦ Spray penetration is slightly affected by the engine speed ♦ the ratio of the injection momentum to the intake air momentum at 750 rpm is approximately 1:13 which decreases with increasing the engine speed
Alger et al (1998)	• Zexel swirl injector • 60 degree spray angle • 4-valve optical engine • flat piston • tumble air flow • Mie scattering • fuel vapour measurements using fibre optics	♦ in the instance of injection while the intake valve are fully opened, the fuel spray immediately deviated and the intake side of the cone is suppressed and the exhaust side is entrained down towards the piston ♦ the amount of tumble in the cylinder has a significant effect on the local equivalence ratio.
Rudolf et al (1998)	• 4-valve DISI optical engine has a tumble air motion with an installed swirl injector centrally located. • the study includes only injection during induction stroke 90 ° ATDC • laser sheet droplet imaging (Mie scattering) • One component LDV across one section of the cylinder/	♦ the intake flow deflects the fuel spray and deforms the cone shape. This deflection significantly augments as the engine rpm increases. ♦ during the injection event and for a short time thereafter, the air flow velocities influenced by the injected spray, however the axial velocity during the intake stroke appears to be dominated by the intake flow. ♦ during the compression stroke the mean axial velocities of the flow for injection and non-injection cases start to deviate from each other as the piston approaches the TDC. The axial velocity becomes less with the injection case.

Author(s)	Experimental Set-up + Technique	Main Findings
Lee et al (2000)	- firing optical single cylinder equipped with swirl and/or tumble generation cylinder flow - pressure swirl injector - PLIF technique is used - exhaust hydrocarbon measurements	- reduction in Hydrocarbons was observed during the engine cold start when introducing tumble and/or swirl during the intake stroke. - the best cold start in terms of the accumulated hydrocarbons is achieved with tumble flow followed by the swirl and finally without both. - the optimum ignition for lowest accumulated hydrocarbons is delayed by 15 degree with swirl and tumble than that without any swirl or tumble. Therefore more time becomes available for fuel vaporization.
Baumgarten et al (2001)	- test bench and real engine - LIF, PIV - swirl injector, rail pressure is controlled by engine rpm - forward tumble motion and specific piston geometry - FEV combustion system	- Tumble air motion shows substantial benefits with regard to emission behaviour at part load as well as under full load especially for HC & soot. 13% fuel reduction was achieved compare to the corresponding PFI whilst meeting EU4 emission targets. - exhaust gas after treatment was used.
Koga et al (2001)	- single cylinder with shell-shaped cavity piston to control mixture formation - high pressure fan-shaped injector - first nine cycles are studied	During cold start - the engine out HC is mainly due to cylinder wetting by the high boiling point components of the fuel. - increasing the injection pressure reduces the cylinder wetting due to the improvement in fuel atomisation. - injecting fuel late during the compression stroke reduces the required injection volume and cylinder wetting and accordingly HC. - injecting heated fuel and late opening of the intake valves reduces the injection volume and HC
Ortmann et al (2001)	- single cylinder optical engine and also real single cylinder engine - multi-hole and swirl injectors - wall , air and spray guided systems - LIF - Schlieren - light scattering technique - shadowgraphy	- multi-hole injector produces constant spray geometry and only varies in penetration when the ambient conditions changes - in wall guided system, the higher air entrainment of the multi hole injector leads to less soot formation at all injection pressures. It also causes wider ignition window when using air guided spray system with injection pressure above 5MPa

CHAPTER 2

NOZZLE AND NEAR-NOZZLE FLOW VISUALISATION

2.1 INTRODUCTION

There are a limited number of studies published in the open literature on the flow characteristics in the injection system of DISI engines, especially inside the nozzle and near the nozzle exit of a swirl injector. Most of the experiments have focused on characterising sprays from different injectors under different injection pressures and air ambient conditions, but with limited investigation of the injection conditions at the nozzle exit. All existing mathematical models that simulate the DISI spray have assumed previously known initial conditions from either simple analysis of swirling flows (Lefebvre, 1989; Cousin et al., 1998) or by using two-phase flow CFD complicated models (Arcoumanis and Gavaises, 2000). However, there is very little experimental information available to validate these models. Therefore, the purpose of this study is to identify experimentally the flow conditions and the film thickness at the nozzle exit, created as a result of the centrifugal forces associated with the swirl motion of the liquid inside the injector nozzle. This study will provide a better understanding of how the flow develops inside the injector nozzle and will identify the initial conditions for the conventional CFD spray models. Another major purpose is to provide a quantitative analysis which could allow verification of a complete mathematical model that includes a one-dimensional flow sub-model inside the injection system, a three-dimensional single phase sub-model for the flow inside the injector and a two-phase flow sub-model inside the injector nozzle. This study is also expected to be of use to the engineers interested in improving the design of the injector nozzle since it includes results on the effect of some design parameters and operating conditions on the nozzle flow and spray characteristics.

This chapter presents a description of the injection system, instrumentation, imaging system and the control unit. Following this description, the flow conditions are

presented inside the injector nozzle and very near to the nozzle exit at different times from the start of injection by means of direct imaging using a CCD camera and high magnification lenses. This method has no effect on the flow conditions, however special attention has been paid to consideration of the errors associated with light path distortion due to light refraction when using fluids with different refractive index than the transparent nozzle material (Perspex). The mean axial velocity at the nozzle exit has been calculated by means of the film thickness and the average injection flow rate. In order to calculate the swirl velocity component, the flow direction has been visualised by considering slightly longer exposure times and focusing on the outer surface of the liquid flow. Through this method, the flow angle inside the transparent nozzle and at the nozzle exit has been identified which allows the swirl velocity component to be calculated. The liquid film profile has also been identified as well as the spray angle at different times and injection pressures. The atomisation of the liquid film has also been discussed and the pressure fluctuations inside the common rail system have been measured and analysed.

2.2 INJECTION SYSTEM AND THE TRANSPARENT NOZZLE

The injection system used in this study is a common rail type connected to four pressure swirl injectors. In this system the fuel is pumped into a common rail pipe and the fuel is injected when the needle opens. The system used is a prototype, which simulates the production system incorporated in engines when the electrically driven pump is replaced by the engine's camshaft. A transparent nozzle hole extension has been attached to the end of the injector nozzle in order to visualise the flow prior to injection.

2.2.1 Mechanical Components of Injection System

A schematic diagram of the mechanical connections of the system is shown in Figure 2-1. A common rail pipe (1) is connected to four swirl injectors (2). The fuel is supplied from the fuel tank (3) via a filter (4) by means of gravity to a three-piston type pump (5) which is driven by a 2800 rpm single phase induction motor (6). The common rail pipe is connected to the pump through a flexible high-pressure pipe.

The pressure in the common rail system is controlled by a mechanical pressure regulator (7), which could be set manually to a certain value according to the reading of

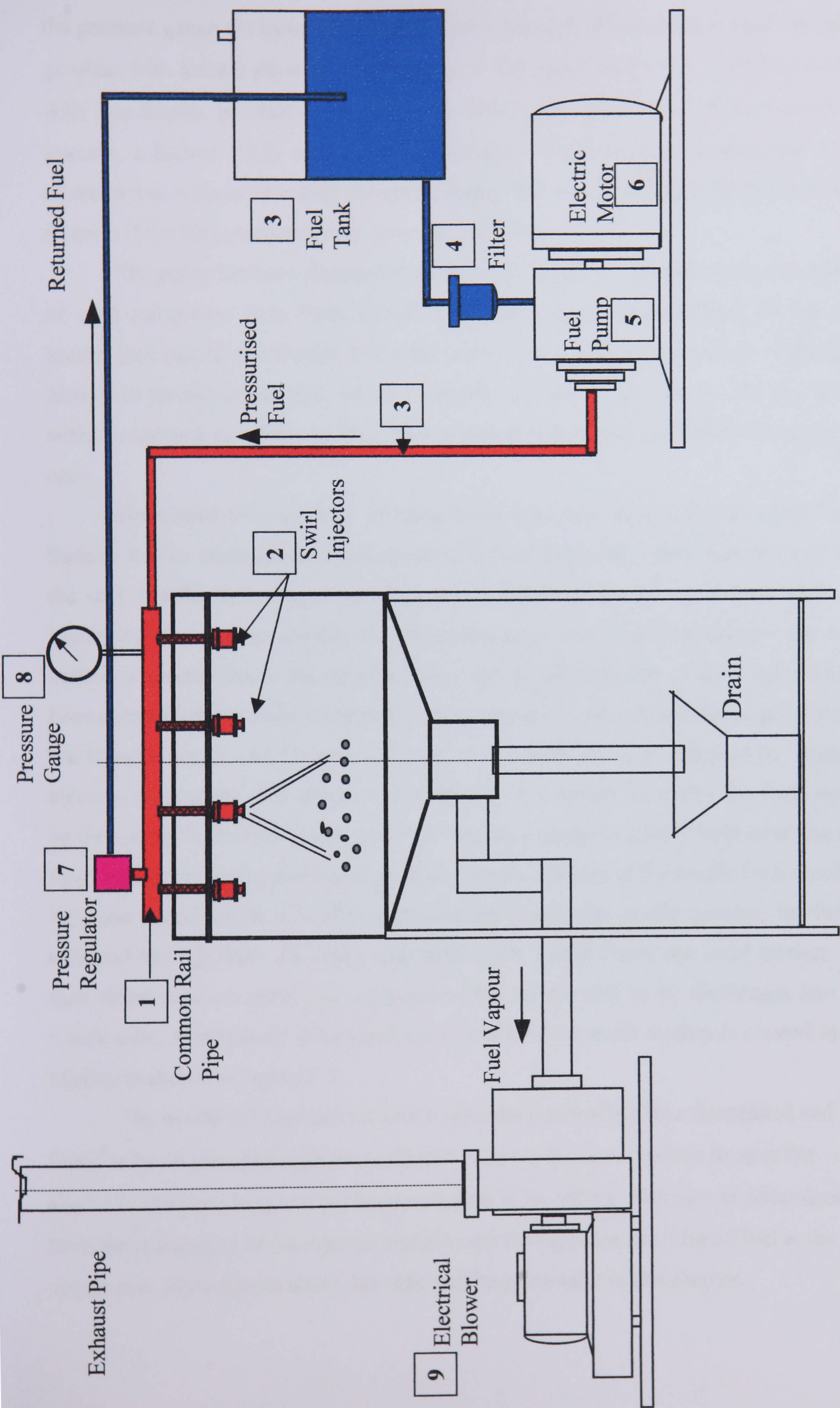


Figure 2-1 Mechanical arrangement of the common rail injection system

the pressure gauge (8) installed on the common rail pipe. White spirit is used instead of gasoline with known physical properties ($\rho = 785 \text{ kg/m}^3$ and $\mu = 9.4 \times 10^{-3} \text{ kg/ms}$) but with less hazard in health and safety. To collect the white spirit injected from the injector, a blower (9) is installed which creates a small vacuum downstream of the injectors but without affecting the spray shape. The power supply of both the blower motor and the high-pressure pump motor is 220 Volt.

The pump has been dismantled as shown in Figure 2-2, and the main dimensions of each component have been identified for the purpose of modelling the injection system (not part of this thesis). When the pump cam rotates, it pushes one of the three pistons to pressurise the fuel, which is directly admitted at the pump exit. The spring with the cap acts as a force on the piston to maintain continuous contact with the pump cam.

Four swirl pressure type injectors have been investigated in this study. Each injector has its enclosed electromagnetic coil controlling the needle opening and thus the start and the end of injection. The main components of the injector are shown in Figure 2-3. In order to identify the dimensions and also to understand how the swirl motion is created inside the injector before the nozzle exit, one of these injectors has been dismantled and cross-sectioned along a symmetry plane. The detailed geometry of the injector nozzle can be seen in Figure 2-4, which has been obtained by using an electron microscope. The needle of the injector is normally closed by the force acting by the spring. When the injector coil is energised, a magnetic field is built up acting as a force against the spring and resulting in an inwards opening of the needle for a specified injection period which is flexibly programmed. During the needle opening, the fuel is admitted through four off-centre tangential holes which create the swirl motion, and then flows into a conical slot upstream of the nozzle seat to be discharged into the nozzle exit. A schematic demonstration shows how the swirl motion is created in the injector is shown in Figure 2-5.

The needle lift has been measured after the injector had been dismantled and found to be $60 \text{ }\mu\text{m}$. Although the needle lift is small, the time required to open the needle is relatively long and has been estimated to be $300 \text{ }\mu\text{s}$. This time is determined from the rising edge of the demand signal to the first appearance of liquid fuel at the nozzle exit. More details about this time will be given later in this chapter.

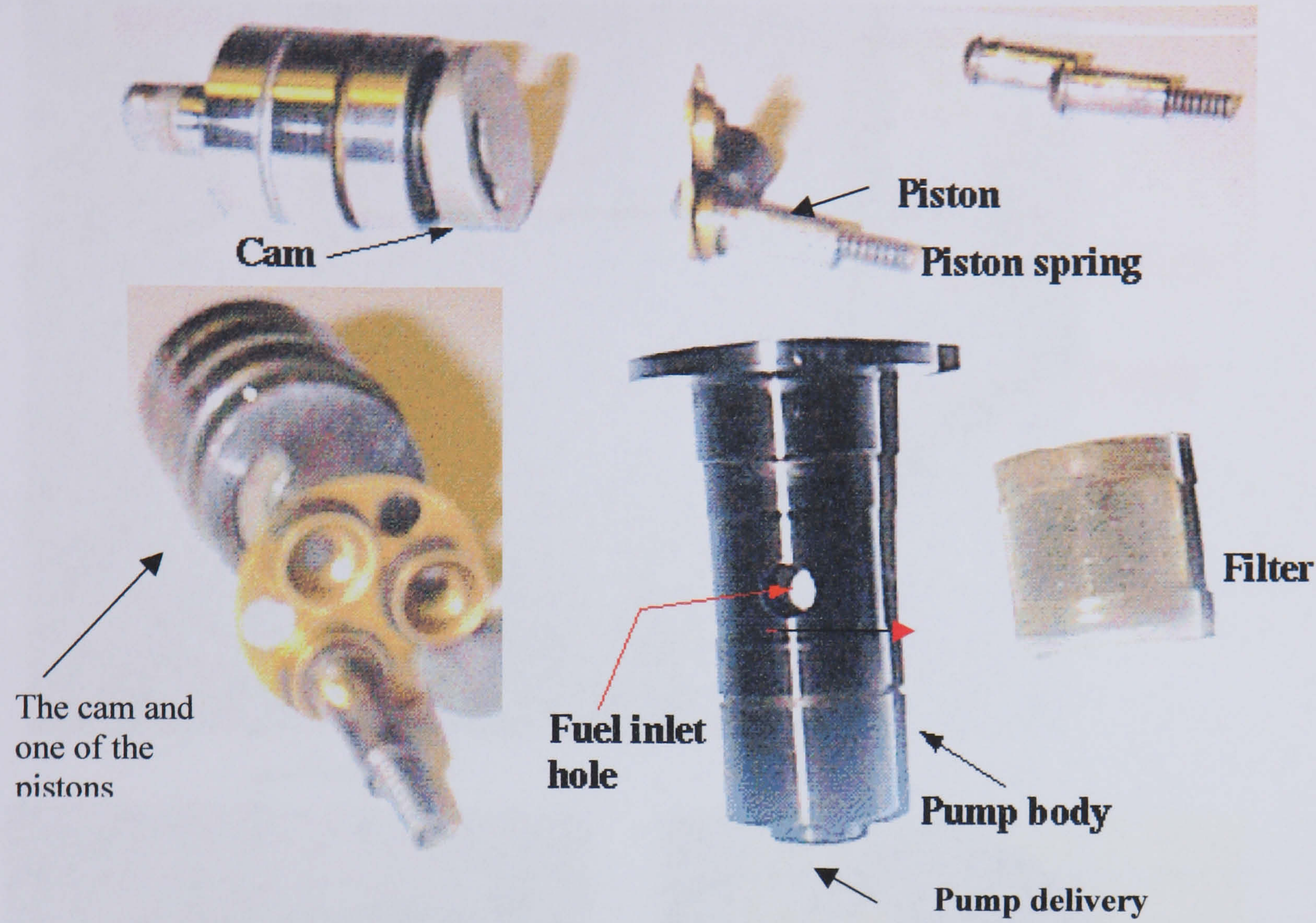


Figure 2-2 Main components of fuel pump

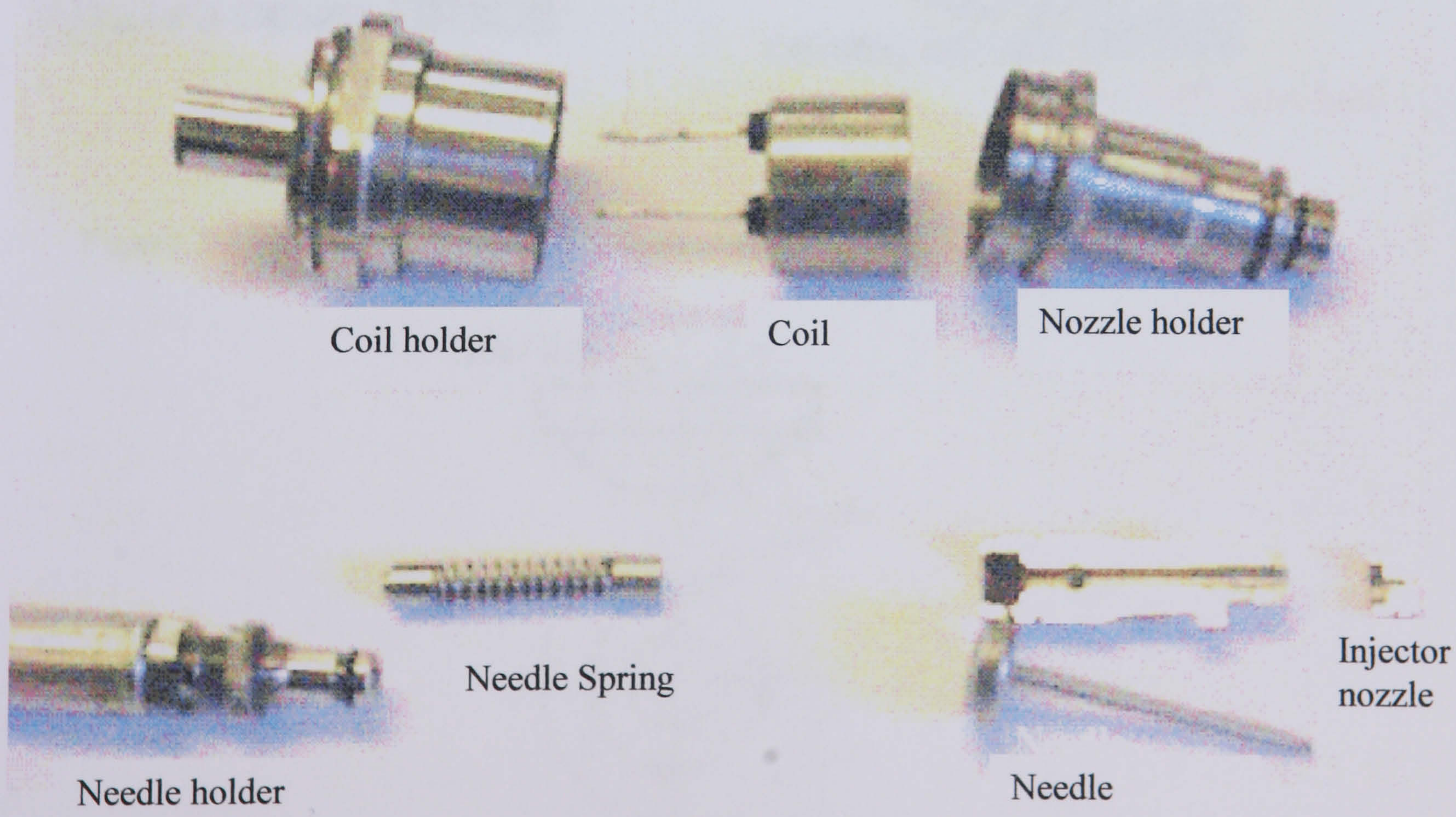


Figure 2-3 Main components of swirl injector

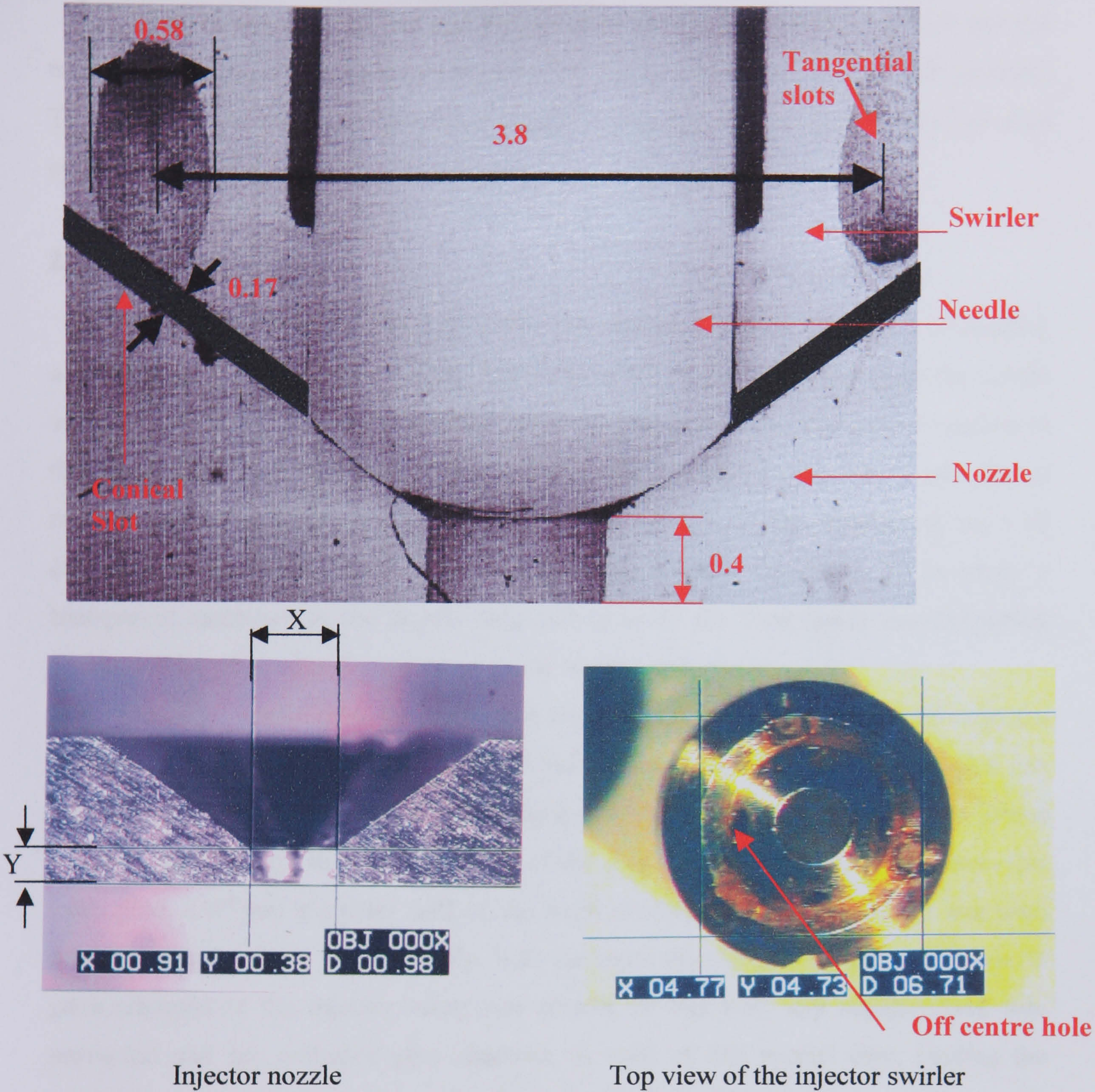


Figure 2-4 Magnified picture of the swirl injector nozzle with dimensions (mm)

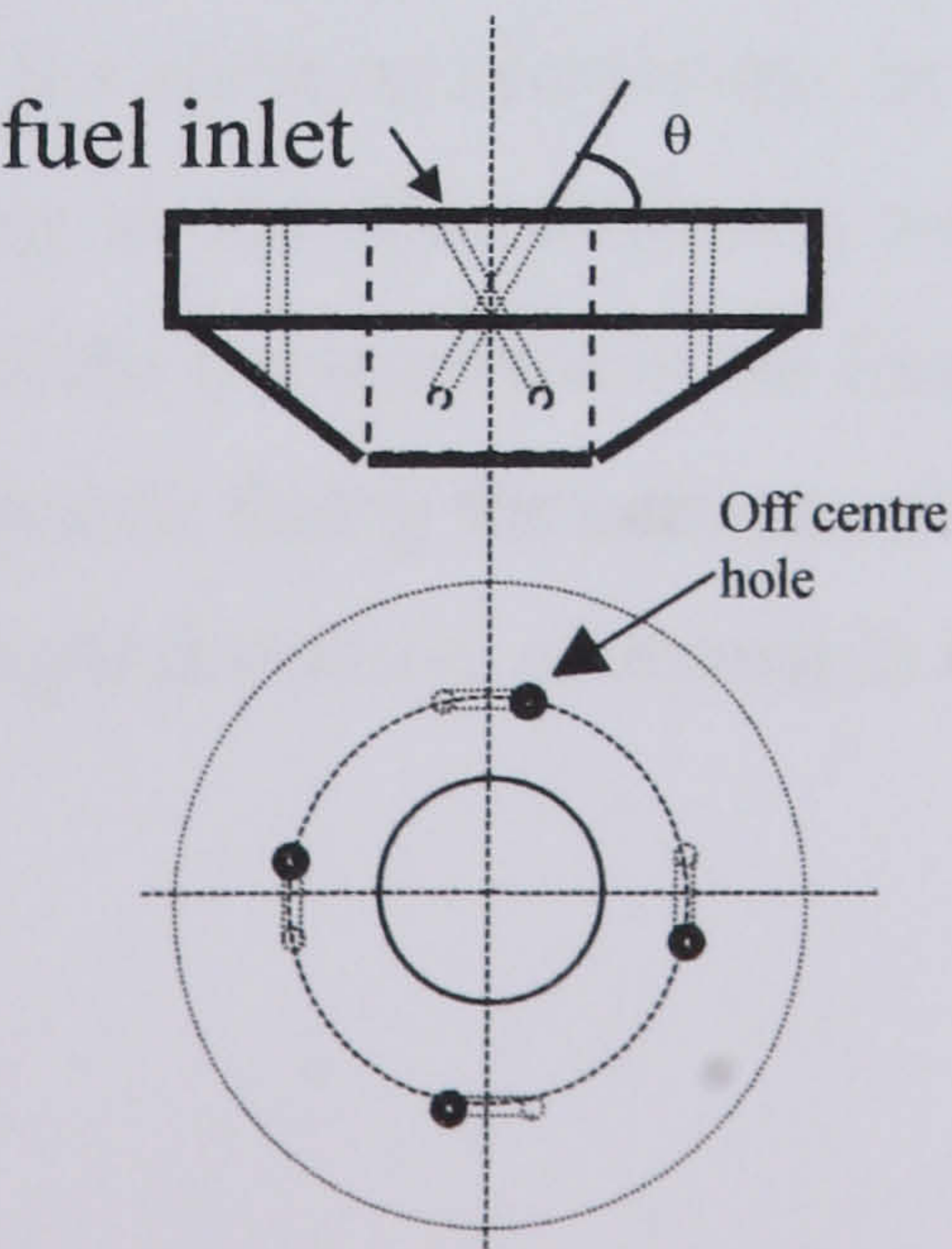


Figure 2-5 A schematic of the injector swirler

The common rail system and the injectors have been clamped in a 2-D traverse mechanism in order to have small incremental displacement and allow easier focusing. The fuel tank and the pump have been located separately. A photograph of the injection system and the traverse is shown in Figure 2-6.

2.2.2 Transparent Nozzle

In order to visualise the flow inside the injector nozzle just before it is injected, a transparent nozzle had to be made. Since the nozzle length downstream of the needle seat is rather short, 0.4 mm, it is quite a difficult job to install a transparent window in that short length. It is also very difficult to replace the main nozzle with a transparent one because of possible leakage and the lower strength of the transparent material compared to the steel nozzle. The most practical solution proved to be to attach a transparent extension to the nozzle hole and to study the flow inside this extension, which will be referred to from now on as the transparent nozzle.

In order to minimise the errors that arise from the presence of gaps between the transparent nozzle and the main nozzle which could cause flow disturbance, glue has been applied to both sides and then the transparent nozzle and main nozzle were joined leading, unavoidably, to some extrusion of the glue inside the hole. To eliminate this extra glue, a 0.9mm diameter drill of the same diameter as the nozzle hole was used after the glue has been dried. Finally, both the main nozzle and the transparent nozzle were clamped to the injector using two screws. In this way, any misalignment was prevented and any extruded glue removed. In spite of this special care, binding the transparent nozzle to the main nozzle allowed some flow disturbance to occur due to the interface. This was clear in the images of the film and has been confirmed when the interface was checked under the electron microscope, as shown in Figure 2-7. In order to avoid any errors occurring in the film thickness measurements due to this flow disturbance, the first 0.2 mm from the interface in the film image has been ignored. The two sides of the transparent nozzle facing the camera and the light have been made flat in order to minimise any the light distortion, as shown in Figure 2-8.

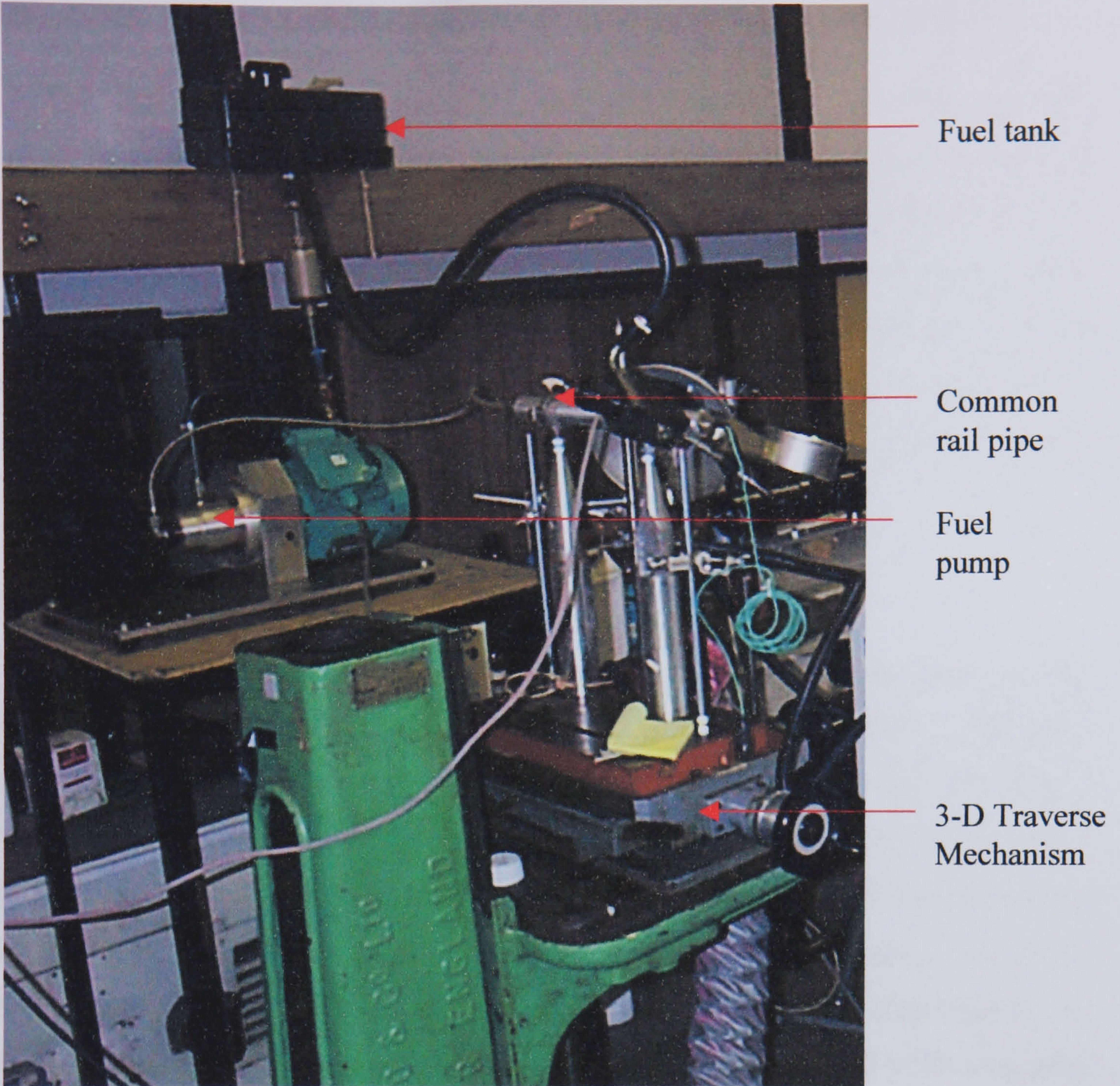


Figure 2-6 Photograph shows the common rail injection system installed on a 2-D traverse mechanism

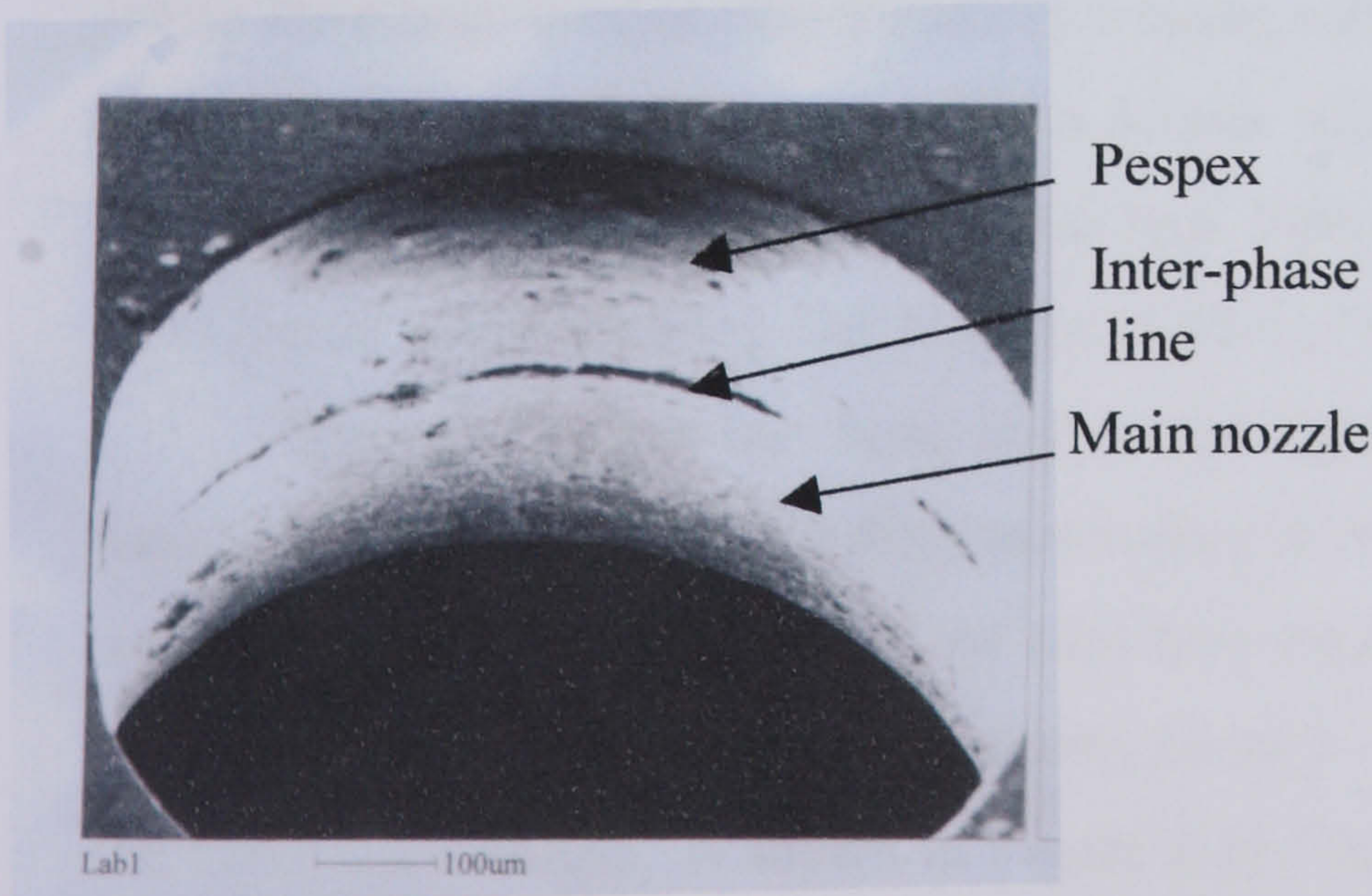


Figure 2-7 Photograph of the interface between main nozzle and transparent nozzle

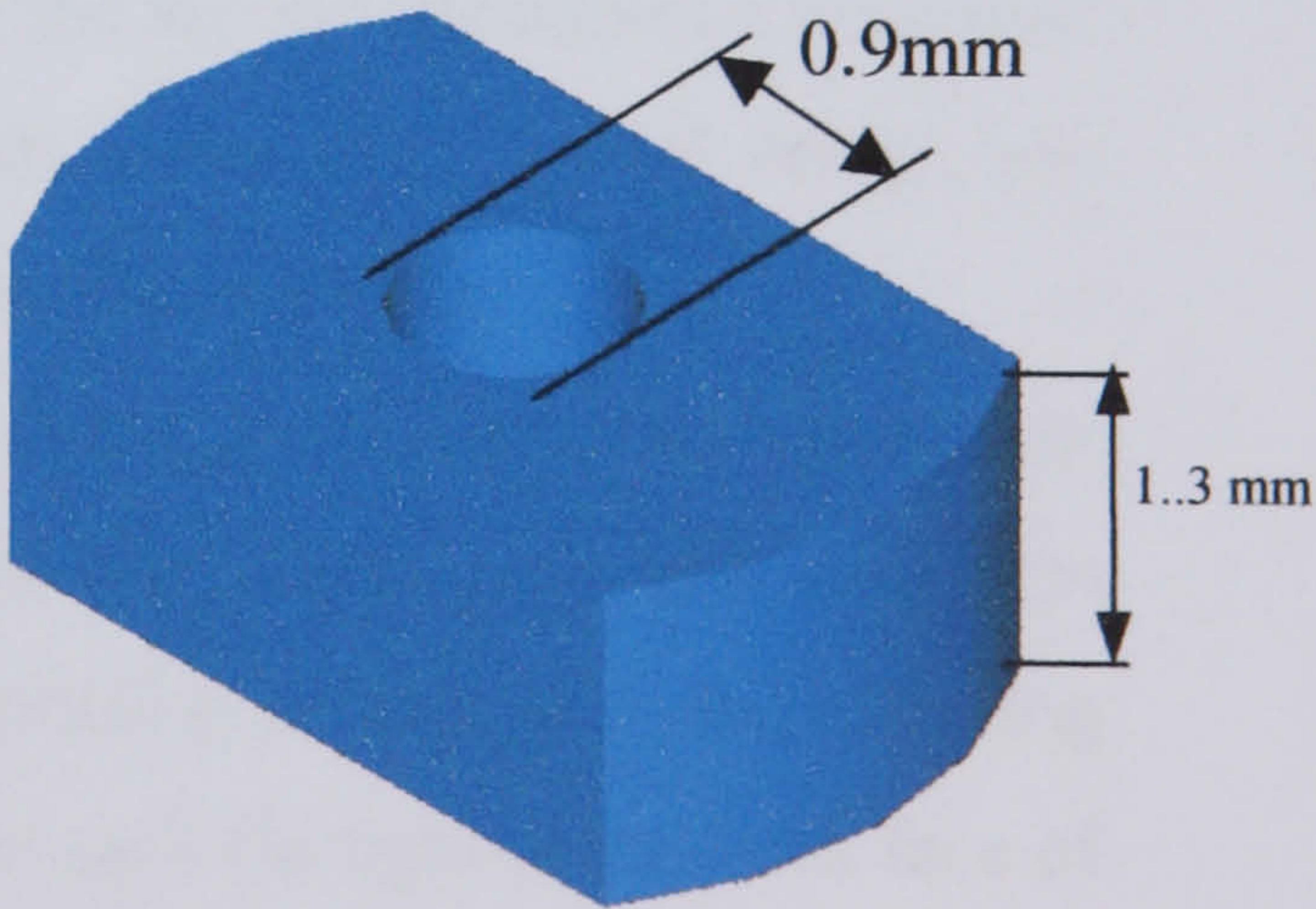


Figure 2-8 A schematic of the transparent nozzle

2.3 IMAGING SYSTEM, INSTRUMENTATION AND CONTROL UNITS

The fuel inside the transparent nozzle and near the nozzle exit has been visualised by an imaging system, which will be discussed in the following section. Also a fast data acquisition system was installed in order to acquire the pressure fluctuations inside the injection system. To allow control of the injection system and the triggering signals and also of the data acquisition system, a control unit has been designed and constructed. Figure 2-9 shows the arrangement of the instrumentation, imaging system and the timing system used in this study. The following subsections will describe each part in more detail.

2.3.1 Imaging System

The imaging system consists of a CCD camera with microscopic lenses, a light strobe unit, a PC with an interface card and relevant software. A SenciCam 12 bit CCD camera has been used in all the visualisation work presented in the thesis. The camera has an internal focusing system and a resolution of 1280 (H) x1024 (V) pixels with a minimum exposure time of 0.1 μ s. The camera is connected to a PCI-interface-board installed in a PC via a fibre optical cable to achieve a maximum transfer rate of 132 MByte/s. Light intensity, exposure time and mode of operation were controlled by the interface software. The strobe light and the camera are triggered by a 5-volt rising edge signal produced by the timer card. The front of the camera is connected to a long distance microscope whose magnification varies by changing the front lens. Two front lenses have been used in film thickness imaging called CF2 and CF3; the first is used to visualise the image field from 2.2mm to 3.5mm while the second visualises a minimum object of 1.6 mm. The strobe light has two 10mm-diameter fibre optical cables and front lenses with a focusing ring to allow control of their focal length.

Special attention has been paid to the light distribution along the transparent nozzle, which has to be as uniform as possible in order to minimise any light distortion and reflections. Better images of the film have been obtained when using backlighting through a copper plate as a light reflection mirror to forward the light to the back face of the transparent nozzle, as shown in Figure 2-10. Therefore, the two fibre optical cables have been located very carefully in front of the copper plate and their position changed until the best image could be obtained.

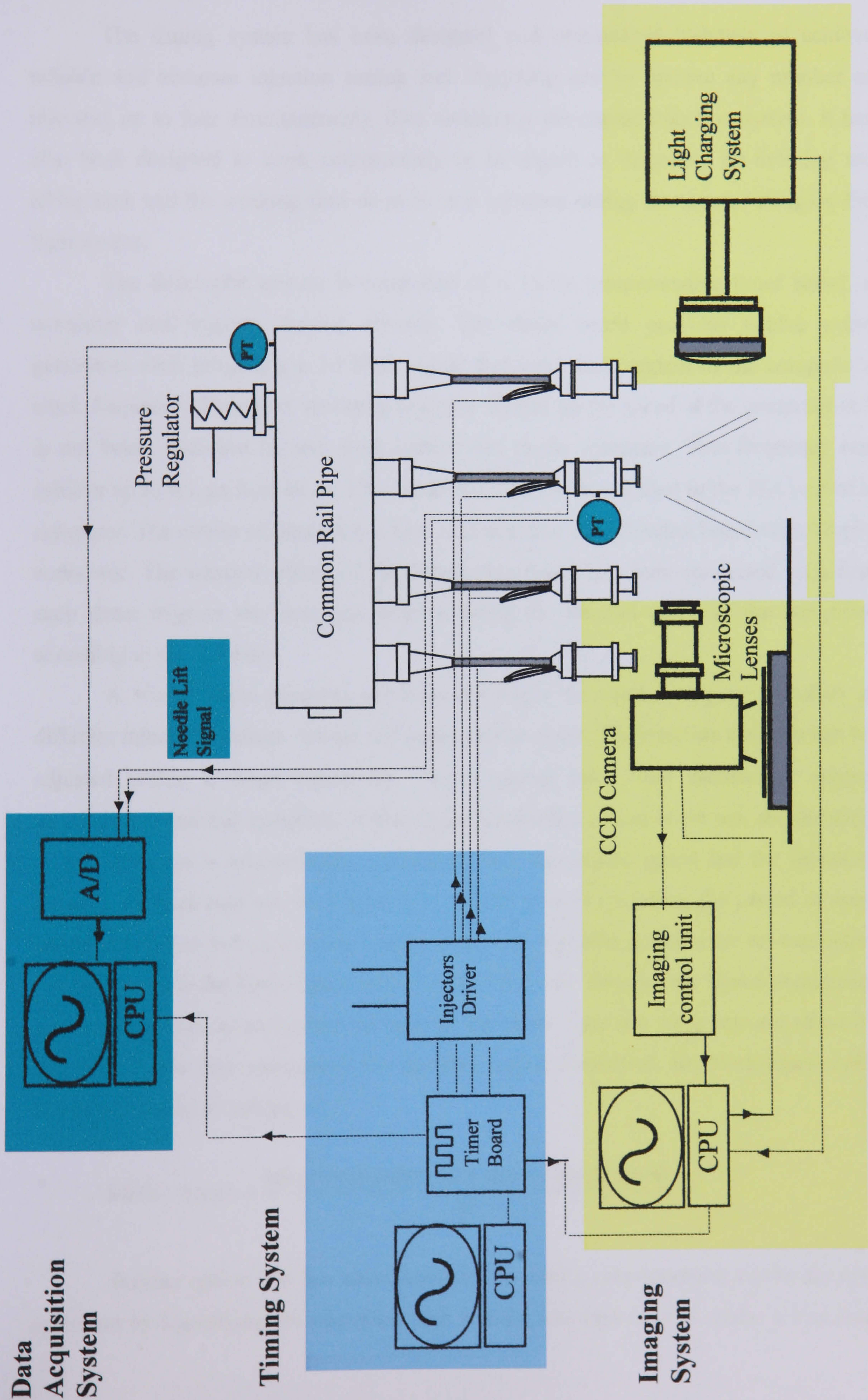


Figure 2.9 A schematic of instrumentation, imaging and controlling systems

2.3.2 Timing System

The timing system has been designed and constructed in-house to achieve reliable and accurate injection timing and triggering and to operate any number of injectors up to four simultaneously, thus simulating the engine injection system. It has also been designed to work continuously as in engine or discretely by defining the idling time and the working time so as to stop injection during the time of charging the light source.

The developed system is composed of a 16-bit programmable timer board, a computer and injector driving circuits. The timer board provides twelve pulse generators each producing a 10 MHz signal frequency independent of the computer's clock frequency. Therefore, its signal does not depend on the speed of the computer and is not being disturbed by any work carried out in the computer. This frequency can achieve up to 0.1 μ s time steps. This timer board has been installed in the ISA port of a computer. The output of the card has been connected to a distribution board via a 78-pin connector. The internal wiring of the distribution board has been connected such that each timer triggers the next one without using the internal clock of the computer according to the software.

A Visual Basic program has been developed to simulate engine operation at different injection timings, speeds and injection durations. The injection duration can be adjusted within a 0.1 μ s sensitivity. Two options have been considered: engine simulation or manual operation. If the engine simulation option is chosen, the duration of non-injection is automatically calculated from the engine speed and the injection duration for each injector, see Figure 2-11. Under manual operation, the period of non-injection between two successive injections and the injection duration are set manually. For both options the Mark-Space ratio, which is the ratio between the injection duration and the total time between two successive injections from the same injector (time of two cycles), has been calculated. For the case engine simulation, the Mark-Space ratio can be calculated according to:

$$\text{Mark - Space} = \frac{\text{injection duration (s)} \times \text{engine speed (rpm)}}{30}$$

Another option that has been considered is continuous operation versus discrete operation by identifying the operation time and the idle time. In this study, a 1 second

operation time has been set while a 35 seconds idle time has been considered as corresponding to the time needed for charging the light source. The advantage of this discrete operation is saving fuel and minimising the quantity of fuel (white spirit) drained to the exhaust gas pipe as well as the quantity of fuel being evaporated in the laboratory during the period of charging the light source. It also has an advantage in reducing the time required to clean the windows of the constant volume chamber (see Chapter 4) during spray visualisation, since the quantity of fuel injected is adjusted for only one injection.

The triggering signal can be continuous or single pulse. In order to avoid any injection distortion which could occur in the first few cycles either from the mechanical response device or from the electronic devices, a number of cycles before triggering starts have been bypassed; this number has been set in the timer card (3 cycles). The delay time as defined from the rising edge of the injection signal to the rising edge of the triggering signal could be set up in the software with a sensitivity of $0.1 \mu\text{s}$. The developed design also considers another triggering signal which could be extracted from the distribution board which has a pre-defined delay time from the rising edge of the first triggering signal; therefore, two consecutive images could be achieved with a certain delay time. Appendix 2-A shows the interface software panel indicating all the parameters which could be changed to control the injection process and the triggering signal.

The distribution board is connected to the injectors driving unit via BNC cables. The main function of the driving unit is to provide adequate current and voltage to charge the solenoid valves of the injectors. The output of the driving unit is 70 Volts, which drives the injector solenoid valves; the power supply of the driving unit is obtained from a 12 Volt-DC transformer with a rectifier circuit.

2.3.3 Data Acquisition System

Two piezo-resistive transducers (Kistler type 4067 A1000) have been used to measure the pressure fluctuations in both the common rail and above the swirler of the nozzle. The natural frequency of the transducer sensor is more than 100 kHz and its sensitivity is 10 mV/bar. A signal conditioner has been used to achieve temperature compensation and linearization for the pressure signal as well as stabilising the power supply. To measure the pressure above the swirler in the nozzle, a side hole has been

drilled using an electric arc; this technique has an advantage over normal machining in minimising stresses which could result in nozzle deformation.

A data acquisition system composed of a 16-channel data acquisition board with a 12-bit resolution has been employed to transfer the pressure data to the computer; more details about the system could be found in Fairbrother (1994). Although the system has 16 channels, only three channels have been used with a sampling frequency of 250 kHz. The first channel measures the signal from the common rail pressure transducer; the second channel measures the signal from the nozzle exit pressure transducer while the third channel monitor the signal of the injection signal.

Although the range of operation of the two pressure transducers is large (0-1000 bar) compared to the maximum range used in this study (0-100 bar), the measuring system proved to be linear and accurate. The degree of linearity has been estimated from a static calibration of the transducers and the data acquisition system using a dead weight tester; Figure 2-12 shows the linear calibration relationship.

2.4 FLOW CHARACTERISTICS INSIDE THE INJECTOR NOZZLE

2.4.1 Injection Mass Flow Rate

Before starting the flow imaging work, the injector was installed and the mass injected from the injector measured for different injection durations under three injection pressures of 20, 50 and 70 bar. The volume of the fuel injected over a 30-60 seconds period has been collected in a test tube and divided by the number of cycles in order to calculate the average mass flow rate for an assumed fuel density of 750 kg/m^3 . Figure 2-13 shows the relationship between the injected mass for different injection duration; Figure 2-13b shows the first 1ms in the form of three stages (A,B,C). The first stage (A) corresponds to the delay time, which has been estimated to be about 0.3 ms; during this period there is no fuel is delivered. The delay time is defined as the time required to start the opening of the needle and to allow the fuel to start flowing. This time could be divided into a mechanical delay time and an electrical delay time that is required to build a sufficient magnetic field to push the needle against its weight and the fluid pressure difference applied on it. The mechanical delay time is required for the needle to open adequately for the liquid to flow through the gap. Identifying this delay time is very important for having comparable results with similar injectors but it is also

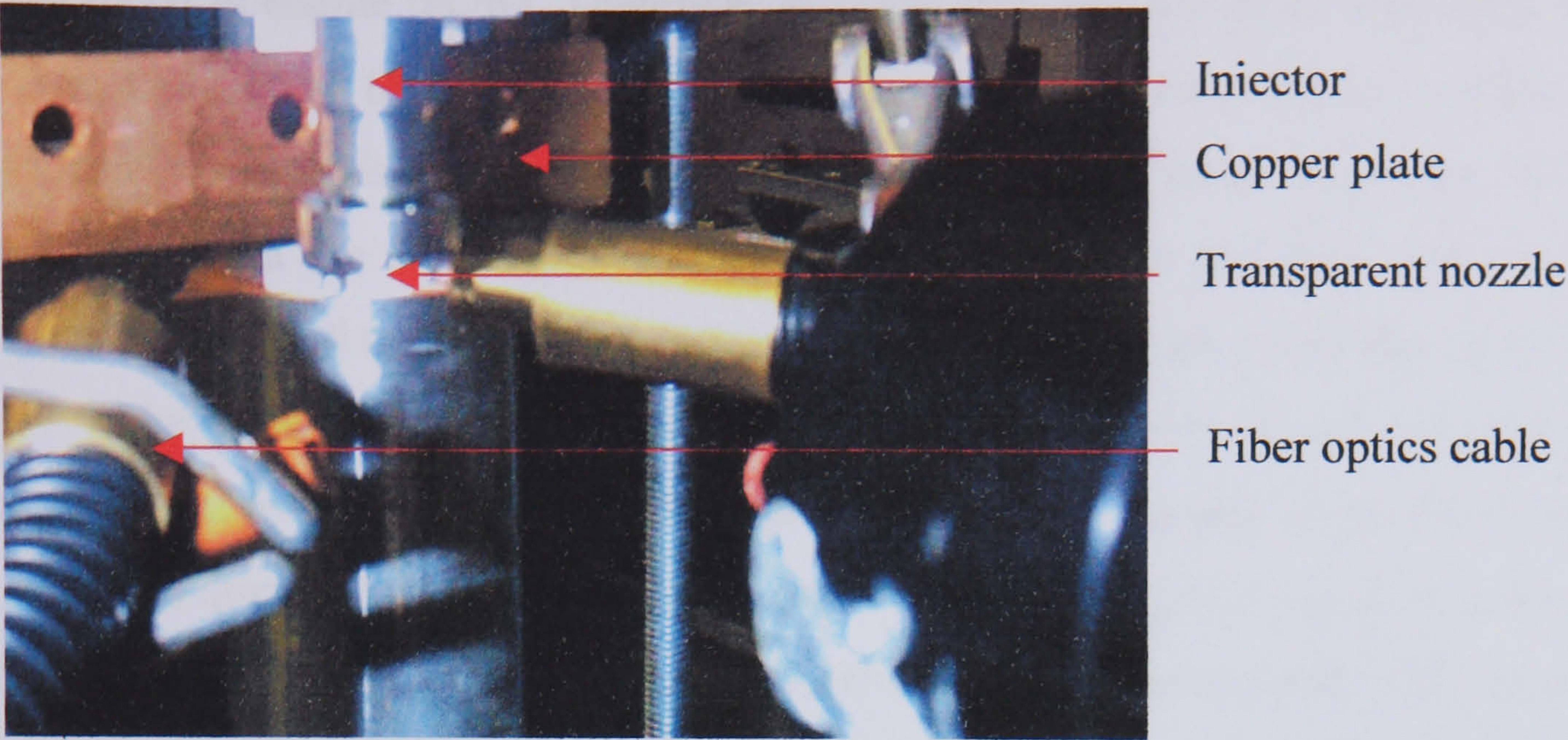


Figure 2-10 Back lighting arrangement for film thickness imaging.

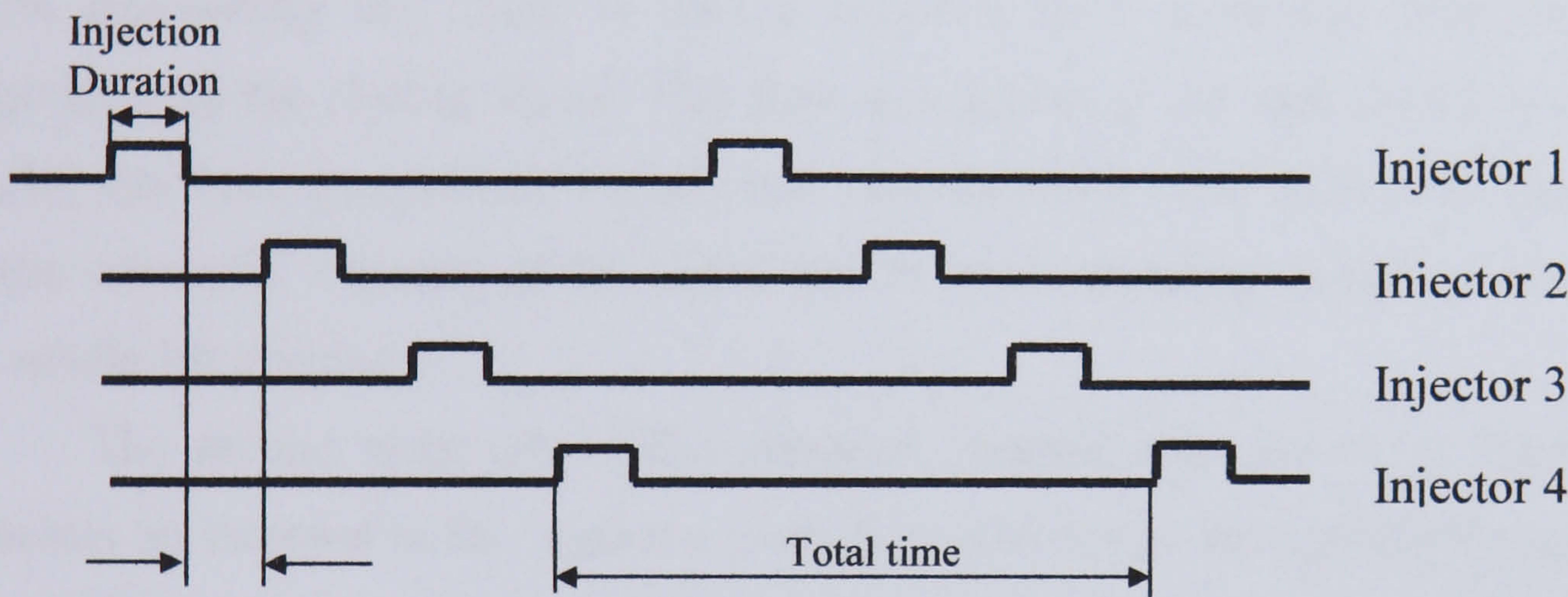


Figure 2-11 Output signal from the timer card for operating the four injectors.

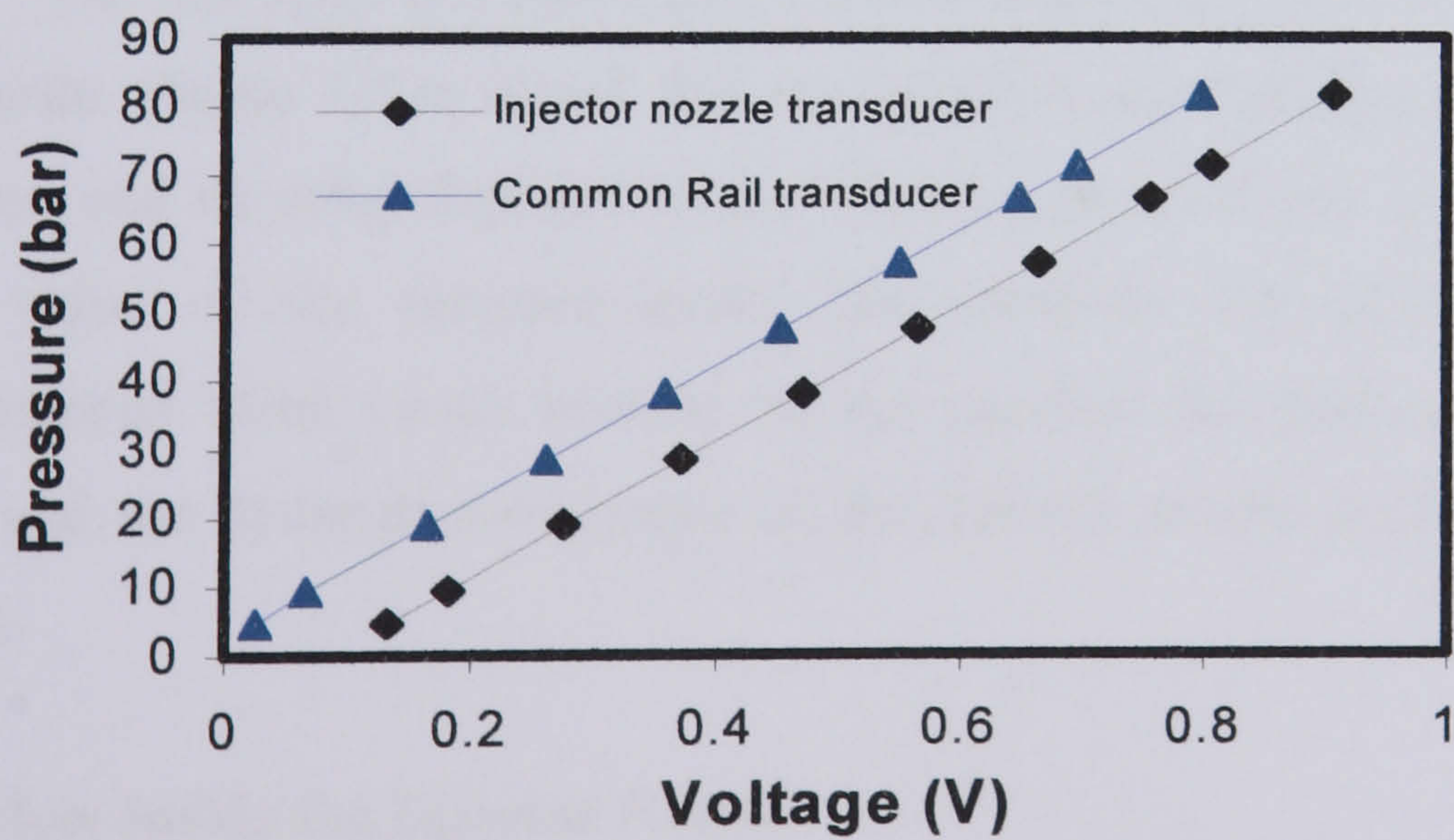


Figure 2-12 Calibration graph of the pressure transducers.

very crucial in engine timing. Therefore, this time has been identified accurately and all the times presented from now on in this thesis will be referred to the start of the actual injection process, which is delayed from the start of the injection signal by this delay time. In general the delay time is defined as the time from the start of the injection demand signal until the first appearance of liquid fuel just at the exit of the nozzle.

The delay time has been determined in this thesis by direct imaging of the nozzle exit at every 0.02 ms until the first drop of liquid appears at the nozzle exit, as shown in Figure 2-14. This delay time has been plotted against injection pressure in Figure 2-15; this figure shows a slight increase in the delay time with decreasing injection pressure which could be attributed to the lower injection velocity and, thus, the increase in the time needed for the fuel to travel from the needle to the nozzle exit. It is worth mentioning that there is also a response time associated with the needle responding to the closing signal. This time in addition to the time for complete valve closing has been found to be slightly less than the delay time (0.25 ms). Figure 2-16 shows schematic variation of the signal and the corresponding theoretical variation of the needle lift opening.

The second stage (B) of the averaged injected mass shown in Figure 2-13b indicates an increase in the injection mass flow rate due to the continuous opening of the needle until it reaches its maximum lift at about 0.25 ms. This opening causes pressure waves which travel forward and backward along the injection system with the speed of sound, as it will be discussed later.

The last stage (C) represents the fuel injected during the full opening phase of the needle. Figure 2-13a shows that the injection rate remains almost constant during that time and its value depends on the injection pressure and in particular the constant mean value of the pressure inside the common rail despite the fact that the instantaneous value varies because of the pressure fluctuations within the injection pump and the dynamic movements of the injector needle according to Cousin et al. (1998).

2.4.2 Flow Inside the Injector Nozzle

During the early stage of the needle opening, fuel flows axially inside the transparent nozzle with almost zero swirl velocity. As the needle opening increases, an annular flow starts to be formed under the effect of the centrifugal forces until the

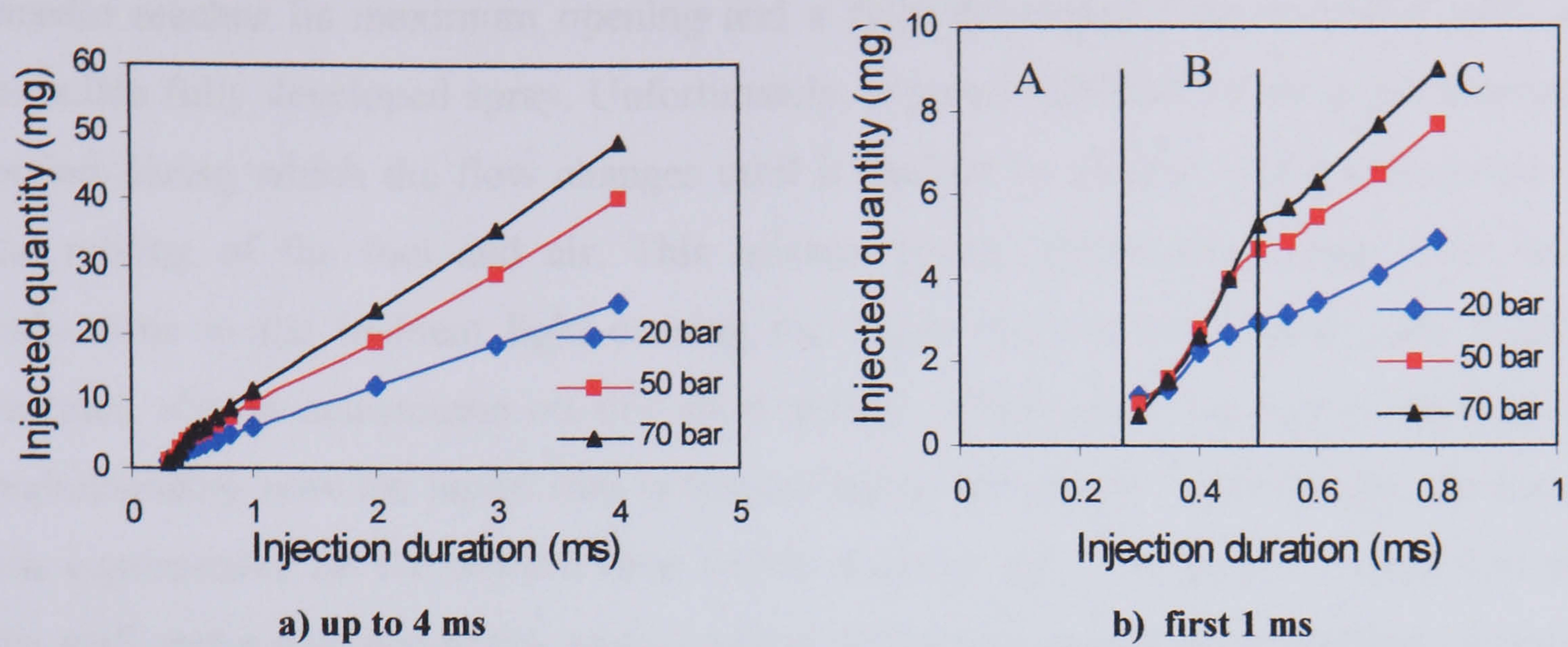
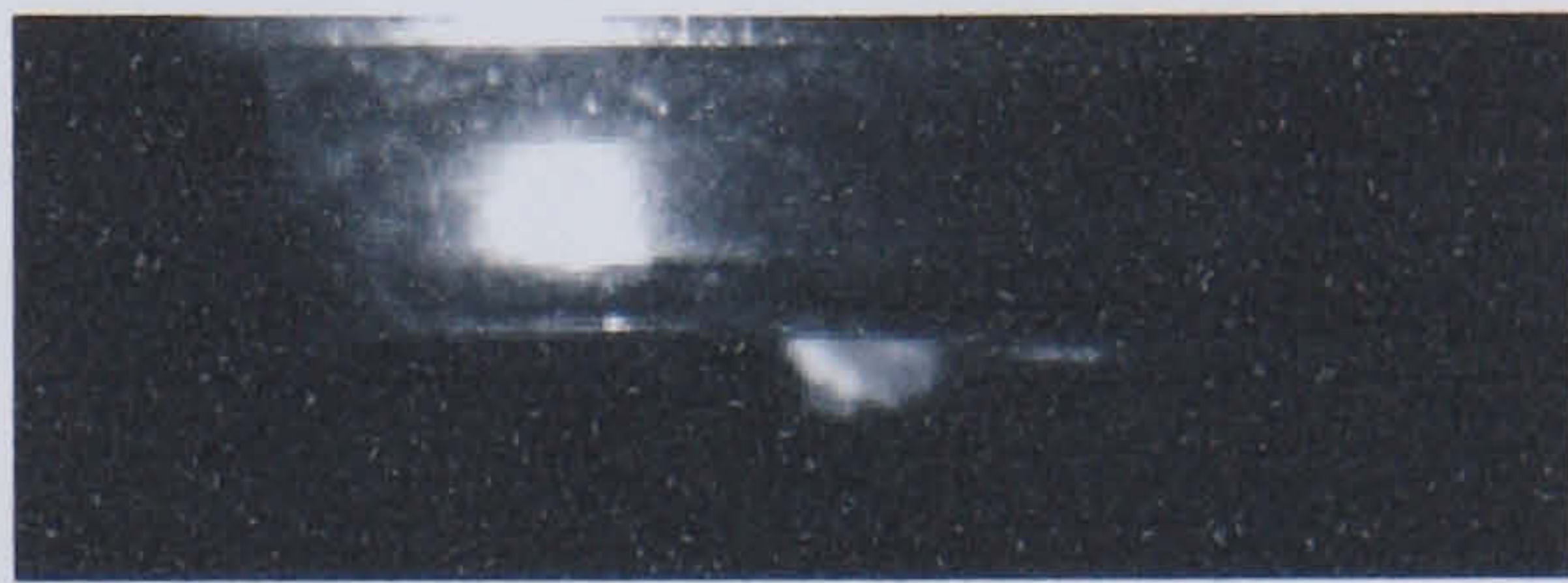


Figure 2-13 Effect of injection duration on quantity of fuel injected for different injection pressures.



10 bar- 335 μ s



70 bar- 305 μ s

Figure 2-14 Images of first appearance of fuel at the nozzle exit.

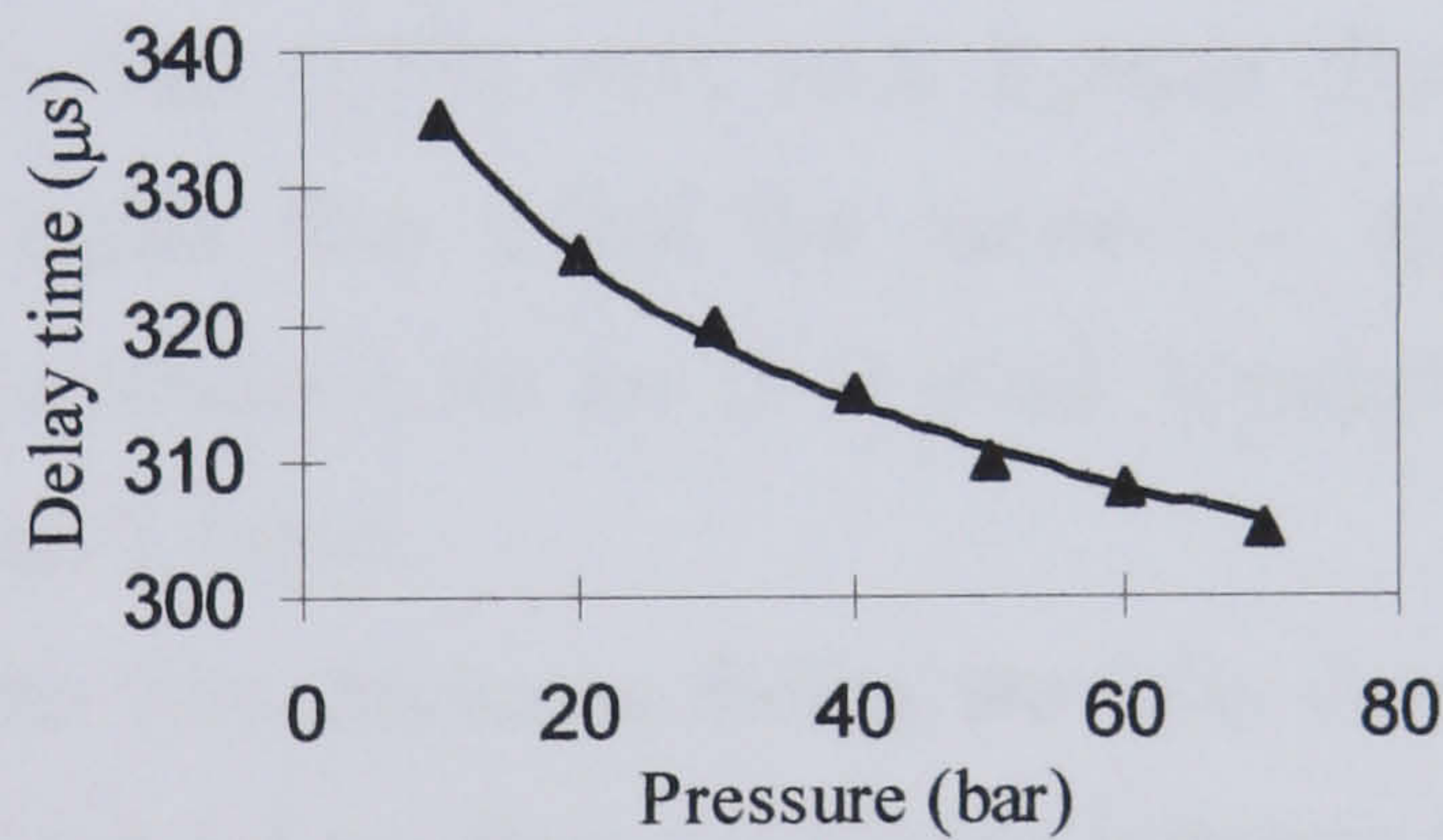


Figure 2-15 Effect of injection pressure on the start of fuel appearance.

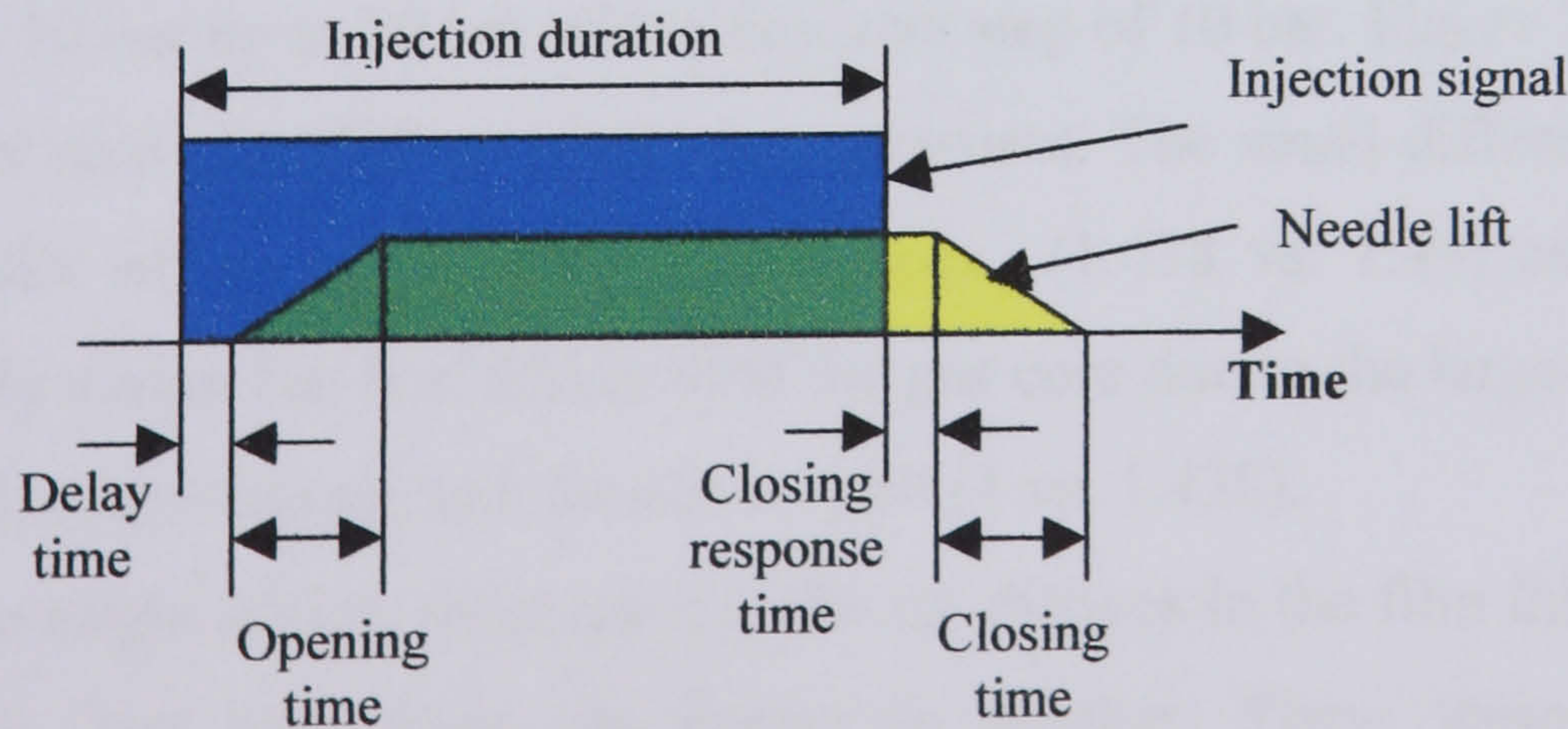


Figure 2-16 A schematic sketch of the needle lift opening and closing upon signal activation.

needle reaches its maximum opening and a fully developed film is created with an associated fully developed spray. Unfortunately, it proved difficult to study the transient period during which the flow changes until it reaches its annular structure because of the mixing of the fuel and air. This mixture gives, unfortunately, rise to internal reflections to the incident light causing the image to look completely dark. Future research should concentrate on this short period, which could be very interesting in understanding how the liquid film is formed inside the nozzle. In this study, the focus has concentrated on the annular flow which is composed of an annular liquid film on the wall and a gas core in the central region of the transparent nozzle during the fully developed stage. Having identified the film thickness and the flow rate, it is straightforward to calculate the liquid axial injection velocity from the continuity equation. Both the axial liquid velocity and the film thickness are the main input parameters to the atomisation sub-model, as will be discussed in the next chapter. Knowledge of the variation of the film thickness and the flow rate with different flow and geometric parameters is useful in predicting the quality of the atomisation process.

2.4.3 Film Thickness Imaging

A high magnification lens (CF3) with back lighting illumination has been employed to visualise the liquid film inside the transparent nozzle, as discussed previously. Images have been obtained for the flow inside transparent nozzles having different lengths of 0.5, 0.9 and 1.4 mm.

In order to identify the film thickness during the fully developed stage of the spray, images have been taken at 1.5 ms from the start of injection for a 2 ms injection duration. A number of 50 images have been collected for each injection pressure starting from 10 bar up to 70 bar with a constant step of 10 bar. Figure 2-17 shows some of the images taken for different injection pressures. The small difference between the refractive index of the white spirit and Perspex (1.438 vs. 1.49) causes the film to appear slightly darker but less darker than the gas core due to the large difference in the refractive index between air and the white spirit (1 vs. 1.438).

As we might notice, there are significant changes in the film thickness along the nozzle length, and also from one image to another. These temporal and spatial variations in film thickness are attributed to the dynamic interaction between air and liquid at the inter-phase and also due to pressure fluctuations in the liquid flow itself. In

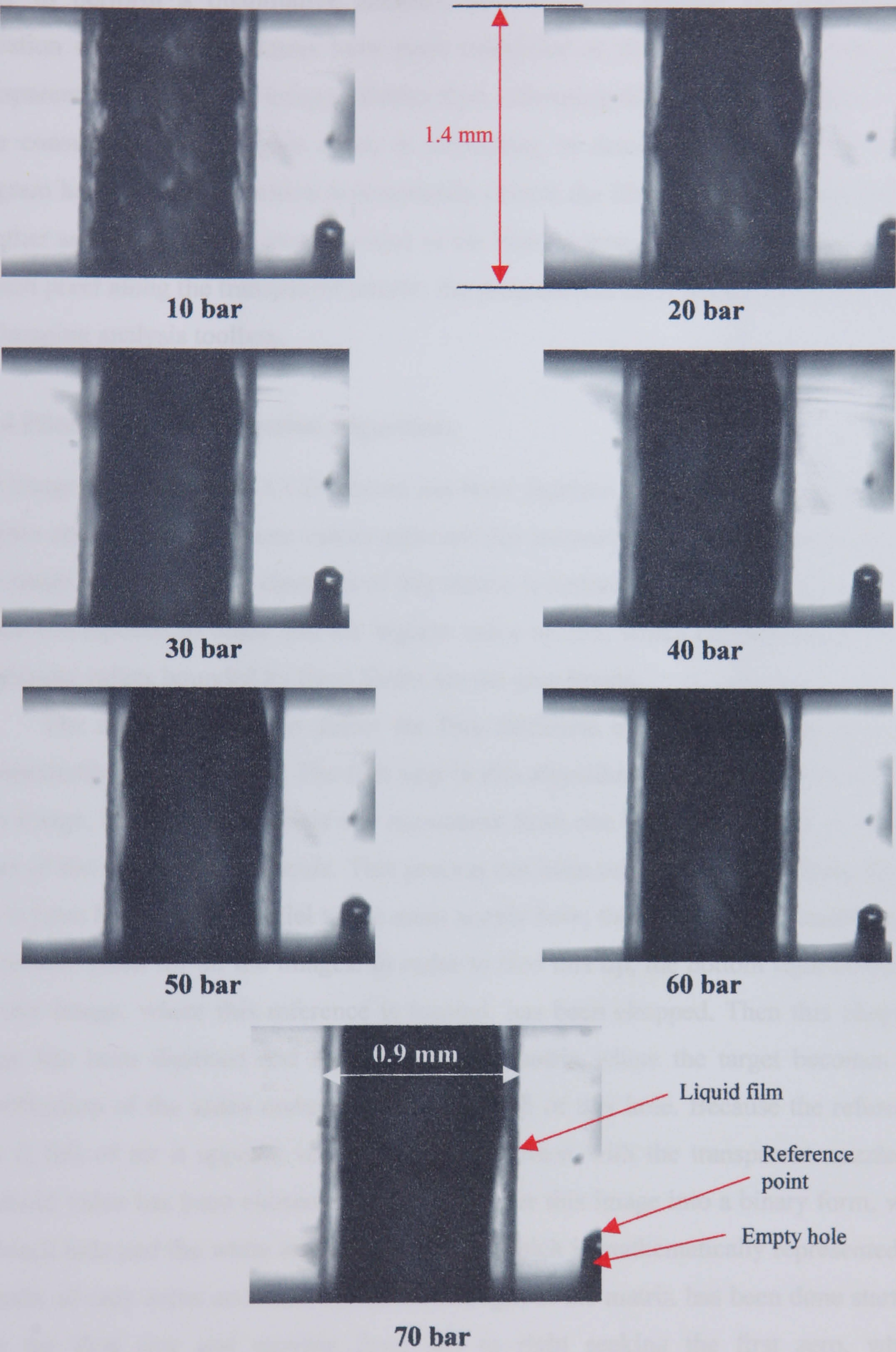


Figure 2-17 Typical liquid film images inside the transparent nozzle under different injection pressures.

order to perform a quantitative analysis, the ensemble average and the standard deviation of the film thickness have been calculated at different locations along the transparent nozzle over 50 images. Rather than estimating them manually which is very time consuming and can give errors in attempting to detect the film edge by eye, a program has been written which automatically detects the film edge, adds all the images together and calculates the average value of the film thickness and its standard deviation at each pixel along the transparent nozzle; the program has been written in Matlab using the imaging analysis toolbox.

2.4.4 Film Thickness Detection Algorithm.

The image captured by the CCD camera has been digitised and expressed in a matrix of m rows and n columns whose values represent the intensity of each pixel in the picture. The range of values of the elements of this matrix is limited by the lowest value of zero, which corresponds to black and the highest value of 255, which corresponds to white. Brightness values bounded by these limits are the grey-levels.

The algorithm used to detect the film thickness over 50 images is described schematically in Figure 2-18, The first step in this algorithm is to adjust the location of each image, in order to eliminate any movement from one image to another, so that all edges of the nozzle hole coincide. This process has been achieved by identifying the tip of a 0.1mm hole drilled parallel to the main nozzle hole; this tip has been considered as a reference point for all the images. In order to find this tip, the bottom right corner of the raw image, where this reference is located, has been chopped. Then this chopped image has been digitised and represented in a matrix where the target becomes the identification of the index corresponding to the tip of this hole. Because the reference hole is full of air it appears very dark in comparison with the transparent nozzle. A threshold value has been chosen in order to transfer this image into a binary form, with the black hole and the white transparent nozzle, which is mathematically represented by a matrix of only zeros and ones. A search throughout the matrix has been done starting from the first row and moving from left to right seeking the first zero, which corresponds to the tip of the hole. The index (I, J) of this zero in the binary matrix gives the location of the pixel that corresponds to the reference point.

Having identified the location of the reference point of each image, all images are shifted and relocated so that their reference pixels coincide and then chopped around

the nozzle hole in order to minimise the computational time and image noise. All 50 new images resulting from this adjusting process have the same number of pixels in the horizontal direction (m_1) as well as in the vertical direction (n_1), which could be represented by a matrix ($m_1 \times n_1$).

Different methods have been tried (Sonka et al., 1999) in order to detect a very clear edge. Canny method (Canny et al., 1986) has been found to be the best method to detect the liquid film edge because of its reduced sensitivity to noise. The method has been applied to two chopped images around the two edges for better detection of the edges and minimization of any errors. After applying this method a matrix of only zeros and ones is produced where the locations of the zeros represent the location of edges and all the other pixels have a corresponding value of one. By finding the index of the matrix elements whose values are zeros and by knowing the scale of the image (pixels/mm) it was easy to identify the location of the edge in mm. Having specified the location of the left and right edges, the film thickness could be calculated at each pixel along the transparent nozzle for each individual image. The film thickness is calculated by subtracting the inner diameter from the nozzle diameter and then dividing the resultant by two. Therefore, the film thickness is assumed to be axisymmetric. This is an acceptable approximation for the purpose of averaging, however we have to keep in mind that the film thickness varies in the circumferential as well as in the axial direction.

The edge in some cases was not continuous due to a very blurred or noisy image resulting in film thickness getting the value of zero, which is clearly not true. In order to eliminate this problem, searching for the index of the matrix elements corresponding to the edge always starts from side located closer to the nozzle axis and moving horizontally along the same row in the matrix until reaching the film edge. The same way of searching for the film edges in the following rows of the matrix is carried out. A condition of at least 30 % from the previous searched thickness has to be achieved to consider it as a right value. This percentage has been chosen just to avoid the zero thickness when no edge was detected.

2.4.5 Effect of Nozzle Length on Film Thickness

Since it is difficult to detect the trend of film thickness variation along the nozzle length from an individual image, an ensemble average of 50 images was used to

reveal this trend. Figure 2-19 shows the variation of the ensemble average and the standard deviation of the liquid film thickness along the nozzle length for a 70 bar injection pressure and a fully developed film (1.5 ms from the start of injection). As the nozzle length increases, the ensemble averaged film thickness also increases. That increase in film thickness is mainly attributed to the friction losses between the wall of the transparent tube and the liquid film. The aerodynamic forces created from the recirculation zone and the low pressure region developing at the centre of the injection hole could also contribute to the observed increase in film thickness, as predicted by Arcoumanis and Gavaises (2000); both forces decelerate the liquid flow velocity, thus increasing the film thickness while the average flow rate remains constant, causing a decrease in the injection axial and swirl velocities.

2.4.6 Effect of Injection Pressure on Film Thickness

To examine the effect of injection pressure on film thickness, the same algorithm for calculating the film thickness ensemble average and its standard deviation has been employed for a range of pressures (10 to 70bar) with a 10 bar step. A similar trend of increasing film thickness with nozzle length has been observed as shown in Figure 2-20 for 20, 50 and 70 bar injection pressure. The effect of injection pressure seems to have only a slight effect on the ensemble averaged film thickness. These results are in agreement with the early mathematical analysis performed by Lefebvre (1989) for a swirl injector and reported in Appendix 1-A for a non-viscous flow. The same analysis has been applied by Han et al. (1997) and Fan et al. (1997) in their spray models after introducing a correction factor to fit their experimental results. It has also been used more recently by Cousin and Nuglisch (2001). Datta and Som (2000) whose numerical analysis was not directly applicable to DISI injectors but rather to a Simplex injector have also predicted that the film thickness decreases with increasing flow rate only at very low rates while at relatively higher flow rates, the film thickness remains constant. Another numerical study by Keller (1998) shows a very slight (2%) increase in film thickness as the injection pressure increases from 20 bar to 120 bar. The individual images show surface ripples on the free film surface, which are more pronounced as injection pressure decreases. This reduction of the ripples at higher injection pressures could be attributed to the increase in the tangential velocity

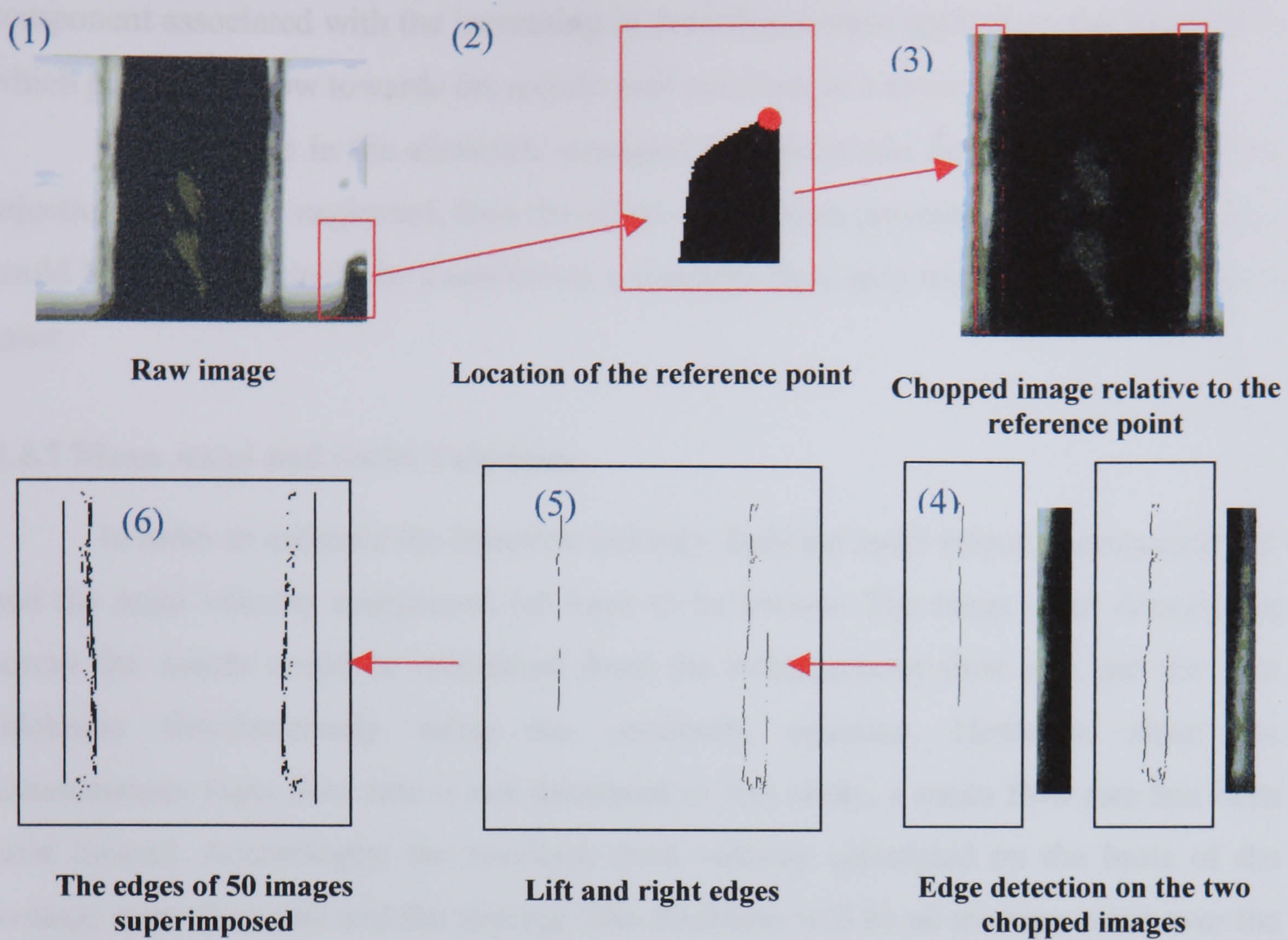


Figure 2-18 Algorithm of liquid film edge detection.

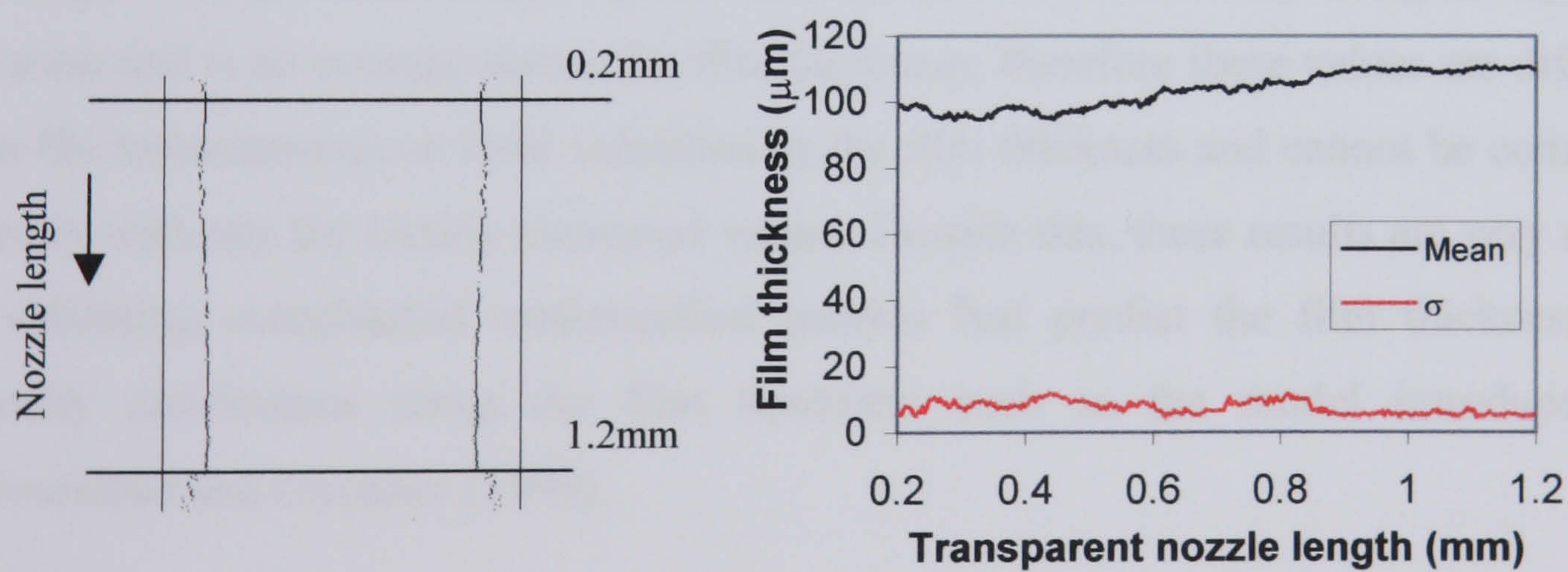


Figure 2-19 Variation of the ensemble average and standard deviation of the film thickness with nozzle length.

component associated with the increasing in centrifugal force applied on the liquid film, which pushes the flow towards the nozzle wall resulting in a more stable flow field.

If the change in the ensemble averaged film thickness due to the change in the injection pressure is neglected, then the effect of injection pressure on injection velocity could be predicted easily by considering a constant flow area with different mass flow rates.

2.4.7 Mean Axial and Swirl Velocities

In order to estimate the injection velocity, both the swirl velocity component (w) and the axial velocity component (u) have to be known. The mean axial velocity (u) across the nozzle could be calculated from the instantaneous flow rate and the film thickness simultaneously using the continuity equation. However, since the instantaneous mass flow rate is not measured in this study, a mean flow rate has been used instead. Accordingly, the resultant axial velocity calculated on the basis of the average mass flow rate and the average film thickness will be an average value over the whole injection duration. This average temporally and spatially axial velocity could be calculated at any section inside the transparent nozzle along the nozzle hole. The mean axial velocities close to the nozzle exit under 20,50 and 70 bar injection pressures are given in Table 2-1. It is important to keep in mind that this calculated velocity is an ensemble averaged value assuming the injection flow rate is constant along the injection duration and is an average across the film thickness; therefore these values are different than the instantaneous or local velocities in the film thickness and cannot be compared directly with any the locally measured values. Despite this, these results are very useful in validating complicated mathematical models that predict the film thickness and velocity distribution along the film thickness such as the model introduced by Arcoumanis and Gavaises (1999).

Table 2-1 The effect of injection pressure on mean film thickness and axial velocity

	$\dot{m}$ (g/s)	Film thickness (μm)	Axial Velocity (m/s)
20 bar	6	87-101	31-35
50 bar	10	87-101	52-60
70 bar	12	87-101	63-71

The swirl velocity is commonly estimated in the literature from the slope of the swirl disk holes or by measuring the spray angle and considering the swirl velocity equal to $u \tan(\theta)$ where (θ) is the spray angle. This assumption is based on the theoretical calculations proposed by Lefebvre (1989), which are only valid if the flow is fully atomised at the nozzle exit; clearly this is not the case here as a liquid film penetrates outside the nozzle exit before starting to disintegrate.

In this study, the swirl velocity component has been measured from the marks evident on the images obtained when focusing the camera on the outer film surface using high magnification lenses. Figure 2-21 shows some of these images where marks of the liquid flow are noticeable with their slopes varying along the nozzle length. The marks located at the nozzle centre represent approximately the projection of the flow velocity at this location, which can be resolved into axial and swirl velocity components. The axial velocity is assumed to be equal to the ensemble averaged velocity estimated previously from the average injection flow rate and film thickness. The angle between the flow direction and the nozzle centre line is defined as the flow angle (α) .

In order to identify the flow angle accurately, a window of 0.1mm side length on the centre of the nozzle hole has been chopped from the main images in order to analyse these marks using the edge detection technique. The slope of these marks, which represents the slope of the flow angle (α) , was easily measured in the way shown in Figure 2-22. This flow angle (α) is more accurate than the spray angle to use in order to calculate the swirl velocity component $(w) = u \tan(\alpha)$. The average value of the flow angle, as calculated from five images, has been considered to represent the flow angle.

2.4.8 Temporal Variation of Liquid Flow Inside the Transparent Nozzle

As the needle starts to open, liquid gradually enters into the discharge hole with almost zero swirl velocity through the needle seat passage, thus creating a poorly atomised ‘pre-spray’ concentrated at the centre of the injection hole. Following this initial stage, a gradual increase of swirl is taking place in the conical slot forcing the main bulk of liquid entering the hole to rotate and to gradually move towards the wall of the discharge hole as a result of the action of the centrifugal force associated with swirl. In order to visualise the flow development inside the nozzle from the start of injection until it reaches its fully developed stage, images have been taken every 20 μ s

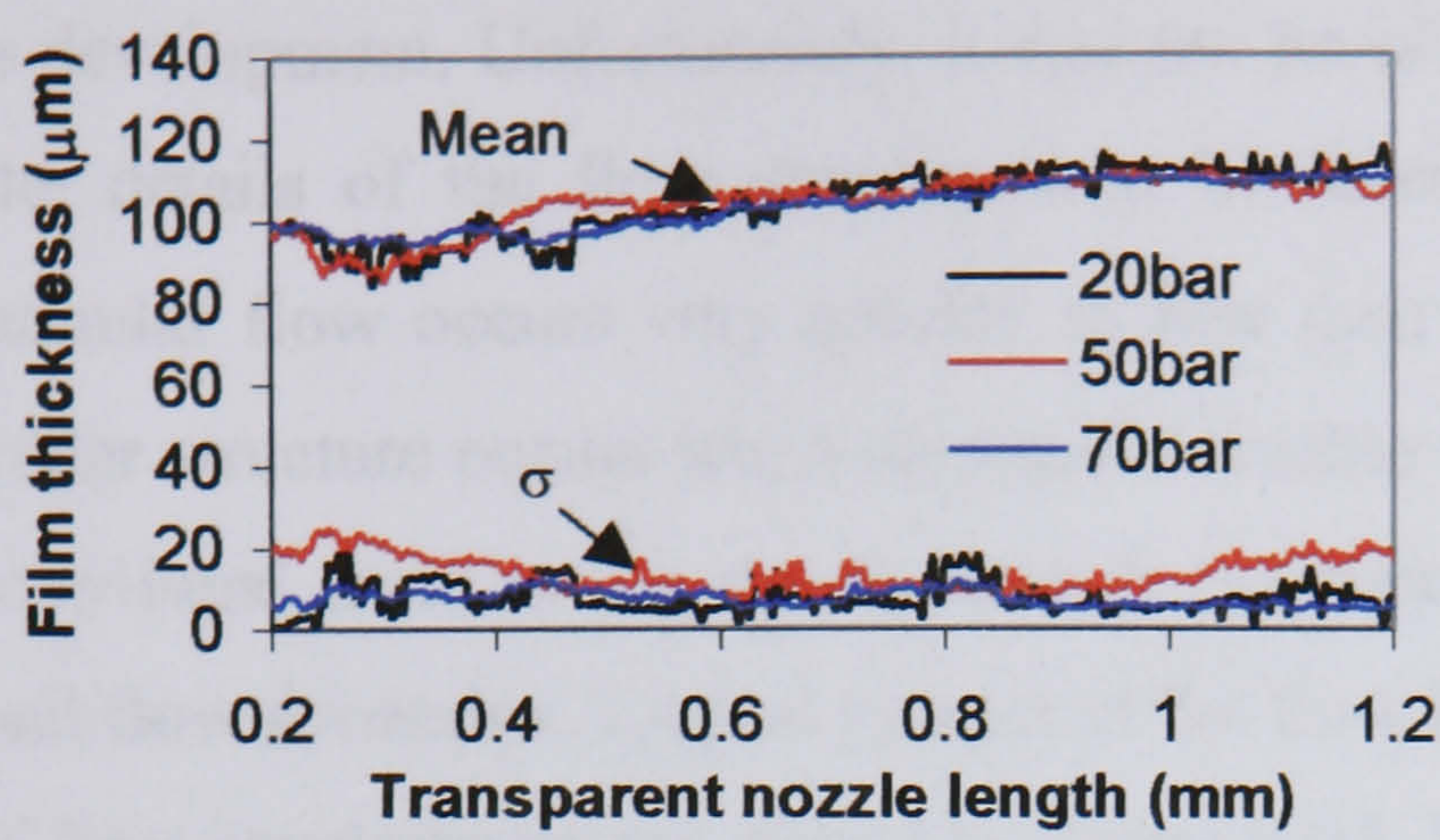


Figure 2-20 Variation of the ensemble average and standard deviation of the film thickness with nozzle length for different injection pressures.



Figure 2-21 Images show the flow direction inside the transparent nozzle for 70 bar injection pressure and 1.5 ms from start of injection.

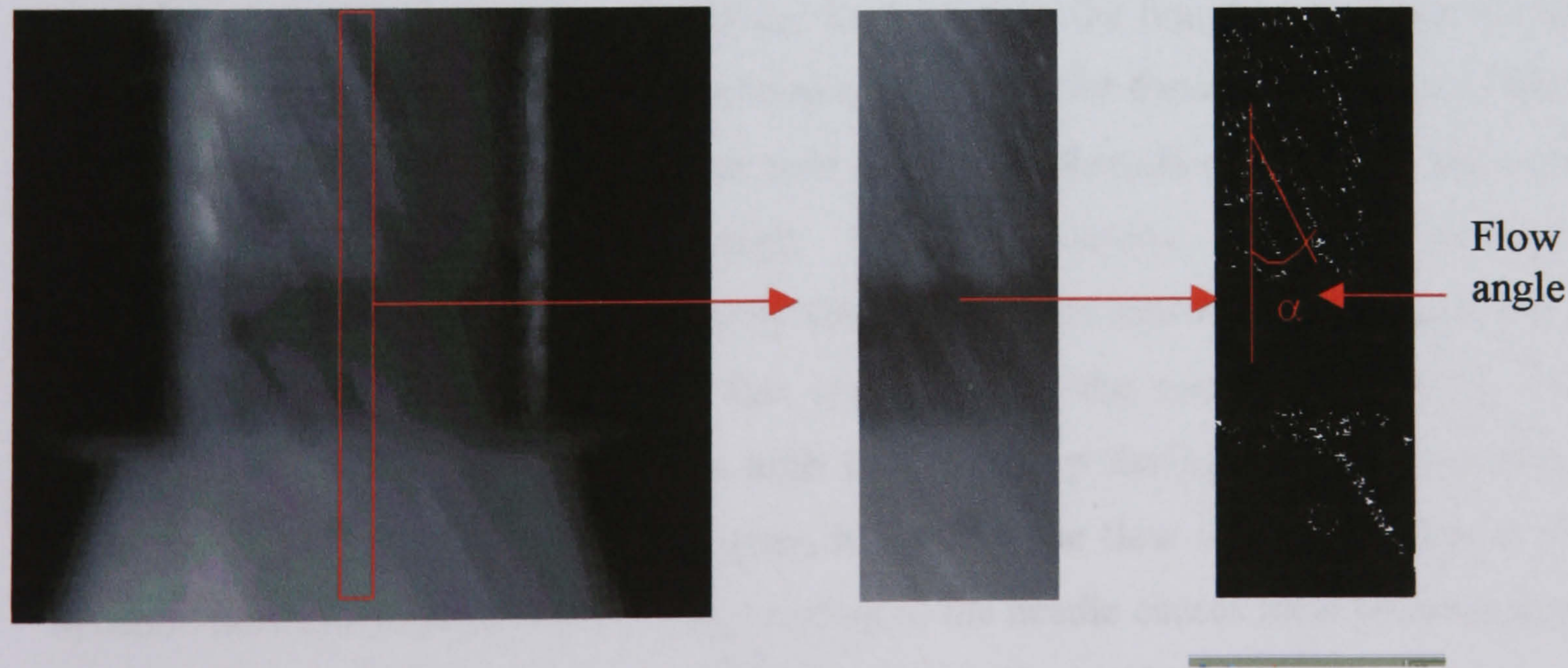


Figure 2-22 Images processing of flow direction inside the transparent nozzle for 70 bar

until the spray angle reaches a constant value. This investigation is aiming to identify the film thickness development. Unfortunately, it was not possible to obtain any useful information on the details of the flow development, however it was clear that the transition to an annular flow occurs very quickly in less than 20 μs . It is also more likely that the annular structure occurs when the needle reaches its maximum lift, since before that the centrifugal force is not strong enough to create a film because of the frictional losses and flow dynamics. Typical images of the flow inside the nozzle during the early stages of film development are shown in Figure 2-23. The reason for having a complete dark area within the nozzle hole is the presence of air trapped inside the liquid as a result of the local flow dynamics.

In order to have some information on how the flow is developing and the direction of the flow, six images have been taken of the outer surface of the flow at each time step for equal time steps of 10 μs from the start of injection; some typical images are shown in Figure 2-24. Analysis of these images has been performed using the edge detection method in order to calculate the flow direction and its variation with time from the start of injection. The increase in flow angle with time starts approximately after 250 μs from the first appearance of fuel at the nozzle exit, which is approximately the time corresponding to the opening of the needle.

As mentioned before, the liquid film starts to appear when the ratio between the swirl to the axial velocity reaches a certain value and when the swirl velocity is large enough to create the associated centrifugal force to push the liquid towards the nozzle wall. Although these two factors are necessary conditions for formation of a liquid film, they are not the main parameters in our case where the injection pressure and the swirl to axial velocity are comparatively high. In fact the hollow cone spray could be established at very low injection pressure (3bar) and confirmed at lower swirl-to-axial velocities by Kashdan (2002). This fact could lead to the conclusion that the low injection pressure resulting from the high pressure drop during the needle opening period is not responsible for the pre-spray, it is rather the flow disturbance due to the dynamic movement of the needle. The opening of the needle causes local pressure drop and fluctuations which can lead to a disruption of the film inside the nozzle.

Avoiding any sharp edges and restrictions in the internal design of the nozzle which could cause flow disturbances may assist in creating a film during needle opening. The velocity profile of the needle during its opening is another factor in the

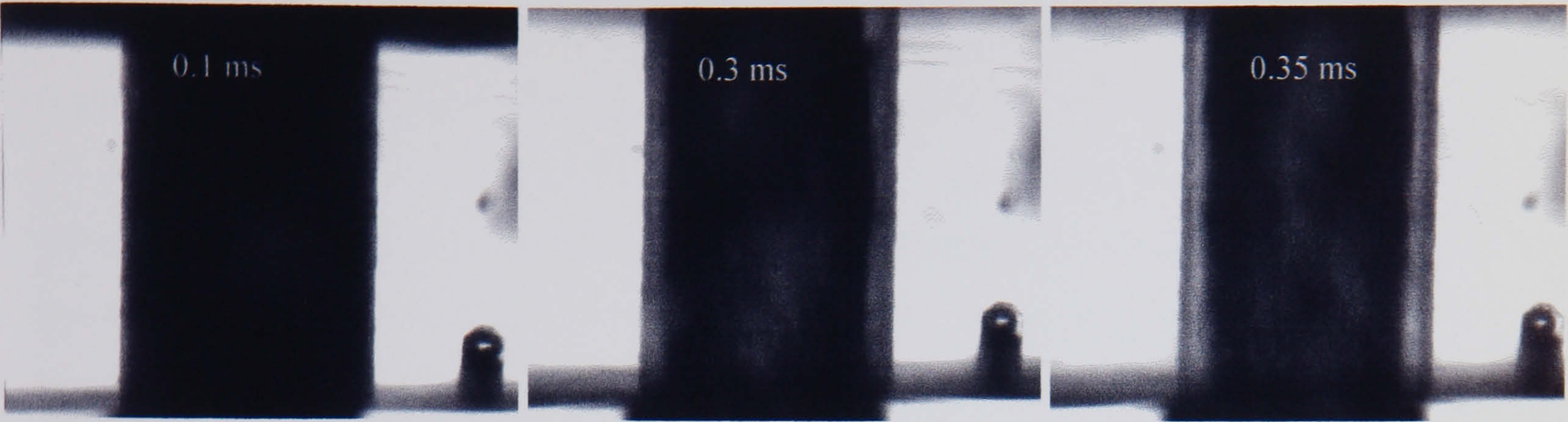


Figure 2-23 Temporal variation of liquid film inside the transparent nozzle.

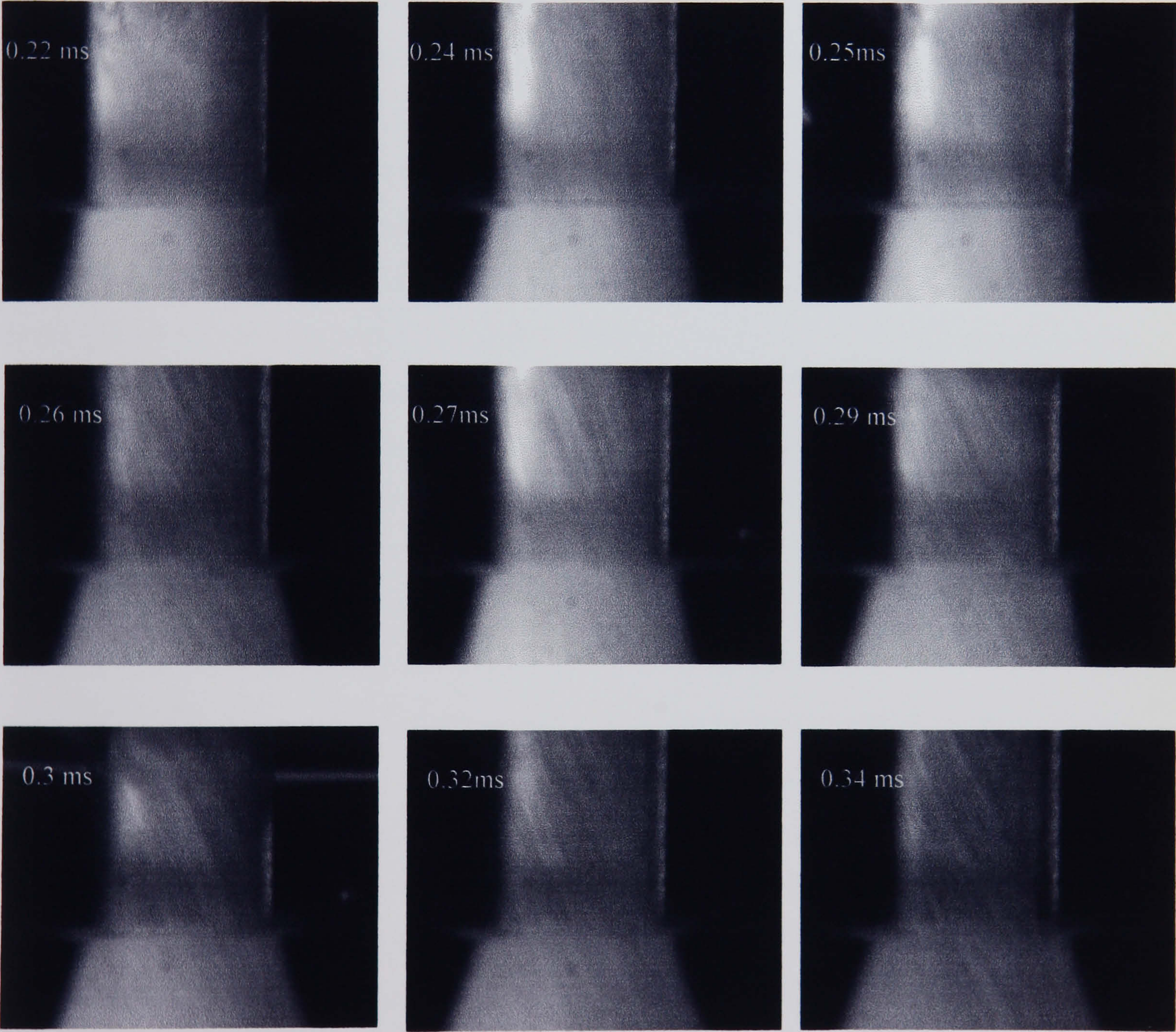


Figure 2-24 Temporal variation of flow angle inside the transparent nozzle.

design. Another direct way of decreasing the pre-spray is by means of decreasing the duration of needle opening; however this could lead to increased amplitude of pressure fluctuations and damping of these fluctuations could take much longer time.

More recent research in DISI injectors has proposed to use outward opening injectors where the film is forced to be formed even during the needle opening period, however the penetration of the spray from this injector could be relatively long and the pressure drop significant.

2.4.9 Errors in Film Thickness Measurements

The image obtained by the CCD camera is two-dimensional, although the light received by the camera comes from a 3-D object. This transformation from 3-D to 2-D depends on a number of independent factors such as surface reflectance properties, illumination properties and surface orientation with respect to the camera and the light source. In order to minimise errors arising from this transformation, the depth of field has to be as minimal as possible. Although, in this study the depth of field ($100\text{ }\mu\text{m}$) is relatively small when using a CF3 lens, there is still some image distortion mainly due to the change of the light pathway which is attributed to the curvature of the interphases and the difference in the refractive indices of the white spirit, Perspex and air (1.438, 1.49 and 1, respectively). In order to eliminate this error a fluid with a similar refractive index as to that of Perspex has to be used. A mixture of 31.8% of Tetraline and 68.2% oil of Turpentine at $25\text{ }^{\circ}\text{C}$ has proved to have the same refractive index as Perspex of 1.49 (Nouri et al., 1988). However, this mixture attacks any rubber material or gaskets which makes it unsuitable for use inside the injection system. Since it was not possible to eliminate the error in this way it became necessary to quantify it. A separate simple experiment aiming to form a static film thickness in a non-operating transparent nozzle, instead of the actual dynamic film, was thus designed to quantify this error.

A transparent calibration nozzle similar to that attached to the injector has been constructed with four different diameters (0.6, 0.7, 0.8 and 0.9 mm) and is shown in Figure 2-25. This construction allows examination of film thicknesses of 150, 100 and $50\text{ }\mu\text{m}$ by imaging the transparent nozzle after inserting a pin of 0.6, 0.7 or 0.8 mm diameter respectively. The distance between the inserted pin and the 0.9 mm hole diameter is filled either by white spirit or by the mixture. By using the same imaging technique and the same magnification, the distance between the pin and the hole edge,

which represents the film thickness, could be measured directly from the images. The difference between the film thickness when using the fluid mixture and the white spirit provides an estimate of the error arising due to the light distortion under static conditions.

Figure 2-26 shows typical images of the film using white spirit and the fluid mixture in case of the 100 μm film thickness. It is clear that there are some noticeable differences in the light intensity between the Perspex and the mixture. This difference is attributed to Perspex surface finish which gives some light reflection, and also to the small difference in the refractive index which arises due to the small change in the laboratory temperature of 23 $^{\circ}\text{C}$ from the temperature at which the refractive indices of the mixture and Perspex are matched (25 $^{\circ}\text{C}$). Similar images for 50 μm and 150 μm film thickness are presented in Figures 2-27 and 2-28. In order to achieve more accurate results, the average thickness has been calculated along the nozzle length so that only one value of the film thickness for each image is estimated and then the average of five images is estimated and presented assuming uniform film thickness around the nozzle hole.

This error for 50 μm , 100 μm and 150 μm film thicknesses proved to be an overestimation by 10.5, 8.2 and 4.4 % respectively, of the actual film thickness measured for the fluid with the same refractive index. Some other errors have also arisen from the experiment itself which include eccentricity in the holes, the temperature effect on the refractive index and errors from the edge detection method itself, which can be significant especially in the smallest film thickness case of 50 μm . Therefore analytical calculations could be very useful to prevent these experimental errors and allow estimation of the error in film thickness for any refractive index fluid and for any film thickness.

In order to find a relationship between the error in the film thickness and the film thickness itself, analytical analysis has been performed based on the light pathway inside the transparent nozzle (i.e. geometric optics). Because the light source used in these experiments is not coherent, it is difficult to start following the light pathway from the light source towards the object, on the other hand the light received by the camera has been analysed by three different methods. Each method depends on assuming a certain direction of the light source when arriving at the camera front lens. The first method considers the light passing through the transparent nozzle and arriving at the

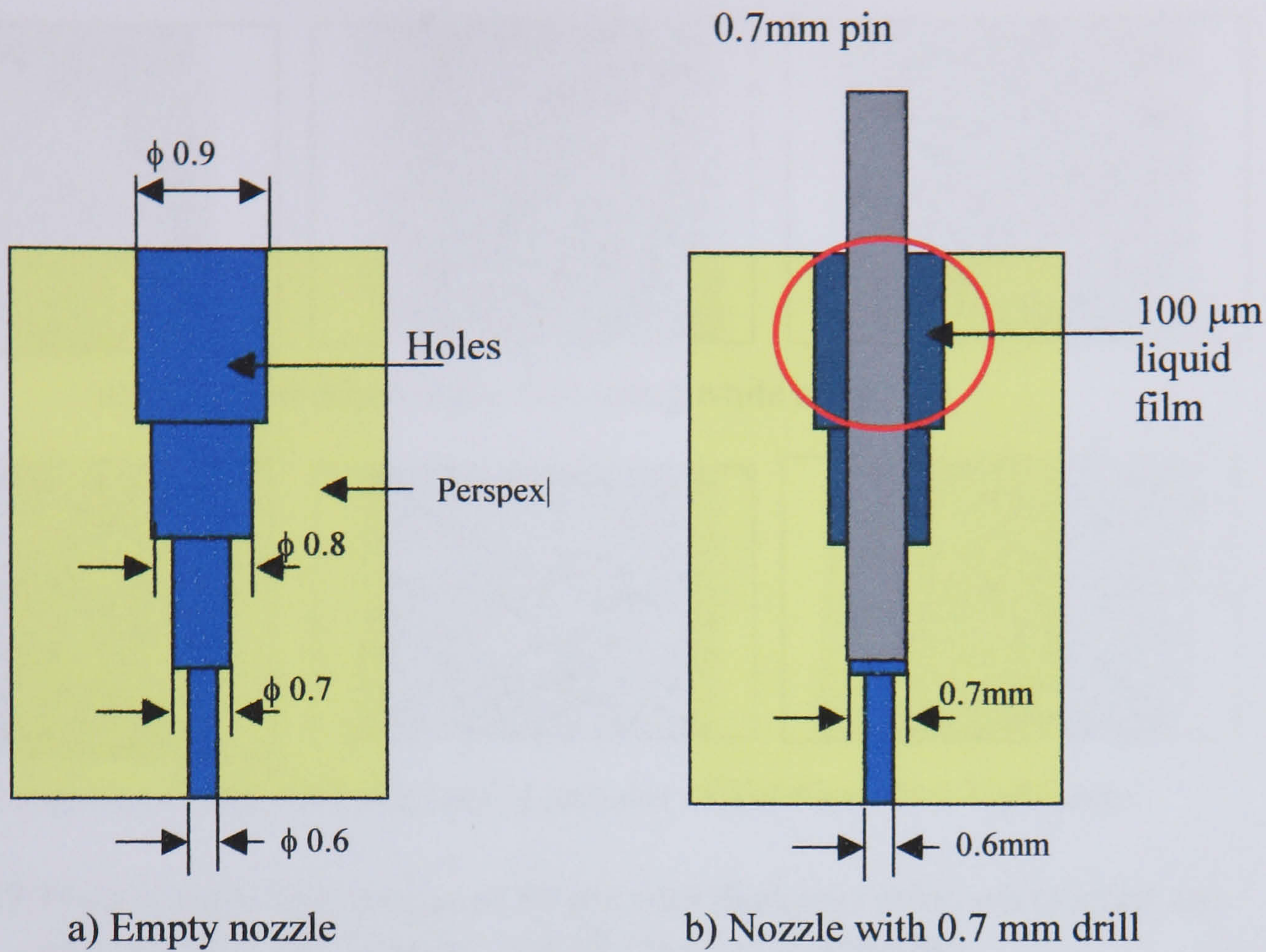


Figure 2-25 A schematic showing the construction of the calibration transparent nozzle.

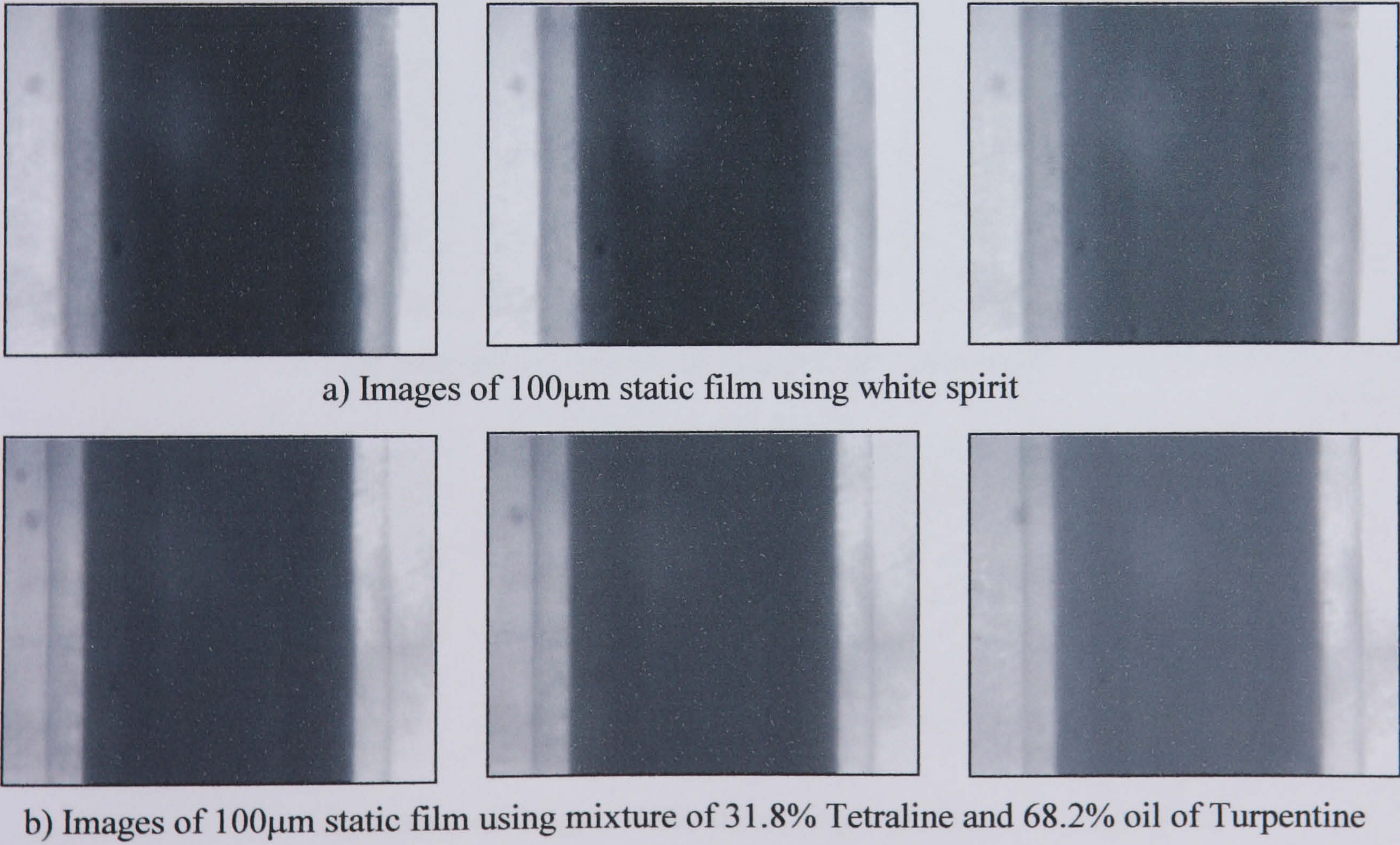


Figure 2-26 Typical static film images of 100 μ m film thickness using white spirit and mixture of 31.8% Tetraline and 68.2% oil of Turpentine.

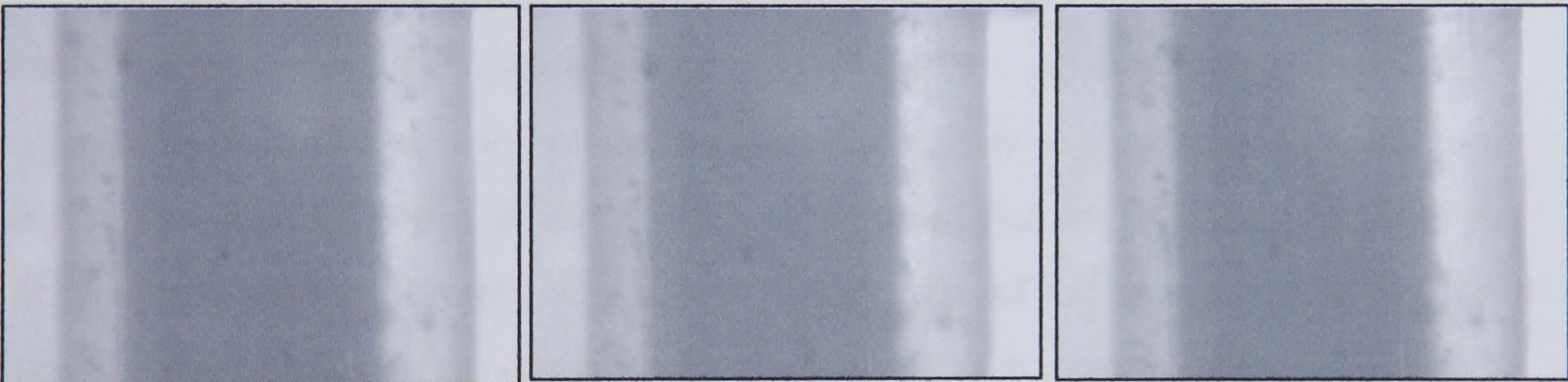


a) Images of 50µm static film using white spirit



b) Images of 50µm static film using mixuter of Tetrolene and Turbintene

Figure 2-27 Typical static film images of 50 µm film thickness using white spirit and mixture of 31.8% Tetraline and 68.2% oil of Turpentine.



a) Images of 150µm static film using white spirit



b) Images of 150µm static film using Tetrolene and Turbintene

Figure 2-28 Typical static film images of 150 µm film thickness using white spirit and mixture of 31.8% Tetraline and 68.2% oil of Turpentine.

front lens parallel to the centre of the camera lens. The second method considers the light travelling from the object to arrive at the centre of the front lens of the camera (convergent). The third method assumes that the image is printed on the whole front lens of the camera which implies that the light is diffused from the object until it reaches the front lens (divergent); in fact, the light arrives at the front lens from all directions. These three methods of study thus cover all other directions for light to be received from the object.

In order to perform the above analysis, the geometric distances have to be known, as shown in Figure 2-29, for both convergent and divergent light pathways. The distance between the camera and the nozzle centre is denoted by (L), the front lens diameter by (D_1). The camera has been located so that its centre is coincident with the centre of the nozzle hole cross section. The transparent nozzle (centred at point o) has a hole of radius (R) equal to the injector nozzle radius (0.45mm) and its front face is located at a distance (w) from the centre of the hole. The film thickness is calculated as the difference between the nozzle hole radius and the inner radius (r) of the liquid film. The refractive indices of white spirit and Perspex are defined as n_l and n_g , respectively.

Having defined all the geometric distances, an analysis has been performed to transfer the location of the outer edge (hole edge) of the image to the real nozzle and a similar analysis has been done to transfer the other points in the image. In order to do that, the locations of points B and C have to be identified by following the light ray ABCD which passes through the cylindrical hole edge of the transparent nozzle and enters the front lens at a pre-specified distance (y) from the centre of the camera. The mathematical analysis of the light pathway inside the transparent nozzle and towards the front lens of the camera for the convergent light method is described in appendix 2-B. A similar analysis has been performed for both parallel and divergent light in order to identify the light pathway and allow the calculation of the error in the film thickness measurements.

Figure 2-30 shows the light pathway for both convergent and divergent light beams. In both cases, the light diverges as it passes from the white spirit to the transparent nozzle on its way to reach the camera with a thickness larger than the actual thickness inside the nozzle. This deviation in film thickness at the image becomes larger as the film thickness decreases. In order to identify the error for different any film thickness and fluids a program has been established using Matlab to calculate the error for different input data.

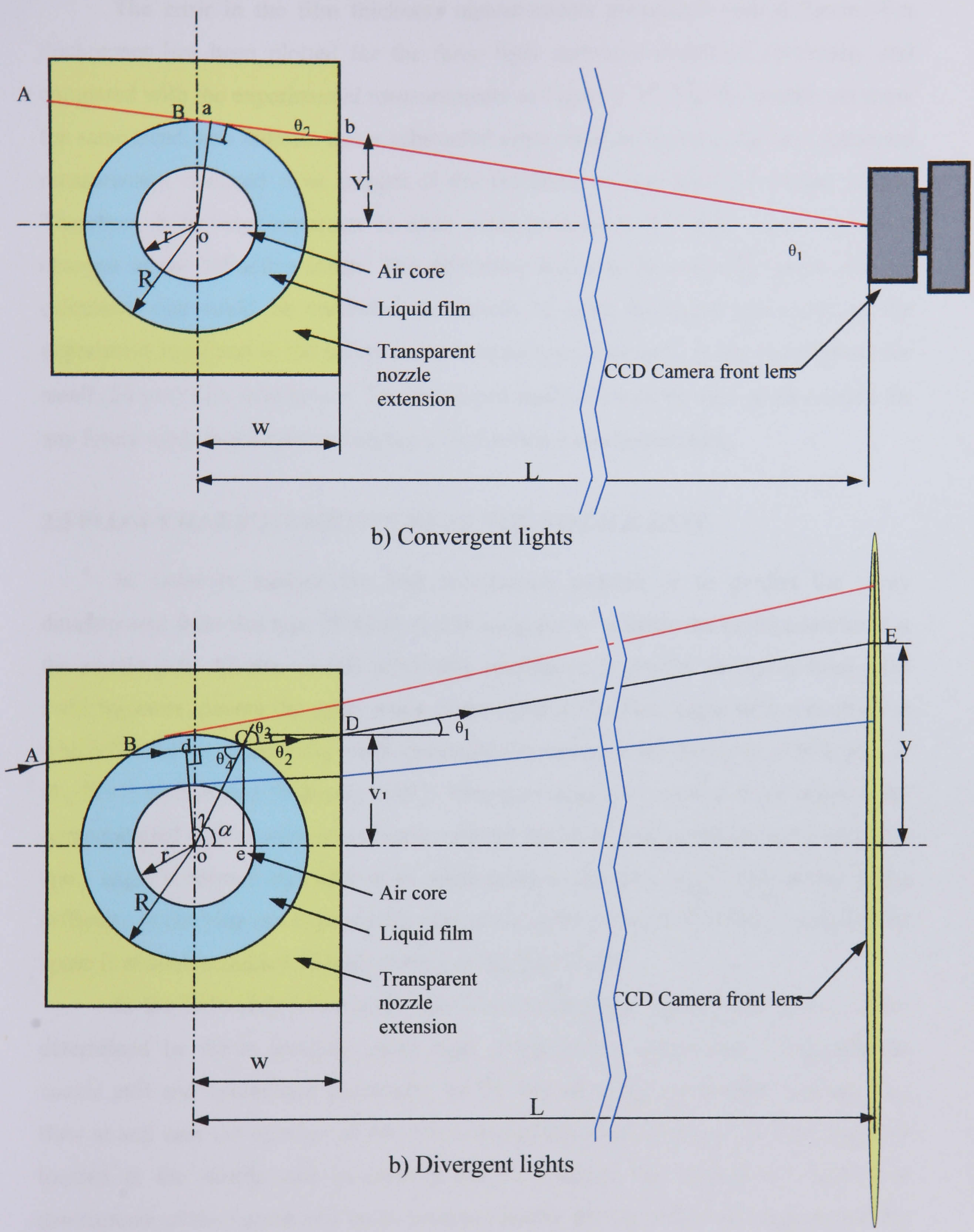


Figure 2.29 Geometrical analysis of the light pathway for both convergent and divergent light rays.

The error in the film thickness measurements associated with different film thicknesses has been plotted for the three light pathways discussed previously and compared with the experimental measurements in Figure 2-31. The three methods show the same trend, and indicate that a substantial error could be found in the film thickness measurement deduced from images if the thickness is less than 0.1 of the radius. Therefore, it is very important in such measurements to consider even very small changes in the refractive index. The difference between the measured error and the calculated one could be attributed to sources of error discussed previously in the experiment itself and in the image analysis technique; especially in the experiments for small (50 μm) film thicknesses. The developed analysis could be used and modified for any future work that involves imaging a film within a transparent tube.

2.5 FLOW CHARACTERISTICS NEAR THE NOZZLE EXIT

In order to analyse the fuel atomisation process or to predict the spray development from this type of injector, it is compulsory to know the initial conditions at the nozzle exit. All the models previously introduced to predict the spray from DISI swirl injectors assume the spray angle to be equal to the flow angle with a coefficient which fits the experimentally measured angle (for example see Ren et al., 1997; Han et al., 1997; Cousin and Nuglisch, 2001). The spray angle is included as an input to the computational model without any experimental evidence and justification for how this spray angle is formed and what is its relationship to the flow angle. This is due to the difficulty in carrying out experiments very close to the nozzle exit where a very limited space is available and a very high density of the fluid exists.

In the following subsections, the flow conditions at the nozzle exit have been determined by direct imaging using high magnification lenses just underneath the nozzle exit and subsequent processing by the Matlab image processing toolbox. The flow at and near the nozzle exit has been divided into three regions: the first region is located at the nozzle exit to identify the flow angle, the second is located just downstream of the nozzle exit up to 1mm to identify the liquid film divergence, and the third is located between 1mm and 5mm downstream of the nozzle exit to identify the nozzle spray angle. The experimental results presented in the following sub-sections have been used in Chapter 3 to predict the liquid film trajectory and also used as input data for the atomisation model based on a linear instability analysis.

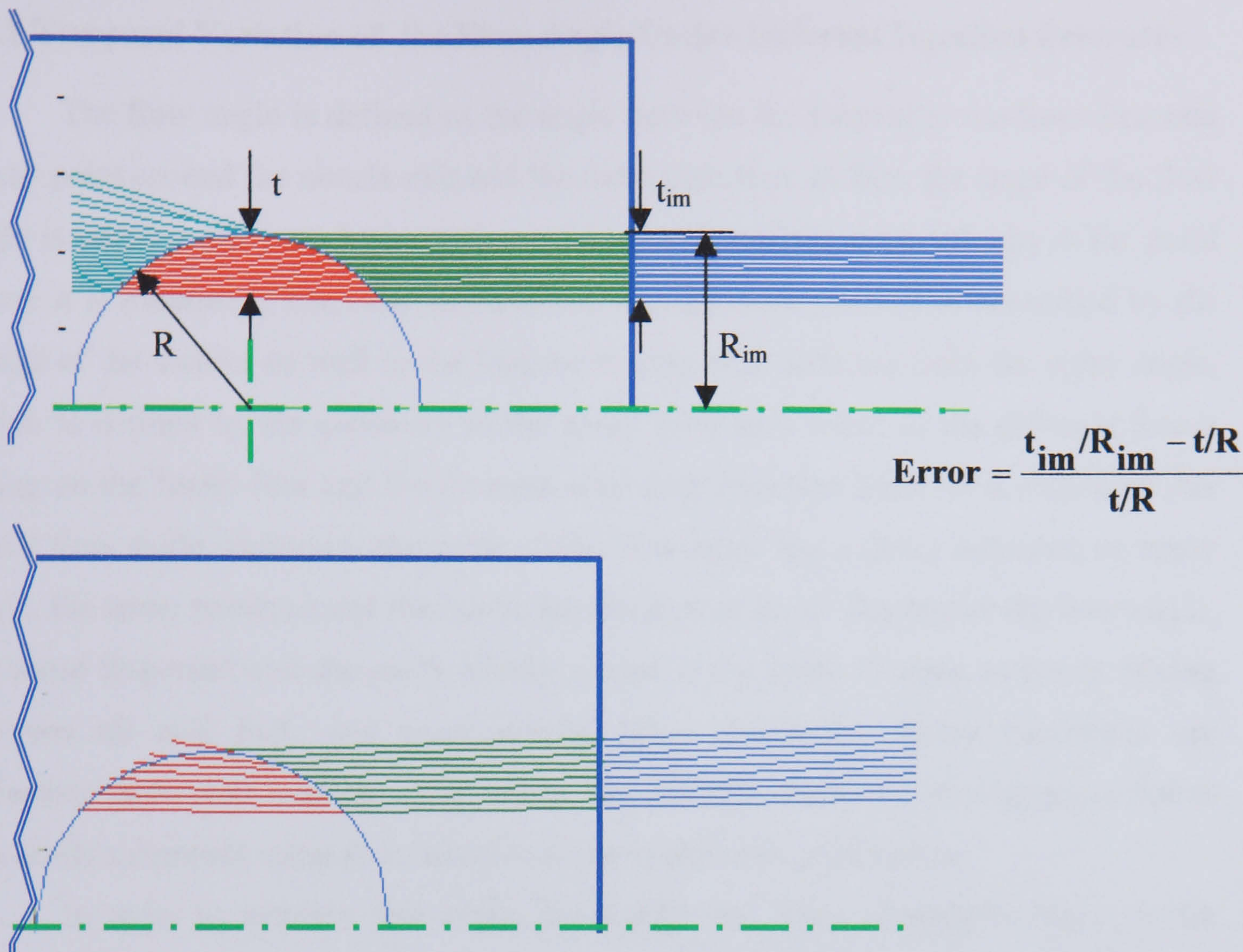


Figure 2-30 Top view of convergent and divergent light pathway inside the transparent nozzle when using white spirit.

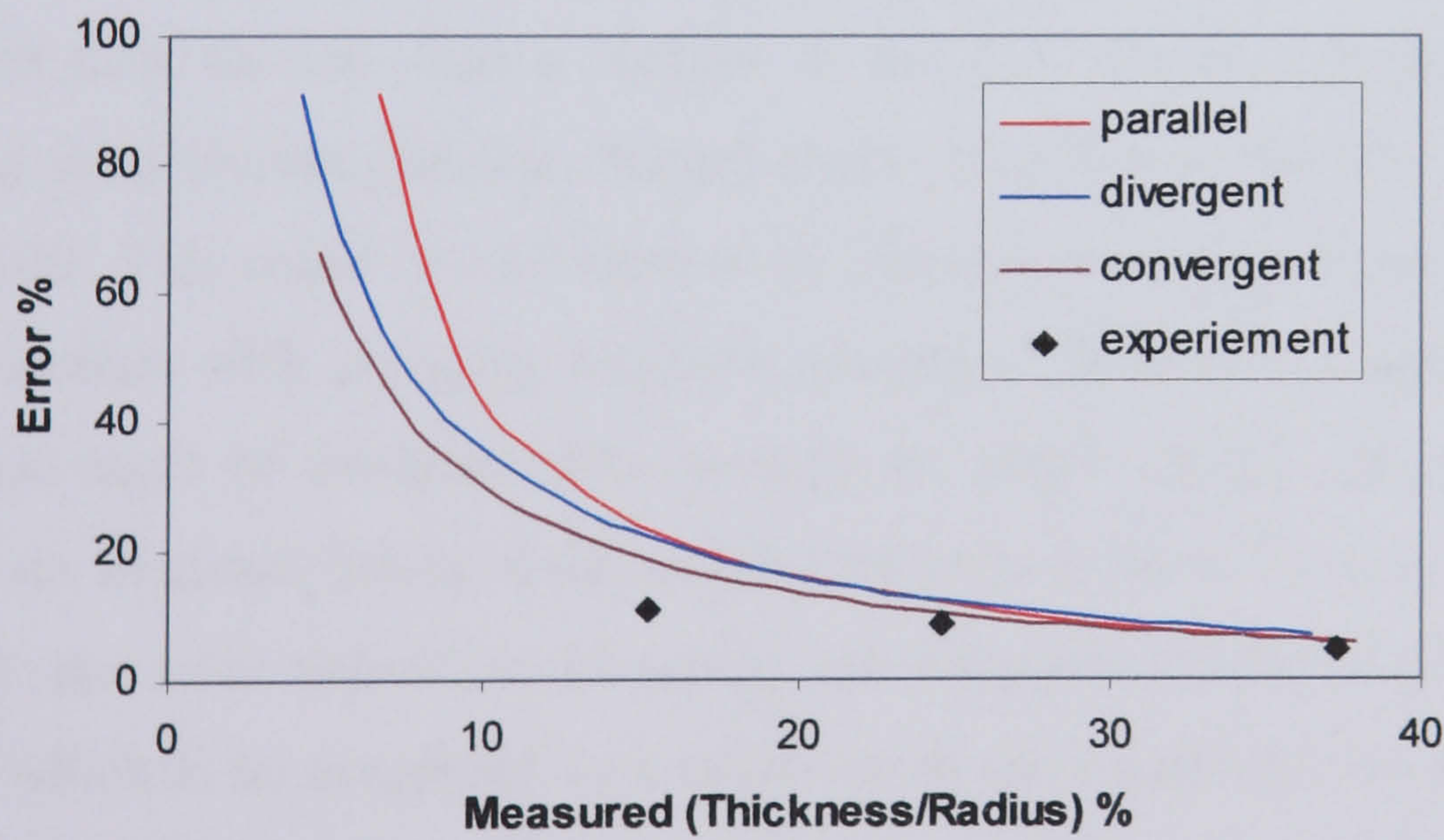


Figure 2-31 Variation of the error with film thickness when using white spirit.

2.5.1 Temporal Variation of the Flow Angle Under Different Injection Pressures

The flow angle is defined as the angle between the tangent to the flow direction at any point around the nozzle exit and the axial direction. In fact, the slope of the flow angle is equal to the ratio between the swirl velocity and the axial velocity at the point where it is measured. The flow angle is a design parameter which is controlled by the design of the swirler as well as the injector nozzle. It is different from the spray angle, which is defined by the geometry of the spray itself as a result of the different forces acting on the liquid film and the droplets after their injection from the nozzle with this initial flow angle. However, the value of the flow angle has a direct influence on spray angle, the spray structure and the whole atomisation process. The higher the flow angle, the more dispersed and the more widely spread is the spray, leading to better mixing between air and fuel, and consequently, lower emissions. These conditions are necessary under full load operation where the mixture should be homogeneous but at low loads a compact spray is essential in order to achieve stratification.

In order to measure this angle, the camera has been adjusted to focus on the outer surface of the annular flow just underneath the nozzle exit using forward lighting. The images obtained using this technique showed marks that clearly indicate the flow direction, as shown in Figure 2-32. These marks appeared on the image by increasing the exposure time so that the time of imaging is longer than the time required to freeze the liquid flow motion, thus a shadow of the flow stream appeared on the images. Changing the injection pressure showed almost no effect on the flow angle as shown in Figure 2-32. This could be attributed to the nearly constant ratio between the axial and swirl velocities with changing injection pressure. This ratio depends mainly on the inclination angle of the holes/slots creating the swirl velocity upstream in the swirler and also on frictional losses in the nozzle that seem to have the same effect in reducing the swirl and axial velocities. However, the reduction in the magnitude of the swirl velocity which is accompanied by a reduction in the centrifugal force causes a delay in film divergent thus, a longer distance downstream of the nozzle is noticed to achieve the same radial divergence as that in the high injection pressure case. This reduction does not affect the spray angle, which is measured far downstream of the nozzle as will be seen in the images obtained for the fully developed spray in Chapter 4

In order to have quantitative data for the flow angle development, images have been taken every 20 μs for an injection pressure of 70 bar, as shown in Figure 2-33. These images have been analysed using the edge detection method discussed before and binary images have resulted from the analysis of chopping windows of 0.15mm width and 0.2 mm length located at the nozzle centre in the original image. The resultant images illustrate marks that represent the flow direction, as shown in Figure 2-34. The slope of these lines has been measured to determine the flow angle for each image and the average flow angle is then calculated out of five different images for each time step. The variation of this average flow angle during liquid film or spray development stage is shown in Figure 2-35. The time required to reach almost the maximum flow angle is found to be almost the same as the time required for the needle to reach its full lift. Therefore it is important to reduce the opening time of the needle in order to reduce the flow angle developing time and hence, the quantity of fuel injected during this period, which is associated with relatively poor atomisation. Reducing the time of the developing stage becomes crucial during the late injection (during the compression stroke) when the quantity of fuel has to be injected very quickly and when the needle opening time (0.25ms) relative to the whole injection duration (1-1.5 ms) can be substantial.

2.5.2 Liquid Film Profile at the Nozzle Exit

According to the experimental and theoretical study performed by Sivakumar and Raghunandan (1997) and Ramamaurthi and Tharakan (1998), the fuel injected from the nozzle goes through two stages as the injection velocity increases before reaching its final annular divergent profile. In the first very early stage of needle opening, the fuel is injected asymmetrically as a bulk of non-atomised liquid and there is no liquid film; in the second stage the liquid film exhibits a closing annular convergent profile as a result of the increase in injection velocity but its associate inertia force is not enough to overcome the surface tension force which pushes the liquid towards the centre of the nozzle to form a closed annular jet. Although the images of the spray during the needle opening period do show these two stages, it is believed that the film is been disrupted inside the nozzle before been injected outside and the shape, which is similar to the second stage, is formed by the droplets which acquired some tangential velocities; the end results is not a hollow structure but a full spray structure. During these two stages

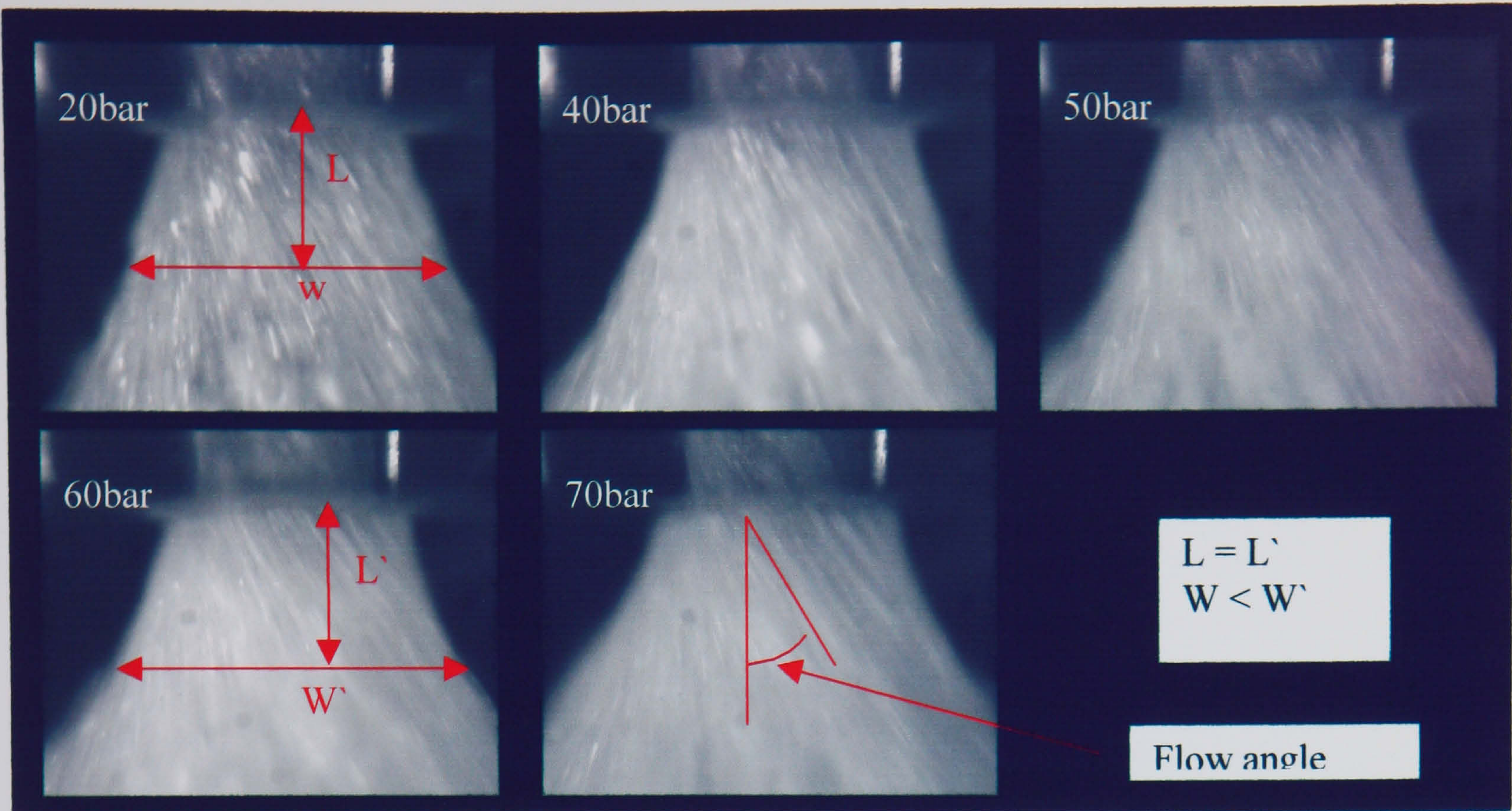


Figure 2-32 Effect of injection pressure on flow characteristics near to the nozzle exit.

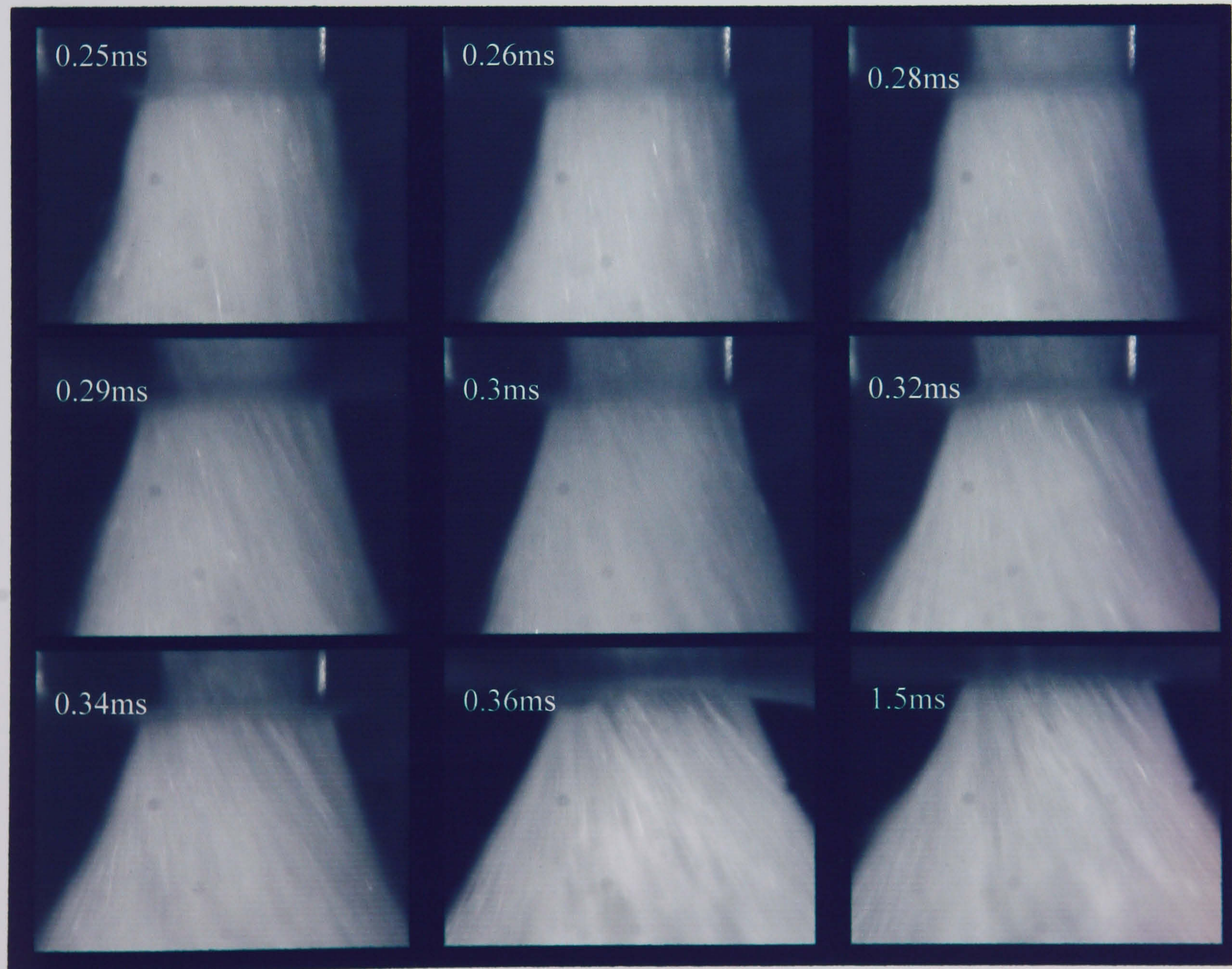


Figure 2-33 Development of the flow angle near to the nozzle exit.

the liquid experiences very poor atomisation due to the dynamic influence of the needle, which affects the flow turbulence level and causes film disintegration inside the nozzle. After these two stages and due to the increase in the injection velocity and the full opening of the needle, the liquid film is formed inside the nozzle and then injected outside the nozzle where it breaks-up forming droplets of different sizes. After the film disintegration, the angular momentum of the liquid film is converted into linear momentum, which pushes the atomised droplets with an initial velocity that depends on the momentum of the liquid film before its disintegration, into the surrounding air. Identifying this initial velocity of the droplets is very important in modelling the spray development irrespective of the spray model used.

In order to identify the initial direction and velocity of the droplets resulting from the liquid film break-up, the injection velocity and the profile of the liquid film have to be known as well as the location of the disintegration or the break-up length. Although it is possible to identify the liquid film profile, it is experimentally very difficult to identify where the liquid film starts to break-up due to the high density of the liquid droplets at this location. These droplets reflect the light in all directions preventing the acquisition of a clear image, moreover the velocity of the droplets is relatively high which requires a very small exposure time and high light intensity. Unfortunately, it proved impossible to obtain any images that show the location of the break-up. However, the overall image of the liquid film profile with the analysis of the centrifugal forces besides other forces such as surface tension, inertia, viscous, etc. which affect the liquid film at this stage can provide some useful information for identifying approximately the location of film break-up.

In order to identify the liquid profile during the developing stages, images have been obtained very near to the nozzle exit, up to 1.5 mm downstream of the nozzle exit, using a CF3 lens at every 10 μ s from the start of injection until the liquid film became fully developed. These images have been analysed by selecting a threshold value converting all the images into binary ones, thus making possible the liquid flow profile at the nozzle exit to be obtained. A sample of these images is shown in Figure 2-36 that reveals the outer profile at different times. Although these images show continuous divergence, further downstream after a certain distance the divergent angle starts to decrease with increasing distance from the nozzle exit during the early stage of spray development. The full picture of the spray confirms that collapse occurs and a closed

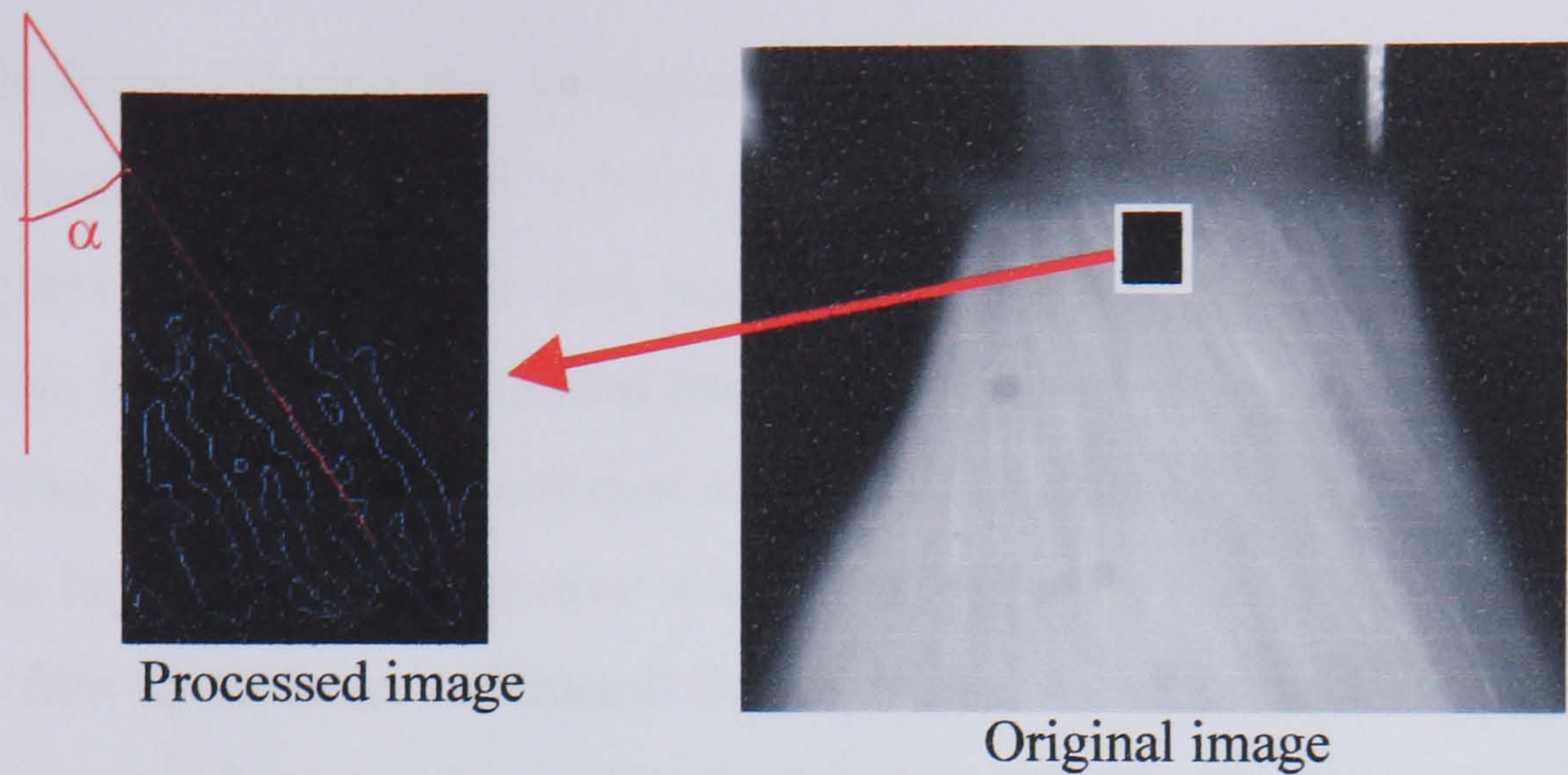


Figure 2-34 Measuring the flow angle by image processing.

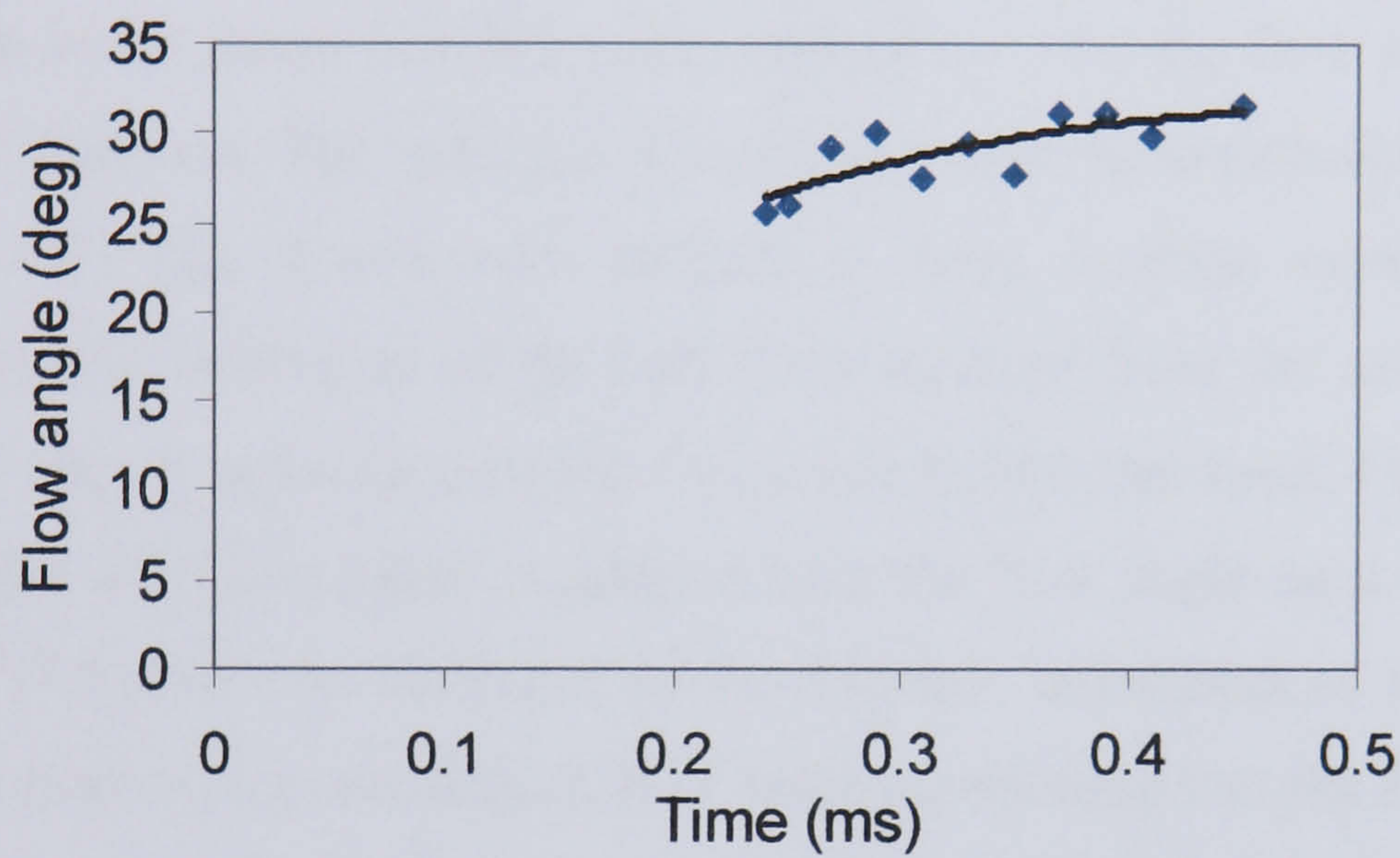


Figure 2-35 Flow angle development at the nozzle exit.

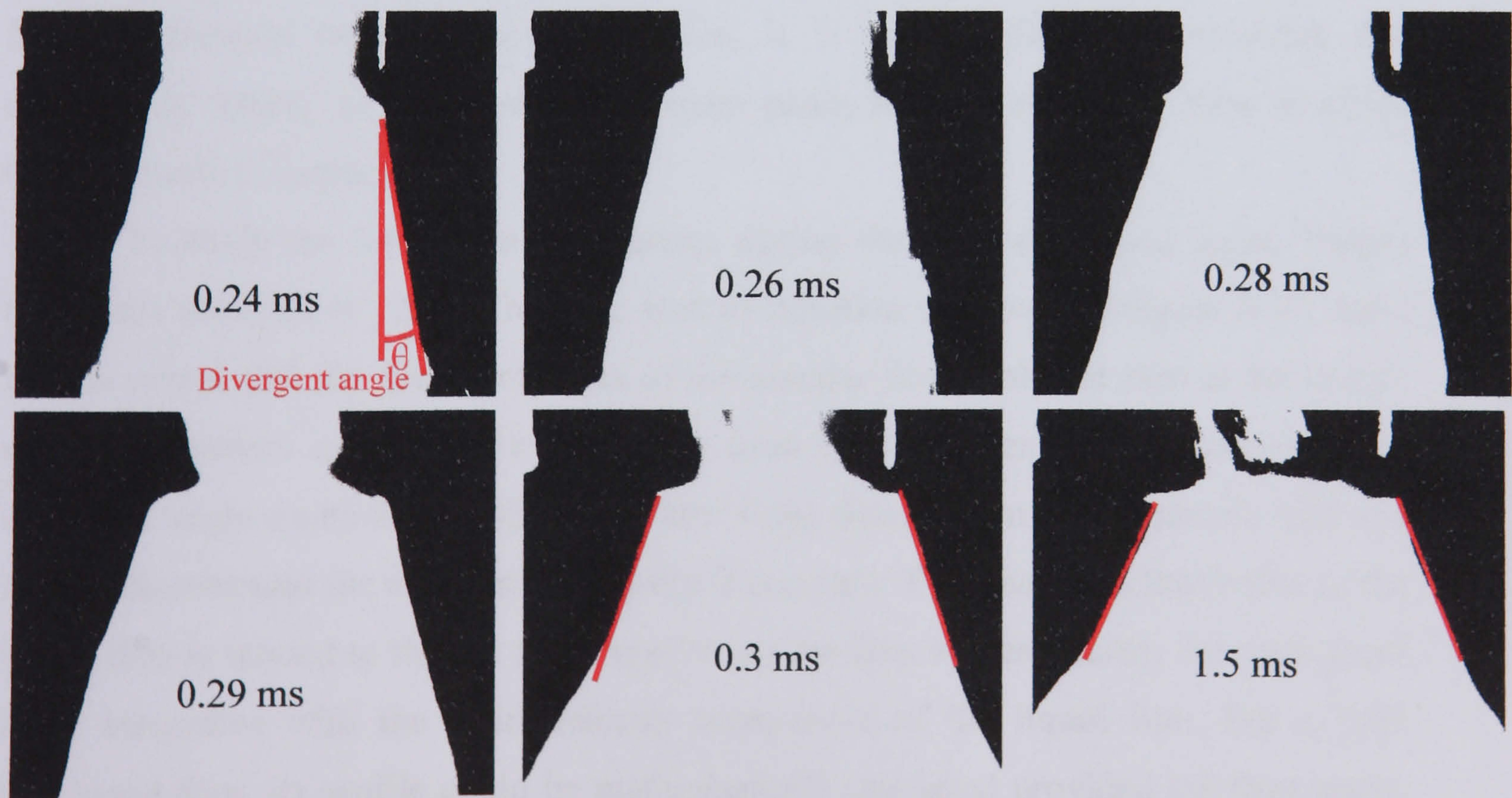


Figure 2-36 Analysed images show the liquid film profile during the developing stage.

annular profile is formed during the developing stage as shown in Figure 2-37. The images of the whole spray are not indicating presence of liquid film but mainly existence of droplets which have been atomised earlier. Therefore, the shape of the full spray is due to the larger droplets injected earlier which move faster than the rest of spray and could also be related to the collapse of the liquid film itself.

In order to have a more quantitative analysis of the liquid film development, the trajectory of the film up to 1mm underneath the nozzle exit has been approximated by linear profile. The angle between this approximated line and the nozzle axis has been measured for five images at each time to calculate the mean angle. Figure 2-38 shows the temporal variation in the mean film divergence angle and also the flow angle after 0.23 ms from start of injection. The standard deviation has also been estimated as 0.6-1.8 degrees. Before 0.23 ms, it was very difficult to have accurate measurements because of the asymmetric behaviour of the bulk flow injected from the nozzle. It is also believed that the film is destructed inside the nozzle before that time. Figure 2-38 also shows that the film divergent angle is delayed from the flow angle until it reached its developed value. This could be attributed to the dynamic movement of the needle that prevents the film from being established. It is worth mentioning that the flow angle represents the liquid film motion in the circumferential direction while the film divergent angle represents the liquid film motion in the radial direction. Although there is a relationship between the two angles, it is rather difficult to establish this relationship during the transient development phase and to identify the time at which the film starts to appear.

To study the liquid film divergence during the fully developed stage, images have been obtained at 1.5 ms from the start of injection as shown in Figure 2-39; these images reveal that the divergent angle of the annular film is almost zero at the nozzle exit and increases as the film moves away from the nozzle exit. This increase in the divergent angle continues to approximately 1mm downstream of the nozzle exit and further downstream the angle becomes almost constant. This change in the profile of the liquid film is related to the net force applied on the film and especially, the centrifugal force associated with the swirl velocity component of the liquid film. For a fully developed film, its profile could be mathematically predicted provided the flow angle, injection velocity and liquid film thickness at the nozzle exit are known.

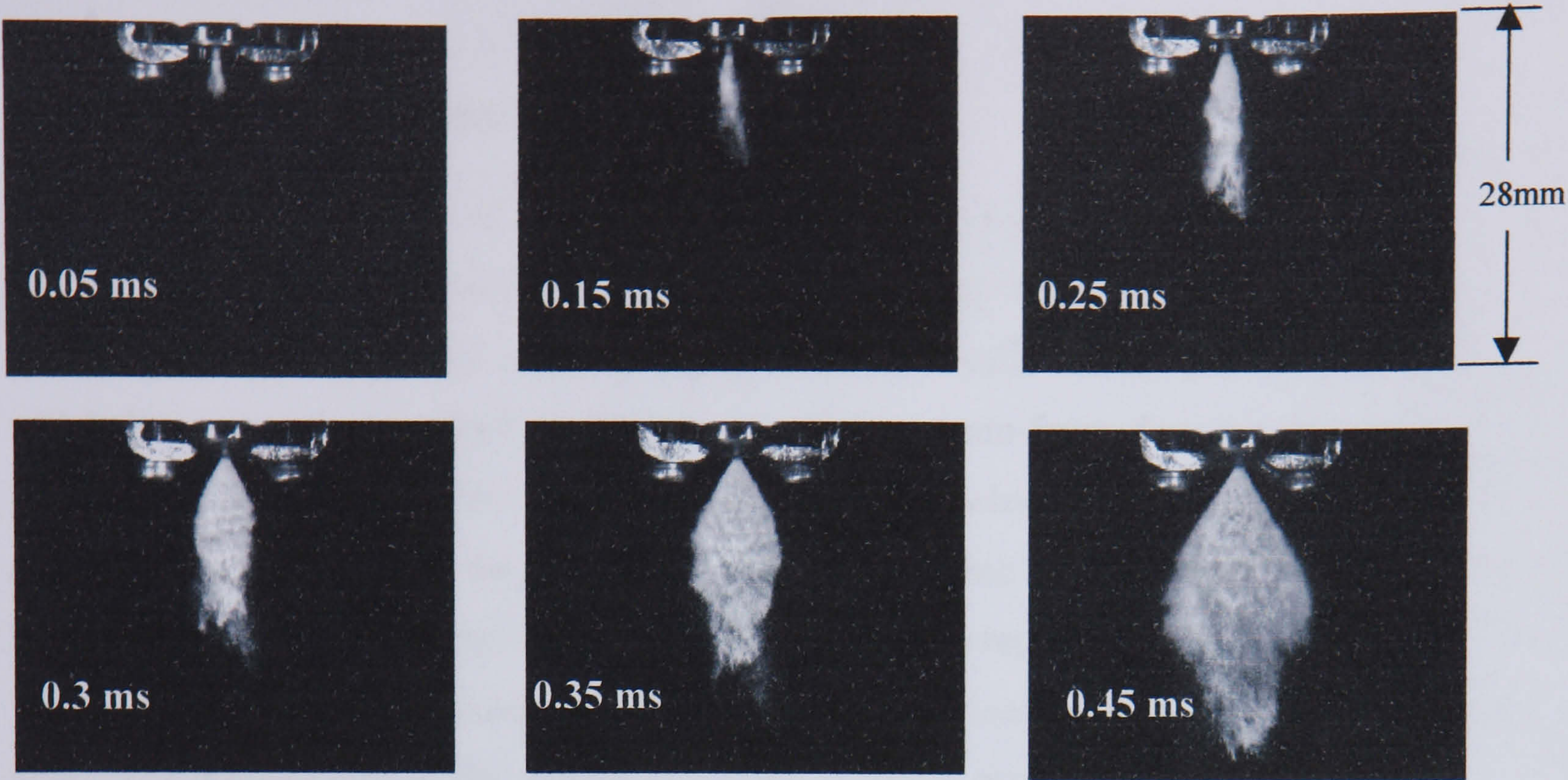


Figure 2-37 Full picture of the spray during the developing stage (70 bar).

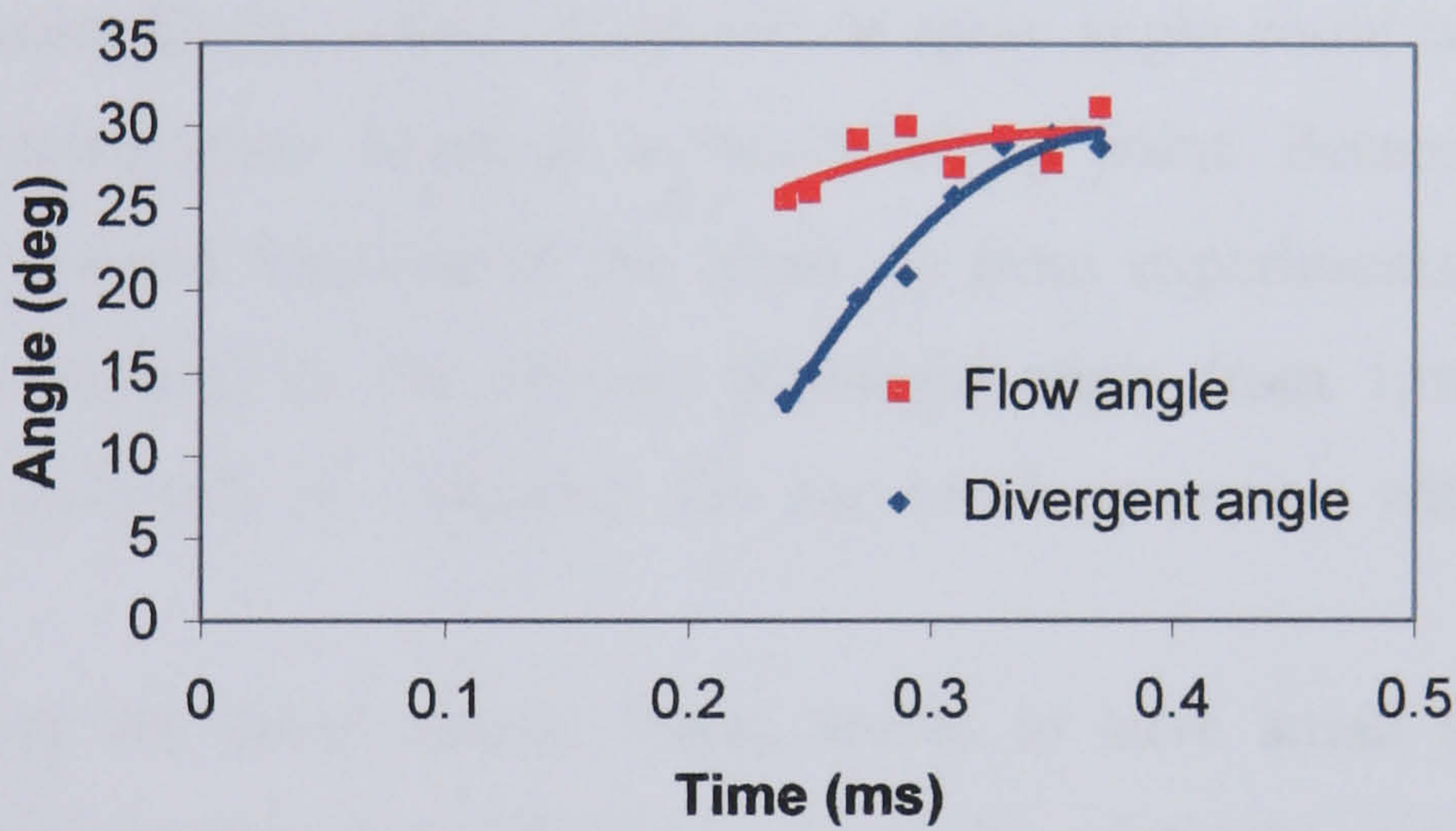


Figure 2-38 Development of flow and divergent angles.

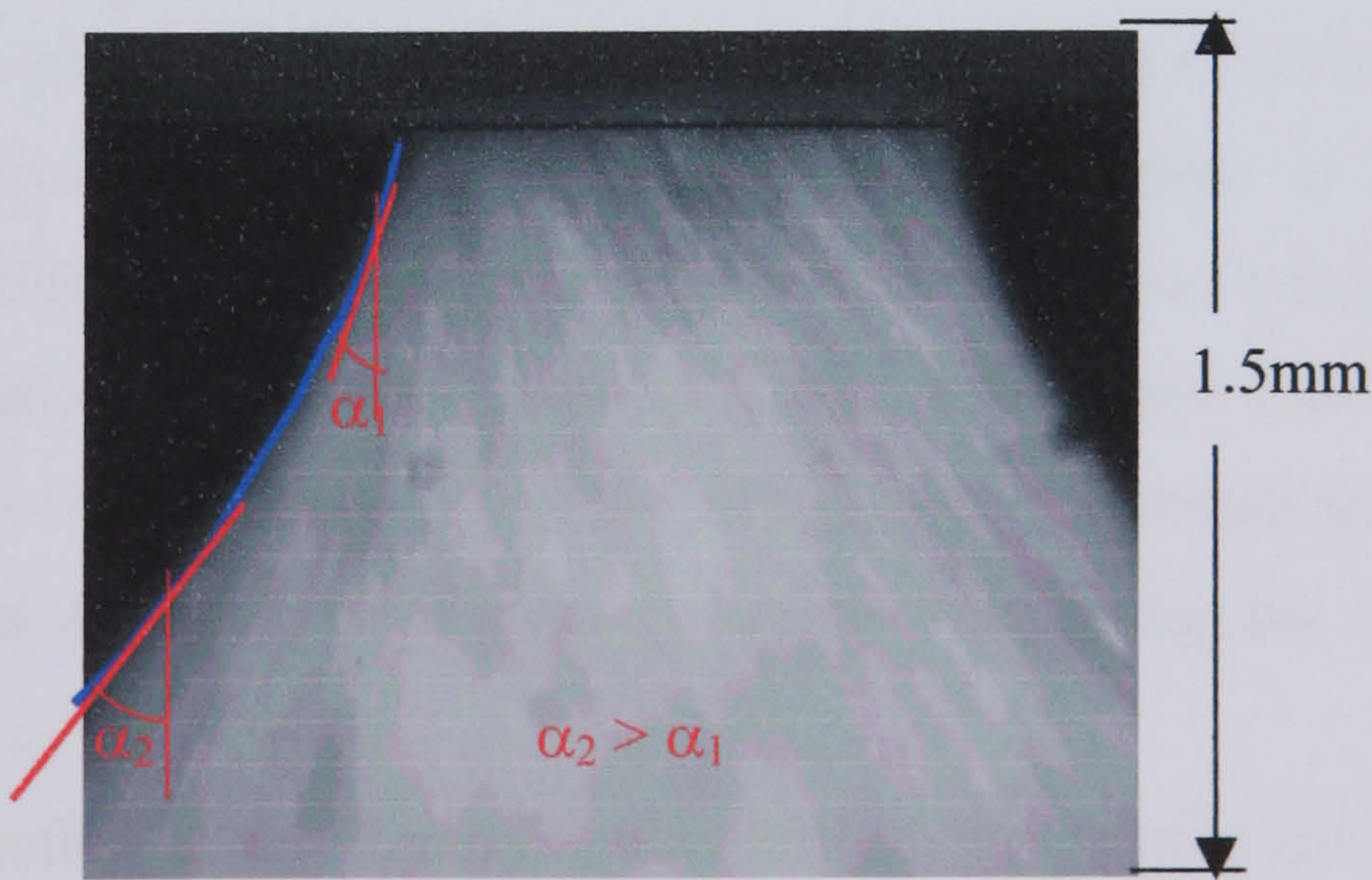


Figure 2-39 Fully developed spray profile underneath the nozzle exit.

2.5.3 NOZZLE SPRAY ANGLE

The spray angle could be defined in different ways depending on the location and the method used to measure it. In this study employing direct imaging method, the spray angle has been divided into two angles. The first is called the nozzle spray angle, which is measured very close to the nozzle exit, i.e. 1mm-5mm downstream of the nozzle-exit and the other is the conventional spray angle, which is measured between 5 and 15 mm downstream of the nozzle. The reason for having these two angles is due to the difference in the major factors affecting each of the two regions.

The liquid film diverges with an angle that decreases or increases as it penetrates downstream of the nozzle exit until break-up occurs to form droplets that start to move linearly according to their initial momenta acquired from the liquid film. This initial penetration of the atomised droplets determines what is defined as the nozzle spray angle. Accordingly, a theoretical nozzle spray angle could be defined as the slope of the projected spray envelope at the break-up point. Because it is very difficult to identify the exact location of the break-up from experiments, the nozzle spray angle has been defined as the average measured angle from 1mm to 5 mm downstream of the nozzle-exit by assuming that the break-up occurs within the first 1mm.

The surrounding air (aerodynamic force) seems to have small influence on droplet direction soon after their formation because of the relatively small drag force compared to the inertia force of the droplet itself. Since the nozzle spray angle is estimated from the initial droplet trajectory, it is mainly controlled by the droplets initial momenta just after the liquid film breaks-up. This initial momentum is mainly controlled by the flow conditions at the nozzle exit, the interaction force between the liquid film and the surrounding air and the break-up length. After some time of the droplets travelling their velocities decrease, thus the effect of the surrounding air on droplet trajectory increases and finally dominates their movement pushing the droplets to follow the air flow-pattern.

The focus in this section is on identifying the nozzle spray angle during the developing stage and its relationship to the flow angle that has been defined in the previous section. Figure 2-37, shown previously, includes typical images of the spray for different times from the start of injection for a 70 bar injection pressure. In order to

have consistent measurements of the nozzle spray angle, the same technique used to identify the film profile has been applied here by analysing the images using a fixed threshold value that provide binary images and, thus, a clear boundary between the liquid spray and the background. The variations of the nozzle spray angle and the flow angle as well as of the liquid film divergent angle with time are shown in Figure 2-40. This figure shows that it takes more time for both the spray angle and the divergent to reach their maximum value, although, the flow angle has a considerably higher value. This is attributed to the lower magnitude of the injection velocity (both the axial and the swirl components), thus a smaller centrifugal force and inertia force act on the liquid film resulting in lower film divergent and spray angles. The figure also shows lower divergent angle than the spray angle, which is attributed to the way the divergent angle was measured. As mentioned before, the divergent angle has been approximately measured as the average angle from the nozzle exit up to 1mm downstream from the nozzle exit while the spray angle is measured between 1mm and 5mm downstream of the nozzle exit. Also from 1mm to 5mm the droplets are moving free from surface tension forces, which act on the liquid film pulling it towards the spray centre.

In order to study the effect of injection pressure on spray development, images have been taken during the developing stages at different times from the start of injection and for different injection pressures as shown in Figure 2-41. Although the images reveal that the injection pressure has no direct effect on spray angle when the spray is fully developed, it does have an influence on the time required for the spray to reach its fully developed stage. As the injection pressure decreases, the spray develops at a slower pace and, thus it takes longer time to reach full development, as shown in Figure 2-42. Again, that could be attributed to the decrease in the injection and the swirl velocity of the liquid film and thus, the reduction of the inertia force of the liquid film and the centrifugal force acting on the liquid film after been injected from the injector nozzle; these two forces are mainly responsible of the liquid film divergence. Although increasing the injection pressure enhances film atomisation and reduces the spray development time, there are practical disadvantages in increasing the injection pressure as related to the fuel pump wear according to Zhao et al. (1999). This is due to the low viscosity of the gasoline grade used in vehicle equipped with current DISI pumps and its inability to adequately lubricate the pump itself and also due to leakage which could take place in highly pressurised equipment. In addition, there is an increase in the

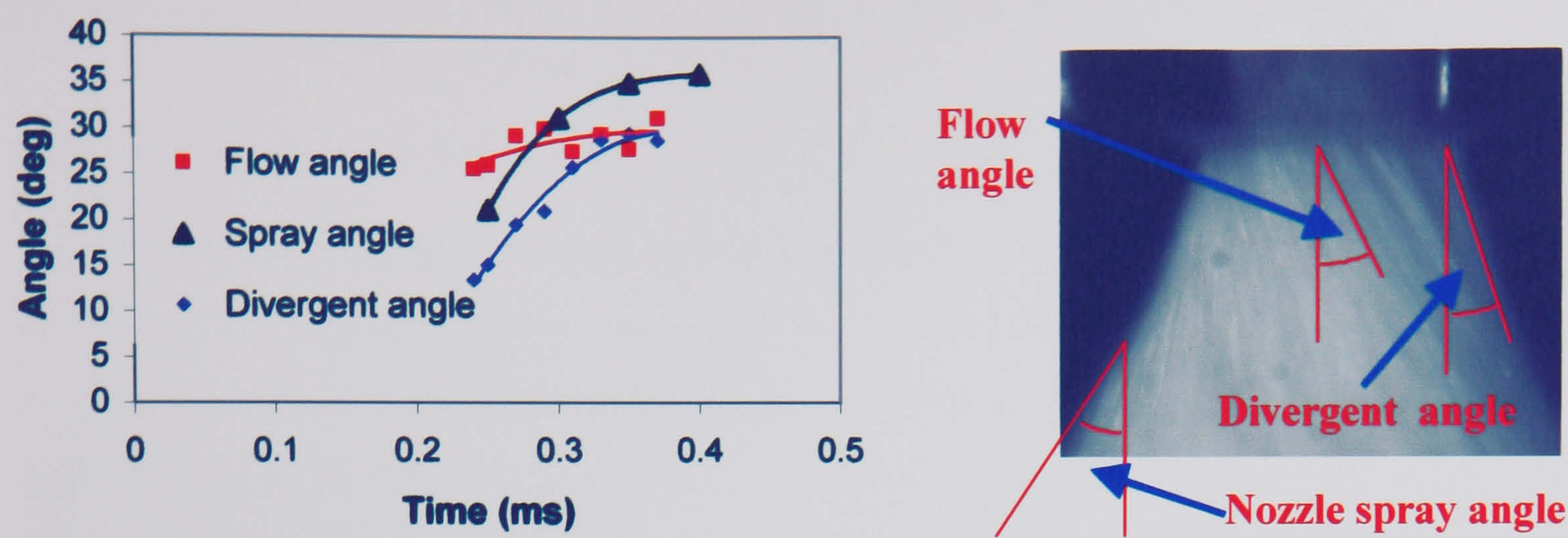
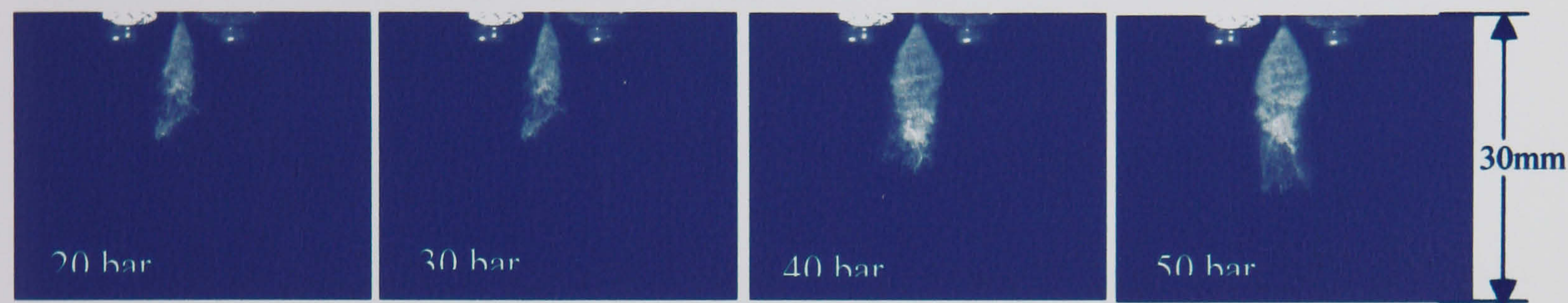
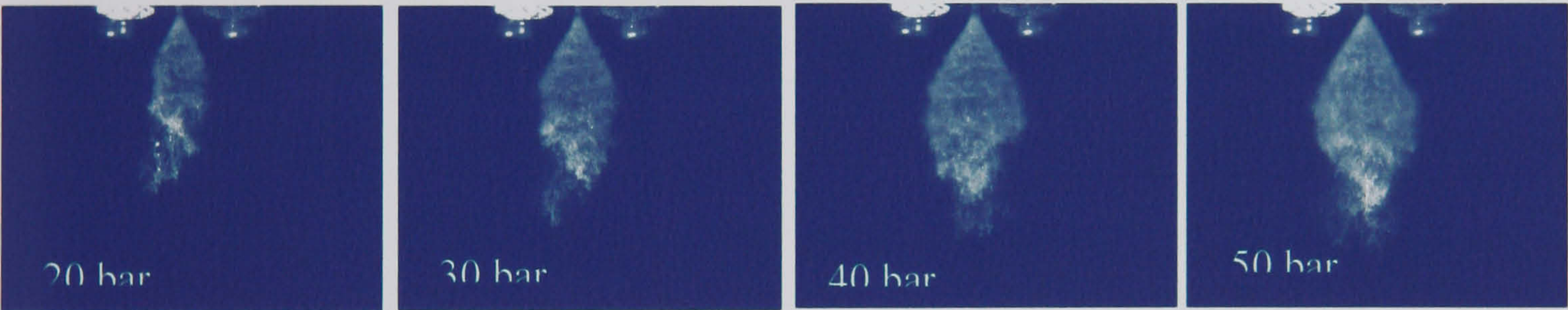


Figure 2-40 Comparison among flow, divergent and nozzle spray angles during spray developing stage



a) Spray images under different injection pressure at 0.3 ms



b) Spray images under different injection pressure at 0.35 ms

Figure 2-41 Spray images at the same times under different injection pressures

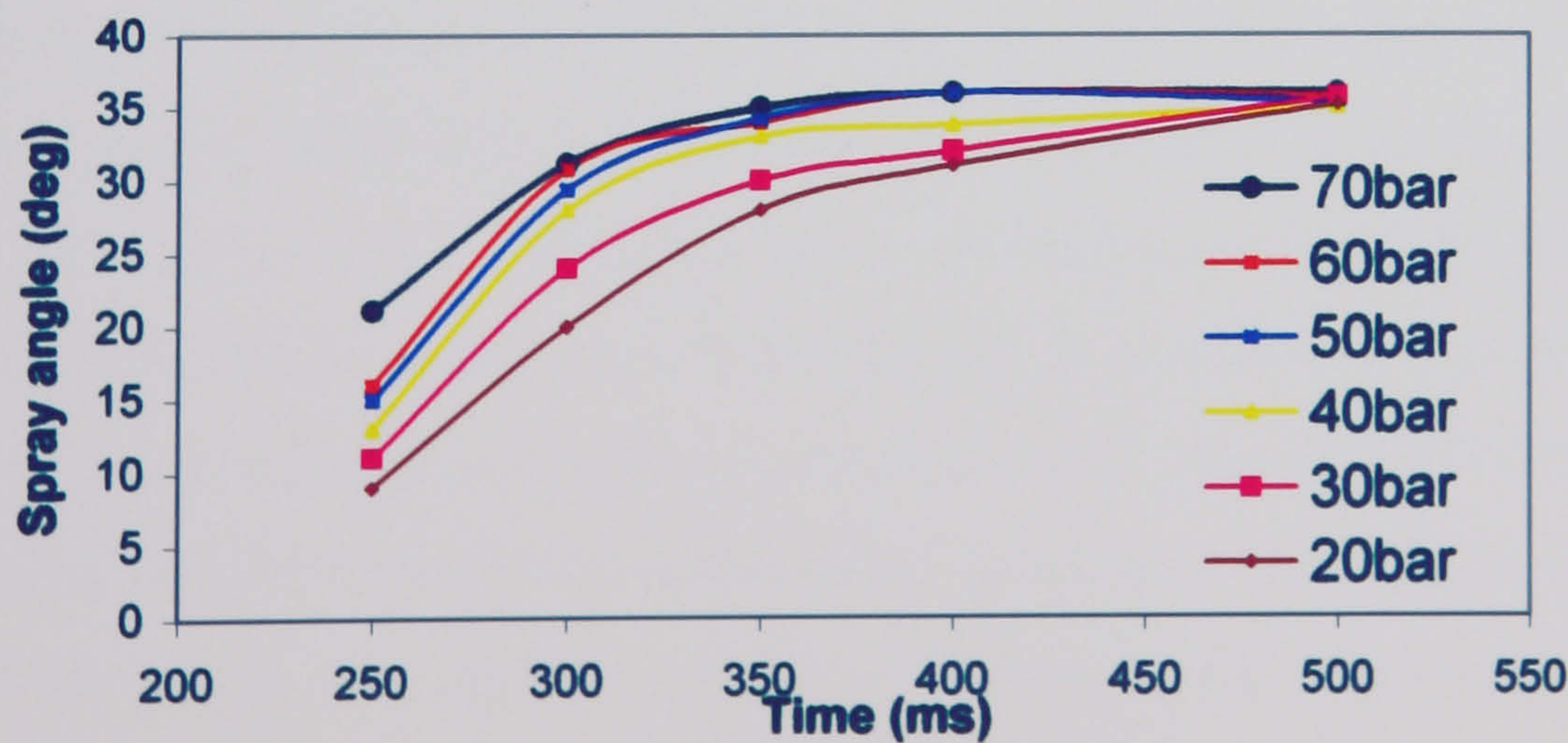


Figure 2-42 Effect of injection pressure on nozzle spray angle during the developing stage

level of noise with increasing injection pressure, in addition to the problem of higher pressure fluctuations developing inside the common rail system and at the nozzle exit.

2.5.4 Atomisation of Liquid Film

Images shown previously reveal that the swirling liquid film is formed inside the nozzle and after being injected outside the nozzle, it is subjected to different forces until it disintegrates. The exact scenario of film disintegration is not clear and it is very difficult to identify it experimentally in terms of quantitative data of both the length and time scales which govern the process occurring due to the high density of liquid droplets in this region. However, some information on how the liquid film breaks-up has been acquired using front light imaging by focusing on the edge of the spray just underneath the nozzle exit at low injection pressures, as shown in Figure 2-43. These images reveal that there are 3-dimensional waves on the film surface whose amplitude increases downstream of the nozzle causing disruption to the circumferential layers and leading to formation of ligaments and, subsequently, droplets. This disruption is enhanced by the internal pressure and velocity fluctuations inside the liquid film resulting from the pressure fluctuations in the injection pressure. Another mode of droplet formation seems to co-exist; it is based on surface waves whose amplitude increases with injection pressure, but this time the droplets are formed by stripping of the wave crests. These droplets are initially connected to the conical liquid film by thin threads which eventually break down as a result of on-going surface disturbances. It is believed that this process of droplet separation continues until the thickness of the liquid sheet becomes thin enough to be perforated into layers leading to droplet formation.

Some images have been taken at higher injection pressure to represent the actual engine conditions, as shown in Figure 2-44, but unfortunately it was not possible to acquire any useful information concerning the atomisation process because of the high density of droplets at the liquid film break-up location which scattered light to all directions resulting in a blurred image. Nevertheless, it is expected that the two modes of atomisation occurring at low injection pressures occur also at higher pressures with obvious differences in the length and time scales, since the overall structure of the spray remains the same. As the injection pressure increases, the atomisation process is enhanced and thus smaller droplets are produced. This is clear from the images shown in Figures 2-43 and 2-44.

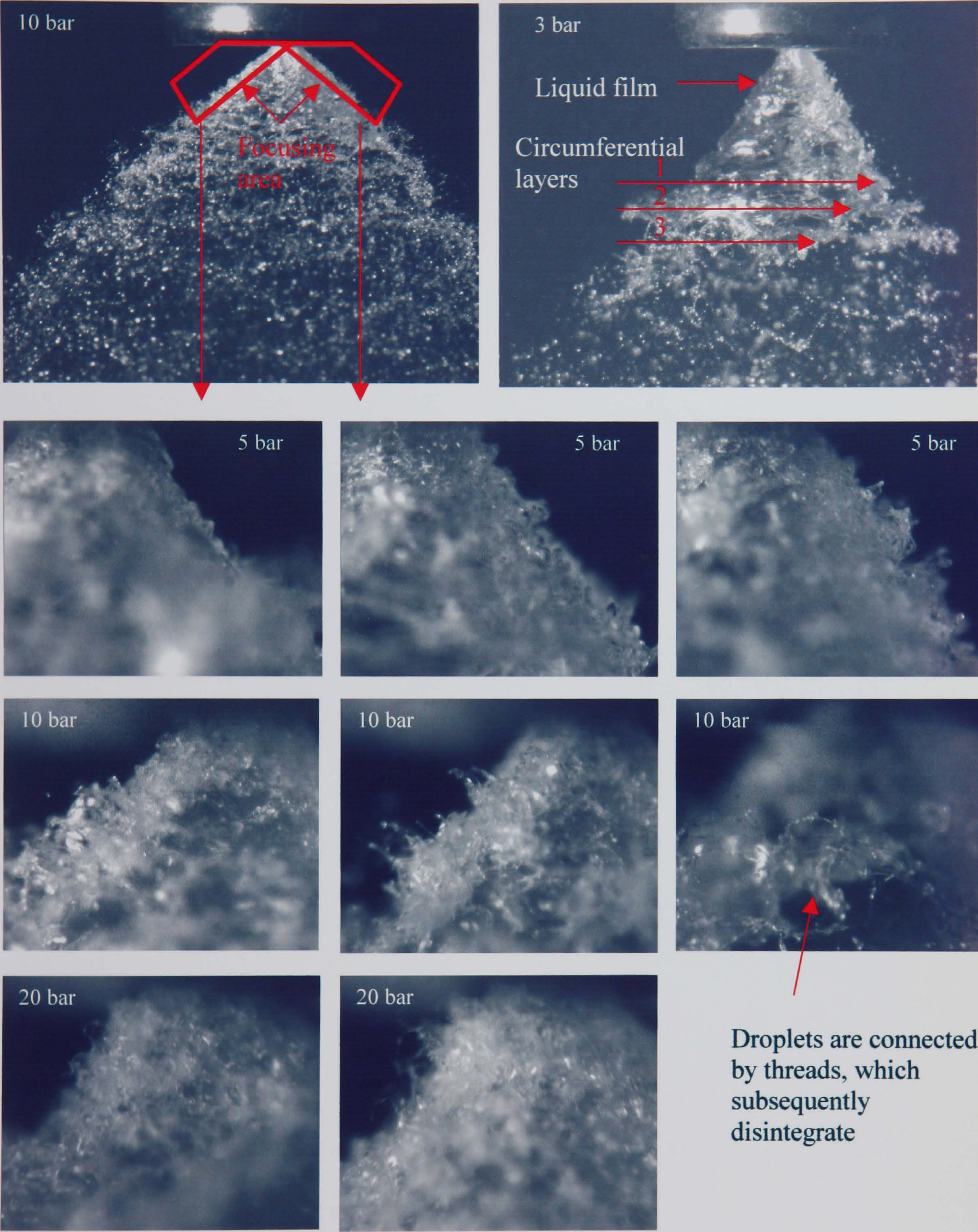


Figure 2-43 Images show the two modes of atomisation of liquid film under low injection pressure at 1.5 ms

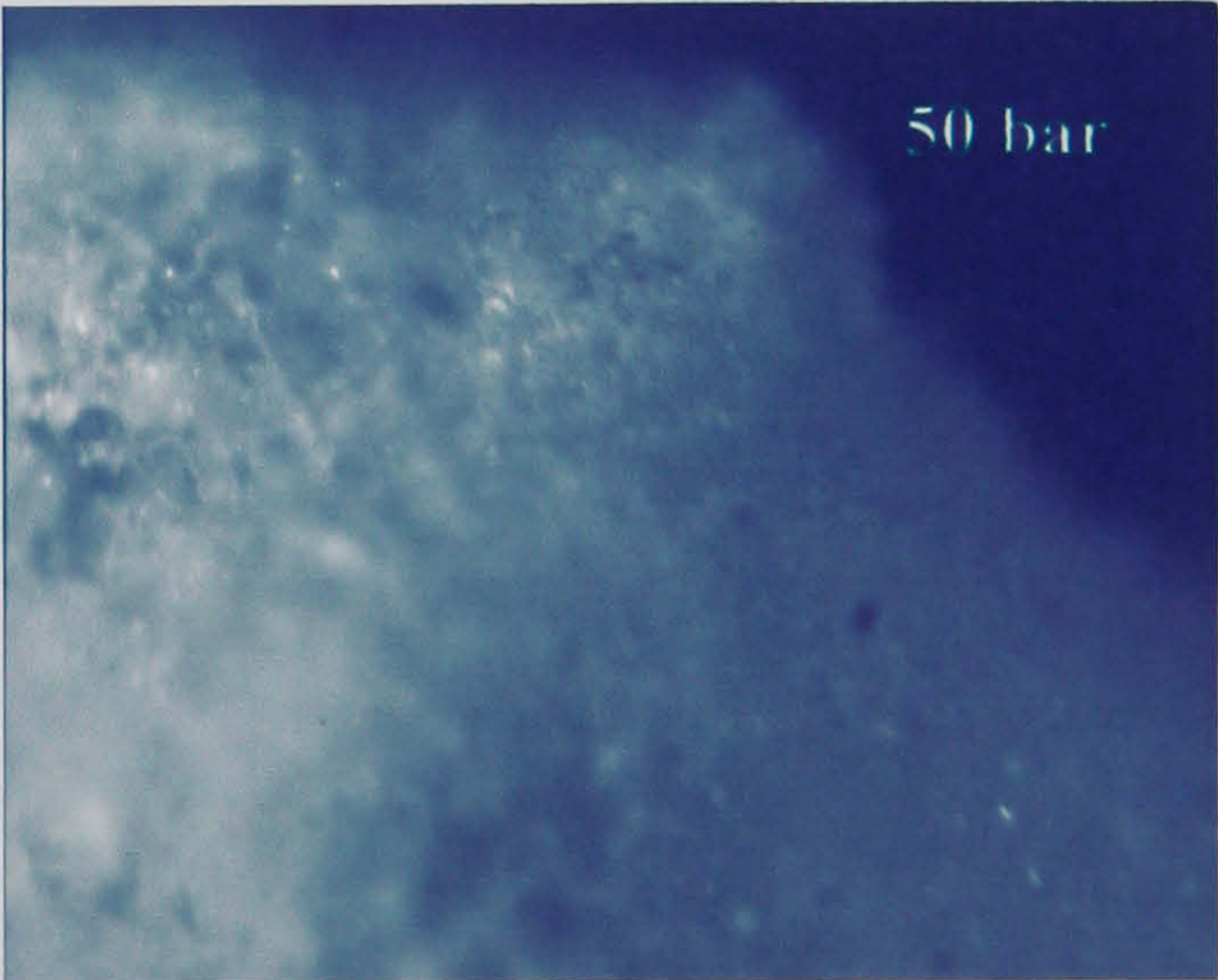
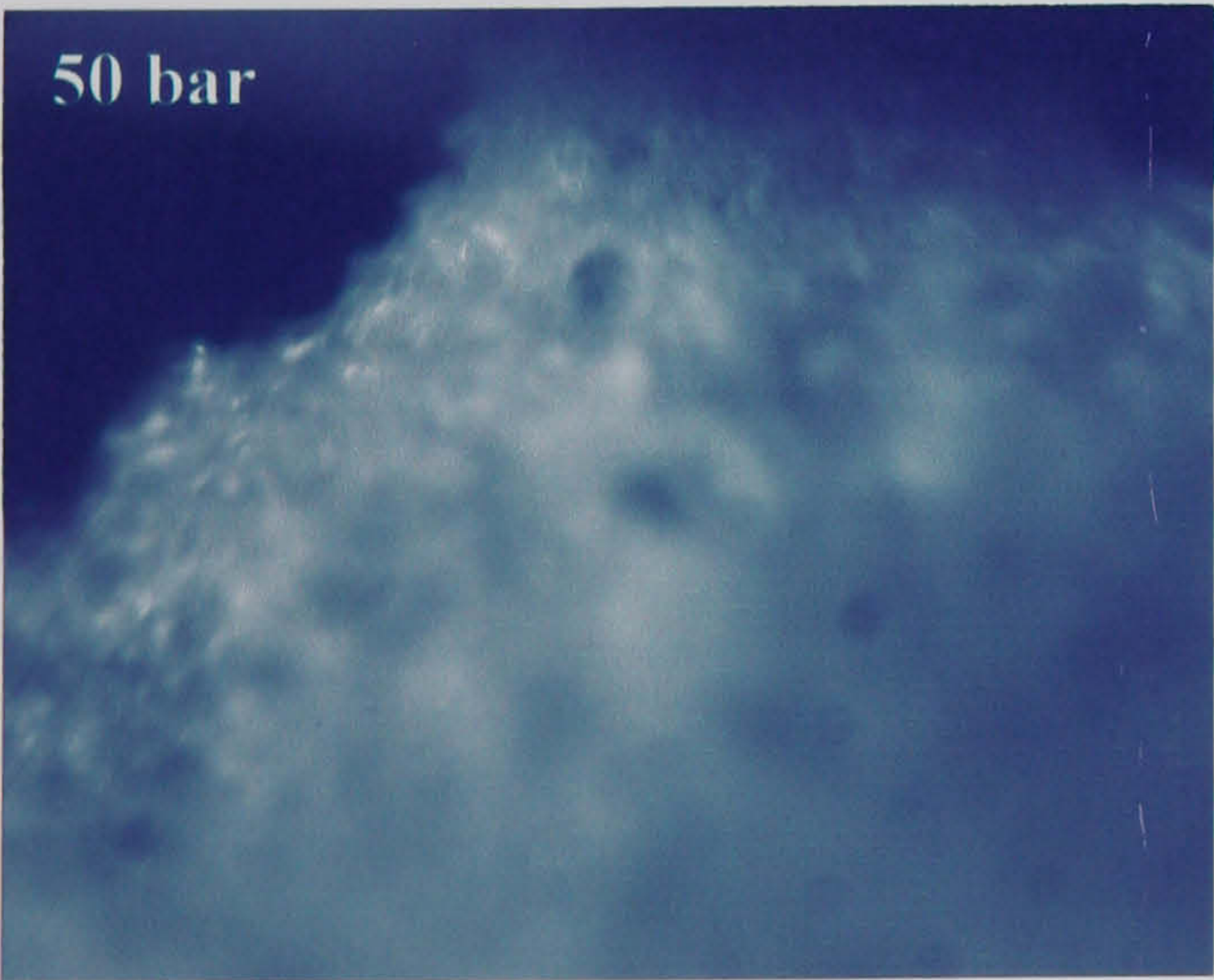
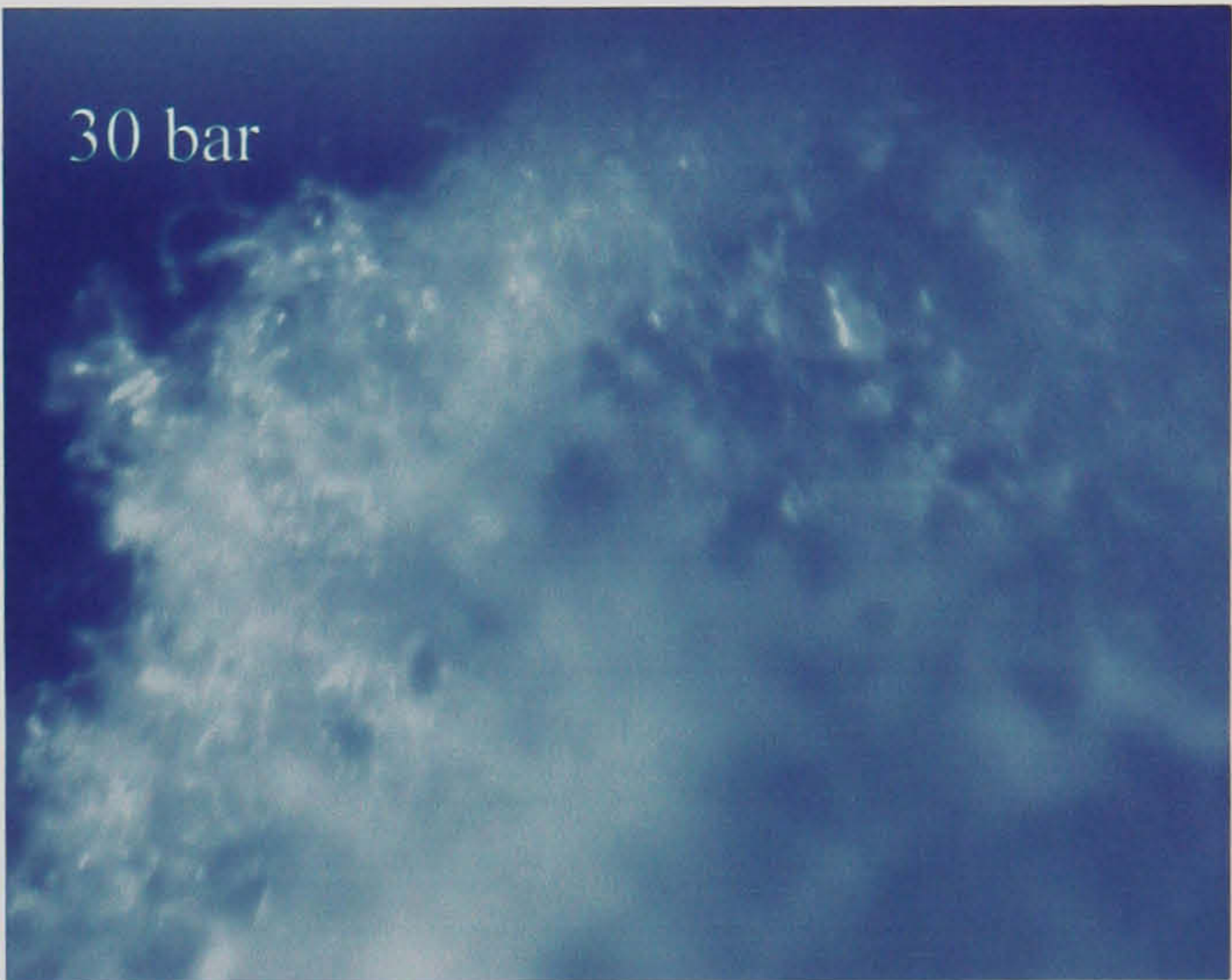
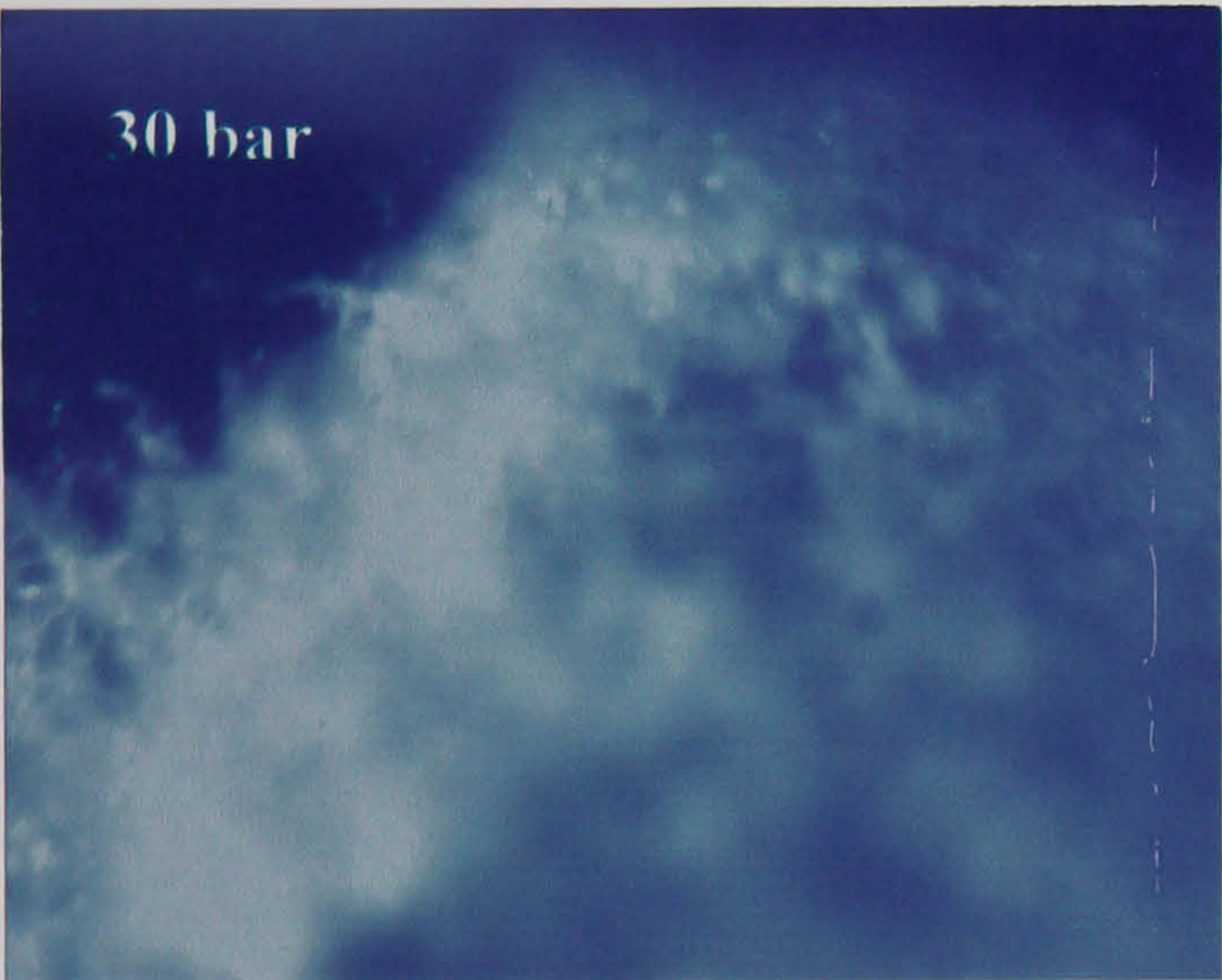
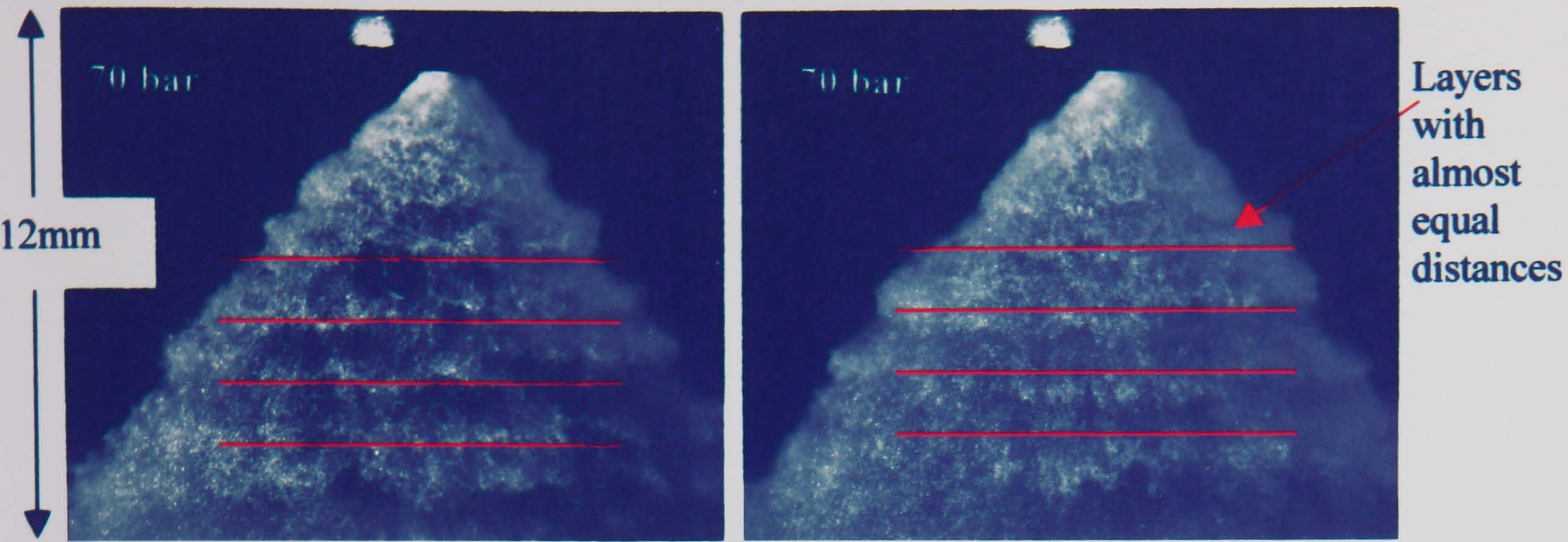


Figure 2-44 Atomisation of liquid film under high injection pressure at 1.5 ms

2.5.5 Effect of Nozzle Length on Spray Angle

Increasing the nozzle length leads to an increase in the resistance of the liquid to flow which is manifested as pressure drop across the nozzle. Due to the radial distribution of the pressure gradient as a result of the tangential velocity component at the discharge plane, the total pressure (ΔP) in a swirl nozzle is the sum of two components: the first (ΔP_w) is the pressure drop along the nozzle wall due to friction while (ΔP_r) is the pressure drop in the radial direction due to the tangential component of the liquid motion at the discharge plane which can have a significant value. Datta and Som (2000) showed that the pressure drop for a nozzle of 4mm length and 2mm diameter with a flow rate of $0.05 \text{ cm}^3/\text{ms}$ can be more than 8 bar. Figure 2-45 shows that increasing the nozzle length leads to a reduction in the spray angle. It has also been observed that the magnitude of the pre-spray is not as significant as in the case of short nozzle lengths. The resulting increase in film thickness and decrease in spray angle could be attributed to the reformation of the velocity profile as the liquid moves towards the exit with the peak of the axial velocity moving towards the nozzle centre, as been indicated by Dumouchel et al. (1993). Although the effect of nozzle diameter has not been studied in this work, Ren and co-workers (1997, 1998) have showed that increasing the nozzle diameter leads to an increase in the spray angle.

2.6 PRESSURE FLUCTUATIONS INSIDE THE INJECTION SYSTEM

Several studies have used constant fuel pressure at the inlet to be admitted to the injector by connecting it to a vessel which is pressurised by nitrogen gas (Nouri et al., 1999; Shelby et al., 1998; Kubo and Sakakid, 2001). Other studies have used an actual injection system (for example Kirhara et al., 2000; Preussner et al., 1998). However most of these studies have focused on the spray characteristics and do not show the dynamic variation of the pressure within the injection system and its effect on spray characteristics.

As already mentioned, production fuel injection systems are subjected to pressure fluctuations inside the common rail even without any injection. Figure 2-46 shows an example of pressure fluctuations inside the common rail pipe; These fluctuations have a frequency equal to the frequency of the pump due to the pulsating

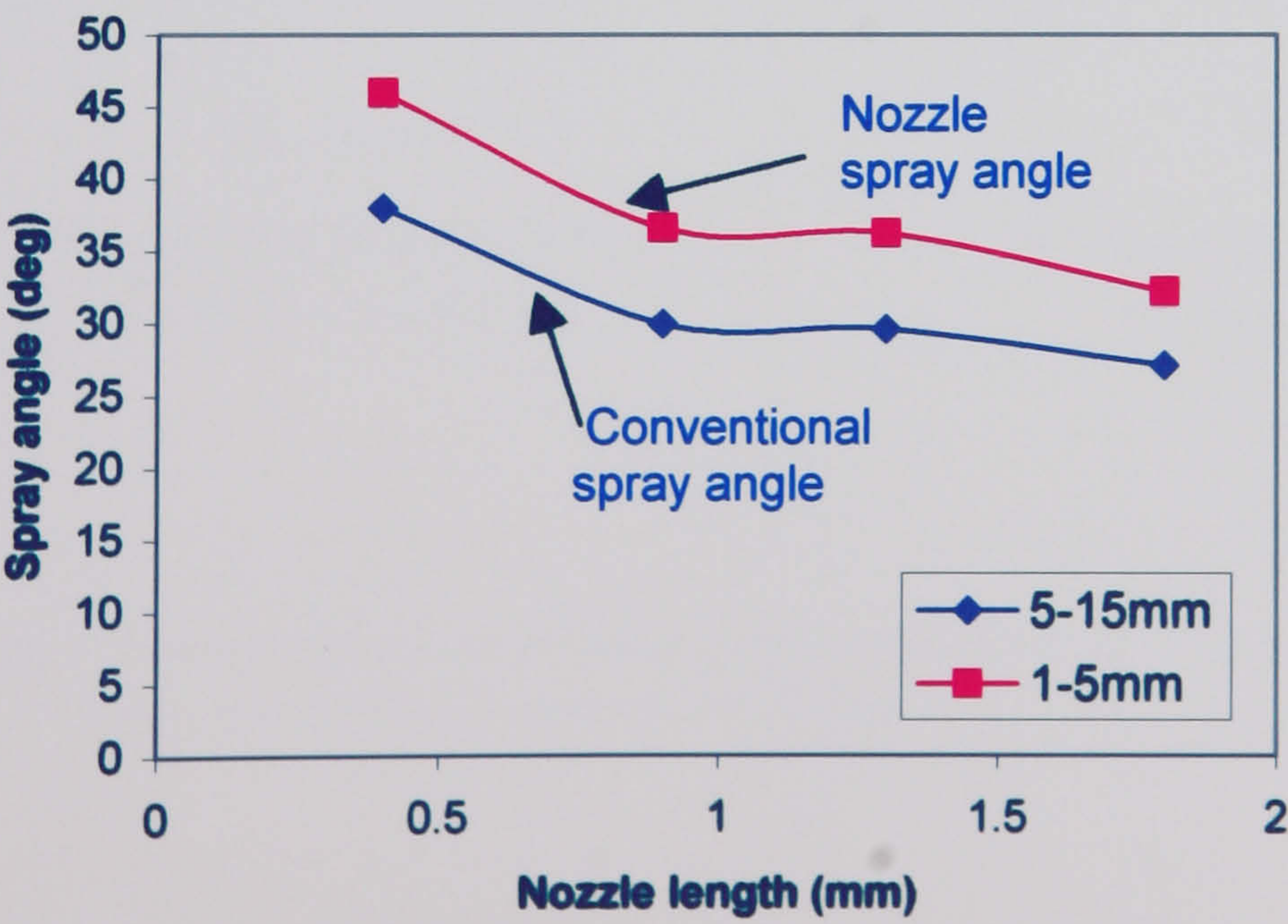
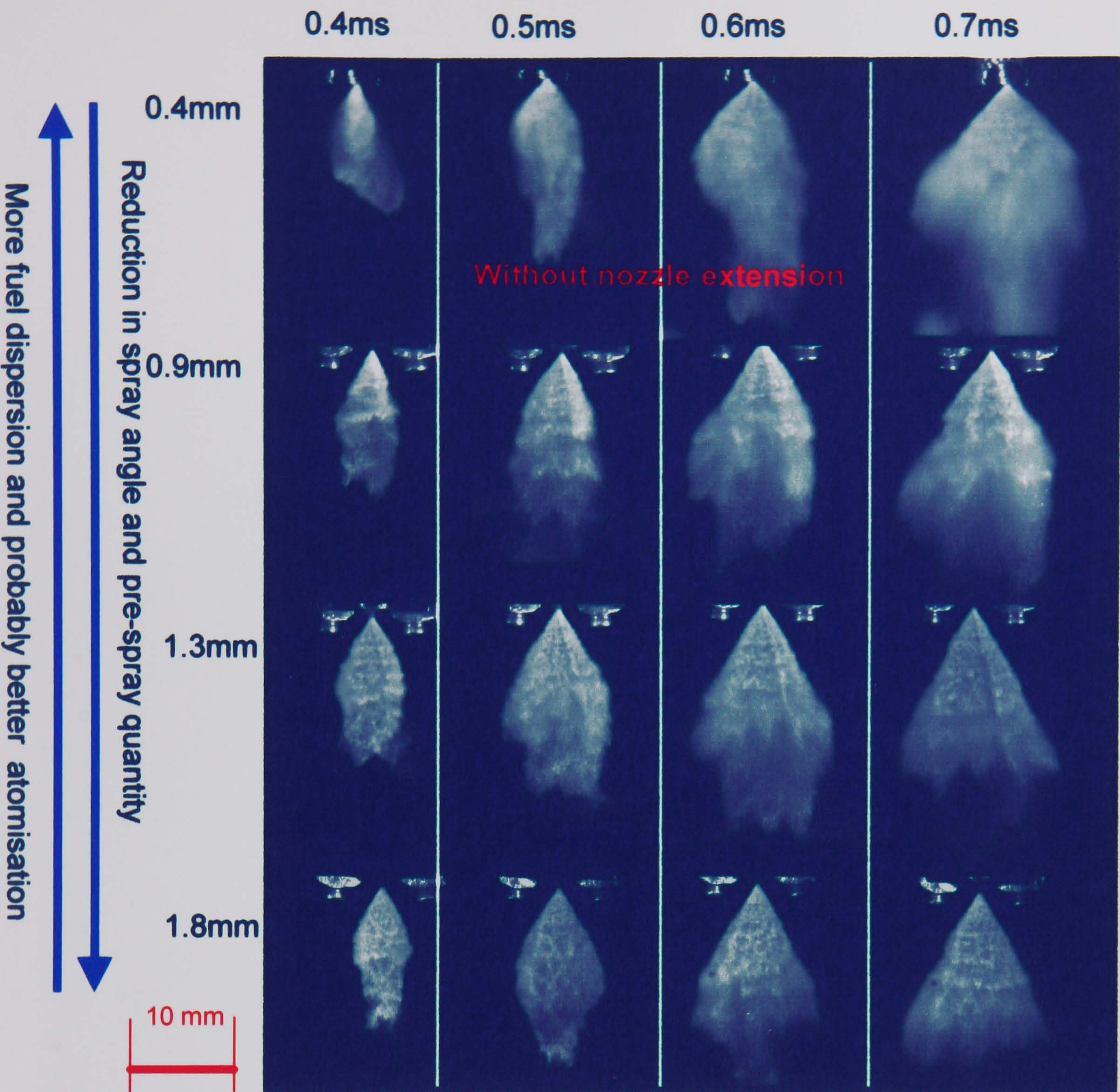


Figure 2-45 Effect of nozzle length on spray angle

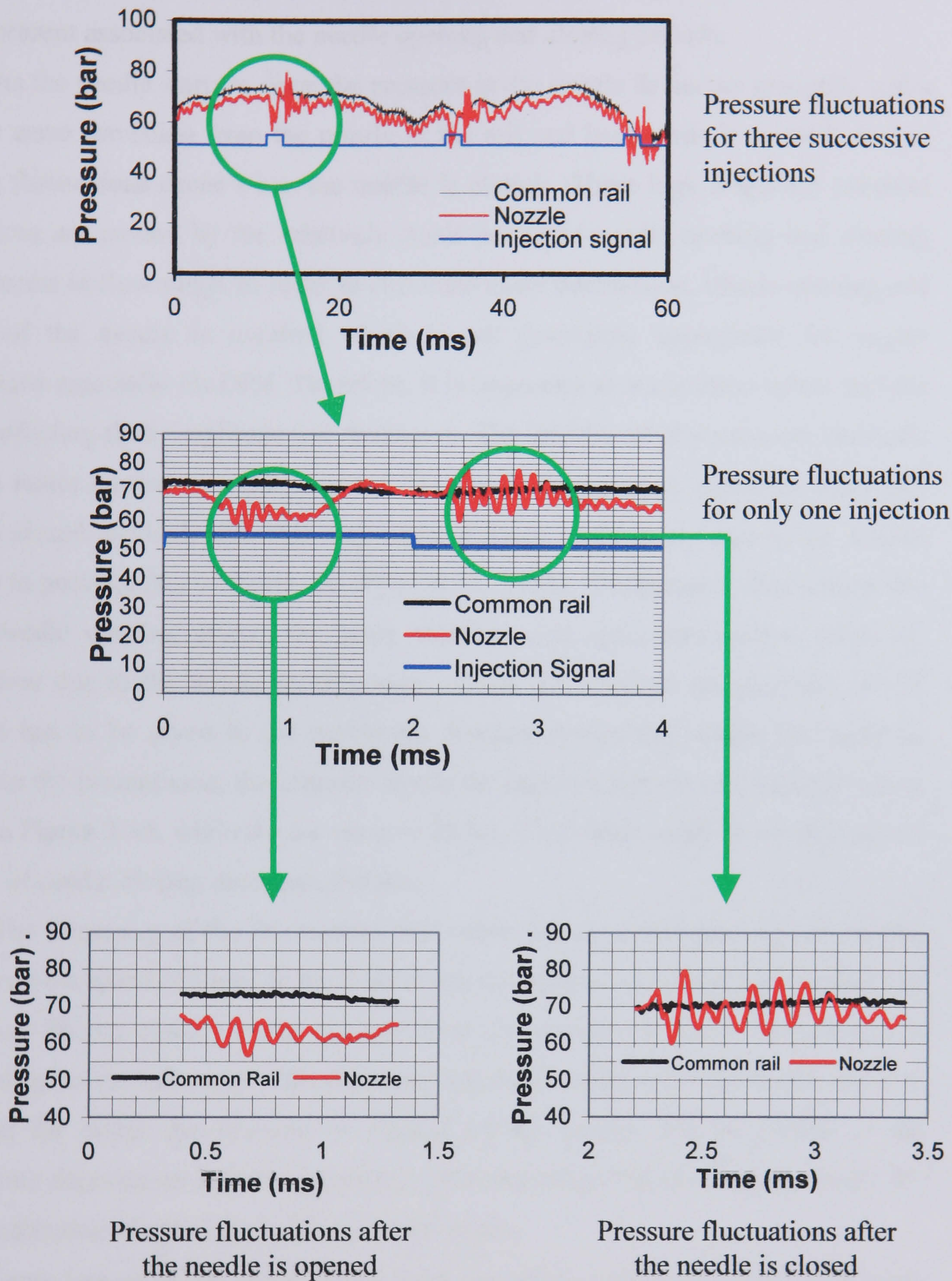


Figure 2-46 Pressure fluctuations inside the common rail pipe and the nozzle for injection pressure setting of 70 bar

flow induced by the piston type pump. Another superimposed pressure fluctuation can also be present associated with the needle opening and closing periods.

As the needle starts to open the pressure in the nozzle decreases gradually and a pressure wave travelling from the nozzle to the rail and backwards is created; similar pressure fluctuations occur when the needle is closed. These high frequency pressure fluctuations are caused by the relatively small period of needle opening and closing, which results in flow surge. In order to eliminate these fluctuations, slower opening and closing of the needle is required which is not practically appropriate for engine applications especially for DISI. Therefore, it is important to study these waves and the factors affecting their amplitude and frequency. The analysis of this transient hydraulic problem better known as water hammer has always happened in pipelines when one valve is closed suddenly and that may cause damage of the pipes due to the sudden increase in pressure (for example see Wylie et al., 1993). The pressure fluctuations due to the needle opening affects the spray structure and spray atomisation while the fluctuations due to the needle closing more affects the injection system; thus special attention has to be given to the maximum pressure developing within the injection system. In the present case, the pressure inside the injection system can reach 80 bar as shown in Figure 2-46, while the set value is 70 bar. This value could be much higher if the time of needle closing decreases further.

The frequency of the fluctuations that occur due to needle opening and closing depends on the speed of sound in the fluid inside the injector and the maximum distance they travel (in our case, it is the distance from the position at which the pressure is measured up to the rail pipe). This frequency has been estimated to be 9 kHz and it is the same for either the opening or closing of the needle. The amplitude of the fluctuations depends on injector geometry, fluid properties, friction losses, needle lift and time duration of opening or closing of the needle.

Spray images of the overall spray have showed that the spray experiences some discontinuities and successive layers appearing along the spray axis at almost equal distances between each other. It is generally believed that this discontinuity and the appearance of these layers are connected to the pressure fluctuations in the flow due to the needle opening. This explanation is supported by observations that the number of layers distinguished on the spray image is the same as the number of fluctuations on the pressure curve.

2.7 CONCLUSIONS

The following conclusions could be drawn from the analysis of the flow images obtained inside the nozzle and near the nozzle exit:

- By imaging the flow inside a real size transparent nozzle attached to the injector nozzle, it was possible to measure both the temporal and the spatial variation of the liquid film thickness along the transparent nozzle hole in terms of mean value of the film thickness over 50 images for each case. This mean value showed that the pressure has a very small effect on the film thickness. The images also revealed that the film is formed very quickly in about 20 μ s after the needle is fully opened providing valuable information for validating flow models inside fuel injection systems.
- The errors in the film thickness measurement have been identified both experimentally, by imaging static films of known thickness and analytically by following the light path from the camera to the film. The results showed that imaging close to the edge of the hole can give large error of 40% or more than the actual film thickness in cases when the film thickness is less than 10 % of the cylinder radius.
- The mean axial velocity of the liquid film at the nozzle exit has been calculated from the mean film thickness assuming constant injection flow rate. This value has been used with the flow direction, which has been identified from imaging of the outer surface of the liquid film at the nozzle exit, to determine the mean value of the swirl velocity. Both the swirl velocity and the axial velocity have been used to calculate the mean injection velocity of the liquid film. These results have been included in the atomisation model presented in Chapter 3
- Three angles have been defined to characterise the flow close to the nozzle exit; the flow angle, the film divergent angle and the nozzle spray angle. Their temporal variations have been demonstrated and their influence on spray structure have been discussed.
- Although the injection pressure has no effect on the fully developed spray angle it seems to be having an influence on the time of development of the spray; as the injection pressure increases, the spray develops faster. Images also show that increasing the injection pressure enhances the atomisation process and

minimises the time of spray development. However special attention has to be given to the design of the pump lubrication system and means to suppress the higher pressure fluctuations which occur inside the injection system during and after the needle opening and closing. Decreasing the opening time of the needle reduces the developing time of the spray during which poor atomisation occurs, however higher pressure fluctuations occur inside the injection system which may again leads to increased development period due to the dynamic flow which may collapse the film.

- It has also been revealed that there is a three-dimensional wave on the liquid film surface after injection from the nozzle exit whose magnitude increases downstream of the nozzle. These waves cause two modes of droplet formation, through ligaments in the direction of injection and through wave crest stripping perpendicular to the liquid film
- Pressure fluctuations inside the common rail pipe have also been measured and also above the tangential holes of the swirler. The rail pressure was found to be fluctuating with a frequency equal to the frequency of the rotating pump with a peak-to-peak value approximately 30% more than the nominal pressure value.
- The fact that the hollow cone spray can be formed at very low injection pressure (3bar), it can be concluded that the destruction of the liquid film inside the nozzle is responsible for the presence of the pre-spray observed during the spray development rather than the pressure drop at the needle during its opening. The reason for having such flow dynamics effects at the nozzle during is attributed to the pressure fluctuations associated with the dynamic movement of the needle.

CHAPTER 3

ANALYSIS OF THE NEAR-NOZZLE FLOW STRUCTURE

3.1 INTRODUCTION

In the DISI common rail injection system equipped with swirl injectors, the fuel is admitted from the fuel pump into the common rail pipe and then to the injector where a swirl motion is initiated by means of tangential slots and is finally discharged through the outlet orifice of the nozzle. Due to the tangential entry of the fuel, an air-cored vortex flow of liquid is formed as a thin film which diverges downstream of the nozzle. The inherent hydrodynamic in addition to the aerodynamic instability disintegrates this liquid film into ligaments and then into drops to form a well-defined hollow cone spray. In an attempt to simplify the mathematical model representing the whole injection process, the flow within the injection system has been divided into different regions which deserve individual attention.

The first region represents the fuel flow inside the fuel injection equipment (FIE). This region can be modelled satisfactory using a one-dimensional model such as those developed for diesel injection systems by Fairbrother (1994) and Gavaises (1997) and modified for DISI injection systems by Arcoumanis et al. (1999). The second region represents the flow inside the injector itself. In order to take into account the effect of the injector geometry and to represent the swirl motion, a 3-D model has been established. The third region represents the flow inside the nozzle where a liquid film is formed with an air core, thus establishing a two-phase flow regime. Solving this problem is very time consuming especially during the developing stage of the liquid film when the needle is opening and its effect has to be taken into account. To mathematically model this process a two-phase flow model, using the volume of fluid (VOF) method, has been developed by Arcoumanis and Gavaises (2000). Although the model uses a two-dimensional analysis, its results proved to be close to the experiments presented in Chapter 2. Simpler models (zero-dimensional models) were also presented

by Cousin and Nuglisch (2001) in order to evaluate the film thickness, however these models are only providing an average value of the film thickness under steady-state conditions, without taking into account the effect of the air flow pattern or any time development. Moreover their results are very limited to the film thickness inside the nozzle, contrary to the VOF method whose solution could be extended to evaluate the film even outside the nozzle, especially in the critical near-nozzle region.

The two-phase flow models inside the nozzle varied between complicated VOF models (Arcoumanis and Gavaises, 2000; Kubo and Sakakid, 2001) to simpler ones such as Datta and Som (2000) to the very simple model of Cousin and Nuglisch (2001). These models succeeded in achieving their purpose of predicting the film thickness with different degrees of accuracy. In this work, the liquid film thickness inside the nozzle has been identified experimentally through imaging analysis, as presented in Chapter 2. The purpose was to study experimentally the effect of different parameters on the film thickness inside the nozzle and to validate the two-phase model developed by Arcoumanis and Gavaises (2000).

This chapter focuses on analysing the flow just outside the nozzle, which could be considered as the region bridging the nozzle internal flow with spray characteristics. The aim is to identify the flow conditions very close to the nozzle exit and to examine the effect of the nozzle flow conditions on spray geometry and droplet sizes. The DISI pressure swirl injector generates, at the nozzle exit, a liquid film of a certain thickness, a specific injection velocity and a flow angle. These three parameters depend on the injector swirler design and the associate flow pattern inside the injector as well as on the injection pressure. It is the aim of this project to develop and test a mathematical model to study the effect of these three parameters besides the fuel properties on the film divergence at the nozzle exit and also to develop a model to represent the atomisation of the liquid film. In order to predict mathematically the film divergence and the atomisation process near the nozzle exit, the film has been analysed in two stages. In the first stage, the film divergence has been modelled assuming there are no disturbances on its surface while in the second stage only surface disturbances that cause film disintegration have been taken into account; this division is considered in order to simplify the problem. Thus, in the first stage the film divergence has been identified by considering the main forces acting on the liquid film itself as it penetrates outside the nozzle, while in the second stage the focus is not on the film itself but on the waves formed on its surface. In fact, it is very difficult and probably inaccurate to solve

simultaneously the film divergence and the surface disturbance numerically since the length scale of the disturbances is relatively small, and therefore, only a very small grid size has to be used which could cause some numerical problems. Moreover, the significant change in the surface tension force due to the dynamic variations on the film surface could cause numerical diffusion problems.

The liquid film exits the discharge nozzle with both axial and swirl velocity components. The ratio between these two velocity components defines the slope of the nozzle flow angle, which cannot directly represent the nozzle spray angle since the swirl velocity is not necessarily equal to the radial velocity. The assumption that the flow angle is equal to the spray angle (Dumouchel et al.; 1993 and Rizk and Lefebvre; 1985) is not accurate and is only valid if the flow at the nozzle exit is fully atomised, as it will be discussed later in this chapter. That explains the introduction of a correction factor in most studies to fit the experimentally measured spray angle with the calculated flow angle. Dumouchel et al. (1993) introduced a factor of 0.556 and the same factor was used by Cousin et al. (1998) in their model. In fact, the relationship between the radial and swirl velocity components depends on the flow conditions i.e. on whether it is fully atomised or continuous flow. In this chapter, an analysis will be presented to calculate the film divergent angle and to relate it to the flow angle; the nozzle spray angle will then be identified at the point of film break-up.

In principle, film divergence could be identified numerically using the VOF method by estimating the film motion after the exit of the liquid from the injection hole and within a very small distance from the hole exit. Unfortunately, the film thickness becomes comparable to the cell size, due to its wide divergence angle, which precludes any realistic predictions at downstream distances according to Arcoumanis and Gavaises (2000). Other studies for example Hassan et al. (1997) and Ramamurthis and Tharakan (1998) have introduced an alternative approach by studying the different forces applied on the liquid film near the annular liquid jet and have been able to identify analytically the shape of the liquid film before it has been atomised, which proved to be in agreement with their experimental results; however, this approach has not been applied to DISI injectors. By applying this approach to the film ejected from the DISI swirl atomiser with the advantage of knowing from images taken near the nozzle exit the initial injection angle and the final film profile of the flow a useful and rather simple engineering sub-model can be developed.

Examination of the shape and divergence of the liquid film is very crucial in determining the spray angle before atomisation occurs and, thus, the initial conditions for any atomisation model. Moreover, the effect of different parameters and operating conditions on the film shape provides useful information for overcoming the major weakness of these injectors which is the poor fuel atomisation during the first stage of spray development. Linking the film divergence with the geometric penetration of a single droplet, in addition to any relevant experimental results, could provide valuable information on the location of the film break-up.

This chapter also includes an analysis of the atomisation process by studying the waves formed on a swirling hollow cylinder subjected to air flow from both sides using the linear instability analysis (LIA) and including its complete derivation and the effect of different parameters. A break-up model has also been proposed in order to estimate the mean droplet size from the wavelength on the cylinder's surface under different operating conditions. The results of the linear instability model with the film divergence can provide important input data for any conventional spray model.

3.2 FILM DEVELOPMENT OUTSIDE THE NOZZLE BEFORE BREAK-UP

This section describes the analysis of the forces acting on the liquid film after exiting the injector nozzle. A differential equation that identifies the film divergence, which is manifested as the film radius at any distance from the nozzle exit, has been derived. The shear forces at the liquid film interface that cause the growth of disturbances or waves on the liquid sheet have been neglected in order to identify more easily the direction of both the surface tension forces and the centrifugal forces during the film development. The aerodynamic drag force affecting the liquid film has also been neglected. A parametric investigation for the effect of different conditions on the film development has been included.

3.2.1 Assumptions and Analysis

A continuous steady-flow of liquid film of density ρ_l and surface tension σ is assumed to exit from the injector nozzle of radius R_0 at constant axial velocity U_j and swirl velocity W_j with an initial flow angle α determined from the flow images obtained at the nozzle exit without any surface disturbances or atomisation. The forces applied on a film annular ring of length dl at distance x from the nozzle exit and distance r from

the nozzle exit moving with axial velocity U and swirl velocity W could be analysed as shown in Figure 3-1 below.

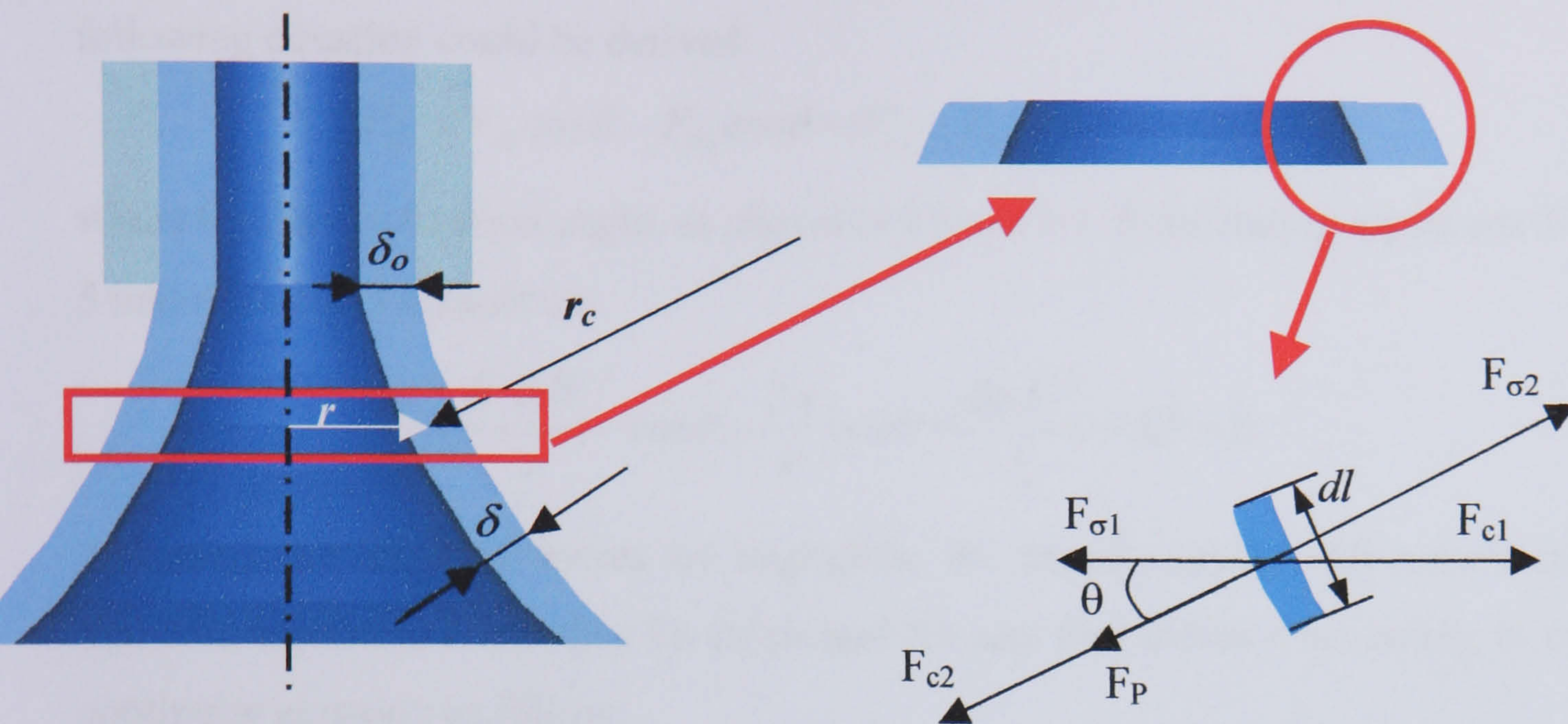


Figure 3-1 A schematic showing the forces acting on the liquid film at the nozzle exit

The following are the main forces acting on the annular ring of the liquid film:

$$F_{\sigma 1} = 2\pi r dl \frac{2\sigma}{r} \tag{3.1}$$

$$F_{\sigma 2} = 2\pi r dl \frac{2\sigma}{r_c} \tag{3.2}$$

$$F_{c1} = 2\pi r dl \rho_l \frac{W^2}{r} \tag{3.3}$$

$$F_{c2} = 2\pi r dl \rho_l \frac{U^2}{r_c} \tag{3.4}$$

$$F_p = 2\pi r dl \Delta P \tag{3.5}$$

where, $F_{\sigma 1}, F_{\sigma 2}, F_{c1}, F_{c2}$, and F_p are, respectively, the surface force due to the circumferential curvature, the surface force due to the axial curvature, the centrifugal force due to the circumferential curvature and the tangential velocity, the centrifugal force due to the axial curvature and the pressure force due to the differential air pressure across the liquid film and the inertia force. r_c in equations 3-2 and 3-4 is the radius of curvature due to the film divergence. Other factors such as the gravity force, the film

internal force and the drag force have been neglected since their values are relatively small close to the nozzle exit. Further more, if the film is assumed to be stable and not flapping then, the net force applied normal to the film should be zero. Accordingly, the following equation could be derived:

$$F_{\sigma 2} + F_{c1} \cos \theta - F_{\sigma 1} \cos \theta - F_{c2} - F_p = 0 \quad (3.6)$$

where θ is the inclination angle, as shown in Figure 3-1. Substituting equations 3-1 to 3-5 into equation 3-6 leads to:

$$\frac{2\sigma}{r_c} + \frac{\delta \rho_l W^2}{r} \cos \theta - \frac{2\sigma}{r} \cos \theta - \frac{\delta \rho_l U^2}{r_c} - \Delta P = 0 \quad (3.7)$$

Assuming the frictional losses are negligible, the liquid velocity will remain constant. The film thickness could thus be calculated for any film radius r according to the flow continuity equation as follow:

$$\delta = \frac{U R_o \delta_o}{U r} = \delta_o \frac{R_o}{r} \quad (3.8)$$

By analysing the injection velocity into a tangential and an axial component, the mean axial velocity could be calculated from the film thickness and the average flow rate as shown previously in Table 2-1. This axial velocity at the nozzle exit could be assumed to be equal to the film flow velocity along its divergence direction and to remain constant by assuming the shear forces to be negligible. The swirl velocity could then be calculated from the flow angle (α) measured from the film images according to the following equation:

$$W_j = U_j \tan(\alpha) \quad (3.9)$$

Because the film diverges as it develops downstream from the nozzle, the swirl velocity does not remain constant and its value has to be calculated from the conservation of the angular momentum, therefore, i.e.;

$$W = W_j R_o / r \quad (3.10)$$

Substituting equations (3.8), (3.9) and (3.10) into equation (3.7) yields:

$$\frac{2\sigma}{r_c} + \frac{\delta_o R_o^2 \rho_l U_j^2 \tan^2 \alpha}{r^3} \cos \theta - \frac{2\sigma}{r} \cos \theta - \frac{\delta_o R_o \rho_l U_j^2}{r r_c} - \Delta P = 0 \quad (3.11)$$

From the film geometry:

$$\frac{dr}{dx} = r' = \tan(\theta), \quad (3.12)$$

and,
$$\cos(\theta) = \frac{1}{\sqrt{1 + (r')^2}} \quad (3.13)$$

The radius of curvature r_c could be presented mathematically in the following form:

$$\frac{1}{r_c} = -\frac{r''}{[1 + (r')^2]^{3/2}} \quad (3.14)$$

Substituting equations 3-12, 3-13 and 3-14 into equation 3-11 and dividing by $\cos(\theta)$, the following equation could be derived:

$$\begin{aligned} \frac{-2\sigma r'' \sqrt{1+r'^2}}{(1+r'^2)^{3/2}} + \frac{\delta_o R_o^2 \rho_l U_j^2 \tan^2 \alpha}{r^3} - \\ \frac{2\sigma}{r} + \frac{R_o \delta_o \rho_l U_j^2}{r} \frac{r'' \sqrt{1+r'^2}}{(1+r'^2)^{3/2}} - \Delta P \sqrt{1+r'^2} = 0 \end{aligned} \quad (3.15)$$

In order to simplify this equation and minimise the number of parameters, a number of dimensionless groups are introduced:

$$\bar{r} = \frac{r}{R_o}, \quad \bar{x} = \frac{x}{R_o}, \quad \bar{\delta} = \frac{\delta_o}{R_o}, \quad \bar{P} = \frac{\Delta P}{(\sigma / R_o)} \text{ and } We = \frac{\rho_l R_o \bar{\delta} U_j^2}{\sigma} \quad (3.16)$$

where, $\bar{r}$ and $\bar{x}$ represent the radius and the axial distance of the film at any location from the nozzle exit while $\bar{\delta}$ represents the ratio between the initial film thickness $\bar{\delta}_o$ and the nozzle radius. The pressure dimensionless group $\bar{P}$ represents the ratio between the pressure and surface tension forces while Weber number We represents the ratio between the liquid inertia and the surface tension forces. Using these dimensionless groups in equation 3-15 with some simple mathematical manipulation, the final equation of the film profile could be derived in the following form:

$$\frac{\bar{r}''}{\left(1 + \bar{r}'^2\right)} \left(\frac{We}{\bar{r}} - 2\right) + \frac{We \tan^2 \alpha}{\bar{r}^3} - \frac{2}{\bar{r}} - \left(1 + \bar{r}'^2\right)^{1/2} \bar{P} = 0 \quad (3.17)$$

By solving equation (3-17), the radius of the axisymmetric annular swirling jet r at any axial distance downstream of the nozzle exit x could be identified for different values of Weber number, pressure differential and initial flow angle and film thickness. Although it might appear that the swirl velocity does not have a direct influence, the slope of the angle α represents the ratio of the swirl to the axial velocities and its effect is very significant, as we will see later. The equation, which is a non-linear, second-order, ordinary differential equation can be solved numerically; a Matlab program has been established to solve this equation using a Runge Kutta 4 method.

3.2.2 Parameters Affecting Film Divergence Outside the Nozzle

From the previous equation (3-17), it is noticed that the factors affecting the film profile could be classified into three groups. The first group represents internal parameters (inside the nozzle) that depend on the injector design and operating conditions. These parameters are the film thickness inside the nozzle relative to the nozzle radius and the flow angle at the nozzle exit as well as the injection velocity. It has been shown before, based on the flow images in the vicinity of the nozzle exit, that the flow angle does not depend on the injection pressure or velocity within the DISI injector range of operation and its effect appears only at the lower injection pressures while its value depends on the swirler and nozzle length.

The dimensionless pressure group defined in equation (3.16) represents the second group and the only external factor affecting the film divergence after the exit from the nozzle. This group represents the difference in air pressure between the two sides of the liquid film whose value increases when the fuel is injected into pressurised gas. It is believed that the gas flow dynamics could have an influence on the liquid film profile; nevertheless to include the effect of the shear force on the film would make the problem very complicated. The effect of shear forces and airflow has been assumed to have an influence on the film surface but not on its internal structure or film direction. The results of Ramamurthi and Tharakan (1998) confirmed no change in the film direction and its development when its surface is disturbed. Although their results

have been obtained at relatively low velocities (16.6 m/s), it could be assumed that the same effect would be applicable at higher velocities.

The third group includes other influential parameters such as the physical properties of the fluid; their effects are represented by the fuel surface tension and density. The fluid viscosity has also implicitly a major influence on the film profile that would appear mainly when calculating the flow conditions at the nozzle exit. In other words, its effect appears on the flow-angle, the film thickness and the injection velocity; however, it may also have a secondary influence outside the nozzle but this has been neglected in this study. In the following subsections the detailed effect of aforementioned parameters on the liquid film profile will be discussed. The input data for the analysis are summarised in Table 3-1; these data are kept constant while changing each parameter to examine its effect on the film profile.

Table 3-1 Input data to the liquid film divergence model.

Initial film thickness	100 μm
Axial velocity	50 m/s
Flow angle	30 deg.
Surface tension	0.02 N/m
Nozzle radius	0.45 mm
Pressure difference across the film	0 bar
Liquid density	750 kg/m^3

3.2.2.1 Effect of injection velocity

The injection velocity depends on the injection pressure which is controlled by a pressure regulator inside the common rail system. Increasing the injection velocity leads to an increase in the quantity of fuel injected and improved atomisation. In order to examine the effect of injection velocity on film divergence, the normalised film radius is plotted in Figure 3-3 versus the normalised distance from the nozzle exit for different injection velocities. This figure shows that the rate of increasing the radius of the film profile remains almost constant when the injection velocity is higher than 8m/s while for injection velocities of 2m/s and 4 m/s the rate is lower. These results agree well with observation based on the spray images and with the conclusion that the spray angle does not depend on injection pressure, in agreement to the results of Nouri et al. (1999). It is

worth stressing again that the spray angle is not the film divergent angle but rather a result of it. The variation in the film divergent angle along the spray axis for different injection velocities is presented in Figure 3-4. This figure shows that the film divergent angle reaches its asymptotic value of 38 deg. at a distance of about 5 times the nozzle radius (~ 2.5 mm). This angle is larger than both the flow angle at the nozzle exit and the measured spray angle and, therefore, it is expected that the film breaks-up before its divergence reaches its asymptotic value; the break-up length is thus shorter than 2.5 mm.

Another interesting result is that at all injection velocities a tulip shape is noticed but with a shorter closing length when increasing the injection velocity, as shown in Figure 3-4. The tulip shape is clearly present at very low injection velocities and it has been studied extensively by many authors such as Ramamurthi and Tharakan (1998); Hassan et al. (1997) and others. The reason for not observing the tulip shape at higher velocities is that the film disintegrates at relatively short distances compared to the closing length of the tulip, and even before it reaches its maximum opening where the conical spray shape is produced. The results shown in Figure 3-3 and 3-4 also showed that a multiple tulip shape occurs at all injection velocities in agreement with the results of Sivakumar and Raghunandan (1997); however, the experiments do not show that shape even at very low injection velocities due to the earlier film disintegration. In order to physically observe a tulip shape, the liquid film has to be injected at velocities less than 2m/s and without pressure fluctuations which may cause collapsing of the film, as shown in Figure 3-3 and 3-4. However this velocity is very low when compared with the velocity of the developed liquid film at the very early stage of DISI spray development. Hence, one can argue that the tulip shape does not exist during the film development in a DISI injector due to the relatively high flow dynamics associated with the needle opening. This argument is supported by the calculations performed using the VOF method by Arcoumanis and Gavaises (2000) who showed much higher axial and swirl velocities of the liquid film during the early stage of film development near the nozzle exit than the previously mentioned transition value. Accordingly, one could conclude that the formation of the pre-spray is not due to the tulip shape of the liquid film but rather it is related to the absence of film formation inside the injector nozzle even after reaching a relatively high injection velocity. That also means that if a steady flow exits the nozzle at even much lower velocity than that occurring during the early stage of DISI spray development, a conical spray can be formed without pre-spray.

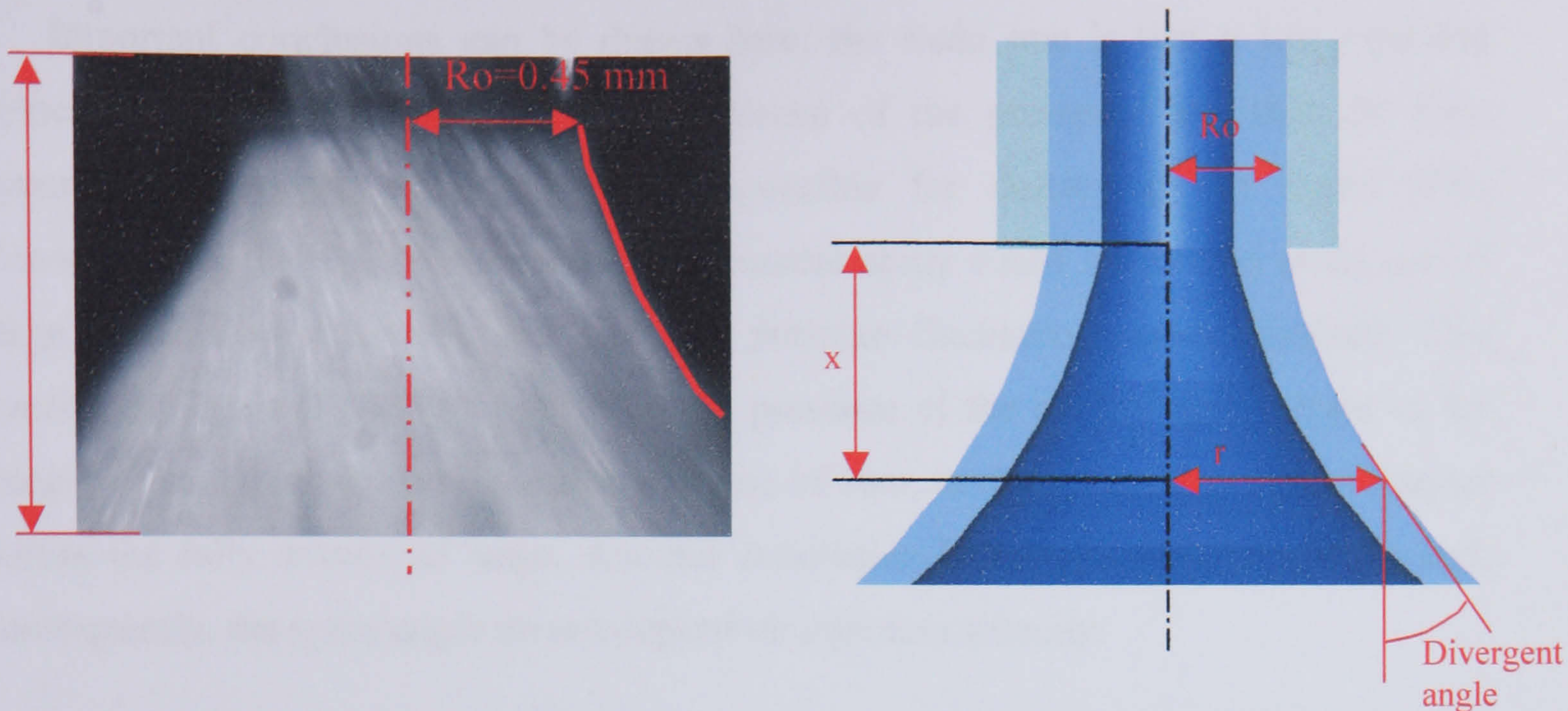


Figure 3-2 Geometry of film profile outside the nozzle exit at 70 bar injection pressure.

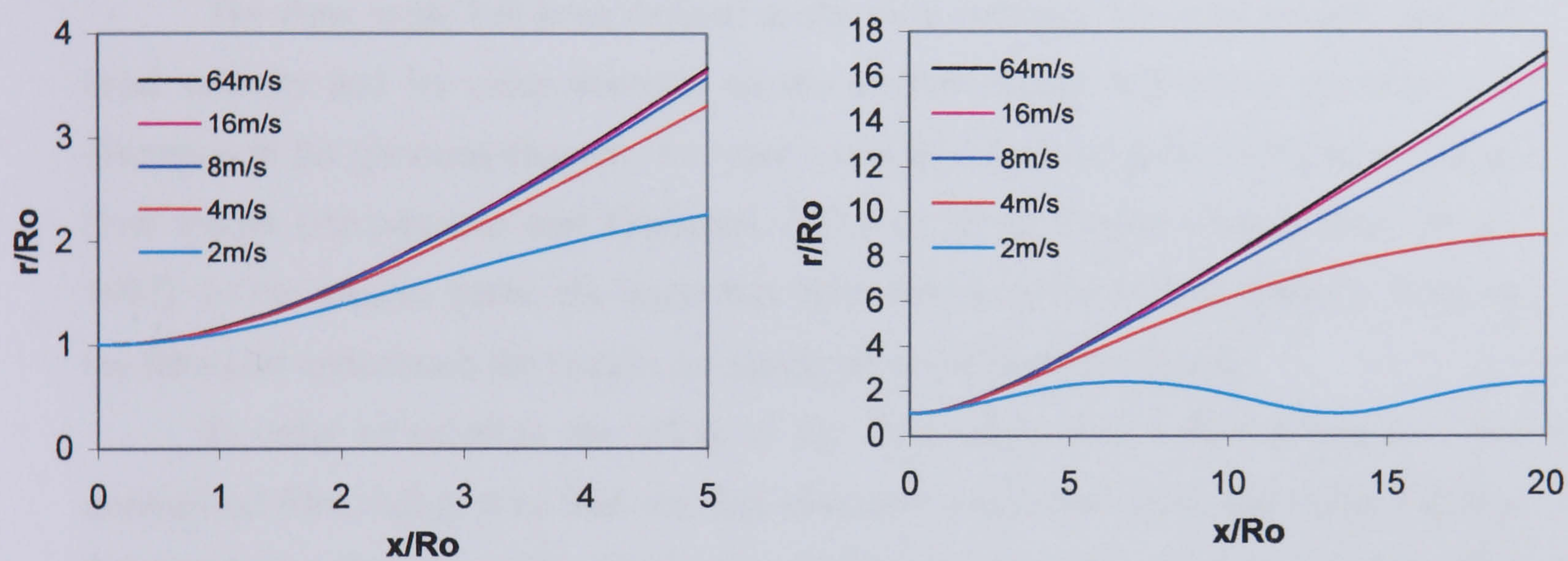


Figure 3-3 Effect of injection velocity on film divergence outside the nozzle.

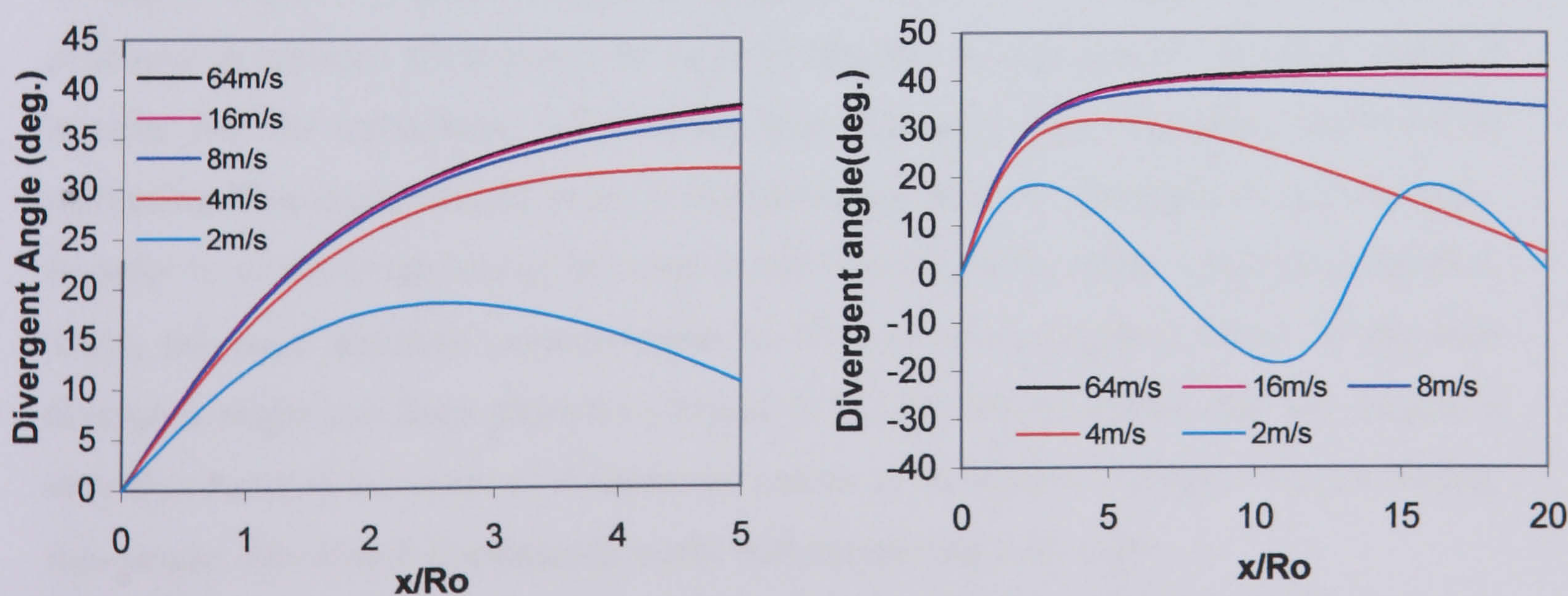


Figure 3-4 Effect of injection velocity on film divergent angle outside the nozzle

Important conclusions can be drawn here; the main one is that a low injection velocity is not a prerequisite for the presence of the pre-spray and that the flow dynamics during the early stage is responsible for destroying the liquid film. Consequently, the film and therefore the conical spray could be formed at an earlier stage if the flow dynamics and associated pressure fluctuations are minimised. This conclusion has been well supported by the presence of the conical spray shape in the images obtained at very low injection pressure of 3bar, as shown in the previous chapter during the fully developed stage. Another conclusion is that the film divergence and, consequently, the spray angle do not depend on injection velocity.

3.2.2.2 Effect of flow angle

The flow angle has been defined as the ratio between the swirl velocity and the axial velocity and its value depends on the swirler design and nozzle geometry, as discussed in the previous chapter. Its value could be calculated either using a two-phase flow model (Arcoumanis and Gavaises, 2000) or using simpler models (Ren et al., 1997). In the present study, the angle has been determined experimentally by imaging the flow just underneath the nozzle, as explained in the previous chapter.

In order to examine the effect of the flow angle on the film divergence, the normalised film radius (r/r_o) and the film divergent angle have been plotted in Figures 3-5 and 3-6 against the axial distance for different flow angles. The results show that the rate of increase of the film divergence increases with flow angle causing the film divergent angle to reach a higher asymptotic value faster. Figure 3-7a shows the relationship between the asymptotic value of the film divergence and the flow angle; It is clear that the asymptotic value of the film divergent angle increases linearly with increasing flow angle and its value is always larger than the corresponding flow angle. In order to evaluate the rate of increase of the film divergent angle with increasing flow angle, the axial distance corresponding to 95% of the asymptotic value of the film divergent angle has been plotted in Figure 3-7b. The figure shows that this length is very short and of the order of 8 times the radius of the nozzle (3.6mm), for the 30 deg. flow angle; this length decreases linearly with increasing flow angle.

Since increasing the flow angle causes an increase in the film divergent angle within a shorter time, it could be concluded that the spray angle will also be increased while the break-up length decreased.

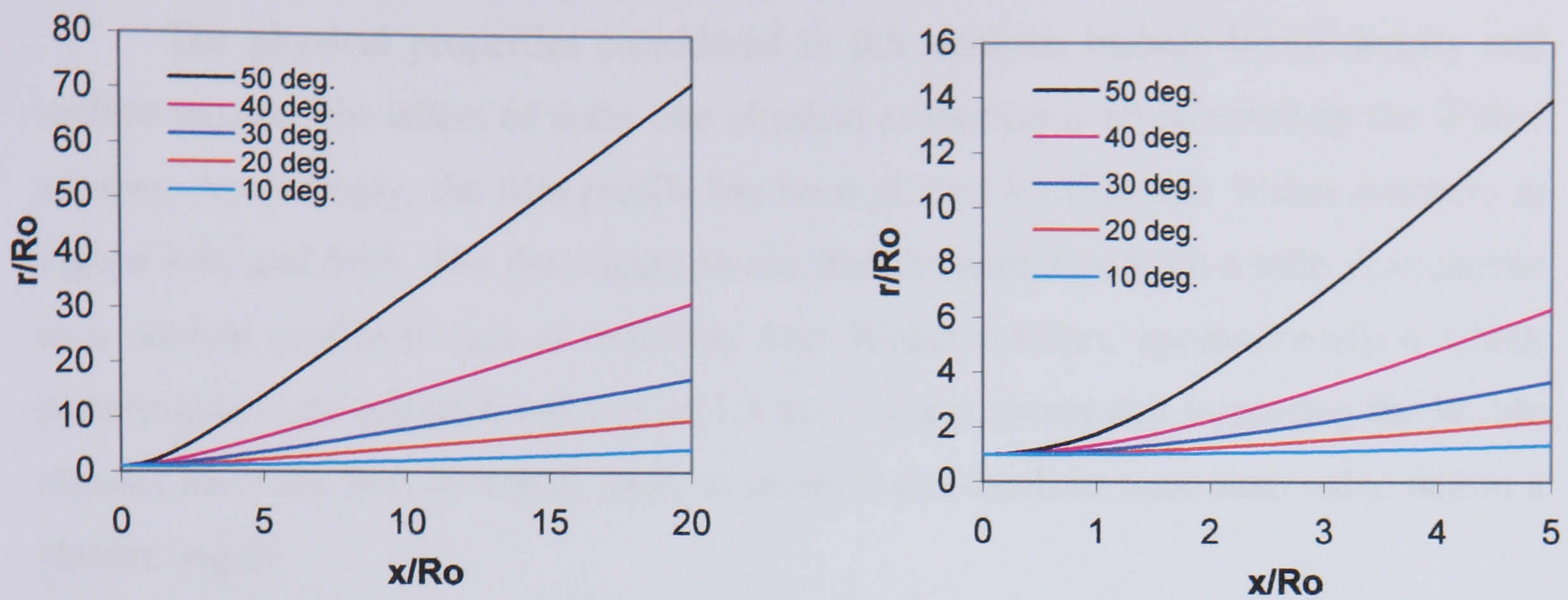


Figure 3-5 Effect of flow angle at the nozzle exit on film divergence outside the nozzle.

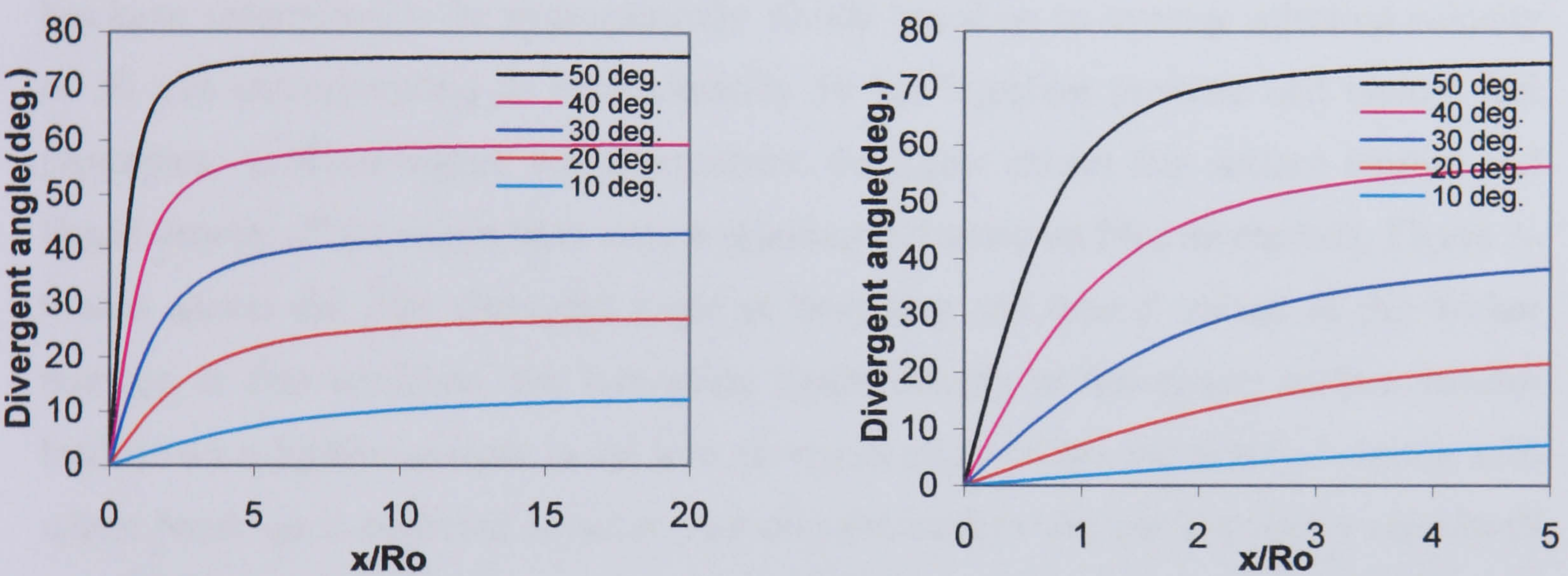


Figure 3-6 Effect of flow angle at the nozzle exit on film divergent angle outside the nozzle

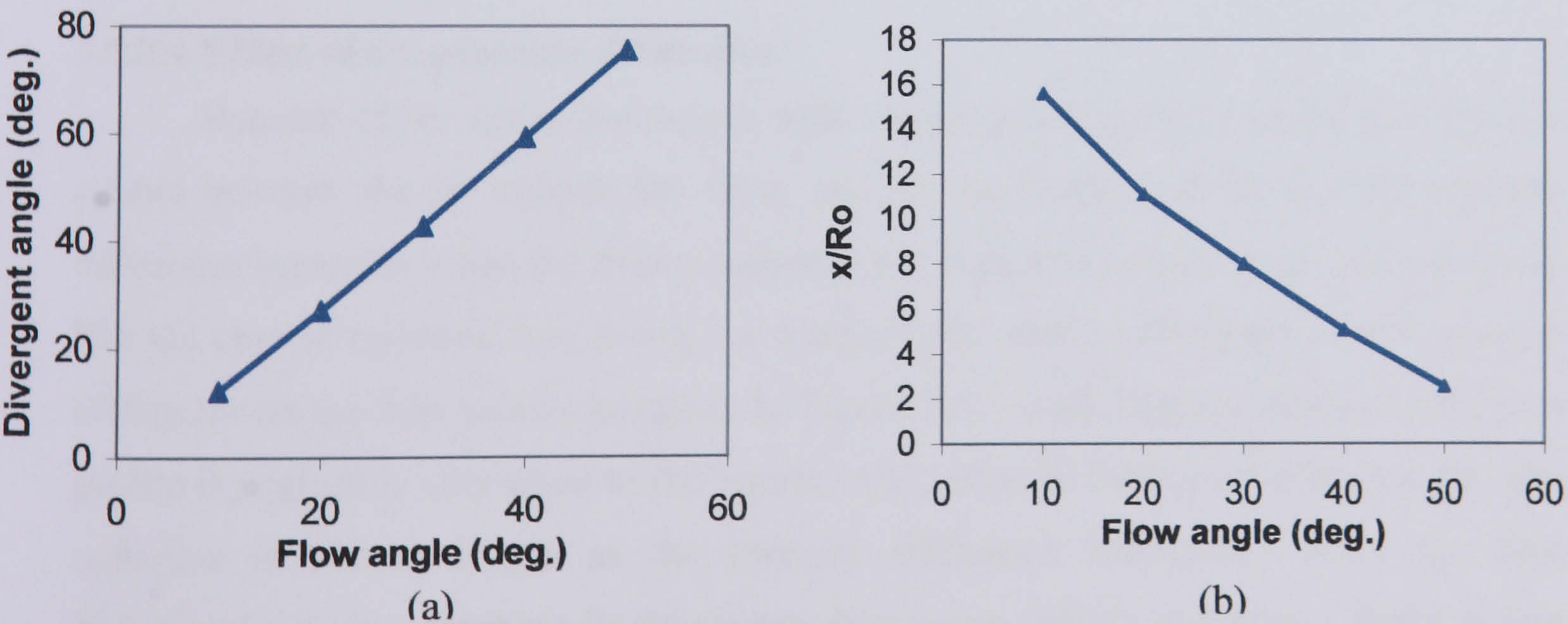


Figure 3-7 Effect of flow angle at the nozzle exit on asymptotic value of film divergent angle and its corresponding distance from the nozzle exit (a value of 95% of the asymptotic value is used)

3.2.2.3 Effect of physical properties

The physical properties considered in this analysis include liquid density and surface tension; the effect of these two physical properties is represented by the Weber number. Accordingly, the film profile has been plotted for different Weber numbers in Figure 3-8a and 3-8b. The first figure shows that the transition from a tulip flow profile to a conical profile occurs at relatively low Weber number, approximately 6 which corresponds to an injection velocity of 1.3 m/s. It also shows that increasing the Weber number leads the film divergent angle to increase and reach its maximum value within a shorter length.

Figure 3-8b shows the profile of the liquid film at higher Weber numbers more representation of sprays in DISI engines. The average Weber number at the nozzle exit has been determined to be approximately 10,000 based on an average injection velocity of 50 m/s corresponding to approximately 50 bar injection pressure and typical fuel properties. At these higher Weber numbers, the figure shows that surface tension and liquid density of the liquid have only a minimal influence on film divergence. Figure 3-9 also shows the film divergent angle at both low and typical values of the Weber number. It also confirms that increasing liquid density or decreasing surface tension leads to a negligible increase in the film divergent angle within the first five nozzle radii where break-up is expected to occur. Actually the surface tension force has a very small contribution on film divergence at such high values of Weber number when compared with the centrifugal forces.

3.2.2.4 Effect of air pressure difference

Because of the spray interaction with the surrounding air, a pressure difference occurs between the air outside the spray and the air trapped inside it. This pressure difference increases when the film is injected into high pressure air in a confined space like the case of injecting fuel during the compression stroke. The effect of the pressure difference on the film profile is shown in Figure 3-10. Although the change in the film profile is negligible very close to the nozzle exit ($x/R_o < 7$), further downstream the film collapses in shorter length as the pressure difference increases. Since the film disintegrates before reaching its maximum divergence, which occurs at a location less than $x/R_o = 7$, the equation representing the film divergence becomes invalid after film disintegration and instead spray penetration has to be considered. However, one can argue that the high density of liquid droplets close to the film disintegration location

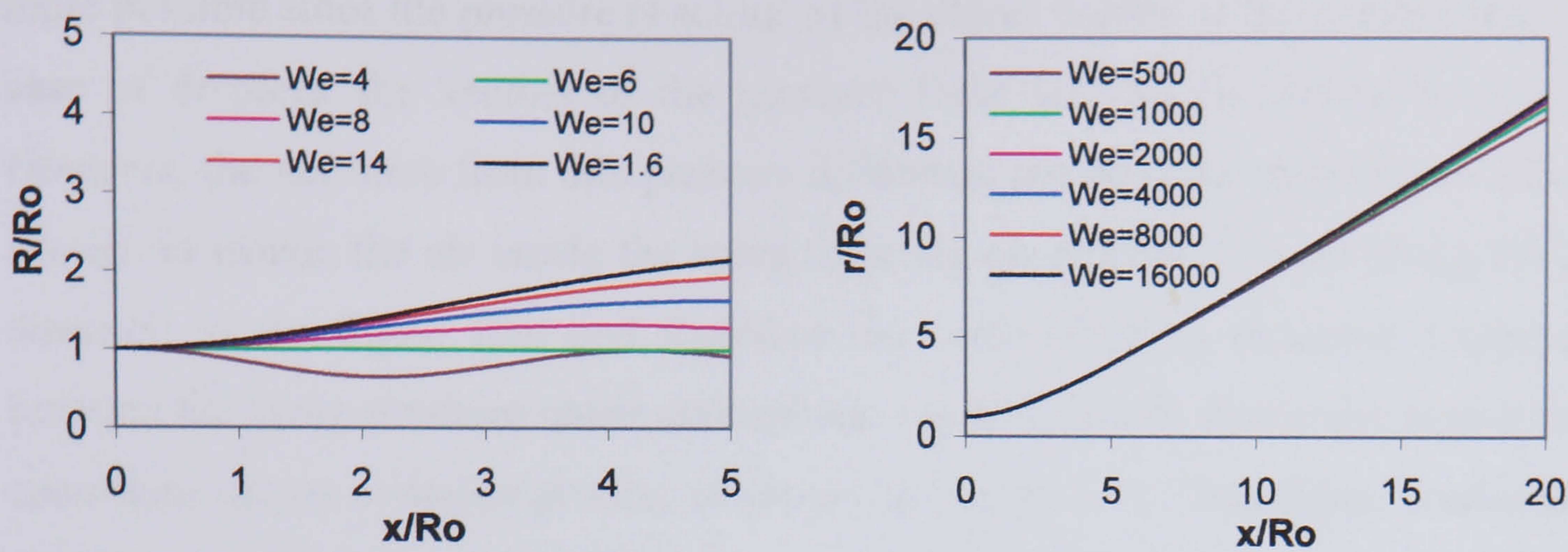


Figure 3-8 Effect of Weber number at the nozzle exit on film divergence outside the nozzle.

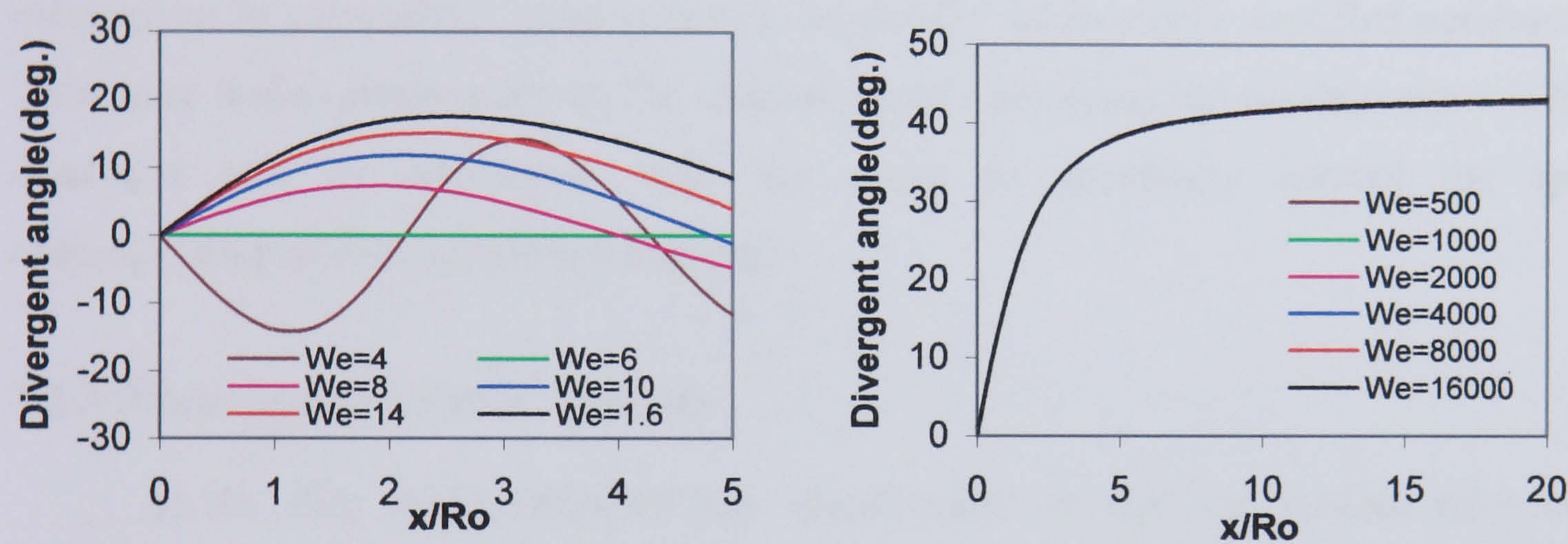


Figure 3-9 Effect of Weber number at the nozzle exit on film divergent angle outside the nozzle.

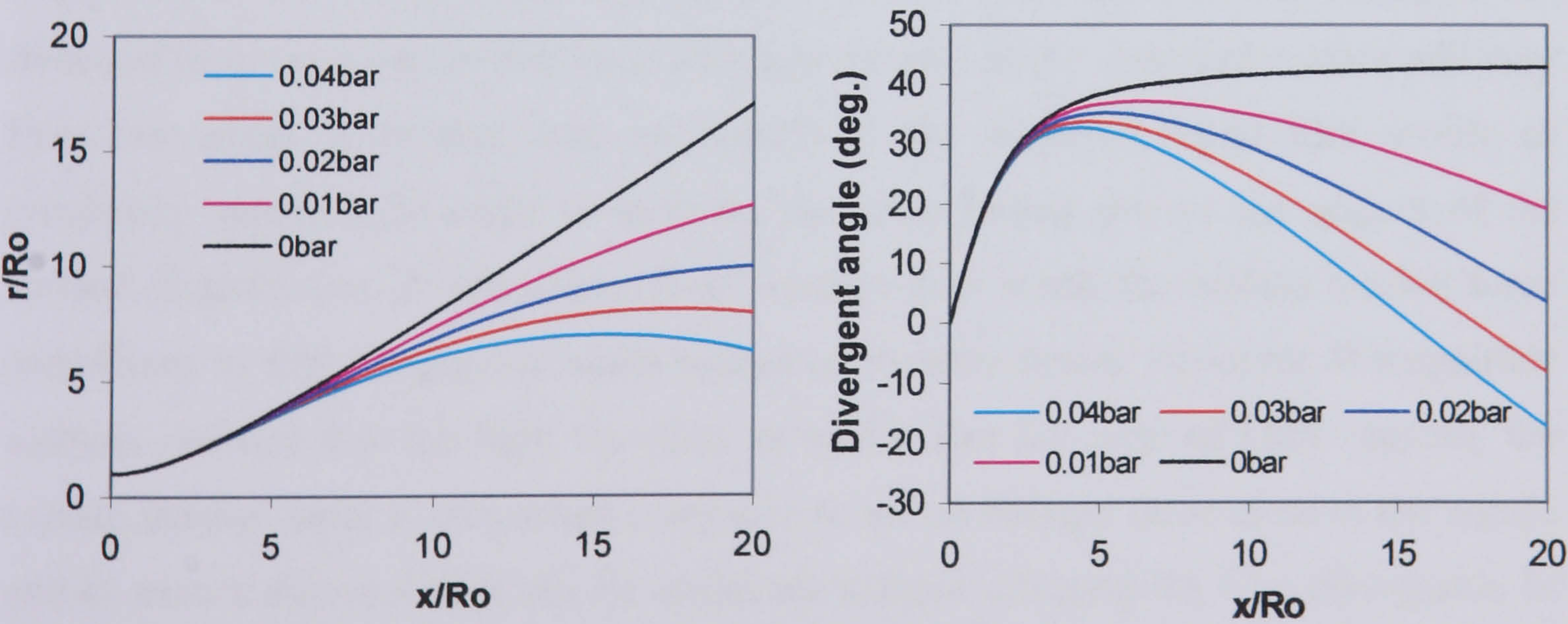


Figure 3-10 Effect of pressure difference across the film on film divergent radius and angle outside the nozzle

makes the droplets behave like a film rather than individual droplets. This argument is more possible since the pressure is acting on the object normal to the surface and, in the case of droplets, the vectors of the pressure force will be all around the droplets. However, the net force from this pressure difference provided the droplets are adequate enough to isolate the air inside the spray from the air outside, will act along the same direction as the liquid film and therefore the same effect is expected. Comparison between the spray structure under atmospheric conditions with that under high-pressure conditions shows a similar profile, as shown in Figure 3-11. This figure confirms that there is no noticeable change in the initial divergent angle and the main effect occurs further downstream as predicted from the film analysis. This considerable change in the film profile due to the change in the pressure difference could be a very useful mechanism in controlling spray structure, especially when stable stratified combustion is the aim. It also emphasises on the importance of identifying the pre-injection air flow dynamics and its interaction with the spray to accurately control the spray characteristics in direct-injection engines.

3.2.3 Discussion and Error Analysis

In the film profile analysis any disturbances on the film surface have been neglected. It could be argued that this assumption would lead to a substantial error especially after looking at the images of the diverged film during the fully developed stage which show significant disturbances on the film surface. Accordingly, the direction of the surface tension force which act normal to the disturbed surface will vary from one point to another and not normal to the proposed liquid film profile as previously assumed. In order to evaluate the error arising due to the neglect of the surface disturbances, it was essential to evaluate how much the surface tension force contributes to film divergence when compared to other forces. An order of magnitude analysis showed that for high injection velocities like the case of DISI engines, the surface tension force is very small compared to the centrifugal force close to the nozzle exit to such a degree that it can be neglected without affecting the film divergence. In fact this is true close to the nozzle exit but less so further downstream; therefore surface tension cannot be neglected if the film does not disintegrate or if the injection velocity is very low. An order of magnitude analysis for the different forces acting on the film, including the gravity force, when neglecting the pressure difference across the liquid

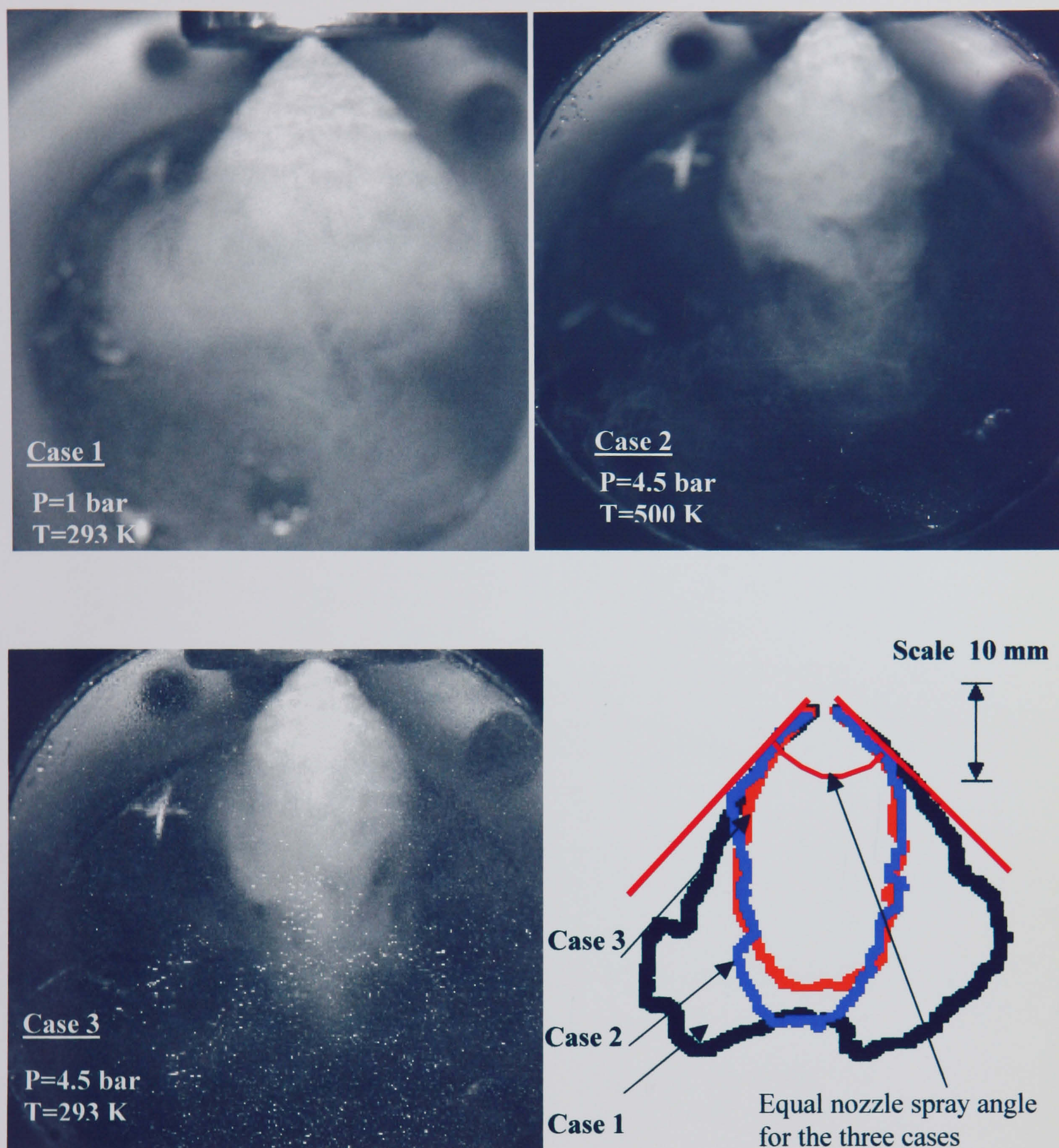


Figure 3-11 Spray images at 1.5 ms show that the surrounding pressure and temperature have almost no effect on nozzle spray angle

film is shown in Table 3-2 in percentage terms. This table shows that for $x/R_o < 5$, the contribution of the surface tension forces do not exceed 1% of the total force. It also shows that the gravity force is very small and thus, could be neglected.

Another interesting point is how much reduction the forces are experiencing as the film develops downstream of the nozzle; the results are presented Table 3-3 which shows the percentage reduction in the centrifugal and surface tension forces when compared to their values at $x/R_o=1$. According to this analysis, only 2.6 % of the centrifugal force is applied on the film after the very short distance of $x/R_o=5$ and therefore the film divergent angle becomes almost constant and the aerodynamic force on the film surface becomes pronounced causing the film to disintegrate.

Table 3-2 Percentage contribution of the different forces applied to the liquid film.

	$F_{\sigma 1}$	$F_{\sigma 2}$	F_{c1}	F_{c2}	F_g
$x/R_o=1$	0.09	-0.03	99.91	-99.97	0.0002
$x/R_o=5$	0.85	-0.08	99.14	-99.92	0.01
$x/R_o=10$	3.85	-0.17	96.12	-99.83	0.03
$x/R_o=20$	15.64	-0.36	84.23	-99.64	0.133
$x/R_o=50$	56.35	-0.95	43.16	-99.05	0.49

Table 3-3 Percentage reduction in the forces applied to the film
as it develops downstream from the nozzle

	$F_{\sigma 1}$	$F_{\sigma 2}$	F_{c1}	F_{c2}
$x/R_o=1$	100	100	100	100
$x/R_o=5$	26.07	8.21	2.62	2.6
$x/R_o=10$	11.5	1.63	0.25	0.24
$x/R_o=20$	5.23	0.25	0.02	0.02
$X/R_o=50$	1.96	0	0.001	0.00

Another issue is the error arising due to the assumption of a uniform velocity distribution across the film thickness. Since it was found that the increase in the

injection velocity does not affect the film profile near the nozzle exit, it can be concluded that the velocity distribution across the film cross section will not affect the film profile. However, it is expected that the velocity distribution affects the surface disturbances and hence, the disintegration process. There is also another one more issue, which is the effect of turbulence on film motion. Rammaurthi and Tharakan (1998) performed some experiments to study the effect of turbulence on the film divergence by changing the liquid film turbulence through varying the surface roughness of the nozzle in a swirl type injector. They found that the surface texture of the sheet becomes more frothy compared to its previous appearance when the surface roughness is increased; however, the transition from tulip to conical shape occurs at the same Weber number and the film shape is not affected.

3.2.4 Spray Angle Calculations

Until now the analysis has been only applied to a continuous liquid film. However, the spray consists of liquid droplets which are formed after the film disintegrates. If a droplet separated from the film is assumed to be moving freely and there are no collisions or agglomeration, the droplet will initially move linearly according to its initial momentum against the aerodynamic force, and will not rotate as the film does until it loses its momentum. If the drag force and all the other forces acting on the droplet are neglected, then the separated droplet will move linearly with a constant velocity. By neglecting any changes in the momentum of the film at the break-up point due to the atomisation process, the velocity of the droplet at this point could be assumed to be equal to the film velocity. As shown previously that the swirl velocity decreases as the film penetrates downstream of the nozzle due to its divergence, the swirl velocity at the break-up length W_b can be calculated according to the conservation of angular momentum as follow:

$$W_b = U_j \tan(\alpha) \frac{R_o}{R_b} \quad (3-18)$$

where, R_b is the radius of the film at the break-up location. Accordingly, this velocity calculated from the above equation is equal to the tangential velocity of the separated droplet; this tangential velocity causes an increase in the radial distance which could be determined geometrically from the droplet linear trajectory between the film break-up location and any axial distance. If the swirl velocity is zero at the location of the film

break-up, then the separated droplet will move linearly at an angle equal to the film divergent angle at the break-up point while the rate of increase in radial distance remains constant. However, if the film breaks at a location where there is a swirl velocity component then an additional increase in the radial distance will result. The ratio between the increased radial distance and the increased axial distance represents the increase in the divergent angle. By adding this increase to the original film divergent angle, the local spray angle can then be identified. In order to identify the spray angle at different locations, the droplet trajectory after it has been separated from the liquid film has to be known.

In order to determine the trajectory of a single droplet after its separation from the liquid sheet, the droplet is considered moving with an axial velocity U_b equal to the film divergent velocity and a tangential velocity W_b equal to the film swirl velocity at the break-up point from the peripheral of a hollow cylindrical liquid film; it is further assumed that the cylindrical liquid film has a radius R_b and a flow angle α_b equal to the film divergent radius and the flow angle at the location of film break-up respectively. Figure 3-12 shows the projection of the trajectory of six droplets (a,b,c,d,e and f) located on the peripheral of a cylindrical liquid film at the break-up location along three perpendicular planes. This figure shows that the droplet (a) moves away from the spray centre with its radial distance increasing by distance y (red line) for any axial distance x . The axial distance x and the tangential distance L after time t could be determined from the following equations:

$$x = U_b t \quad \text{and} \quad L = W_b t \quad (3-19)$$

From triangle (o a₁ a_x) shown in the top view of figure 3-12:

$$L^2 + R_b^2 = (R_b + y)^2 \quad (3-20)$$

since the flow angle represents the ratio between the swirl and the axial velocity then:

$$L = x \tan(\alpha_b) \quad (3-21)$$

Substitute equation (3-21) into equation (3-20) leads to the following equation:

$$x^2 = \frac{y^2}{\tan^2(\alpha_b)} + \frac{2R_b y}{\tan^2(\alpha_b)} \quad (3-22)$$

Equation (3-22) represents the dynamic profile of a single droplet from the start of film disintegration. This equation shows that there is an additional divergence defined by radial distance y for each axial distance x and thus, the spray angle is more than the film divergent angle at the break-up instant. This increase in spray angle depends on the flow

angle or, in other words on the swirl velocity of the liquid film at the point when the film disintegrates. The additional increase in radial distance with axial distance for a flow angle of 30 deg and radius of 0.45mm. is shown in Figure 3-13a. This figure reveals that the rate of increase in the radial distance increases with axial distance until it reaches a constant value. The ratio between the radial and the axial distance represents the droplet divergent angle; Figure 3-13b shows the droplet divergent angle variation as it penetrates downstream of the nozzle, which could be represented by $\frac{dy}{dx}$ where:

$$\frac{dy}{dx} = \frac{2x \tan^2(\alpha)}{(2y + 2R_o)} \quad (3-23)$$

Figure 3-13b shows that the droplet divergent angle increases as it penetrates further downstream of the nozzle until it reaches an asymptotic value. In order to find this asymptotic value of the droplet divergent angle, the second derivative is calculated and equated to zero as follows:

$$\text{For } \left(\frac{d^2y}{dx^2} \right) = 0, \text{ then } \left(\frac{dy}{dx} \right) = \tan(\alpha) \quad (3-24)$$

Equation (3-24) shows that the maximum angle that could be obtained from the penetration of an atomised single droplet from a film having a flow angle α at its break up location is equal to this flow angle.

The conclusion from this analysis is that when an atomised droplet is separated from a film having a flow angle α , it will penetrate away from the film with an angle increasing with downstream distance until it reaches the maximum divergent angle which is equal to the flow angle. Accordingly if the flow is fully atomised at the nozzle exit we would expect that the spray angle will become equal to the flow angle. However, in the case of a break-up length the nozzle spray angle has to be calculated from the film divergence until it breaks up and from the droplet trajectory further downstream. Similarly the nozzle spray angle can be calculated from the sum of the film divergent angle and the flow angle at the break-up point.

3.2.5 Break up Length Calculations

Because of the difficulty to determine the break-up length either numerically or experimentally, the approach proposed here represents an approximation to link the

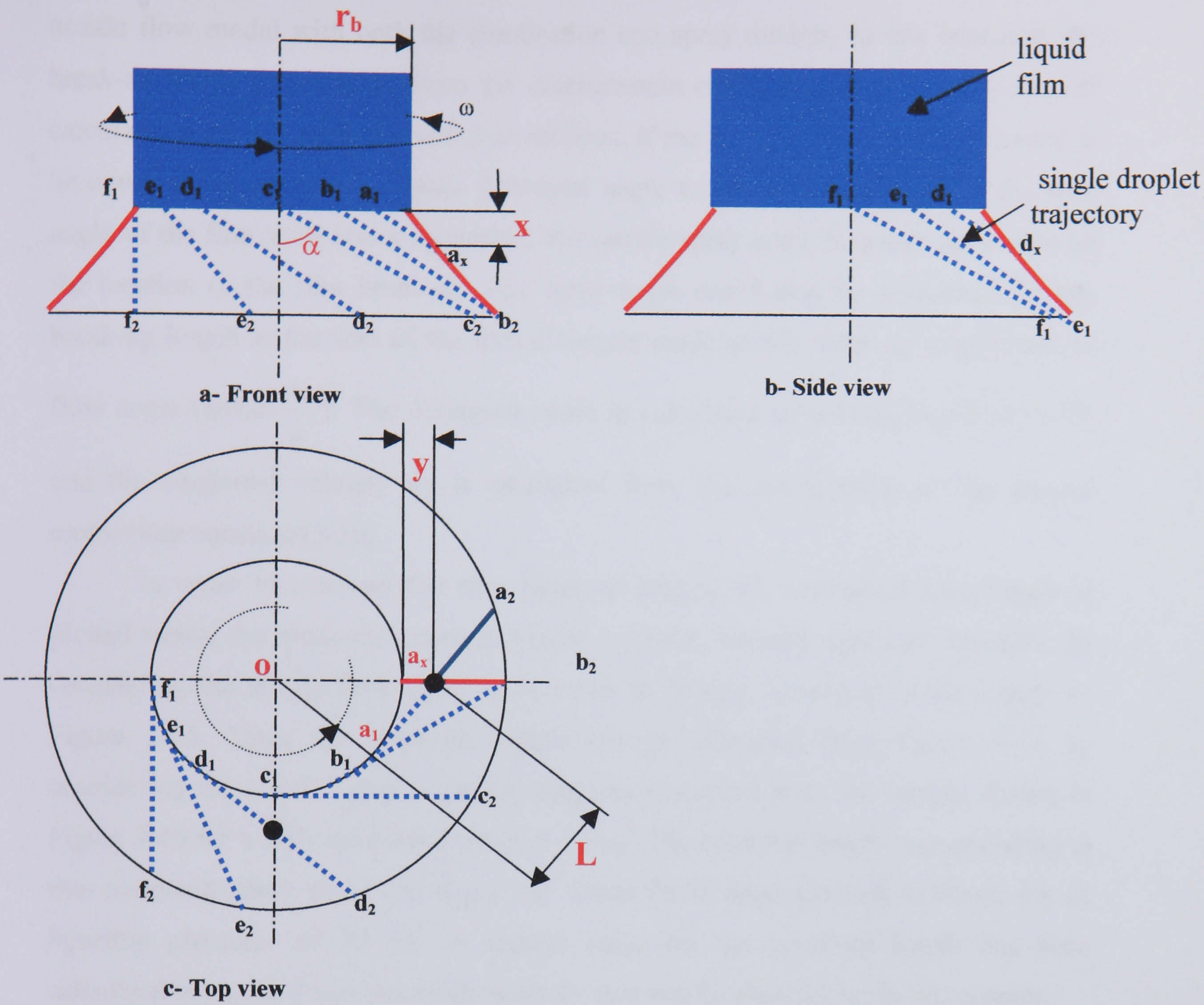


Figure 3-12 A schematic showing the linear trajectory of six droplets moving away from the peripheral of a hollow swirling cylindrical film

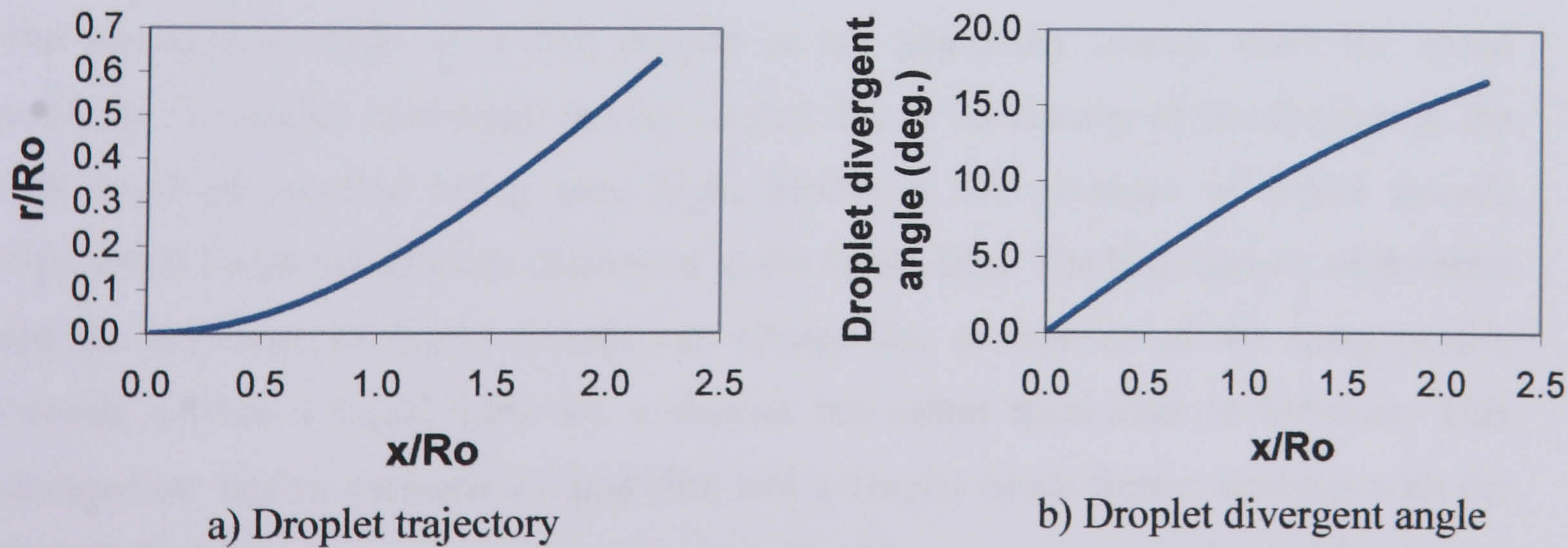


Figure 3-13 A single droplet trajectory and divergent angle for an initial flow angle=30 degree.

nozzle flow model with both the atomisation and spray models. In this approach, the break-up length is calculated from the measurement of the nozzle spray angle and the calculations introduced in the previous sections. If the nozzle spray angle is assumed to be equal to the maximum droplet divergent angle which in turn is equal to the flow angle of the film at its break-up instant, the nozzle spray angle becomes dependent on the location of the film break-up. The spray angle could thus be calculated for any break-up length as the sum of the film divergent angle at this break up length and the flow angle ($\arctan \frac{w_b}{u}$). The divergent angle is calculated by solving equation (3-17) and the tangential velocity w_b is calculated from the conservation of the angular momentum equation (3-10).

In order to estimate the film break-up length, the calculated spray angle is plotted versus the break-up length in Figure 3-14 for different injection velocities, by considering the nozzle flow angle to be equal to 30 deg. according to the images of Figure 3-15. Thus, the break-up length can be identified from Figure 3-14 by considering the nozzle spray angle (36 deg.) as measured from the images shown in Figure 3-15 for a fully developed spray at 1.5ms. The break-up length corresponding to this measured spray angle (36 deg.) was found to be approximately 0.17mm for an injection pressures of 70 bar. A similar value for the break-up length has been calculated using the linear instability analysis as it will be shown later in this chapter.

Although this analysis is simple and is based on several assumptions, it allows researchers to obtain in sight into the various forces and the process of droplet collisions and agglomeration and, through them, to accurately define the main droplet trajectory. The assumption made of a free droplet is not physically correct since the space available for droplet movement is very limited due to the density of the droplets at the film break-up location being very high. Moreover the presence of liquid threads (ligaments) keeps the droplets connected to the liquid film. The high density of droplets and the existence of liquid threads can change the divergence of the spray profile towards neither a liquid film nor a droplet but rather somewhat in between. This intermediate region between a liquid film and a droplet needs further analysis with the aim of obtaining more accurate results. Another interesting point is the swirl velocity, which has been considered decaying according to the conservation of the angular momentum but without considering the losses associated with the aerodynamic force and the atomisation process.

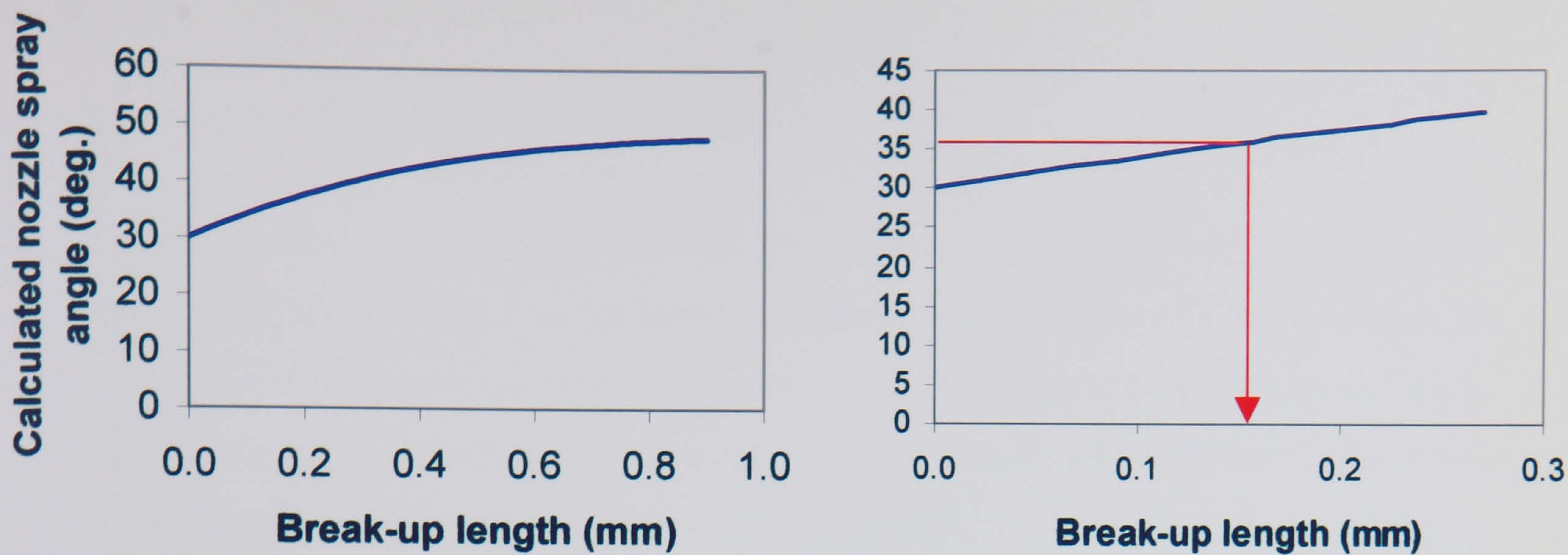


Figure 3-14 Relationship between calculated spray angle and break-up length for a fully developed spray, nozzle length=1.3 mm, initial flow angle=30 and axial velocity=50m/s

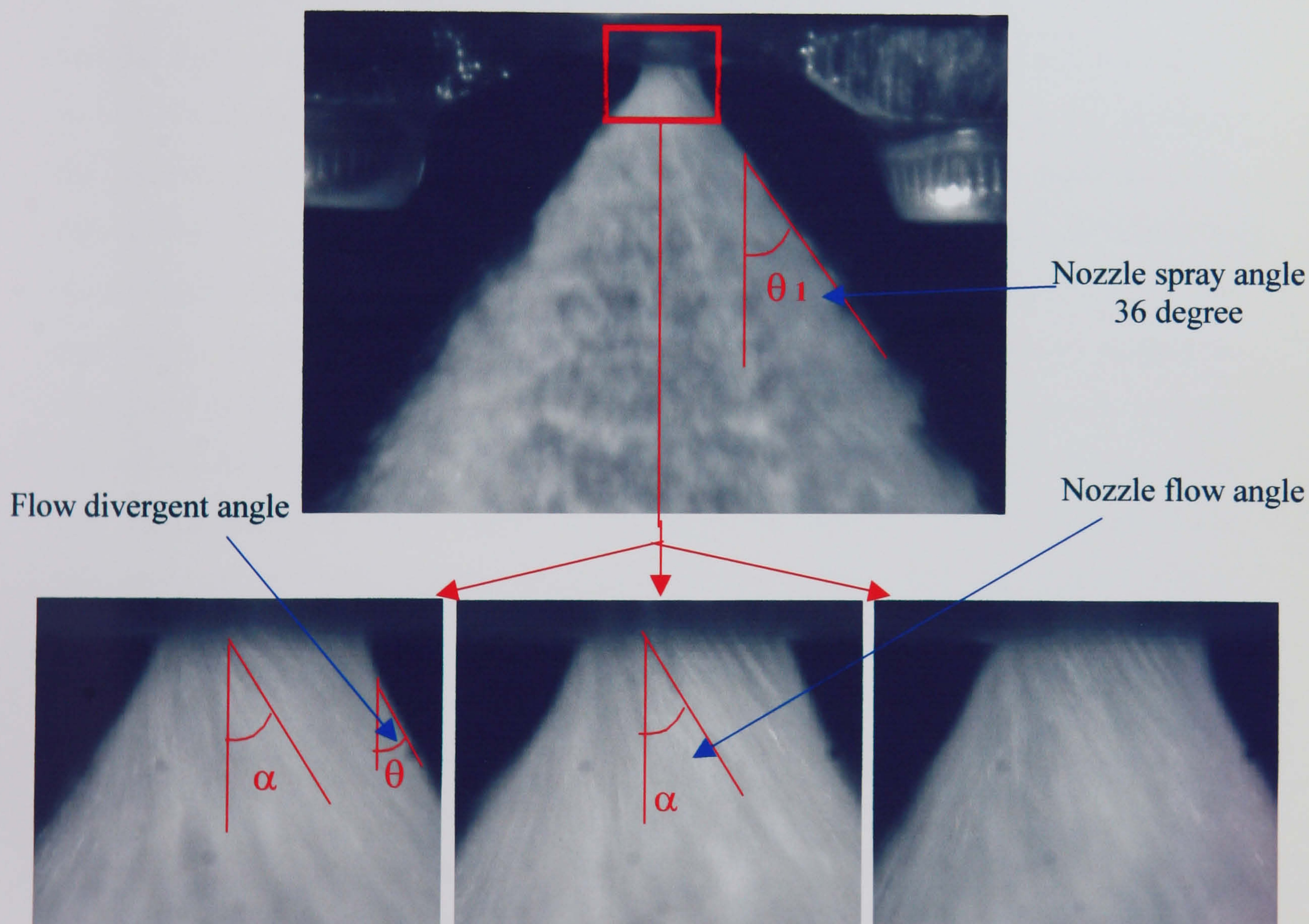


Figure 3-15 Spray images show the flow angle at the nozzle exit and the nozzle spray angle and also the flow divergent angle variation with axial distance at 1.5 ms for nozzle length of 1.3mm and injection pressure of 70 bar.

3.3 LINEAR INSTABILITY ANALYSIS

When the liquid film is injected from the nozzle, its surfaces become non-confined and are subjected to aerodynamic forces which cause 3-D disturbances, as shown in the film images near the nozzle exit presented in the previous chapter. This 3-D, non-linear, two-phase flow problem practically cannot be solved numerically due to the small length scale of the surface waves. Moreover, the initial boundary conditions cannot be well defined in view of the absence of the accurate flow pattern inside the nozzle and the relatively small turbulence length scale. Also, the surface roughness of the nozzle wall may have a significant effect on the film surface disturbances inside and outside the nozzle which cannot be defined accurately.

The solution of this non-linear problem is not unique. This is attributed to the fact that the surface boundary conditions are applicable to a surface which is not defined and cannot be specified until the solution is reached. In order to simplify the problem, the surface location is formulated by considering it perturbed by a mathematically represented wave. This wave is defined by its wave number k which is related to wavelength λ through $k=2\pi/\lambda$. A solution is then obtained by solving the mass and momentum conservation equations for different wave numbers. The wave number at which the amplitude increases faster than others (with the maximum growth rate), is considered as the most favourable mode of sheet disintegration. Thus, disintegration may take place at any wave number depending on the film flow initial conditions (film thickness and velocity, surface roughness of the nozzle wall and fluid physical properties) as well as the conditions outside the nozzle (air flow pattern around the liquid film, liquid film divergent and velocity). In order to derive a mathematical representation of this problem, a simple geometry of liquid film outside the nozzle is assumed with constant air flow velocity.

The following subsections describe the mathematical documentation of an atomisation model for cylindrical liquid sheets using the linear instability analysis. A swirl inviscid hollow cylindrical liquid jet subjected to an axial air flow from both inside and outside, with different velocities is assumed. The model provides estimates of the optimum wavelength and growth rate of the waves assumed to be developing on the surface of the liquid film. A break-up model based on Hagerty and Shea (1955) has also been introduced to estimate the volume mean diameter D_{30} ; this model considers a

cylindrical film rather than a flat sheet as proposed by Reitz et al. (1999), and as such takes into account the effect of the fuel swirl motion.

3.3.1 Model Assumptions

To apply the linear instability theory to the continuous flow of the liquid injected from a nozzle, it is assumed that the conical sheet formed outside the nozzle mainly due to the centrifugal forces, is subjected to an air flow on its sides, as shown in Figure 3-16. The aerodynamic forces acting on the initially disturbed liquid surface cause the unstable disturbance to grow up with its amplitude increasing exponentially with time (Squire, 1953 and Tokuoka et al., 1985). The amplitude of the unstable disturbances (a) increases with a growth rate (ω) so that $a=e^{\omega t}$. Although many unstable waves are present on the conical sheet, there is only one wave with an optimum wavelength (λ_{opt}) which corresponds to the maximum growth rate (ω_{opt}). The amplitude of this wave increases faster than the other waves and thus it is reasonable to consider it as the wave corresponding to film break-up. The liquid film is assumed to break-up when the depth of the disturbance relative to the initial disturbance reaches a certain value to form ligaments which in turn are converted into droplets. The following assumptions have been adopted in order to derive the appropriate flow equations that predict the atomisation process of the liquid film emerging from the injection hole of the pressure-swirl atomiser:

1. The exiting from the nozzle hole liquid sheet has an annular cross-sectional area.
2. A one-dimensional flow is assumed in which the thickness of the annular liquid sheet remains constant. This implies that the liquid film keeps its initial annular shape as it travels downstream the injection hole.
3. The mean velocity of the liquid sheet remains constant, and therefore the mean flow is steady; both the axial and swirl velocity components are considered.
4. The flow is incompressible in terms of both the liquid and the surrounding air.
5. The flow is non-viscous.
6. Gravitational and any other external forces that may act on the liquid film are neglected.
7. Aerodynamic drag effects that may act on the liquid film are neglected.
8. The film disintegrates due to the presence of surface waves developing on its surface with increasing amplitude.

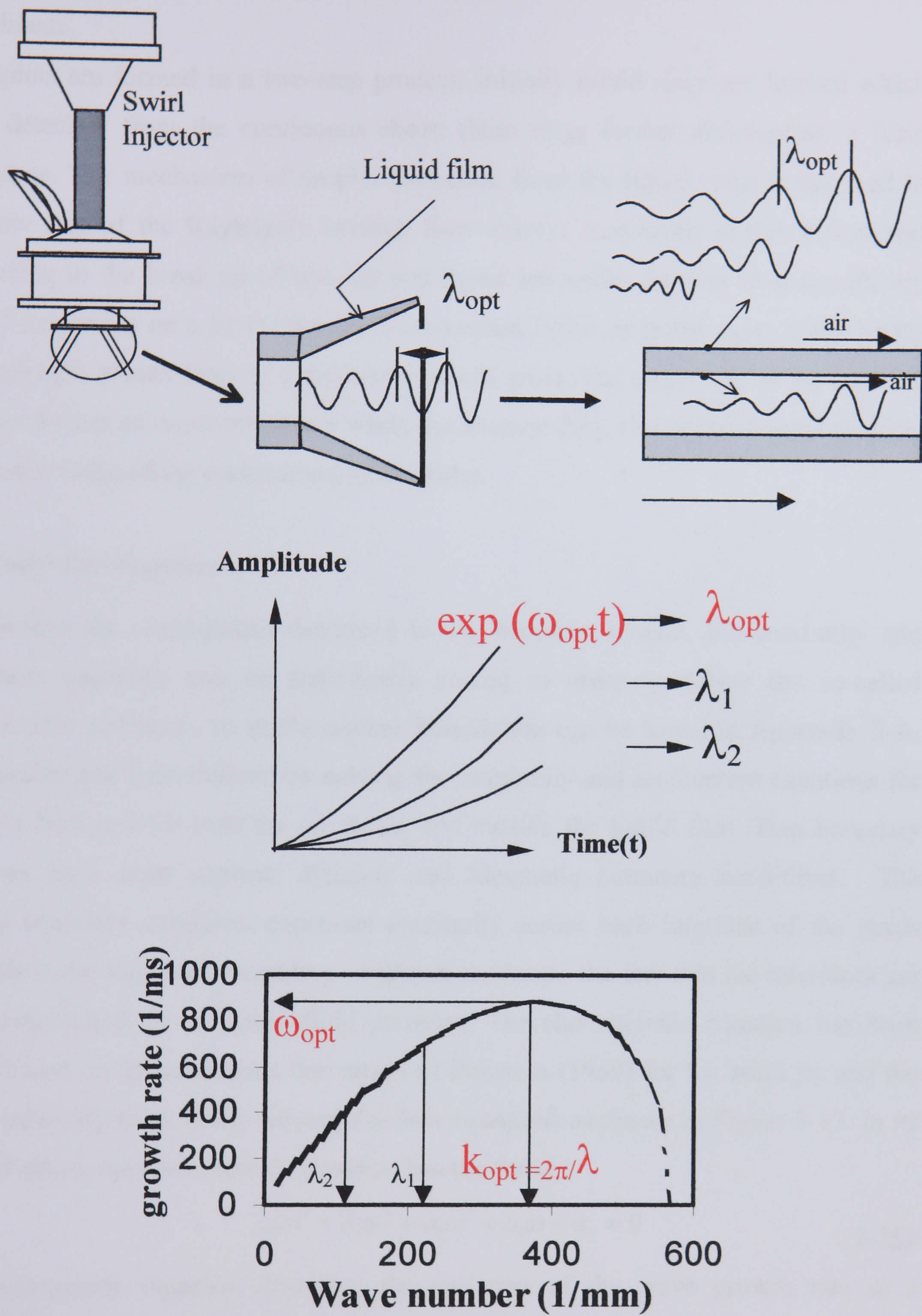


Figure 3-16 A schematic showing the unstable waves that are generated on both sides of the liquid film near the nozzle exit of a swirl injector according to the linear instability analysis

9. The amplitude of the waves is small compared to the wavelength and the film thickness.
10. Droplets are formed in a two-step process: initially liquid rings are formed which are detached from the continuous sheet; these rings further disintegrate to form droplets. The mechanism of droplet formation from the liquid rings is assumed to follow that of the Rayleigh's laminar flow theory. According to this theory and referring to the break up of non-viscous liquid jets under laminar flow conditions, all disturbances on a jet (a circular cross-section ligament in the present case) with wavelength greater than its circumference will grow. The amplitude of the unstable waves follow an exponential law while the average drop size of the formed droplets is nearly twice of the undisturbed jet diameter.

3.3.2 Model Development

Under the assumptions described in the previous section, the continuity and momentum equations can be analytically solved in order to derive the so-called 'characteristic equation'; its mathematical formulation can be found in Appendix 3-A. This equation has been derived by solving the continuity and momentum equations for the liquid film and for both the air inside and outside the liquid film. Two boundary conditions have been applied; dynamic and kinematic boundary conditions. The dynamic boundary condition expresses continuity across each interface of the stress tensor while the kinematic boundary condition expresses the fact that the interfaces are always constituted by the same fluid particles. The characteristic equation has been derived based on the analogous derivation of Ponstein (1959) for the solid jet and the hollow, infinitely-thick jet by solving the flow equations as shown in Figure 3-17. In its final derivation, the characteristic equation has the form:

$$a_4\omega^4 + a_3\omega^3 + a_2\omega^2 + a_1\omega + a_0 = 0 \quad (3-25)$$

This characteristic equation describes the variation of the wave growth rate as a function of the various flow parameters and the liquid and surrounding gas physical properties. Solution of this equation provides the variation of the wave growth rate as a function of the wave number; the growth rate represents the rate of increase of the amplitude of the wavelength that corresponds to this particular growth rate. The maximum growth rate and its corresponding wave number are the necessary

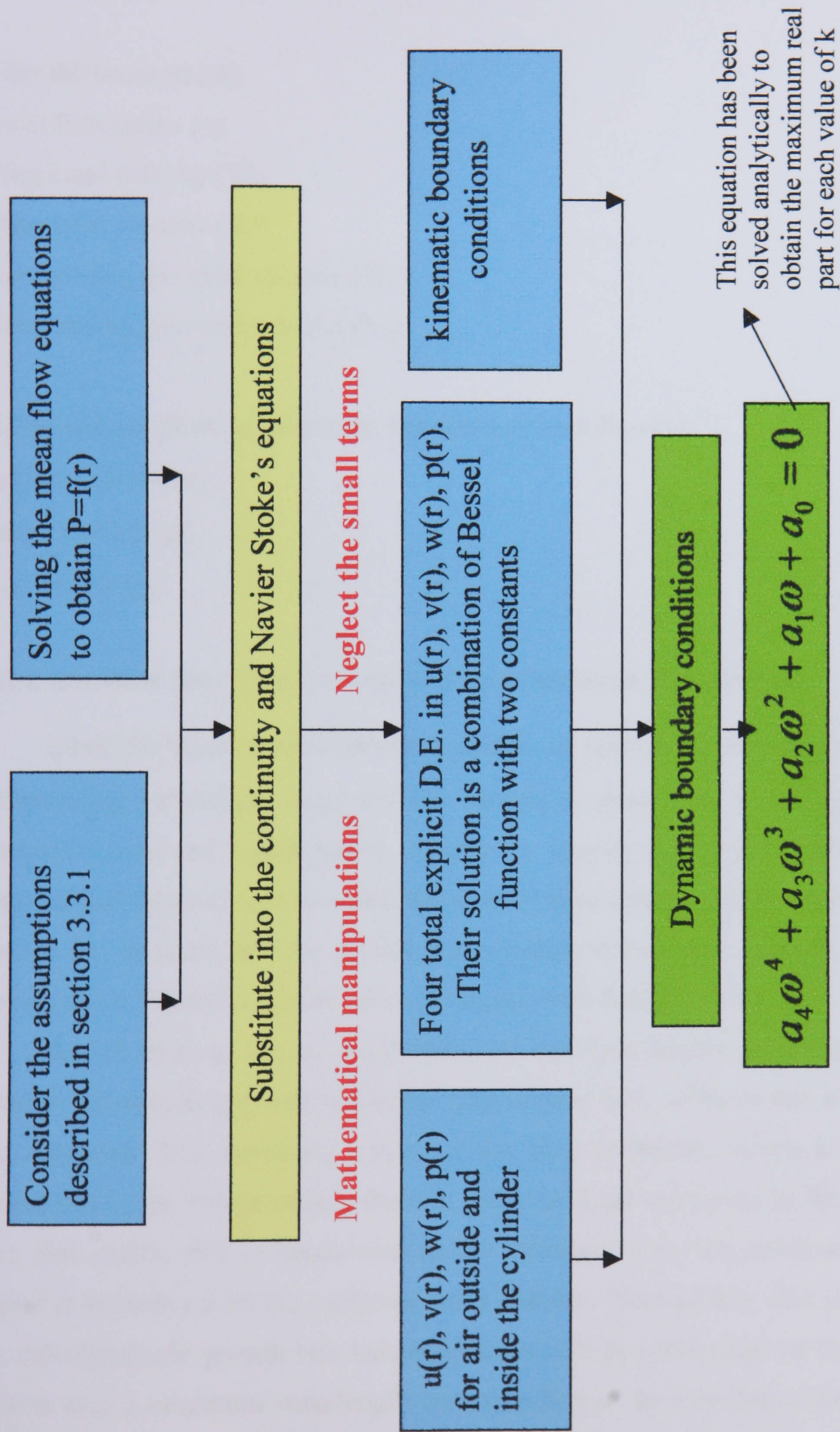


Figure 3-17 A schematic diagram showing the procedures to deduce the final characteristic equation of the waves developing on a swirling hollow cylindrical liquid film subjected to air flow from both sides

information required to apply the film break-up criteria and to calculate the mean droplet diameter.

The flow variables required as initial input conditions to the model are the following :

1. Film thickness (thick)
2. Inner film radius (r_i)
3. Film axial velocity (W_l)
4. Film swirl velocity (V_l)
5. Surrounding gas axial velocity (W_a)
6. Surrounding gas swirl velocity (V_a)

The fuel and gas physical properties required are the following:

7. surface tension (σ)
8. liquid density (ρ_l)
9. gas density (ρ_g)

3.3.3 Parametric Study for Growth Rate and Optimum Wavelength

Using the liquid sheet atomisation model, the optimum growth rate ω_{OPT} and the corresponding wavelength λ_{OPT} are first estimated, since their values determine the atomisation time and length scales. In order to specify the optimum growth rate, a numerical procedure is used in which the wave number increases from zero to its cut off value. It can be found that the growth rate variation versus wave number plot has the form shown in Figure 3-18a for different values of the axial velocity:

As can be seen, for any combination of input parameters, there are waves with different wavelength growing up on the film surface with different rate of increase in their amplitude. For increasing values of the (kr_o) parameter, where k is the wave number and r_o the nozzle radius, there is only one local maximum in the growth rate curve that exists; this is considered as the optimum one; the corresponding wave number is considered as the optimum wave number. Numerically, this is determined after calculating the growth rate values for the whole possible range of (kr_o) numbers. There is also a minimum wavelength corresponding to the maximum $(kr_o)_{max}$ value beyond which there is no growth of these waves, the growth rate becomes again zero

for a critical (kr_o) greater than the maximum value $(kr_o)_{\max}$. For numerical purposes, the more the assumed k values, the better the accuracy in the estimation of the optimum growth rate.

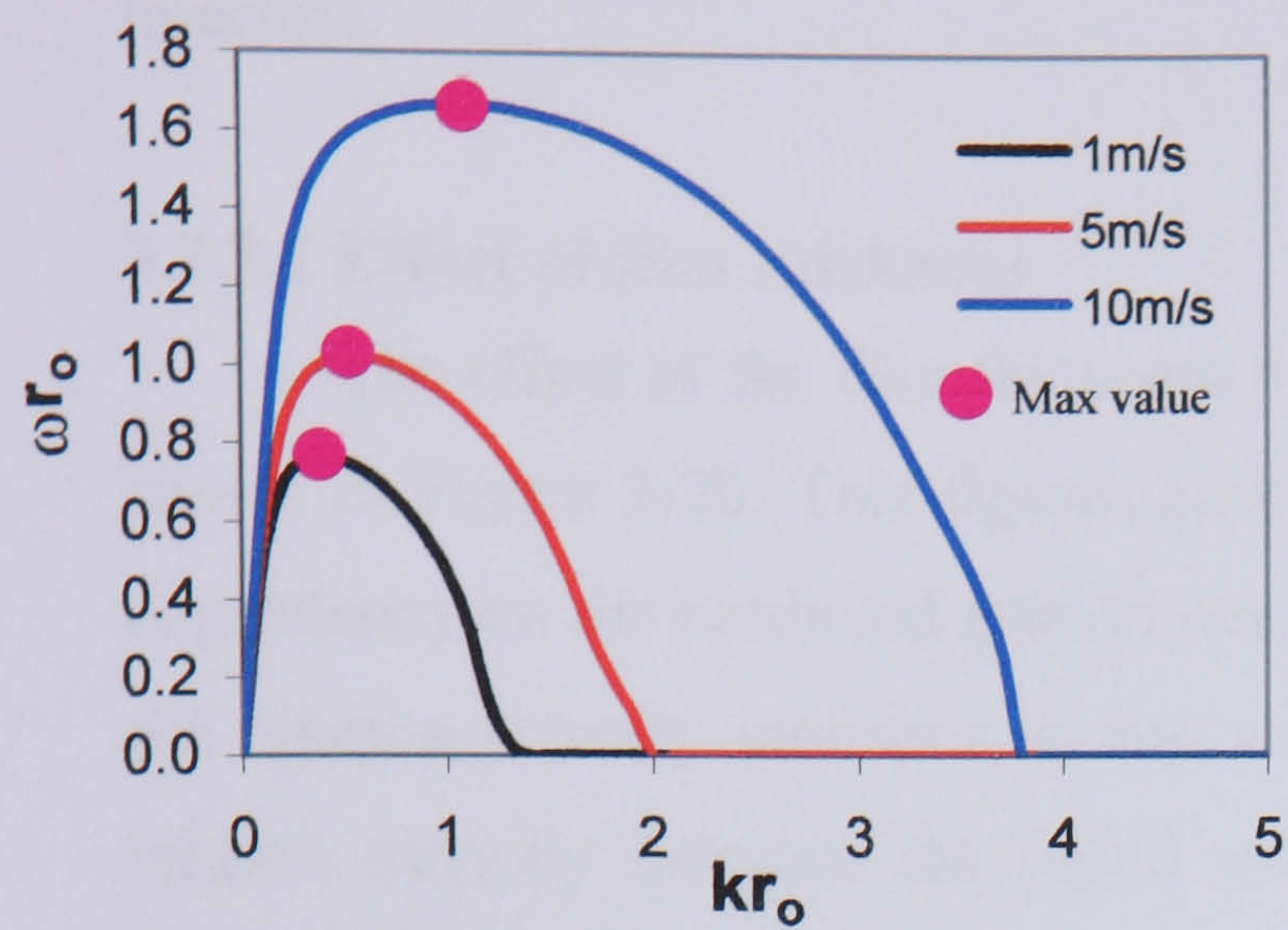
In order to enhance the numerical accuracy with no penalty in the computational time, it has been considered necessary to obtain estimates for the $(kr_o)_{\max}$ value which also varies with the film flow conditions. Therefore, in order to avoid performing calculations within a predetermined and fixed $(kr_o)_{\max}$ value that could cover all cases and which would unavoidably result in estimating the growth rate in a range that is known to be zero, Gavaises and Arcoumanis (2001) proposed the derivation of empirical correlation for the $(kr_o)_{\max}$ value as a function of the flow conditions; the formula proposed has the following form:

$$(kr_o)_{\max} = f U_{REL}^2 \rho_{air} \quad (3-26)$$

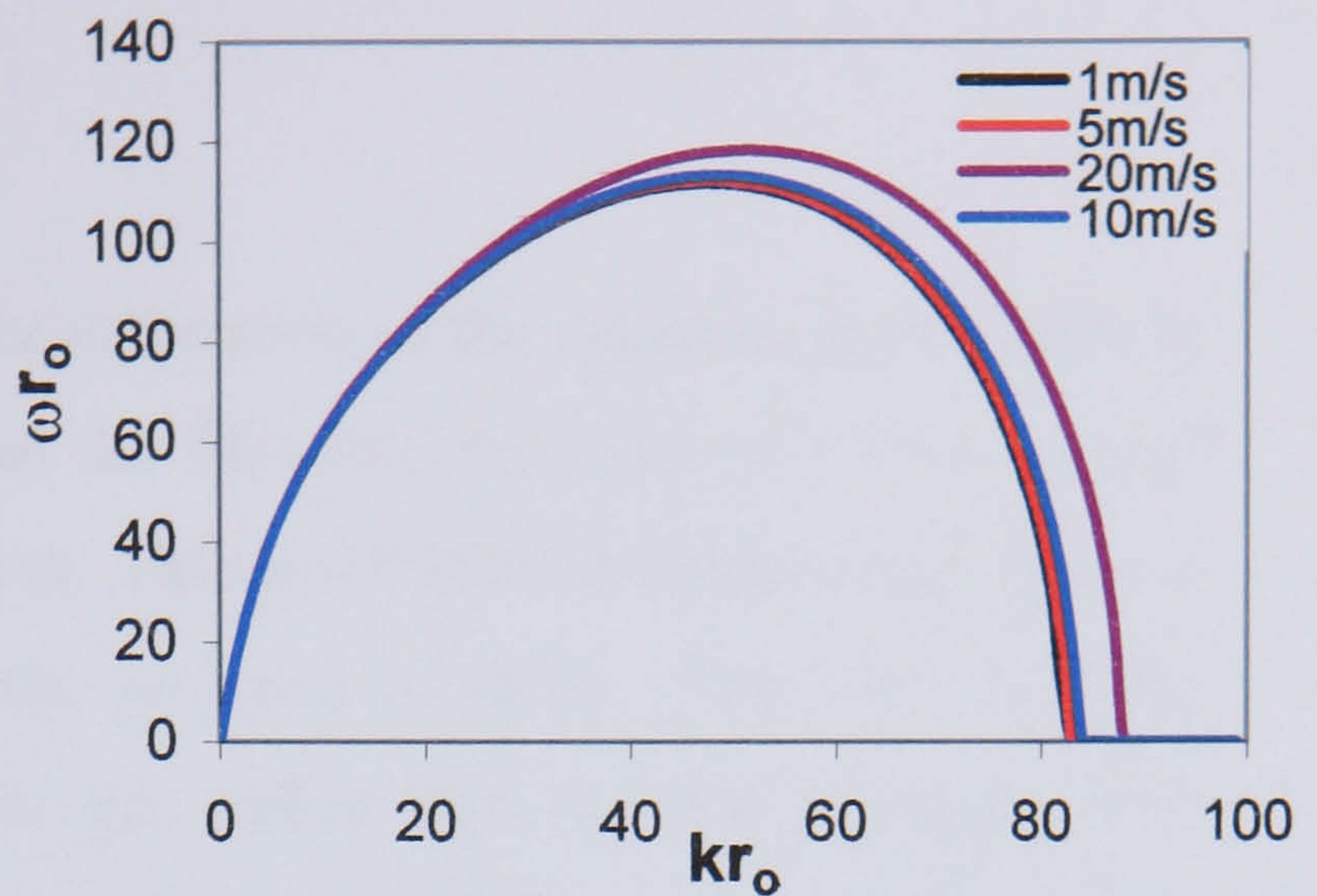
where f takes values of 0.05 - 0.3, depending on the U_{REL} , which is the relative velocity between the liquid film and the surrounding gas, and it is defined as:

$$U_{rel} = [(W_l - W_a)^2 + (V_l - V_a)^2 + (U_l - U_a)^2]^{1/2} \quad (3-27)$$

Figure 3-18b shows the variation of the growth rate with the wave number in the presence of a swirl velocity of 20 m/s. Although the swirl velocity affects significantly the shape of this curve, it is actually the total relative velocity between the liquid and the surrounding gas that determines the shape of this curve. Another point that requires special attention is the maximum possible value of $(kr_o)_{\max}$ that can be numerically used. When this parameter exceeds 700, the estimation of the Bessel functions involved in the various correlations cannot be numerically estimated, since they reach values greater than 10^{256} . Therefore, in the implemented model there is a cut-off value. Its effect is presented in Figure 3-19 where, as can be seen, for cases corresponding to very high injection velocities (200m/s) or for injection against dense gas which would result in similar Weber numbers, estimation of the optimum growth rate is not possible. Physically, this can be justified on the basis that the higher the velocity and the Weber number is the less valid are the assumptions of linear instability theory adopted here are. The model has been tested for injection velocity up to 100 m/s and an air density up to

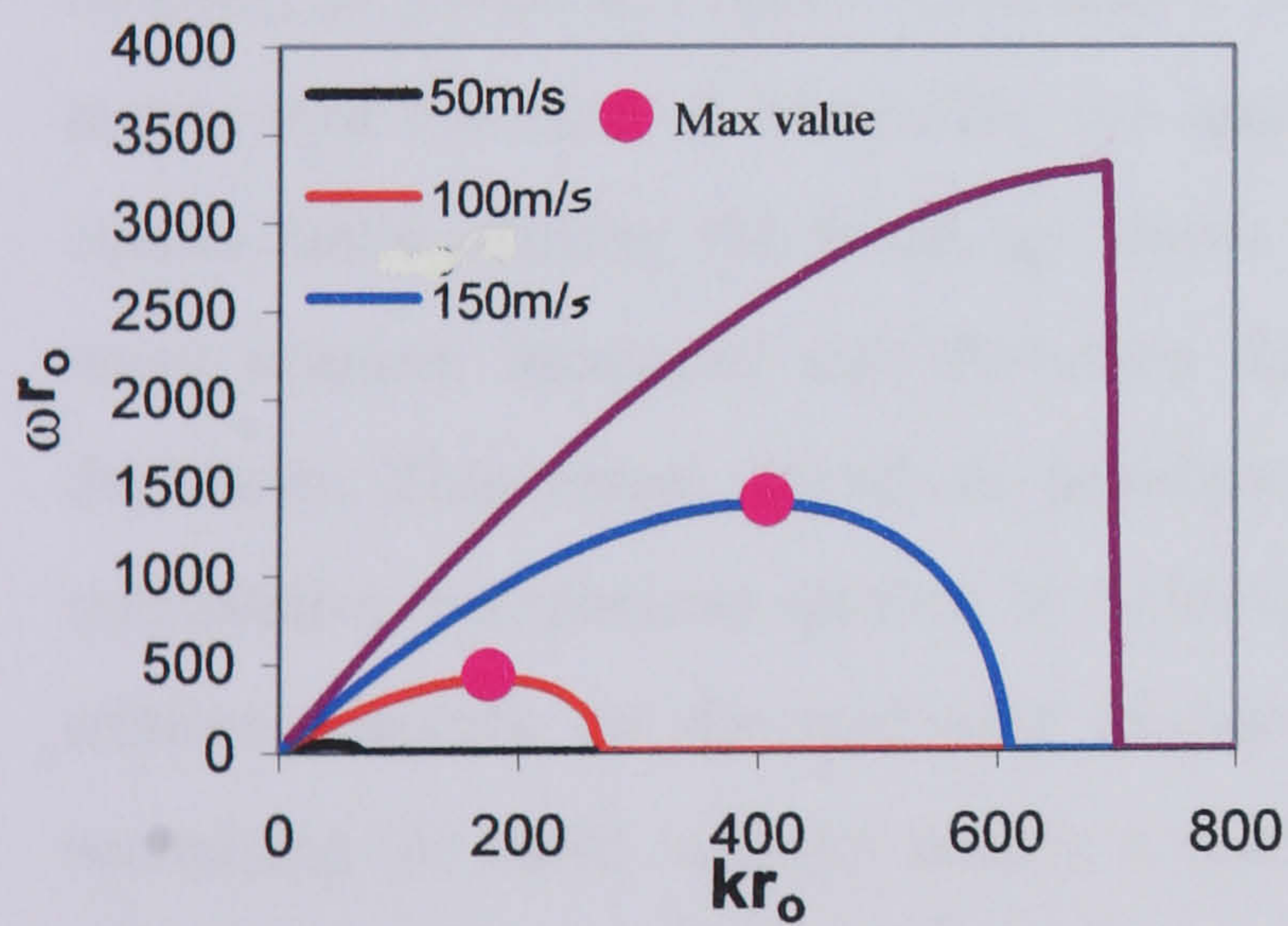


a) Swirl=0 m/s, Film thickness=100 μm

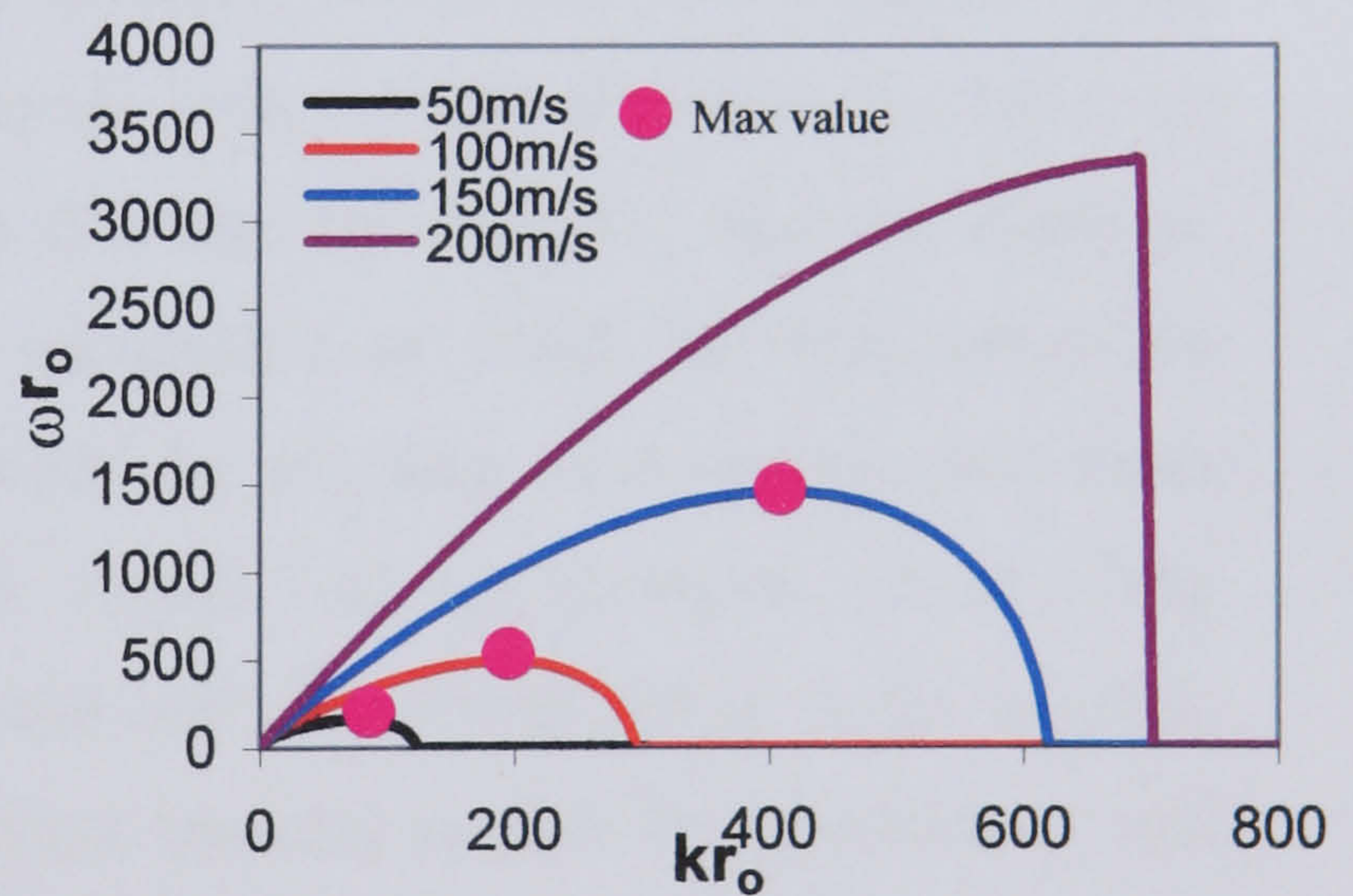


b) Swirl=20 m/s, Film thickness=100 μm

Figure 3-18 Estimation of the growth rate as a function of the (kr_0) parameter for relatively low relative velocities between the injected liquid and the surrounding gas. The optimum growth rate value is denoted with the orange dot as ‘max value’. In addition to the values given on the plots, $r_i=350\mu\text{m}$, $\rho_{\text{air}}=1.2\text{kg/m}^3$, $\rho_{\text{liquid}}=750\text{kg/m}^3$, $\sigma=0.02\text{N/m}$.



a) Swirl=0 m/s, Film thickness=100 μm



b) Swirl=20 m/s, Film thickness=100 μm

Figure 3-19 Estimation of the growth rate as a function of the (kr_0) parameter for relatively high relative velocities between the injected liquid and the surrounding gas. Other flow variables as in Figure 3-18.

8 kg/m³ which are appropriate for modelling the atomisation process of this kind of injectors.

3.3.3.1 Effect of film thickness

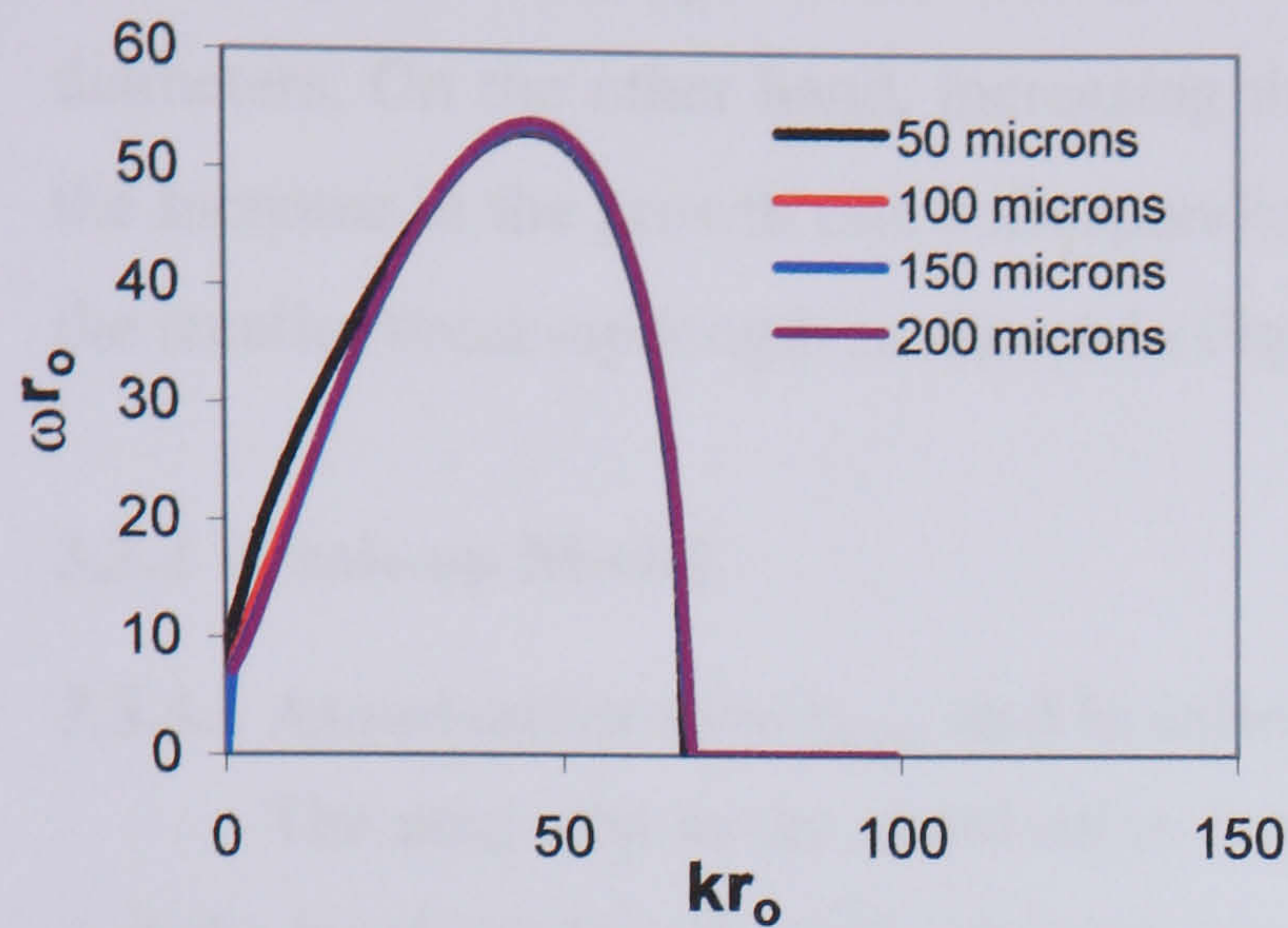
The effect of the film thickness on the estimation of the optimum growth rate is shown in Figure 3-20. This figure shows that the film thickness has only a very small dependency on the predicted growth rate curve. This small effect becomes negligible as the relative velocity increases in and with the presence of swirl. Therefore, it is the relative velocity between the liquid and the gas rather than the film thickness that controls the growth rate value. Nevertheless, the film thickness has a direct influence on the ligament diameter since, the larger size of the film thickness at the break-up point the larger the ligament diameter.

3.3.3.2 Effect of relative velocity

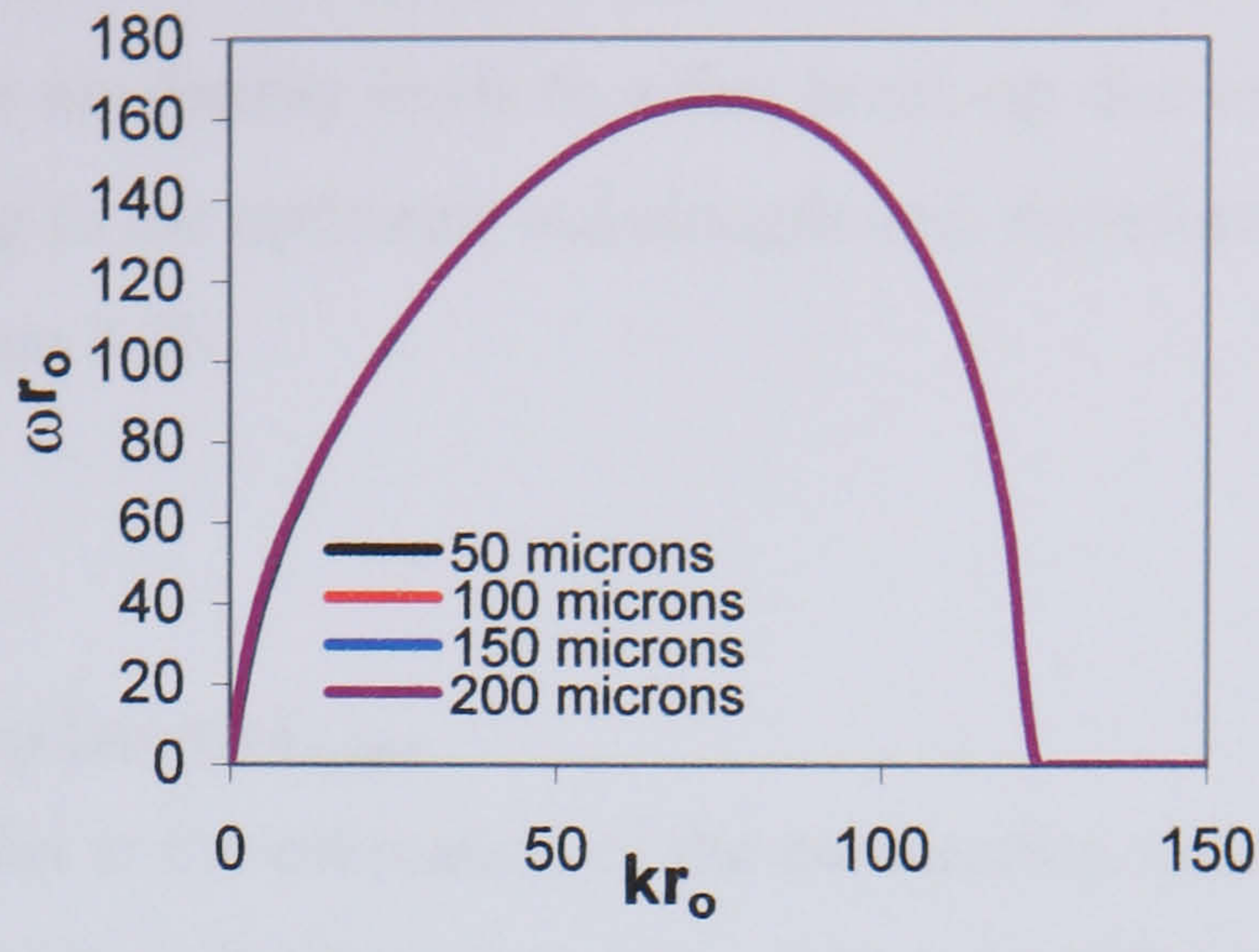
Increasing the relative velocity between the film and the gas causes an increase in the bandwidth of the wavelength of the waves that grow on the film surface, by including shorter waves and that was clear from the increase in the cut off wave number, as shown in Figure 3-21a and b. The optimum growth rate also increases which means that the rate of increasing the amplitude increases and therefore the break-up occurs faster causing the break-up length to dramatically decrease. Also the optimum wave number increases and therefore the wavelength at which the film breaks-up decreases. This result agrees in principle with the physical observations, but direct quantitative comparison cannot be achieved. Figure 3-21 c,d shows the effect of the relative velocity on the optimum growth rate and its corresponding wave number. Increasing the swirl velocity causes a significant increase in both the optimum growth rate and the wave number. This causes a decrease in the break-up wavelength as shown in Figure 3-21d; this significant reduction in the wavelength becomes smaller as the axial velocity increases.

3.3.3.3 Effect of air density

Air density increases the aerodynamic force applied on the film surface. The effect of air density on the growth rate and wave number is shown in Figure 3-22; this figure shows that the optimum wave number and the cut-off frequency increase with

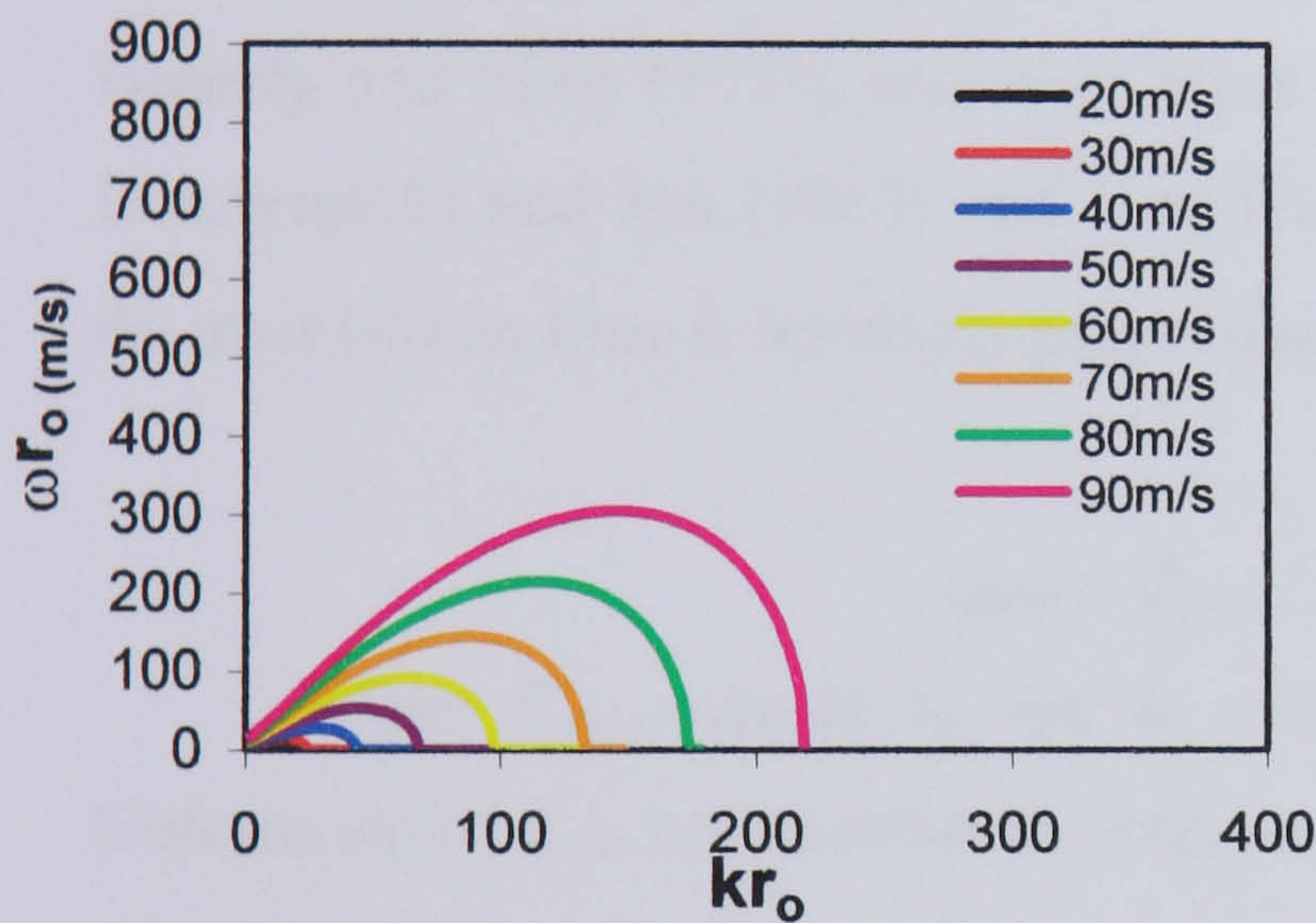


a) Axial = 50m/s , Swirl = 0 m/s

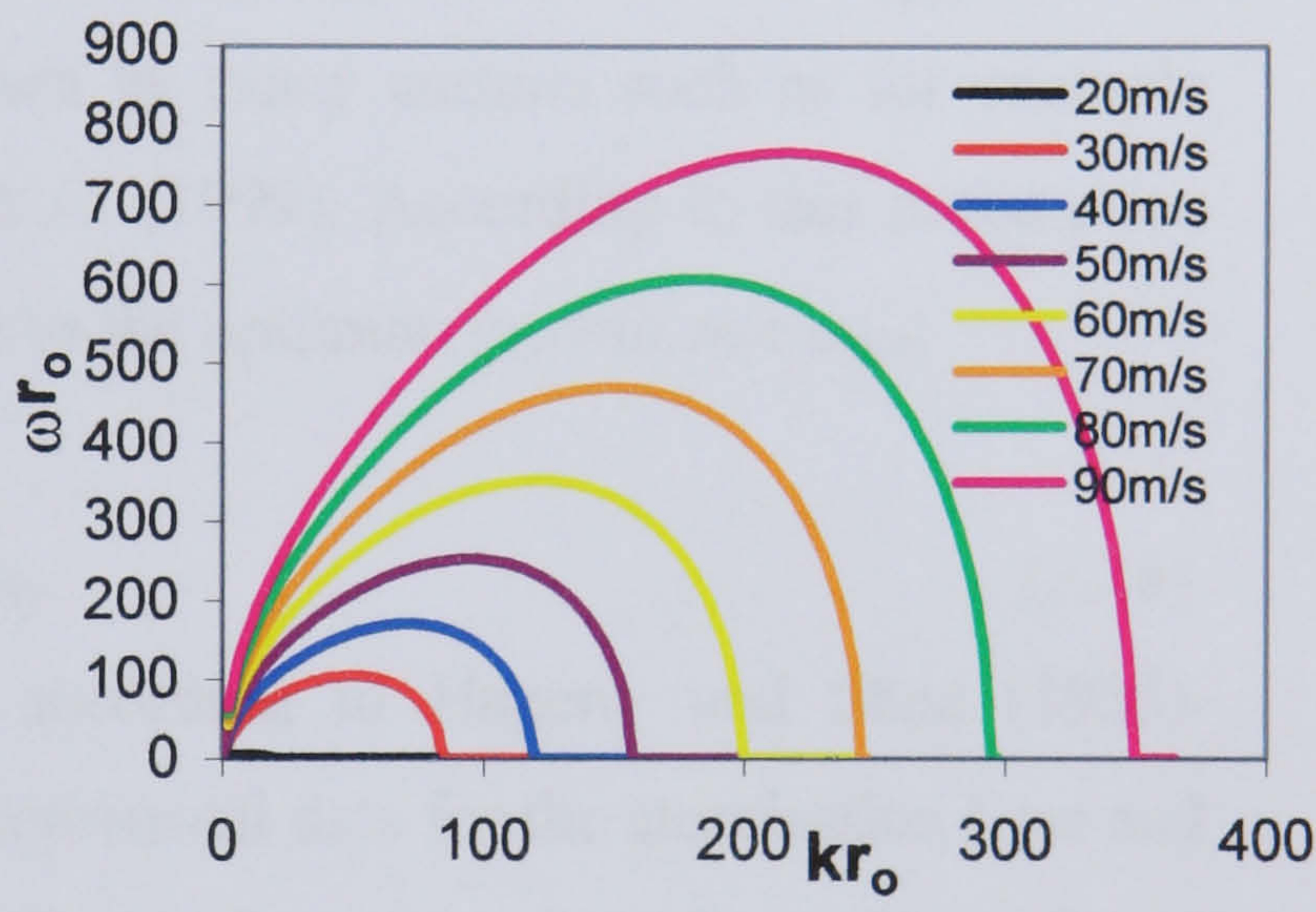


a) Axial = 50m/s , Swirl = 20 m/s

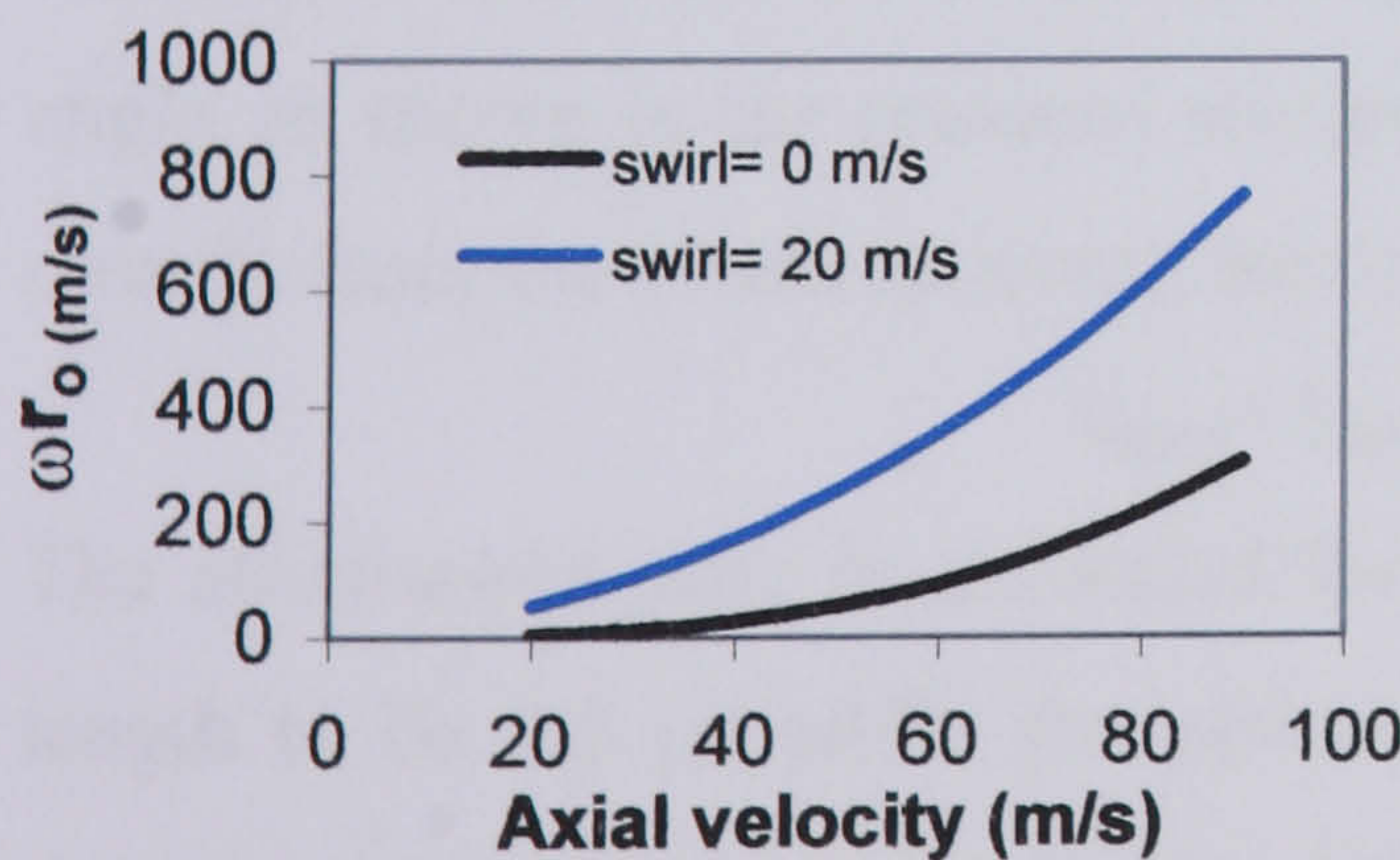
Figure 3-20 Effect of film thickness on growth rate. Other flow variables as in Figure 3-18



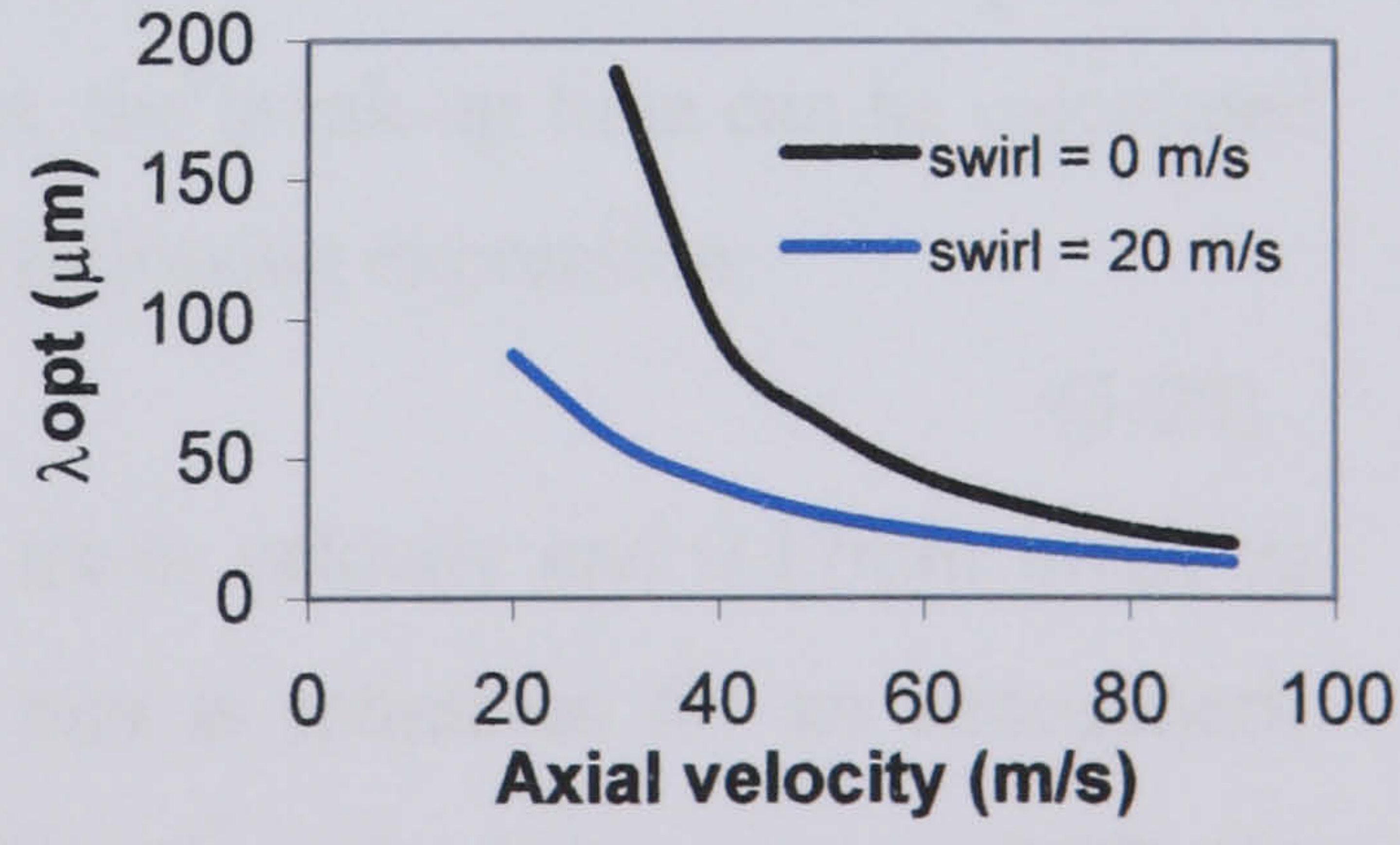
a)Flow angle or swirl =0



b) Flow angle = 30 degree



c) optimum growth rate



d) optimum wavelength

Figure 3-21 Effect of relative velocity on growth rate and its optimum value and also on wave number and its optimum value. Other flow variables as in Figure 3-18

increasing air density which results in shorter wavelength and smaller ligament diameters. On the other hand, increasing the air density leads to a fast break-up due to the increase in the growth rate corresponding to the optimum wavelength and, therefore the smaller break-up length as shown in Figure 3-23.

3.3.4 Break-up Model

3.3.4.1 Atomisation time t_{atom} and break-up length L_{atom}

The next step in the atomisation model is the estimation of the atomisation time and the break-up length. Following film injection, it takes some time which is defined as the atomisation time, t_{atom} until the disintegration of the liquid film into droplets occurs. It is generally assumed that the film starts to disintegrate when the amplitude of the waves reaches a certain value relative to its initial disturbance which is normally taken to be equal to $\exp(12)$ independent of operating conditions as suggested by Hagerty and Shea (1955) and used since then by many authors such as for example Dombrowski and Jon (1963) and Jennifer et al. (1999). According to this assumption the atomisation time is inversely proportional to the optimum growth rate ω_{opt} :

$$t_{atom} = C_{time} / \omega_{opt} \quad (3.28)$$

where C_{time} equals to the $\ln(12)$ according to Hagerty and Shea (1955). Unfortunately, it is not possible to obtain experimental data for the atomisation time and therefore deduce the C_{time} value. In this study the C_{time} value is estimated in order to match experiments and model predictions in an indirect way by comparing the calculated results for the break-up length with the experimental data for the spray cone angle, as shown in the previous section. Therefore, the break-up time can be calculated directly from the break-up length according to the following expression:

$$t_{atom} = L_{atom} / U_{inj} \quad (3-29)$$

The atomisation time is estimated for a 67 m/s mean velocity and 0.17mm break-up length to be 2.5 μs while the optimum growth rate is estimated for an atmospheric density of 1.2 kg/m³ to be 0.96 (1/ μs). By substituting the value into equation (3.28), the C_{time} is found to be $\ln(11.47)$ which is very close to the original value proposed by Hagerty et al. (1955). If the original value $\ln(12)$ is used instead then the atomisation time becomes 2.59 μs which gives a break-up length of 0.173. This constant has been identified based on atmospheric air density but if it is assumed that the value of that

constant does not change with air density, then calculations of the break-up length could be extended to different air densities. Figure 3-23 shows these results where the break up length decreases as the air density increases. Although there is no experimental evidence for the conical spray regarding this trend of the break-up length, it has been considered in many models and also in empirical equations proposed by many researchers such as Elkotb (1982) and Lefebvre (1989). According to the linear instability analysis there is a very slight change in the growth rate when the injection velocity reduced from 70 to 30 m/s corresponding to 70 to 20 bar injection pressure, respectively. However; the break-up length slightly changes with injection pressure within that range. Similar results have been obtained from the liquid film divergent model, when it was commented that the divergence of the film does not change much as well.

3.3.4.2 Mean droplet diameter

The final step in the modelling of the atomisation process is the estimation of the volume mean diameter d_{30} from the disintegration of the liquid film. As already mentioned, the model employed here assumes that at the atomisation time the liquid film disintegrates into an annular ligament ring which, then, forms droplets. The time required for the process of ligament break-up into droplets to take place is neglected.

By considering the approach of Dombrowski et al. (1962) and the non-linear analysis established for a flat sheet established by Rangel and Sirignano (1962), the antisymmetric wave disintegrates at each half-wave length $\lambda_{opt}/2$, where the wave continues growing until the crests are blown out. Accordingly, the diameter of the annular ring could be estimated by equating the volume of this ring to that of the liquid film over a length equal to half the optimum wavelength and for thickness equal to the film thickness (thick) at the instant of its disintegration, as shown in Figure 3-24:

$$D_{ligament} = \sqrt{4 \cdot thick \cdot \lambda_{OPT} / \pi} \quad (3-30)$$

The thickness of the film at the location of film disintegration can be identified theoretically from the break-up length and the continuity equation. However this assumption does not include droplet separation from the film surface before break-up. If droplets separate from the film surface before film disintegration as observed in the images of the previous chapter, then the film thickness has to be less than that theoretically calculated. In order to simplify this problem, an under relaxation-constant

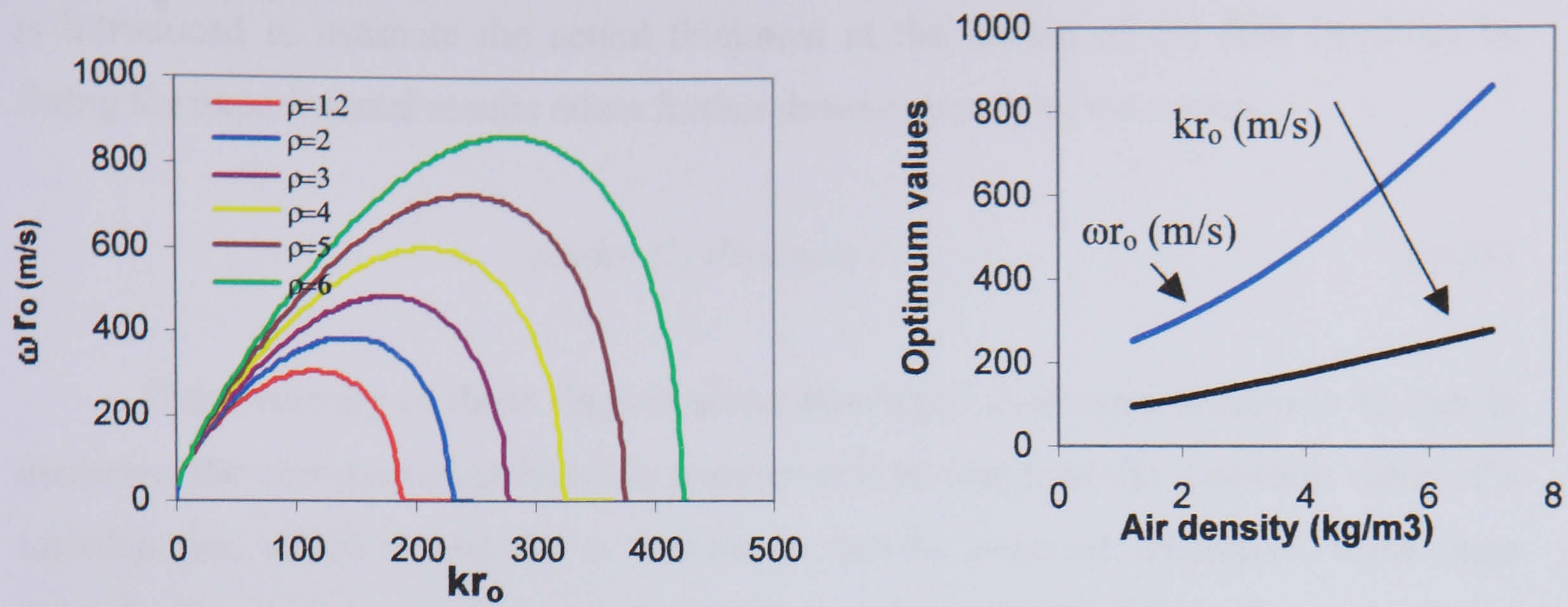


Figure 3-22 Effect of air density on growth rate and its optimum value and also on wave number and its optimum value. Other flow variables as in Figure 3-18

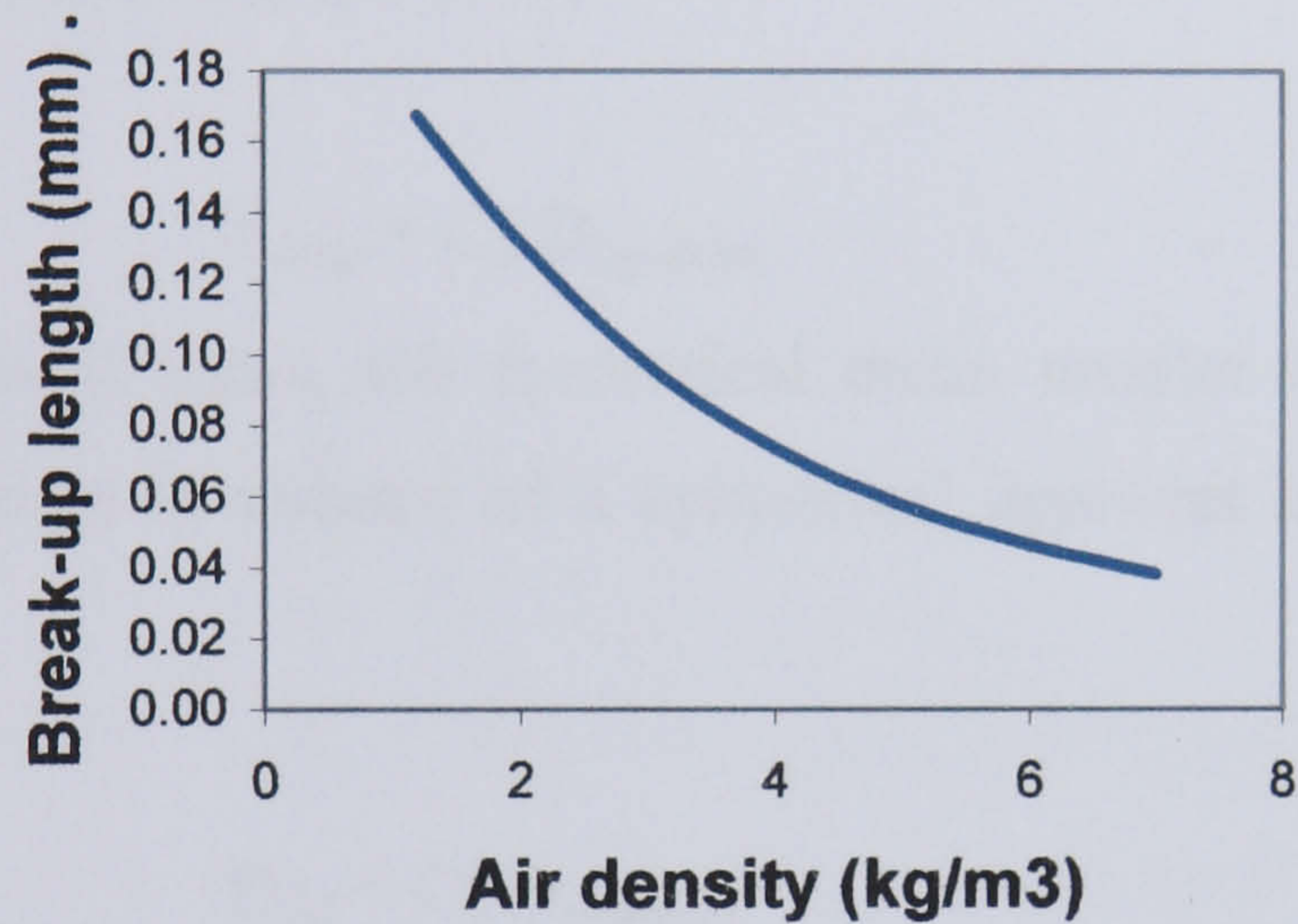


Figure 3-23 Effect of air density on break-up length. Other flow variables as in Figure 3-18

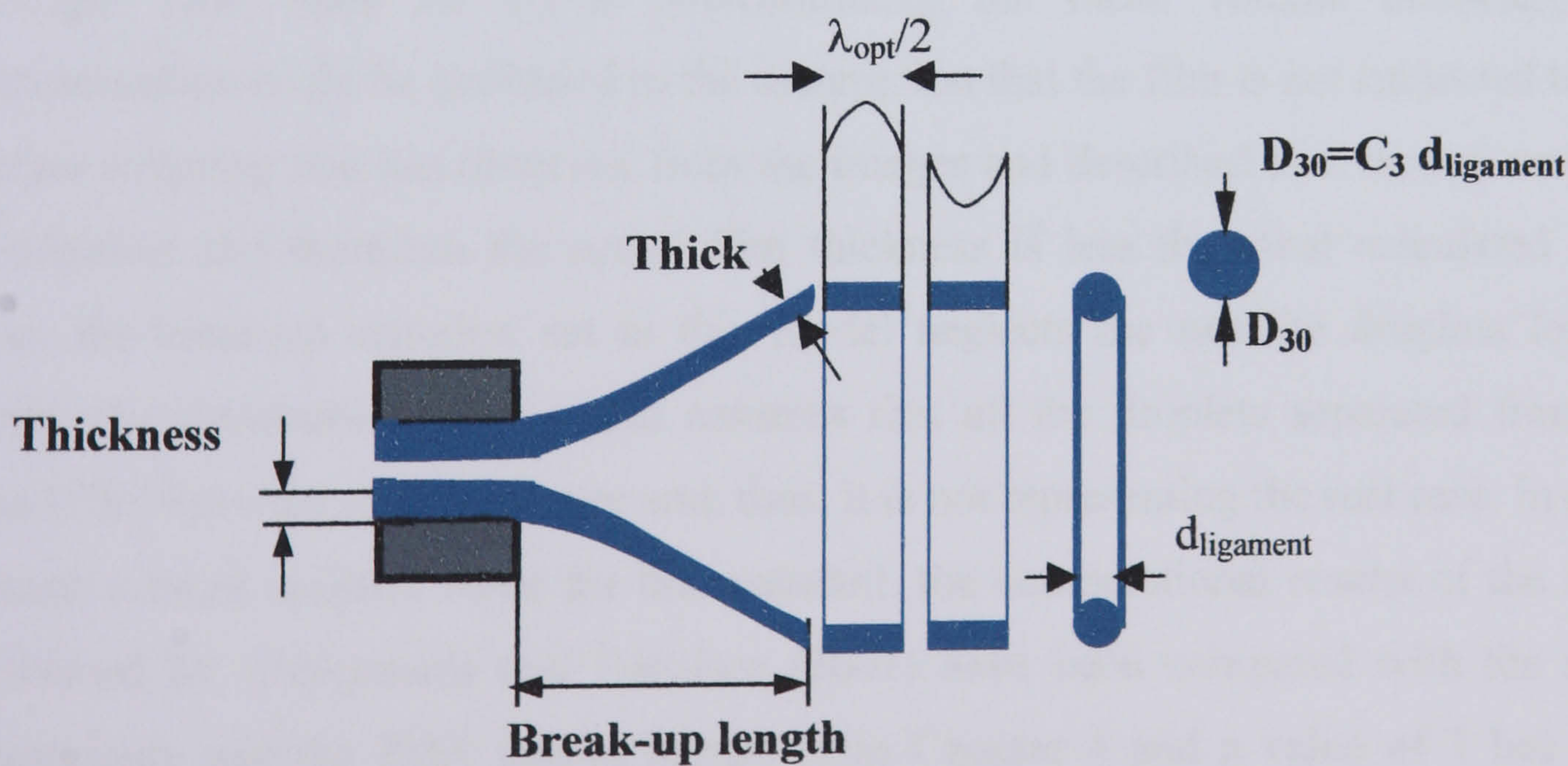


Figure 3-24 A schematic showing the break-up of the conical film into ligaments and droplets.

is introduced to estimate the actual thickness at the instant of the film break-up by fitting the experimental results taken further downstream from the nozzle.

$$thick = C_1 \text{ thickness} \quad (3-31)$$

If the velocity of these rings is about 40m/s and their cross section is 15 mm in diameter, the corresponding Reynolds number will be less than the transition value of a turbulent jet, which means that a laminar jet can be assumed. Therefore, these rings move in the gas flow and disintegrate by the air action or by liquid turbulence according to Weber's results for the surface tension break down. The ligaments fall apart with the most dominant wavelength λ_{drop} given by:

$$\lambda_{\text{drop}} = C_2 D_{\text{ligament}} \quad (3-32)$$

From the conservation of mass, the theoretical mean droplet diameter D_{30} which is resulting from the change in volume of a cylindrical ligament into a spherical droplet leads to:

$$D_{30} = C_3 \cdot D_{\text{ligament}} \quad (3-33)$$

where C_2 , according to Rayleigh's laminar theory (1892), is equal to 4.5 leading to $C_3=1.88$. This value for C_3 is overestimating the mean volume diameter; this overestimation could be attributed to the assumption that the film is not subjected to any surface stripping that has observed from the images and described as a second mode of atomisation and therefore the actual film thickness is less than that calculated here. Also, the break-up criterion set in this model neglects the satellite droplets formed during the atomisation process and assumes that all the droplets separated from the liquid film have the same diameter and, thus, it is not representing the real case. In order to have a more realistic value for this constant, the computational results of the spray performed by Arcoumanis and Gavaises (2001) have been compared with the spray experiments and the PDA results presented in Chapter 4 and a value of 1 has been found to be appropriate for the constant C_3 in order to achieve a good prediction for the spray.

Based on this constant, the effect of different parameters on volume mean diameter after the film breaks-up could be investigated, as shown in Figures 3-25 to 3-28. These figures show that the droplet mean diameter decreases as the axial velocity increases. This trend agrees with the expectation that the atomisation enhances when the injection pressure increases and with the PDA measurements of the spray. In Figure 3-25, the volume mean diameter is shown to be decreasing with the air density. A similar trend has been used for the Sauter mean diameter by Ren and Nally (1998) in their DISI spray model from swirl injectors and leads to a good agreement with the spray experiments under different ambient pressures. It also agrees with the relationship proposed by Abou-Ellail et al. (1978) and Lefebvre (1989) for different geometrical configurations of swirl atomisers. In spite of this agreement, the spray PDA measurements showed the opposite trend for the droplet mean diameters with the air density. However the spray PDA results cannot be compared directly with the results from the atomisation model since the measurement locations are relatively far downstream from the location where the atomisation process occurs. Although the film thickness has almost no effect on the waves developed on the film surface as been discussed before, Figure 3-26 shows its effect on mean volume diameter; the increase in the mean volume diameter is attributed to the involvement of the film thickness in calculating the ligaments diameter as described in equation (3-30). The surface tension of the liquid film damps the waves and causes an increase in the optimum wavelength developed on film surface and leads to the production of larger droplet mean diameter, as shown in Figure 3-27. The same trend has been found in all the relationship reported by Lefebvre (1989) involving the surface tension in calculating the mean diameter of pressure swirl atomisers. The effect of swirl velocity manifested in the flow angle whose tangent is defined by the ratio between the swirl and axial velocity is shown in Figure 3-28. Increasing the flow angle increases the film swirl velocity and therefore a higher relative velocity between the liquid film and the air exists and accordingly smaller optimum wavelengths and smaller diameters are obtained.

Although the effect of different parameters on mean volume diameter of the droplets separated from the liquid film agrees with the trends previously published for swirl atomisers, there is no direct way to compare quantitatively the results from this model with the experiments. This is mainly due to the limiting space and the high density of droplets at the location of the film break-up, which hinder conducting experiments to measure the droplet diameters. Until now, the only way to validate any

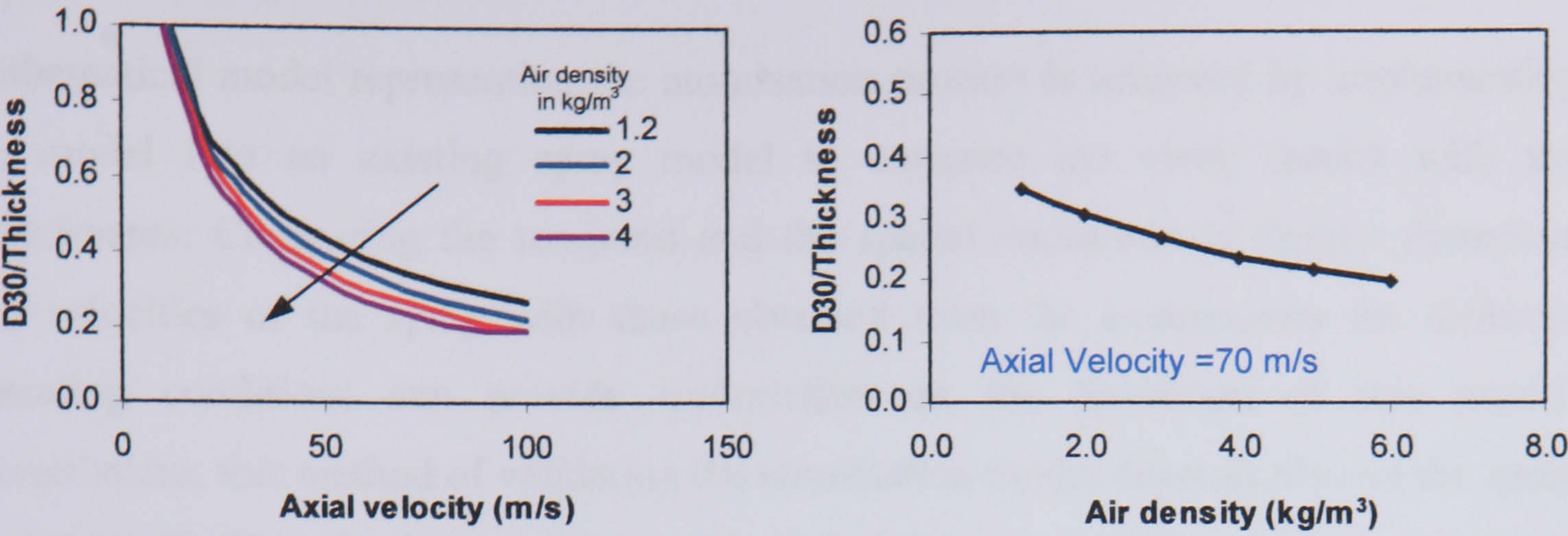


Figure 3-25 Effect of air density on volume mean droplet diameter D_{30} assuming $\rho_f=750 \text{ kg/m}^3$, $\sigma = 0.02$, flow angle = 30 deg., film thickness = 100 μm and nozzle radius = 0.45 mm

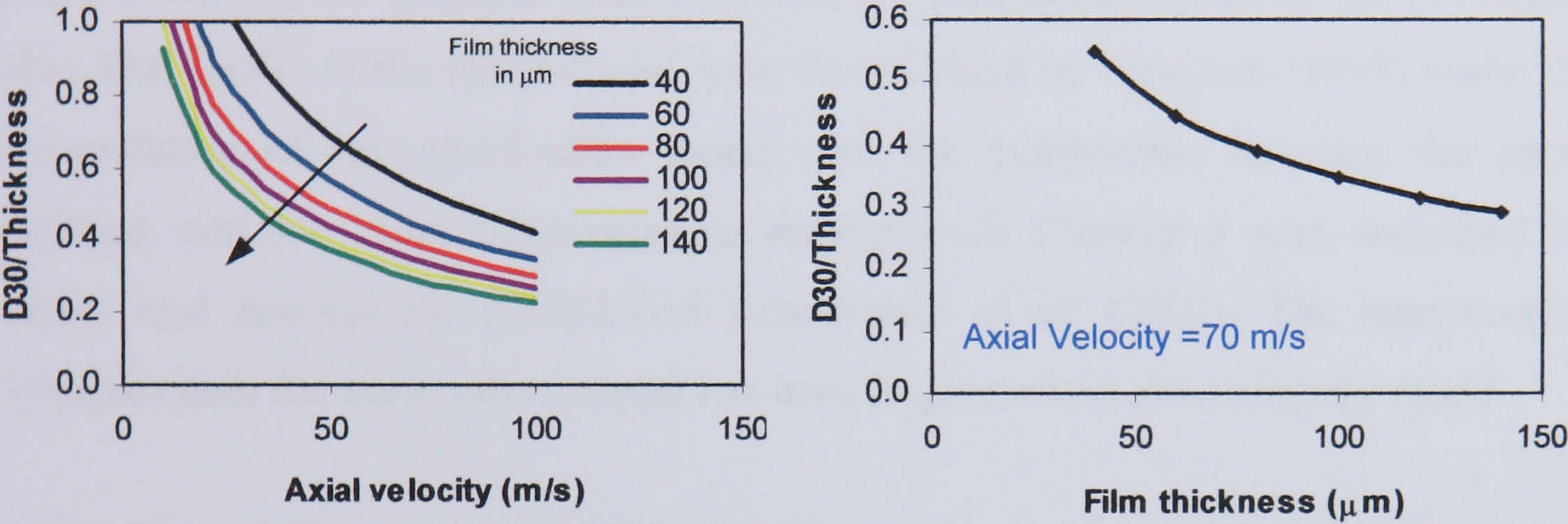


Figure 3-26 Effect of liquid film thickness on volume mean droplet diameter D_{30} for air density = 1.2. Other flow parameters as in Figure 3-24

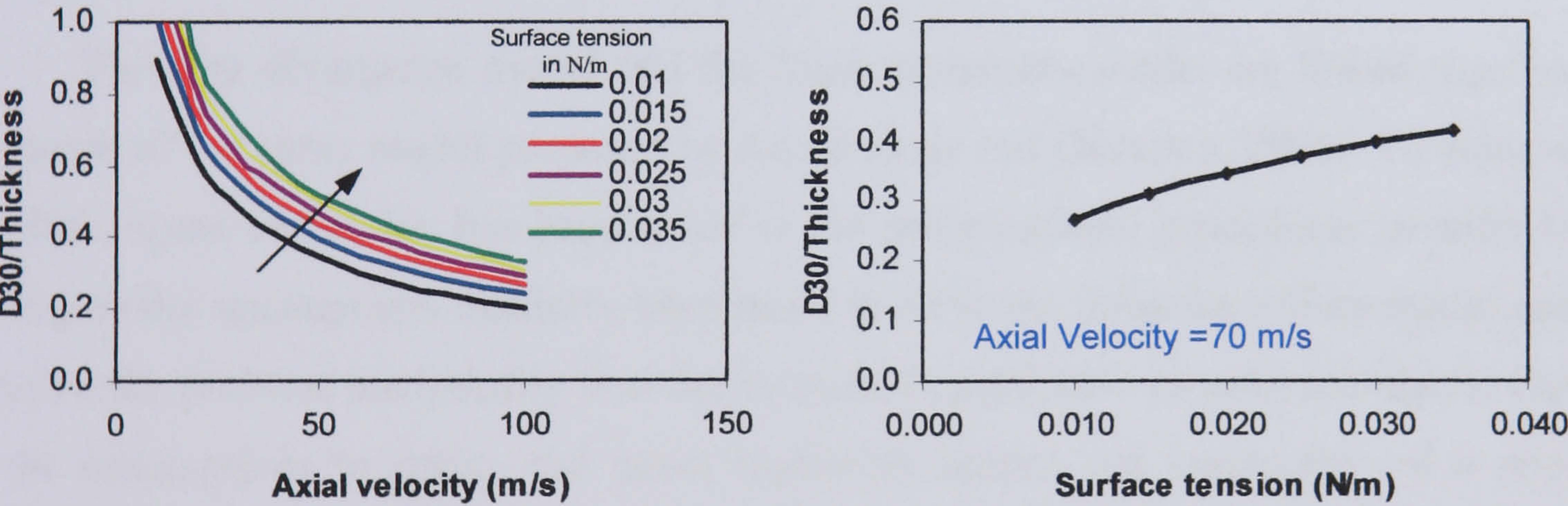


Figure 3-27 Effect of liquid surface tension on volume mean droplet diameter D_{30} for air density = 1.2. Other flow parameters as in Figure 3-24

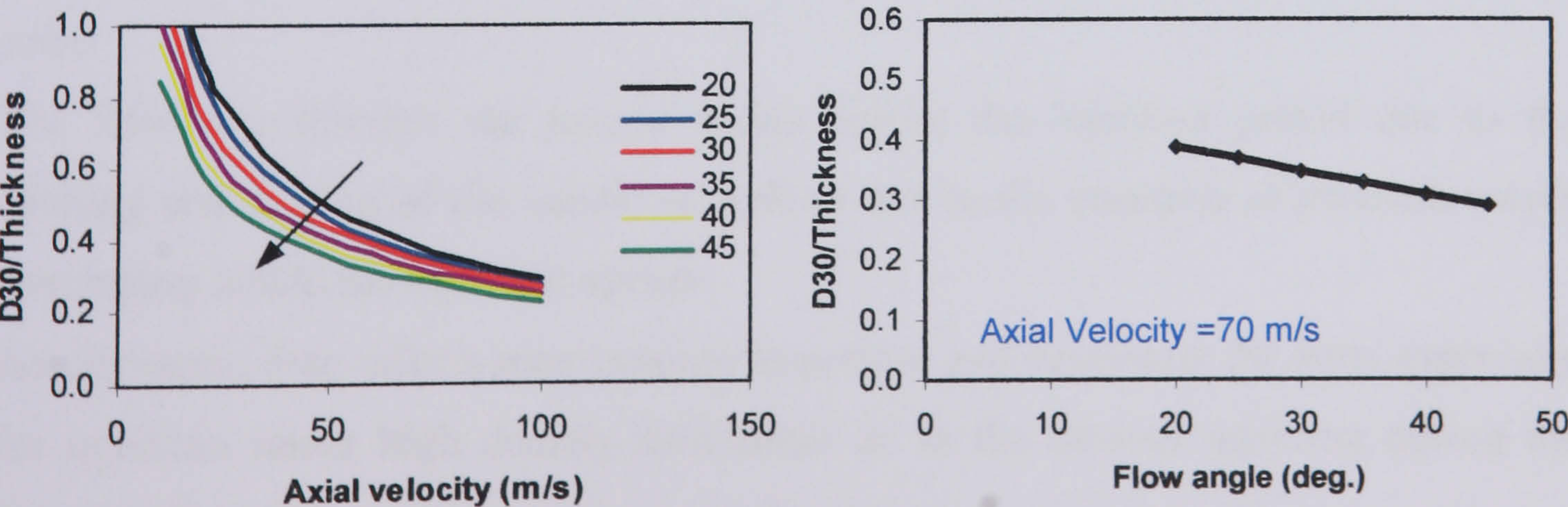


Figure 3-28 Effect of nozzle flow angle on volume mean droplet diameter D_{30} for air density = 1.2. Other flow parameters as in Figure 3-24

mathematical model representing the atomisation process is achieved by implementing the model into an existing spray model to compare the spray results with the experiments. Comparing the temporal and the spatial variations of droplet diameters and velocities of the spray with those obtained from the experiments for different operating conditions can provide information on the limitation of this model. Nevertheless, this method of validating the atomisation model depends also on the spray model itself. The physical phenomena included in this model and the accuracy in predicting the interaction between the droplets and the surrounding air and the mutual interaction among the droplets themselves are all parameters affecting the prediction results. The details of the spray model were documented by Gavaises (1997) while the implementation of the atomisation model and the comparison between the spray predictions and the PDA measurements presented in Chapter 4 was described by Gavaises and Arcoumanis (2001) and Arcoumanis et al. (2002). The next section summarises how the atomisation model has been implemented into the spray model.

3.3.5 Implementation of the Film Divergence and Linear Instability Models into the Spray Model

The film divergence model and the linear instability model are linked together by means of the spray model proposed by Arcoumanis and Gavaises (2000). To achieve this link, some adaptation has been made in the computational procedures in order to overcome the assumptions that have been made to drive the linear instability model and to solve the problem analytically in order to make it applicable to swirl atomizers. Out of the assumptions to derive the linear instability model, the assumption of a one-dimensional flow with constant film thickness and constant velocity is not directly applicable to liquid films injected from pressure swirl atomisers due to the following reasons:

- a. The flow rate through the nozzle varies during the injection period due to the opening and closing of the needle as well as due to the presence of pressure waves developing inside the injection system.
- b. Aerodynamic drag effects may become important and decelerate the film, especially for injection under high density conditions, as in the case of injection during the compression stroke.

- c. After the liquid film exits from the injection hole, it expands along a conical surface due to the action of the centrifugal forces resulting from its swirling motion inside the injection hole. Therefore, the film thickness does vary as the liquid film moves downstream of the injection hole.
- d. Since the film is moving in an expanded conical flow path, both its thickness and its swirl velocity will decrease.

To bypass the first of the above discrepancies between the model assumptions and the actual flow conditions, Gavaises and Arcoumanis (2001) assumed that during the numerical solution the annular liquid film is represented by a discrete number of computational parcels which are moving along the direction of their injection without interacting between each other. Then, for each parcel, the above assumptions are employed as if each of them is following the same flow path as that of the steadily moving annular liquid sheet.

For points b,c and d, it is assumed that at each numerical time step from the time of injection of each parcel until the time of their disintegration, the thickness, the axial and the swirl velocity of the parcel required as inputs to the atomisation model, are those resulting by employing the following assumptions:

- 1- The film thickness is determined by employing the continuity equation; having calculated the radial displacement of the film, the annular cross-sectional area of the liquid sheet at any distance from the injection hole can be determined and, therefore, the instantaneous film thickness can be estimated.
- 2- The axial velocity of the film at every time step is estimated by taking into account the aerodynamic drag forces. For simplification, the aerodynamic drag coefficient is assumed to be proportional to that of a spherical droplet with its radius equal to the instantaneous liquid film.
- 3- The swirl velocity of the film decays as the parcels are moving downstream along a path of increasing distance from the nozzle hole axis of symmetry.

The above assumptions are justified on the basis of the very short atomisation time scales (of the order of a few μs), which practically lead to atomisation soon after the exit of the liquid film from the injection hole. However, their effect on the mean

droplet size as predicted from the atomisation process is numerically estimated. Neglecting viscosity in the derivation of the characteristic equation, in order to simplify the problem, does not seem to introduce much effect on the accuracy of the model. For the application of high pressure swirl atomizer, the inertia force of the liquid is relatively much higher than the internal viscous force of the liquid sheet. Dombrowski and Taher (1977) have shown experimentally using 42 commercial swirl injectors that the viscosity has almost no effect on the droplet size for the same injection velocity. Therefore, viscosity has a major contribution only inside the injector through its effect on the film velocity at the nozzle exit.

In order to start the spray calculation, Gavaises and Arcoumanis (2001) assumed initial distribution of the droplet diameters as a result of the atomisation process. This probability density function that represent the initial size distribution was estimated by applying the approach of the maximum entropy formalism. The data required for applying this approach includes D_{30} and the maximum diameter. The atomisation model predicts the value of D_{30} while the maximum droplet diameter was considered equal to the film thickness.

The following procedure was used by Gavaises et al. (2002) to calculate the mean droplet diameter D_{30} from the linear instability analysis:

- The characteristic equation is solved and the root of the maximum imaginary part, which is the growth rate of the instability wave, is determined for different values of the wave number (k).
- The value of k corresponding to the maximum growth rate is then identified and consequently, the wavelength.
- An empirical formula is used to determine the break-up length; this formula can be used with the spray angle to determine the sheet thickness at the break-up point, using the mass continuity equation.
- The mean diameter of the ligaments is determined assuming that the cylindrical sheet disintegrates at each half wavelength.
- Rayleigh's theory is used to determine the mean droplet diameter of the spray considering the ligament's diameter calculated previously.

3.4 CONCLUSIONS

- A total differential equation representing the liquid film divergence near the nozzle exit before its disintegration under steady conditions has been derived by balancing the forces acting perpendicularly on liquid film as a function of the nozzle flow angle and velocity, the nozzle diameter and the fuel properties as well as the pressure difference across the film.
- After solving the equation governing the film divergence a parametric study has been performed that showed the following conclusions:
 - The injection velocity has very small effect on the film divergence within the practical range of DISI injectors. It also showed that the film could be established at very low injection velocity (3m/s) and will have nearly the same divergence angle if the injection velocity is larger than 8 m/s. Thus, the reason of not having a conical shape during the early stage of spray development (pre-spray) is not due to the low injection velocity but because of the flow dynamic that leads to the prevention of the film formation inside the fuel nozzle.
 - Increasing the flow angle results in an increase in the film divergence angle whose asymptotic value is always higher than the flow angle. It is also shown that as the flow angle increases the film divergent angle reaches its asymptotic value faster.
 - Any small change between the air located outside the conical film and inside causes significant change in the film shape downstream of the nozzle from conical shape to tulip shape while keeping the film divergent asymptotic value the same.
 - The variation in fuel density and surface tension has negligible effect on film divergence within the practical range of DISI injection velocity
- The equation of motion of a single droplet after the film disintegration has been derived which showed that additional radial divergence has to be counted before the spray takes its final spray divergence. Thus, the nozzle spray angle has been identified as the sum of both the film divergence angle at its break-up location and the asymptotic value of the single droplet divergent angle.
- It has been concluded that the nozzle spray angle is only equal to the nozzle flow angle when the fuel is fully atomised at the nozzle exit.

- The location of the film break-up has been identified by matching the measured nozzle spray angle with the calculated one for different break-up lengths.
- From the film and droplet divergent analyses, the relationship between the flow angle, film divergent angle and nozzle spray angle has been identified.
- The break-up length has been calculated by matching the nozzle spray with the asymptotic divergent angle of the single droplet movement after its separation from the film at that specific break-up length.
- A model has been developed to represent the atomisation process from pressure swirl atomisers based on the linear instability analysis that predicts the volume mean droplet diameter as a result of film break-up.
- A parametric study has been performed that showed good agreement for the trend of the predicted volume mean diameter with the previously published data.
- The model has been adjusted and tested after its implementation in a spray model to predict the spray structure form in a pressure swirl atomiser and showed that it provides a good link between the liquid film model and the spray model.

CHAPTER 4

SPRAY CHARACTERISATION

4.1 INTRODUCTION

The fuel spray characteristics are of a significant importance to the direct-injection gasoline combustion systems. Parameters such as the spray cone angle, spray penetration, and fuel flow rate are known to be critical, and the optimum matching of these parameters with the air flow field, piston bowl geometry and spark plug location form the essence of a DISI combustion system. While during the early injection mode (full load), a well dispersed-spray is desirable to ensure a homogeneous charge, a stable and compact spray consisting of small droplets, during late injection (part load), is necessary to contain all the fuel inside the piston cavity. This makes the convection of the fuel by tumble or swirl towards the spark plug easier and repeatable. In addition, it ensures rapid fuel evaporation and adequate mixing with air within the air/fuel ratio window and that is crucially required for stratified combustion. Although some studies (for example Kume et al., 1996; Pontoppidan et al., 1999) have shown the adequacy of the swirl pressure atomizer to satisfy these apparently conflicting requirements, there are on-going investigations regarding the optimum droplet size and velocity distributions which would allow improved fuel atomisation, mixing and subsequent combustion processes. At the same time these investigations can provide the required detailed data for validating existing CFD models and allow development of advanced one. Also, the microscopic structure of the spray is as important as its overall global structure; its droplet size and velocity distributions are the key factors controlling spray evaporation. The fact that a very small percentage of large droplets are capable of degrading engine performance and increasing unburned hydrocarbons is evidence of the long time they take to evaporate.

This study is aimed at identifying the spray characteristics of high-pressure swirl injectors operating under different injection and ambient conditions. This could lead to the development and validation of a complete fuel injection equipment (FIE) mathematical model that will include the injection system, liquid sheet atomisation and spray development. In order to achieve this aim, the transient spray generated by the high pressure swirl injector, used in direct-injection gasoline engines was visualised under both atmospheric and high pressure and temperature conditions inside a constant volume chamber using a high magnification and high resolution CCD-based imaging system. Special attention has been paid to the early spray development, that is the time period starting from the instant the fuel exits the nozzle until the instant of achieving a fully developed spray. The temporal variations of the spray penetration and spray angle have been considered in all cases.

In order to provide a more detailed analysis and better understanding of the microscopic structure of the spray, the droplets size and velocity have been measured using the Phase Doppler Anemometer (PDA) technique. The temporal and spatial droplet size as well as the velocity distributions will be described for both atmospheric and 4.5 bar spray chamber pressure at two gas temperatures (293K and 500K). Results are presented at each measured point and also as an average of the points measured across an axial plane and over the whole spray; this allows an overall picture to be obtained about the effect of the operating conditions on the whole spray. The results of this chapter have been used to validate the spray model developed by Gavaises and Arcoumanis (2001).

4.2 EXPERIMENTAL SYSTEM

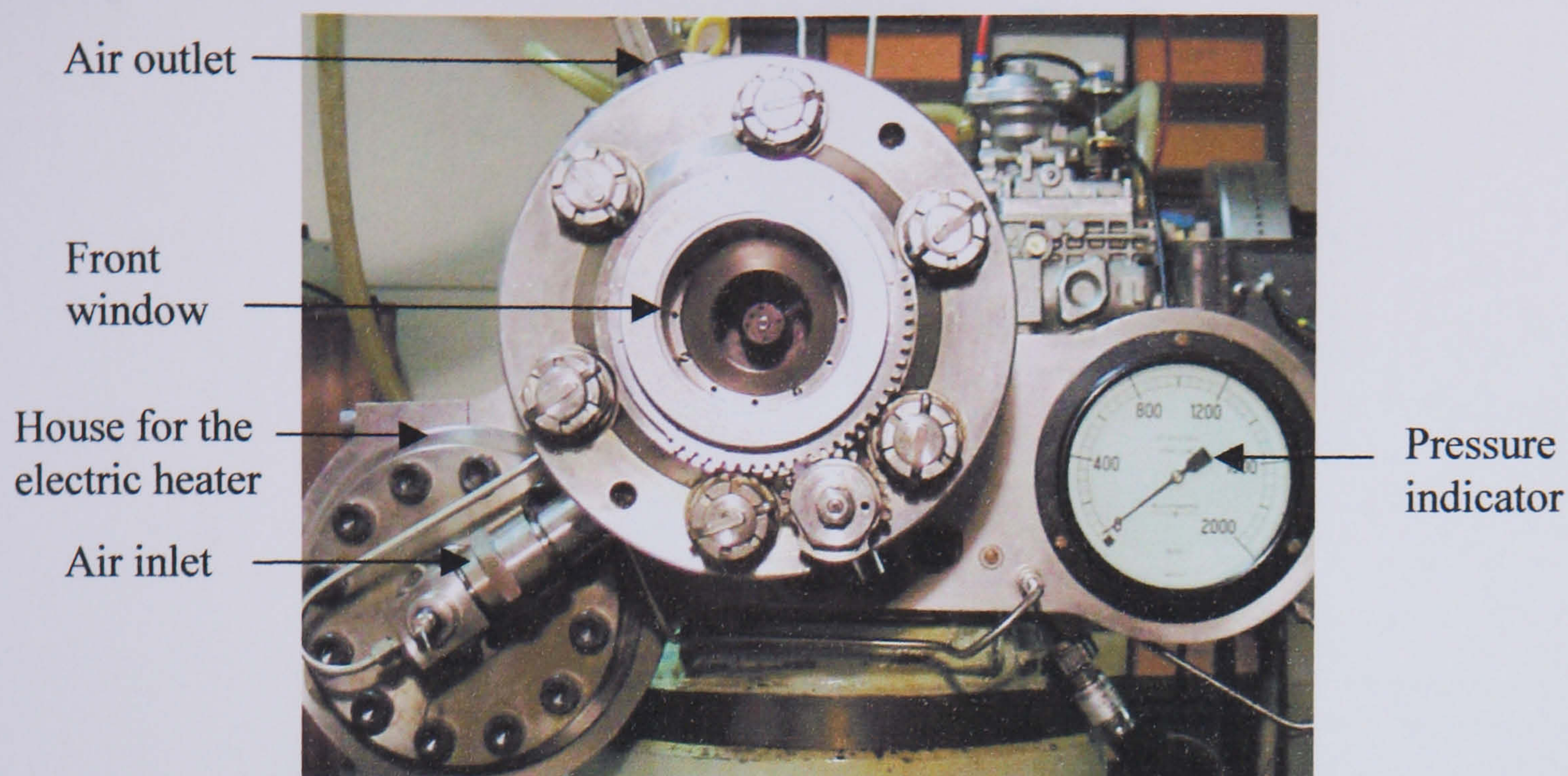
The fuel injection system and the injector used in this study have been described earlier in Chapter 2. In addition, the imaging system is approximately the same, except for the zoom lenses which are used for high magnification to image the flow and film inside the transparent nozzle. These lenses have been replaced by a Nikon 75-300 zoom lens in this chapter to visualise the whole spray. The main parts of the system, not mentioned in Chapter 2, which include the constant volume chamber and the PDA system, will be described below.

4.2.1 The High-Pressure Chamber

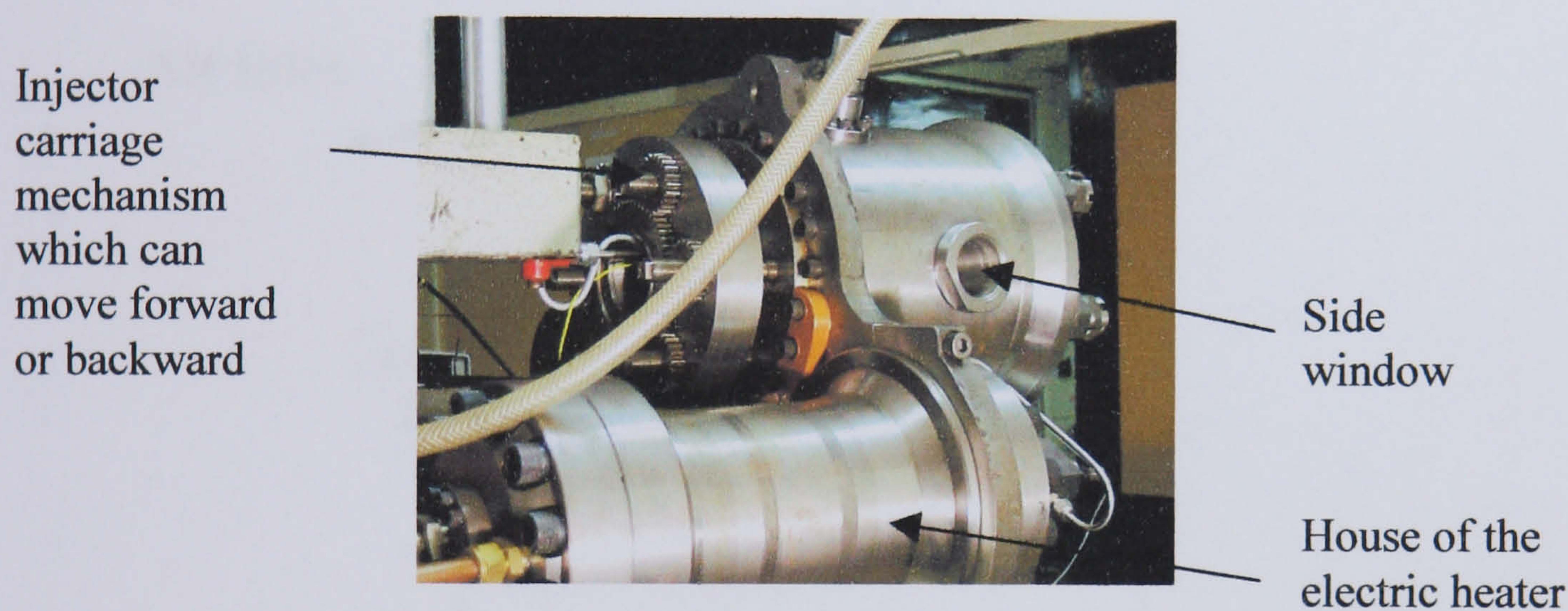
A constant-volume, high-pressure chamber has been designed and built, including the associated mechanical and electrical accessories, by McGlashan, (1997). The chamber is designed to withstand gas pressures and temperatures up to 100 bar and 800 K, respectively. There are four windows in the main body, a front window and three side windows, made of quartz on the chamber cylindrical wall for applying phase Doppler anemometry or any other optical technique, as shown in Figure 4-1. Pressurised gas, introduced from a tangential port, flows along the cylinder into an annular duct through holes in the liner before it exits the chamber through a restriction hole, as shown in the schematic sketch of Figure 4-2. This continuous gas flow has a significant advantage in flushing out the chamber from the fuel after each injection. In addition, it allows the injection pump to operate over a number of injections, when compared to the single-shot constant-volume chambers. The injector has been mounted on a carriage that faces the front larger window for the PDA measurements. This carriage could move forward and backward to adjust the measurement location inside the spray. For the purpose of imaging, an adapter has been designed and constructed to fit the injector into one of the side windows, as it will be seen from the images. The reason for having a different fitting for imaging than for PDA is to allow more optical access for the camera to capture an image of the whole spray from the larger size front window.

A schematic sketch of the airflow feeding system to the constant volume chamber is shown in Figure 4-3. In this system, pressurised air is admitted from a four-stage inter-cooled reciprocating compressor; this compressor, powered by a 35 hp electric motor, delivers pressurised air with a discharge rate of 1 kg/min at a pressure of 270 bar. In order to have continuous flow to the chamber, pressurised air coming out of the compressor is delivered to a 115 litres reservoir (steel bottle) through a non-return valve. From the bottle, compressed air flows through a stop valve to the reducing panel where regulating valves are installed in order to reduce the air admitted from the reservoir at the required pressure into the chamber. After flowing through the chamber, air is delivered to the laboratory exhaust system.

Heating the air that enters the chamber is achieved by fitting an electrical heater with a nominal rating of 36 kW. The heater is installed inside a pressure vessel, which is located upstream of the constant-volume chamber. The temperature of the air is controlled by a closed-loop feed back system, which changes the electric power of the



a) Front view of the constant volume chamber



b) Side view of the constant volume chamber



c) Constant volume chamber

Figure 4-1 Front and side view picture of the constant volume high pressure chamber and its attachment windows

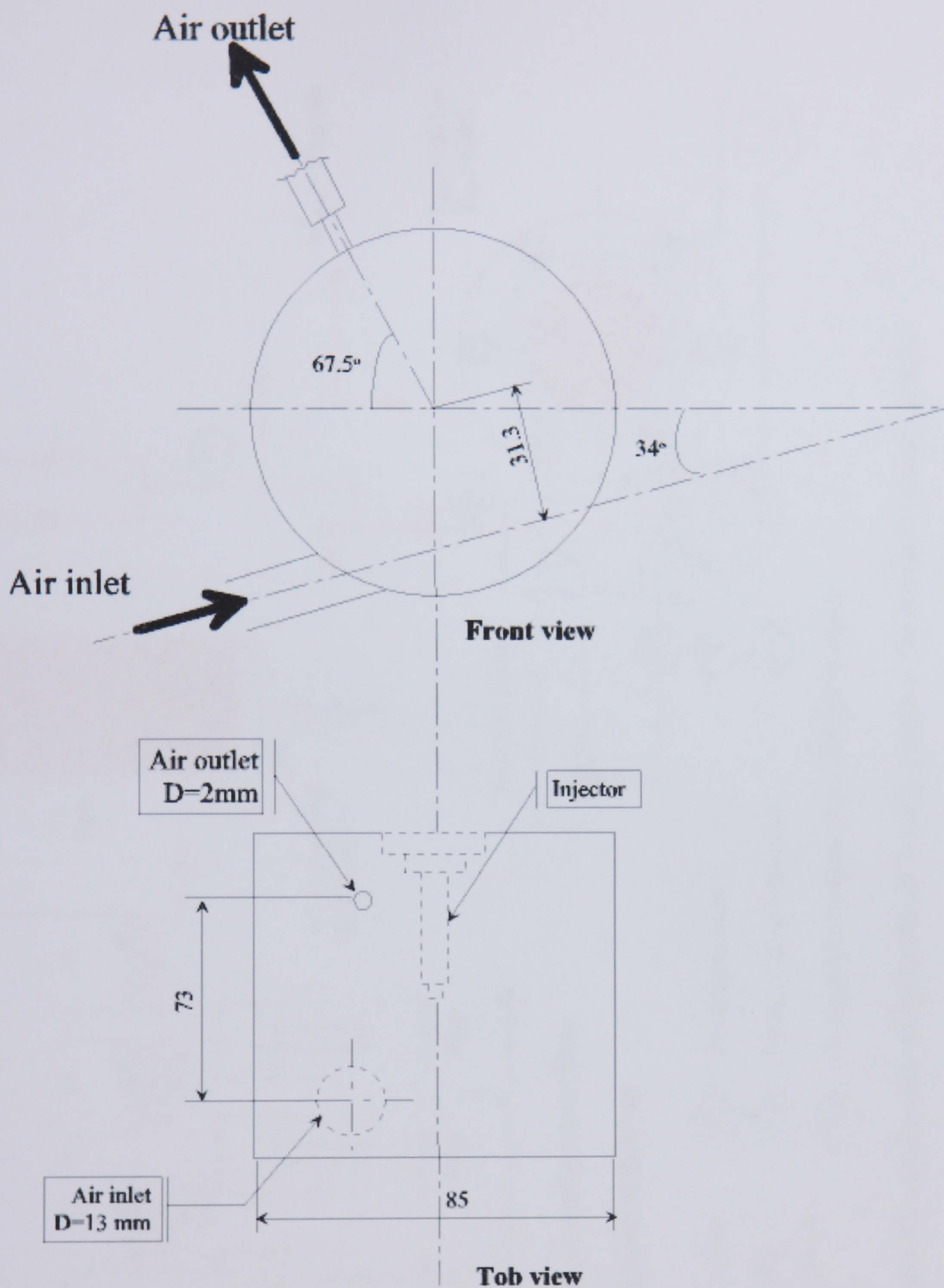


Figure 4-2 A schematic showing the air inlet and outlet with the main internal dimensions of the constant volume chamber (Drawing is not to scale)

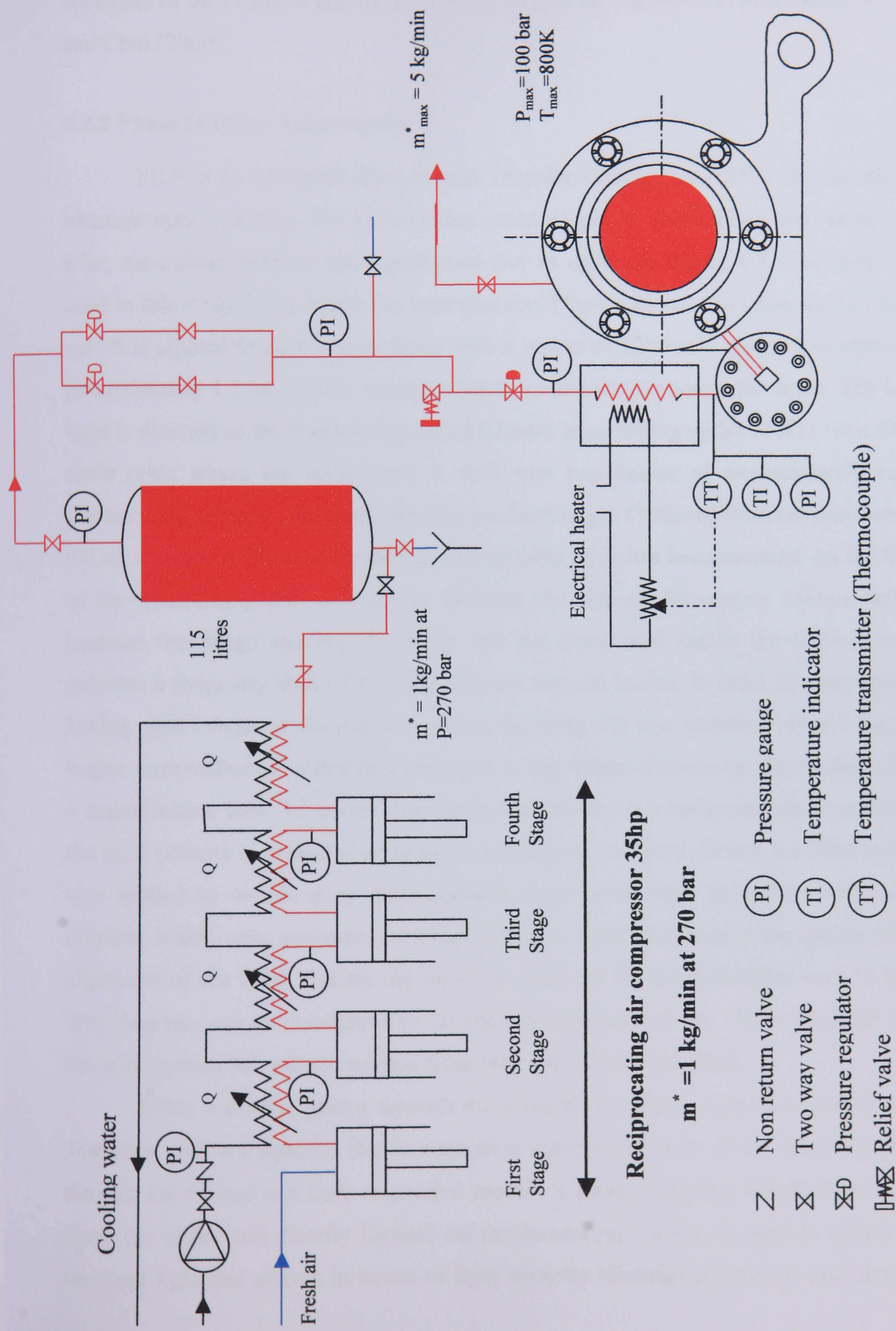


Figure 4-3 A schematic showing the air feeding system to the constant volume chamber

heater according to the difference between the setting value and the actual air temperature as measured by an installed thermocouple. More details on the design and operation of the chamber and its accessories system can be found in McGlashan, (1997) and Choi (2000).

4.2.2 Phase Doppler Anemometer

PDA is an advanced form of laser Doppler velocimetry (LDV) with an almost identical optical setting. The PDA system could simply be divided into four parts; laser tube, transmitter, receiver and signal processor as shown in Figure 4-4. The laser tube, used in this study, is an Argon Ion laser (Spectra Physics Model 164) operating at 514.5 nm. It is aligned to emit a laser beam with a power of 1200 mW and exit diameter of approximately 1.2 mm with a Gaussian intensity distribution across the beam. The laser light is directed to the transmitting optics (Dantec transmitting optics 57X1) via a fibre-optic cable where the laser beam is split into two beams of approximately equal intensity that cross at a distance equal to the focal length (300mm) from the final lens of the transmitter. A beam expander (expansion ratio of 2) has been mounted on the front of the transmitting unit in order to decrease the size of the control volume and to increase the fringe spacing. A Bragg cell has been used inside the transmitter to generate a frequency shift of 40 MHz between the two beams. In order to make sure of having clear fringes at the control volume, the Bragg cell was switched off and the two beams were realigned so that they cross producing fringes that can be clearly seen using a magnification lens. To ensure that the two beams are in a horizontal plane, and thus, the axial velocity is measured accurately a spirit level was used. Before the PDA system was applied to obtain spray measurements, some tests were performed with water droplets, which were generated by a humidifier, in order to guarantee the quality of the alignment of the PDA system; the data-rate achieved for the humidifier was 12 kHz. This data rate can be considered as satisfactory compared to the 11kHz obtained with the same system but without using a fibre optic cable by Choi (2000).

When a droplet passes through the measuring volume, light is scattered in all directions with a frequency that is dependent upon the velocity of the droplet moving through the fringes and their respective spaces. A photomultiplier, installed inside the receiving optics and directly focused on the measuring volume, is used to collect the scattered light that arrives in bursts of light intensity (Doppler bursts) for each droplet,

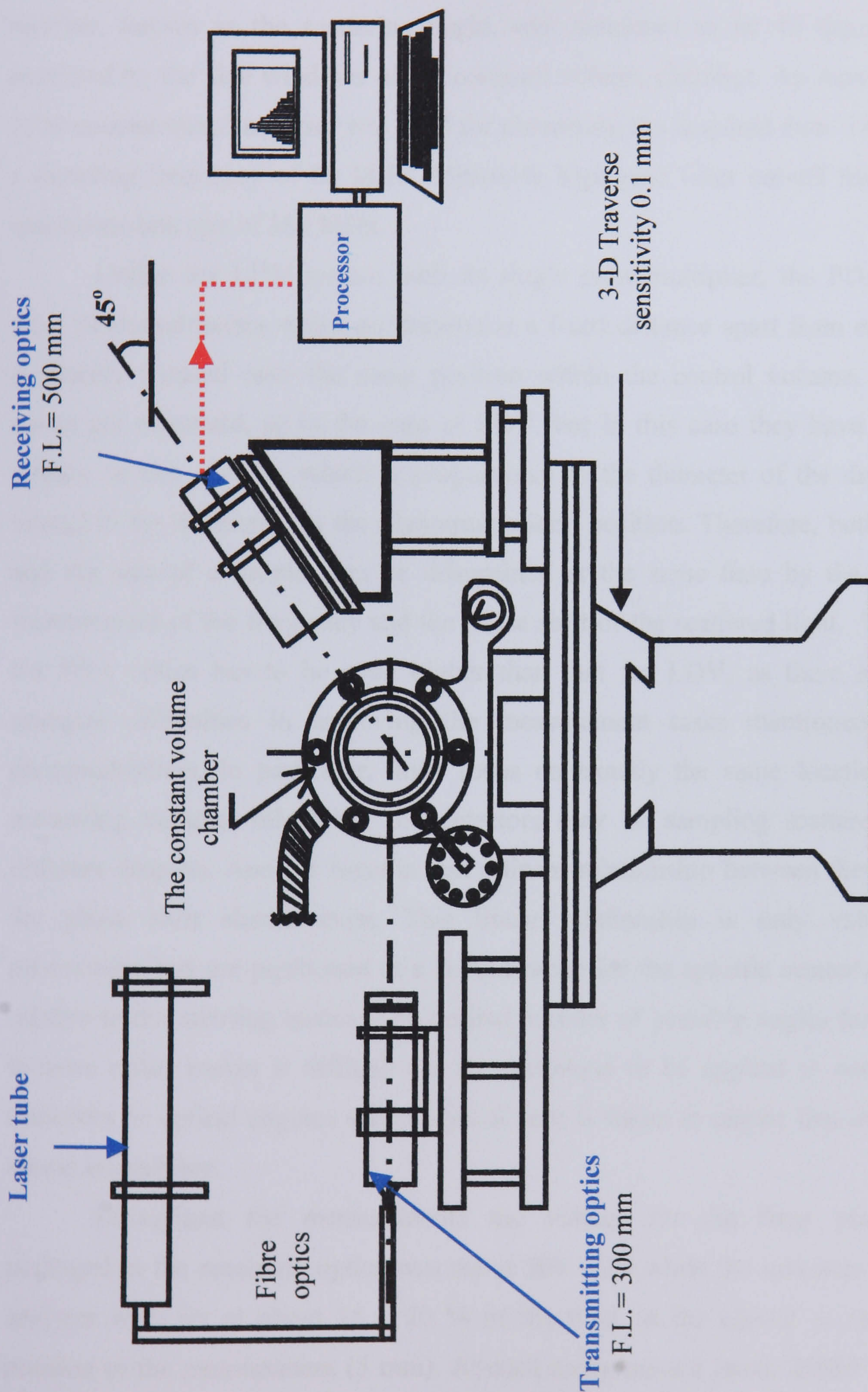


Figure 4-4 A schematic sketch showing the arrangement of the PDA system for spray measurements inside the constant volume chamber

as shown in Figure 4-5. The processor (analyser) is then measuring the frequency of each burst corresponding to the velocity according to the expression given in Figure 4-5. The receiver (Aerometrics RCV 100) has been located so that it faces the transmitter (forward scattering). The angle between the axis of the transmitter and the axis of the receiver, known as the scattering angle, was measured to be 45 degrees, which is restricted by the side windows of the constant volume chamber. An Aerometrics PDP-3100 counter-based analyser was used for processing the acquired data. The counter has a sampling frequency of 80 MHz, adjustable high pass filter cut-off frequency and a maximum data rate of 160 MHz.

Unlike the LDV system with its single photomultiplier, the PDA has two or three photomultipliers which are located at a fixed distance apart from each other and accurately focused onto the same position within the control volume. The Doppler bursts are measured, as in the case of LDV, but in this case they have a phase shift relative to one another, which is proportional to the diameter of the droplet. That is related to the difference in the photomultipliers' position. Therefore, both the velocity and the size of a droplet can be determined at the same time by the simultaneous measurement of the frequency and the phase shift of the scattered light. The quality of the PDA optics has to be even higher than that for LDV, as there are significant practical difficulties in achieving the measurement tasks mentioned above. The photomultipliers, in particular, must focus on exactly the same location within the measuring volume; otherwise, the detectors may be sampling scattered light from different droplets. Another issue is that a linear relationship between the diameter and the phase shift should exist. This linear relationship is only valid when the photomultipliers are positioned at a certain range for the specific scattering angles and relative to the crossing beams. The limited number of possible angles for good signal-to-noise ratios makes it difficult for the technique to be applied to constant-volume chambers or optical engines unless special care is taken to ensure that the appropriate access is available.

Throughout the measurements the voltage for the three photomultipliers packaged in the receiving optics was set at 500 volts while the saturated LEDs on the analyser were set at about 15 – 20 % of the time, at the closest to the nozzle exit position of the measurement (5 mm). At each measurement point, 25000 samples were collected under atmospheric conditions. However, inside the high-pressure chamber, only

5000

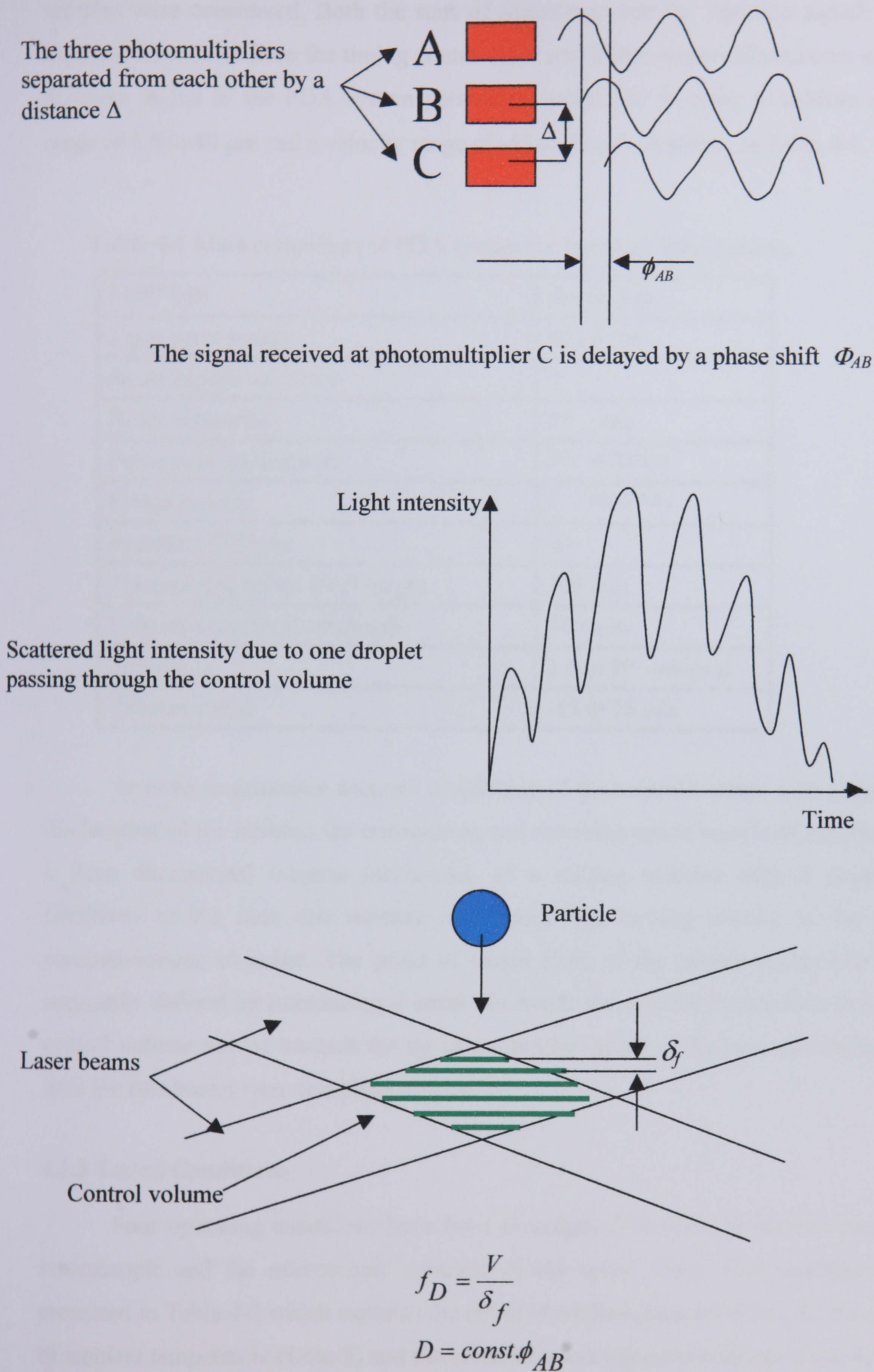


Figure 4-5 A schematic showing the principle of operation of a PDA system

samples were considered. Both the start of acquisition and the injection signals were triggered by a pulse from the timing control unit, while data were collected over a 4 ms duration. A list of the PDA system parameters, employed in order to achieve a size range of 1.6 to 80 μm and a velocity range of -43 to 78 m/s, is shown in Table 4-1.

Table 4-1 Main parameters of PDA system for the spray investigation.

Laser type	Argon-Ion
Laser wave length	514.5 nm
Beam expansion factor	2
Beam separation	38 mm
Probe volume diameter	80 microns
Fringe spacing	2 microns
Number of fringes	40
Transmitting optics focal length	300 mm
Collecting optics focal length	500 mm
Size range	1.6 to 80 microns
Velocity range	-43 to 78 m/s

In order to guarantee accurate positioning of the control volume with respect to the location of the injector, the transmitting and receiving optics were both mounted on a three dimensional traverse mechanism of a milling machine with a positional sensitivity of 0.1 mm; this traverse mechanism was moving relative to the fixed constant-volume chamber. The point of origin (0,0) of the measurements has been accurately defined by introducing a small pin inside the injector nozzle hole then, the control volume moved towards the tip of the pin by means of the traverse mechanism until the two beams were completely blocked.

4.2.3 Tested Conditions

Four operating conditions have been investigated in detail to identify both the macroscopic and the microscopic structure of the spray. These four conditions are presented in Table 4-2 which explains the effect of ambient pressure (Case 2), the effect of ambient temperature (Case 3) and the effect of lower injection pressure (Case 4). The macroscopic structure has been identified by imaging the spray at different times during

its transient development. Both the spray penetration and the spray angle have been measured for the four mentioned conditions at different times, from the start of injection. The microscopic spray structure has been identified using PDA measurements to reveal the temporal and the spatial variation of the mean droplet velocities and sizes. The fluid used in this study is Iso-Octane with a trade name of 2,2,4 Trimethylpentane 99.7 % HPLC grade. It has similar properties to gasoline but it is a single component fuel with a boiling temperature of 99 °C and a specific gravity of 0.69. Its main advantage is the simplicity it offers when used to validate the computer spray model relative to gasoline which is multi-component fuel.

Table 4-2 The operating conditions inside the constant-volume chamber.

Cases	Injection Pressure	Ambient Pressure	Ambient Temperature
1	70 bar	1 bar	293
2	70 bar	4.5 bar	293
3	70 bar	4.5 bar	500 K
4	20 bar	1 bar	293 K

4.3 SPRAY VISUALISATION

4.3.1 Spray Development Under Atmospheric Conditions

Prior to the analysis of the spray images for characterisation of the global spray development during the injection period, the delay time between the demand signal and the first appearance of liquid at the nozzle exit was estimated as a function of injection pressure, as previously discussed in Chapter 2. This delay time represents the combined effect of electrical, mechanical and flow delays. An additional factor is the time required for the fuel to travel from the needle lift passage to the nozzle exit which, say for an average flow velocity of 50 m/s, corresponds to ~20 μs. Overall, the delay time was found to vary between 305 and 335 μs, for injection pressures in the range 10-70 bar. This estimated delay time is very similar to the values reported by Shelby et al. (1998). All the presented results in this chapter consider as starting time the instant the

fuel starts to appear at the nozzle exit, which is delayed from the injection signal by the above mentioned delay time.

4.3.1.1 Spray development stages

The overall spray structure has been revealed by the images taken every 10 μs from the first appearance of the liquid fuel at the nozzle exit. These images were taken for injection pressures of 20, 50 and 70 bars under atmospheric in unconfined space. The above temporal resolution has allowed the transient spray development to be divided into four stages which are described below.

First Stage: It represents the very early stage as presented in Figure 4-6 for a 20 bar injection pressure; the fluid exits the nozzle as a bulk of non-atomised liquid that is located at the centre of the injection hole. This happens because the fluid has not acquired adequate angular momentum to form a conical spray, resulting from the high frictional losses between the nozzle wall and the needle at its early opening phase. Thus, the liquid is poorly atomised and only a small number of droplets is formed. These droplets have very low tangential velocities and appear as a wing attached to the liquid core, while exiting the nozzle. This characteristic wing shape has been observed at all injection pressures, as evident in Figure 4-7 at 50 bar and in Figure 4-8 at 70 bar injection pressure. In these figures, it can be noticed that the droplets are always moving in the same direction relative to the nozzle axis. Although the machining of the nozzle passages could partly be responsible for the observed asymmetry of the exiting liquid jet (Tomada et al., 1997; Cousin et al., 1998), it is more likely that its origin lies in the internal nozzle flow as the fluid starts rotating at the entrance to the hole. Due to the early stage of swirl development, the fluid rotates only for a fraction of a full turn, exiting as an asymmetric jet with an estimated tip penetration velocity between 45 and 75 m/s for nominal rail pressures of 20 and 70 bar, respectively. This first stage of spray development lasts for about 100 μs for injection pressure between 50 and 70 bar while for 20 bar it lasts longer; up to 150 μs .

Second Stage: This stage of spray development starts at about 150 μs after the start of injection for a 20 bar injection pressure, as shown in Figure 4-9. However, for the images obtained at 50 and 70 bar, this second stage seems to start at 100 μs , as shown in Figures 4-10 and 4-11, respectively, and that is due to the higher injection velocity associated with the higher injection pressures. This second stage consists of an asymmetric non-hollow spray that has insufficient swirl to become hollow and

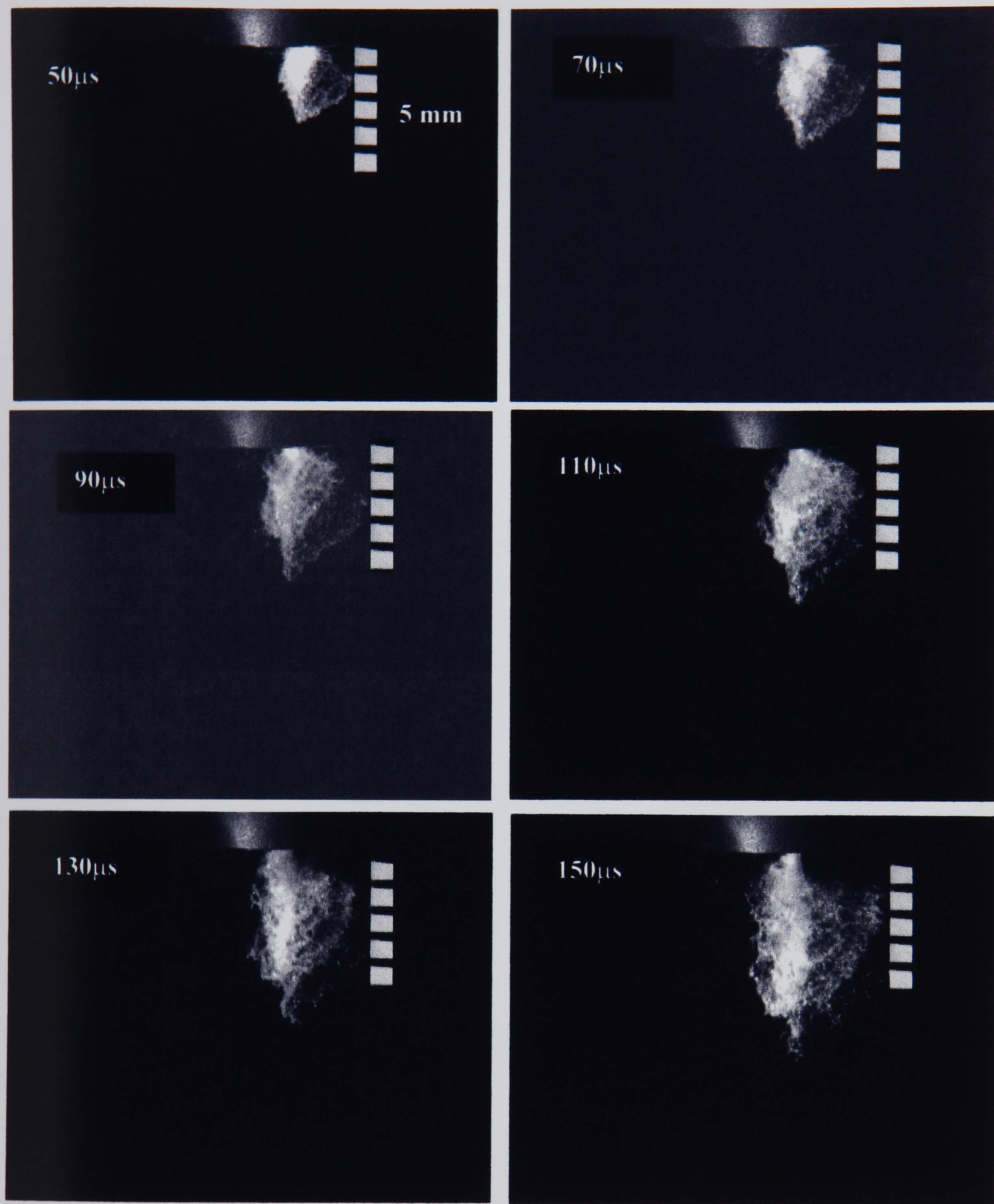


Figure 4-6 Spray images show the first stage of spray development at 20 bar injection pressure and different times from start of injection

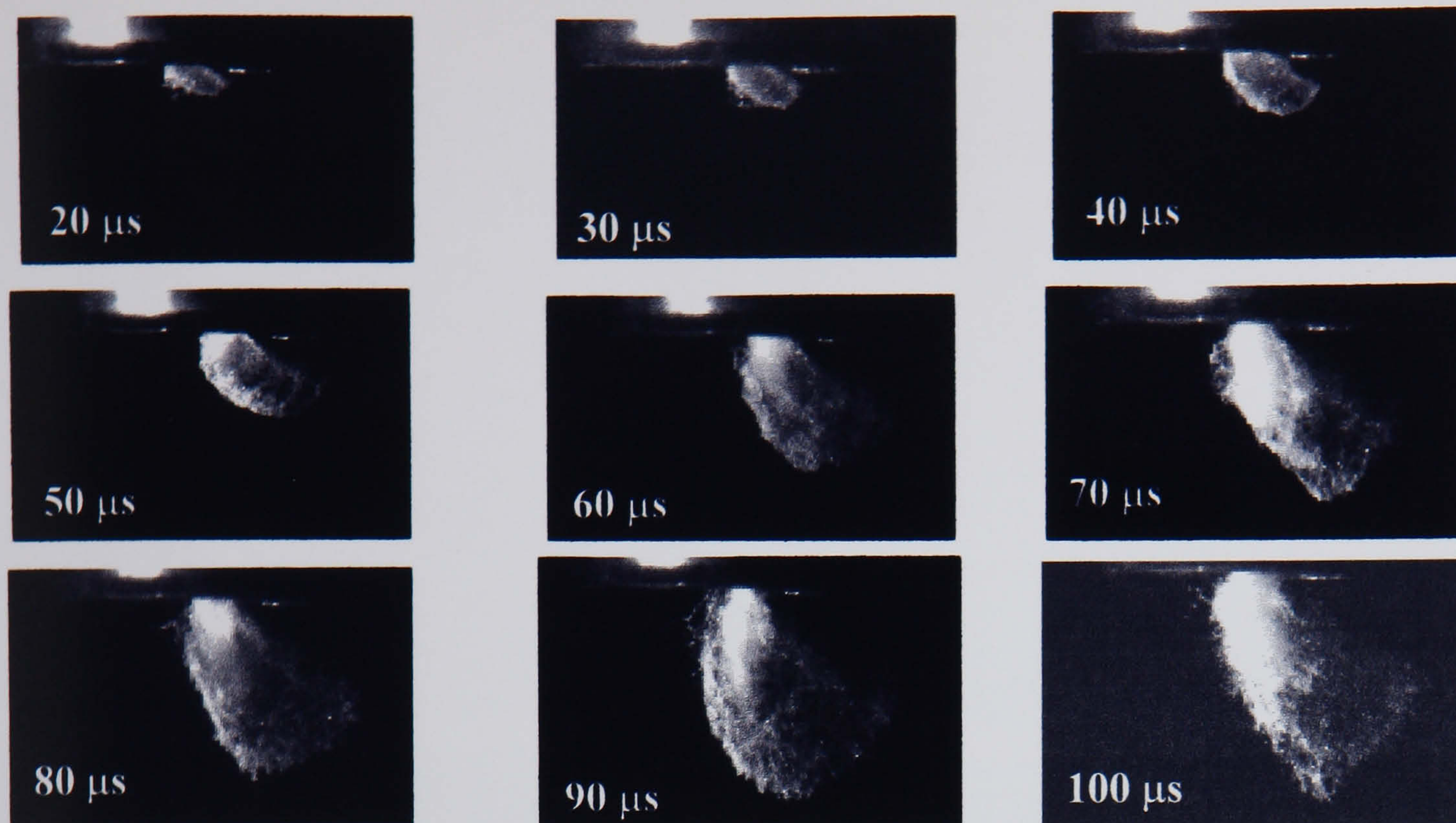


Figure 4-7 Spray images show the first stage of spray development at 50 bar injection pressure and different times from start of injection.

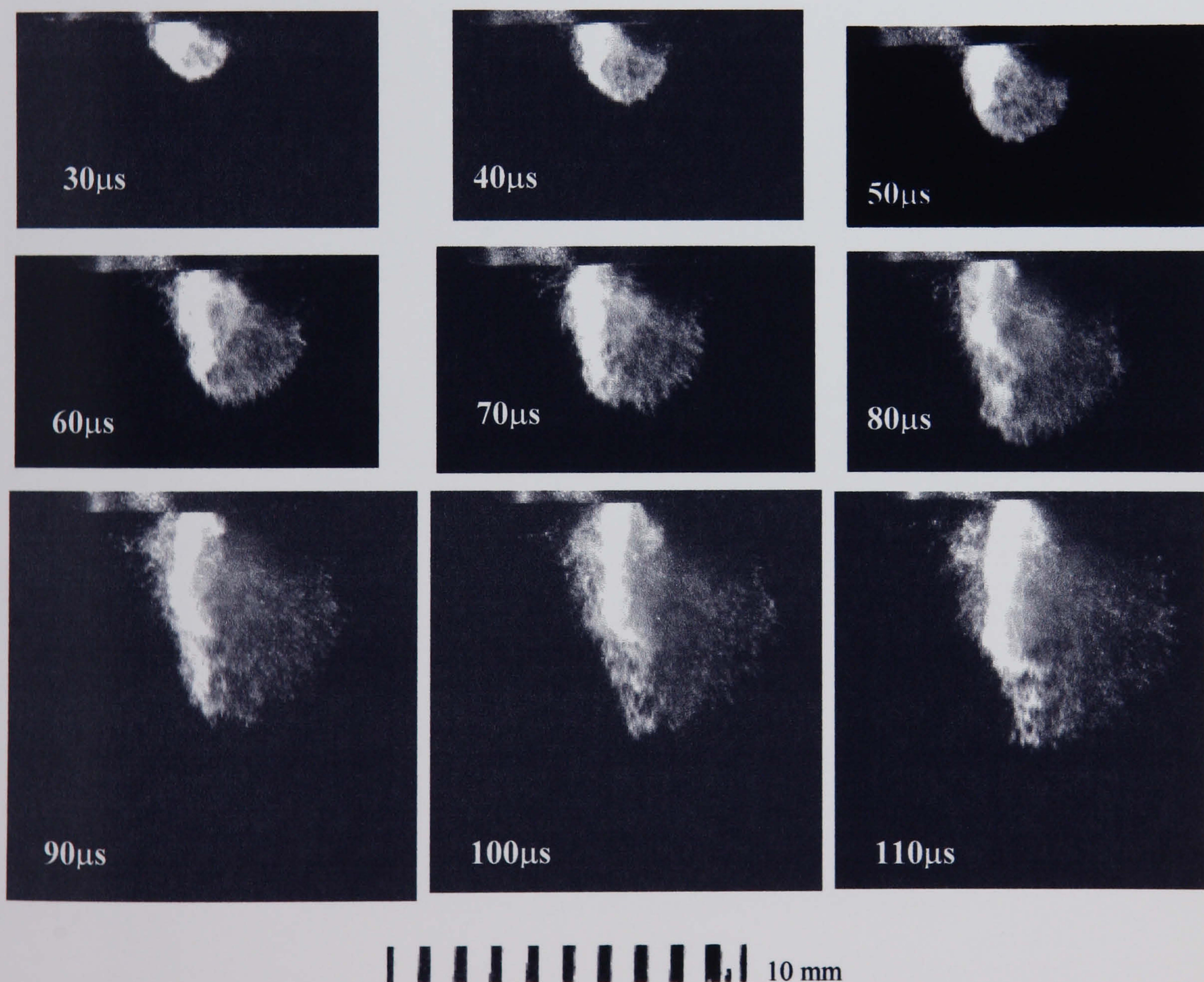


Figure 4-8 Spray images show the first stage of spray development at 70 bar injection pressure and different times from start of injection.

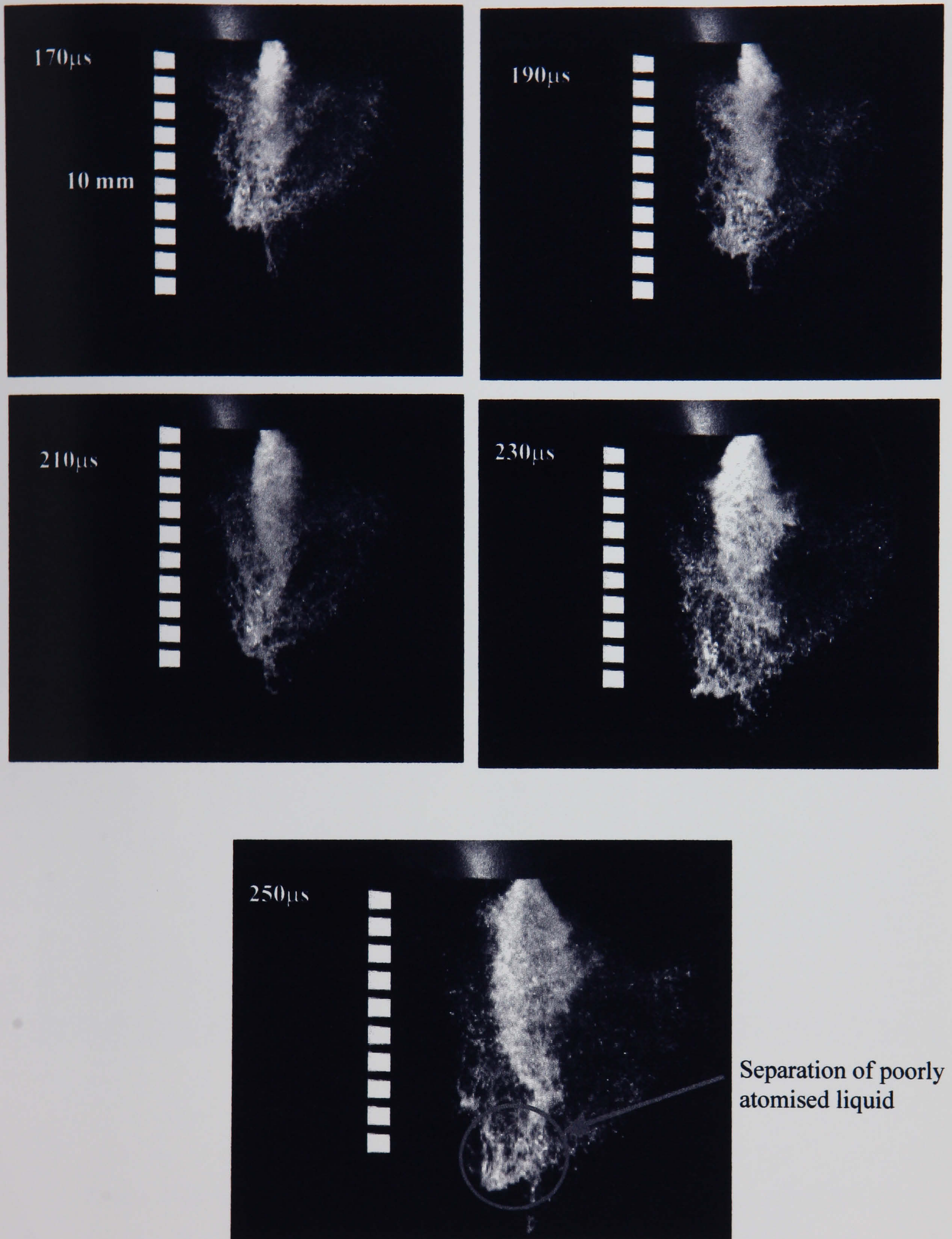


Figure 4-9 Spray images show the second stage of spray development at 20 bar injection pressure and different times from start of injection

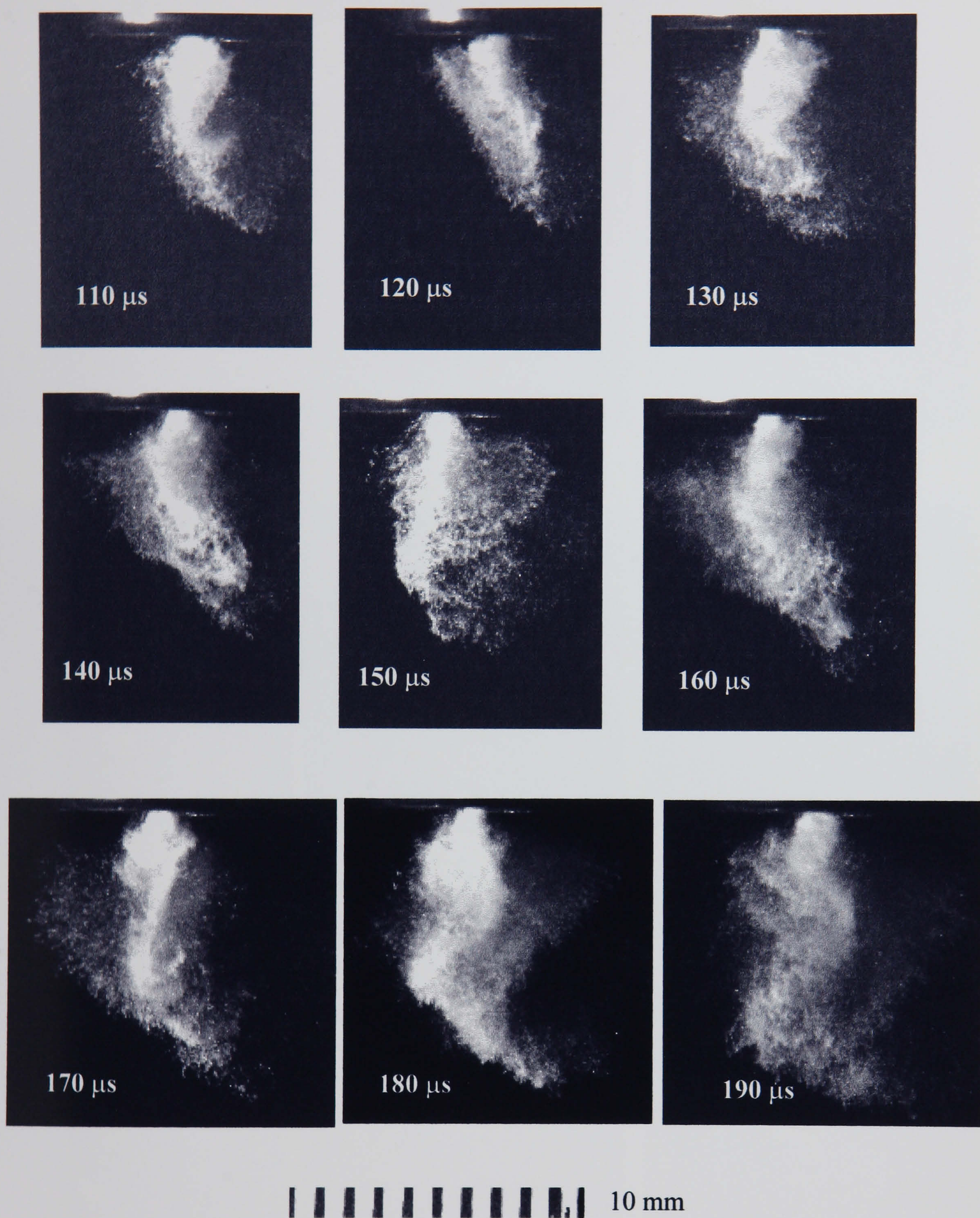


Figure 4-10 Spray images show the second stage of spray development at 50 bar injection pressure and different times from start of injection

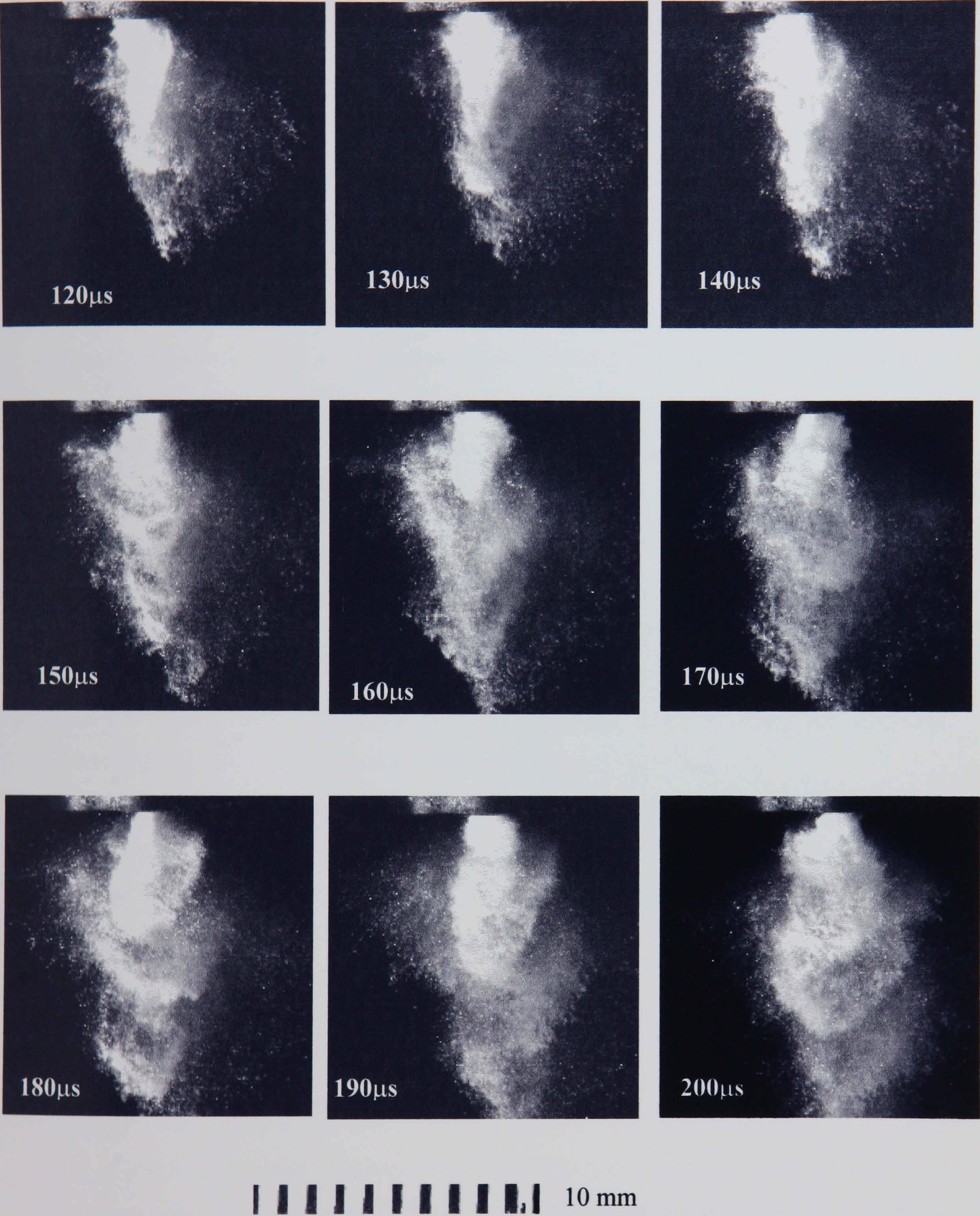


Figure 4-11 Spray images show the second stage of spray development at 70 bar injection pressure and different times from start of injection

resembles a jet with a helical structure. Similarly to the first stage, wings containing a high number of large droplets appear on both sides of the non-hollow spray body, as it rotates during its penetration. However, in this stage the length-to-width ratio of the spray is comparatively larger than that in the first stage, which maintained almost the same ratio. Towards the end of this second stage, which lasts for approximately another 100 μs , the tail of the non-hollow spray tends to separate from the main body, with some evidence of poor atomisation in this relatively small mass of fluid. Although there is a small increase in the spray angle during this stage, it is very difficult to measure it due to the asymmetric shape of the spray and the dynamic variation from one image to another. It is also noticeable that the spray at higher injection pressures has a multi-layer structure towards the end of this stage which could be attributed to the presence of pressure fluctuations in the fuel at the nozzle exit. These fluctuations are associated with the relatively fast opening of the needle as described before in Chapter 2. It is expected that this poorly atomised fuel is mostly responsible for the relatively high hydrocarbons emissions in DISI engines. Although research is still continuing in order to reduce this pre-spray, some researchers such as Xu and Markle (1998) have recommended the use of an outwards opening needle rather than inwards opening one claiming that the droplet sizes from outwards opening injectors during the needle opening and closing period are smaller than those during the main spray; however there are additional problems facing this type of injector which will become clear later.

Third Stage: During this stage the spray transforms from a non-hollow to a conical hollow spray structure with a cone angle increasing with time until it reaches its maximum value of about 80 degrees which consist of a sequence of horizontal layers. This developmental phase is due to the gradual swirl formation within the nozzle, the formation of a liquid film inside the nozzle hole and its development just outside the nozzle, as shown previously in Chapters 2 and 3. The layers noticed in the spray could be attributed to the internal velocity fluctuations associated with the pressure fluctuations that occur at the nozzle exit. Figure 4-12 gives an example of the steps taking place in the spray structure transformation from the asymmetric non-hollow shape to the developed conical hollow spray. The spray development at the 20 bar injection pressure lasts for about 150 μs ; similar images are shown in Figures 4-13 and 4-14 for injection pressures of 50 bar and 70 bar, respectively. Although the spray images do not confirm the hollow structure, another study by Hoffman et al (1997 and

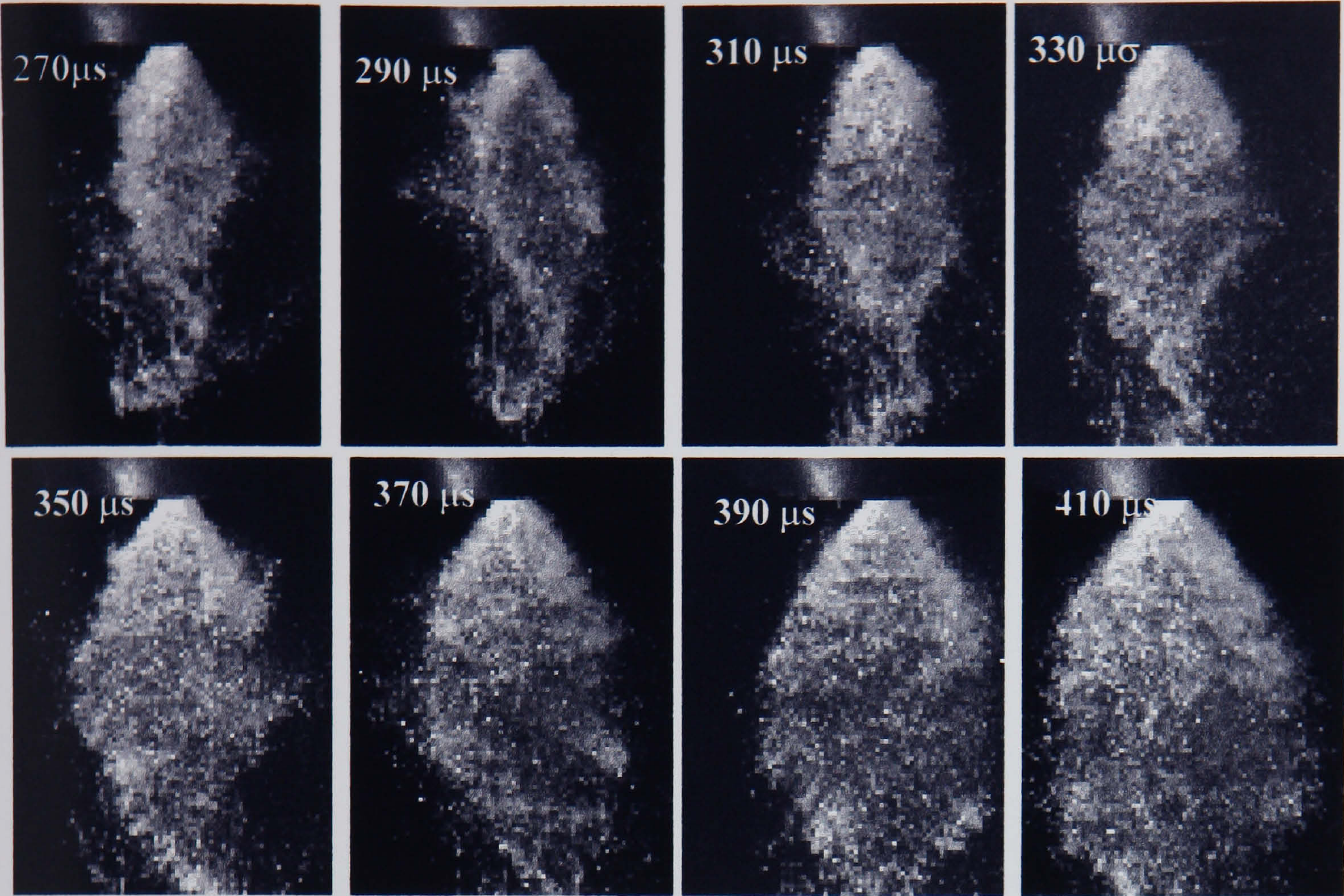


Figure 4-12 Spray images show the third stage of spray development at 20 bar injection pressure and different times from start of injection

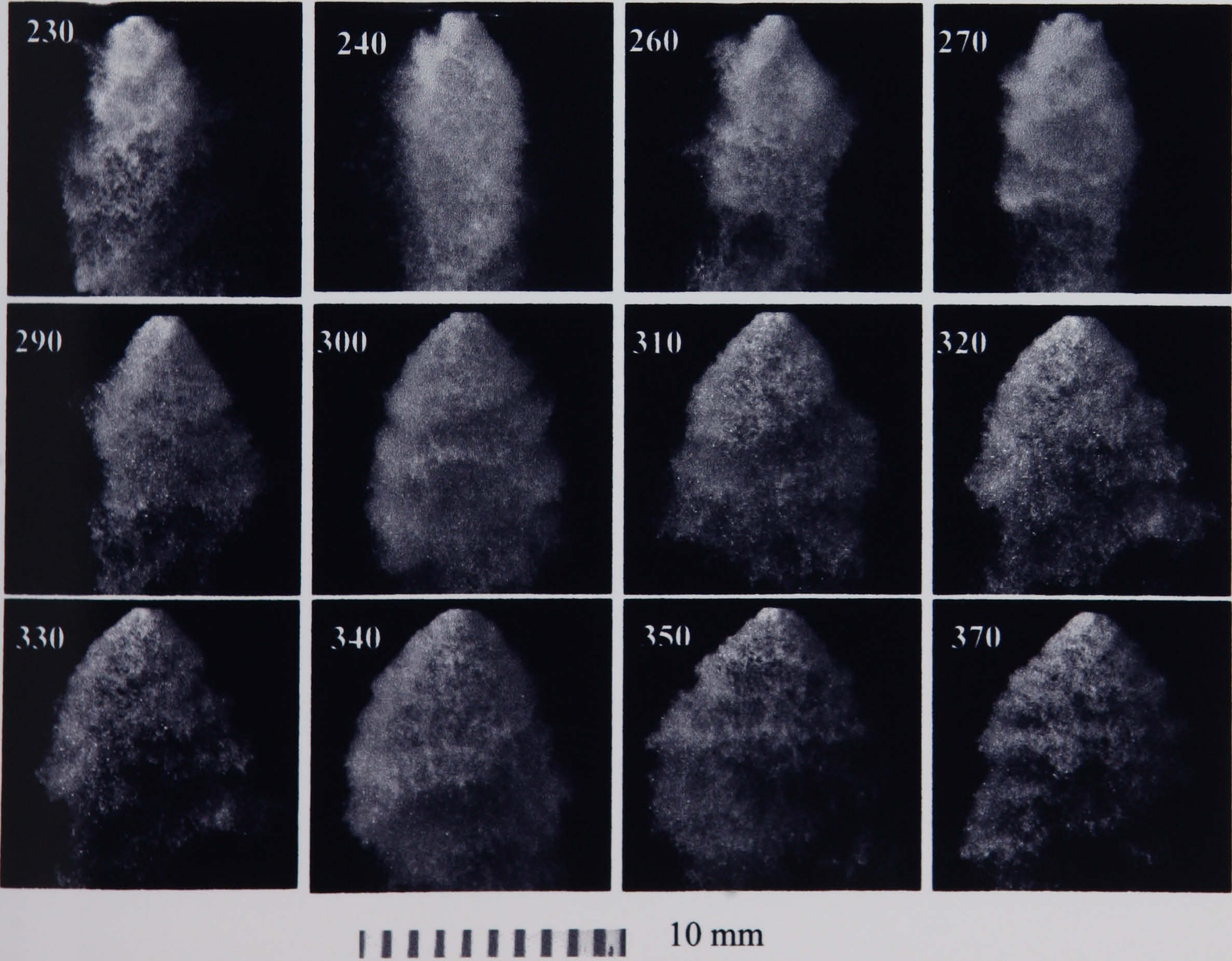


Figure 4-13 Spray images show the third stage of spray development at 50 bar injection pressure and different times from start of injection (μ s)

1998) has provided evidence that the spray has indeed a hollow conical structure by measuring the spray mass flux distribution across the spray using a timed-sampling spray patternator.

Fourth Stage: The fourth and last stage corresponds to the fully developed hollow spray. During this period the spray angle remains constant and the spray-to-spray variation is relatively small at the same time. Typical spray images during this stage, for the 20 bar injection pressure, are shown in Figure 4-15. Spray images for different injection pressures clearly reveal that the degree of atomisation of the fuel spray increases with increasing injection pressure, as shown in Figures 4-16 and 4-17 for injection pressures of 50 bar and 70 bar, respectively, when compared to the 20 bar injection pressure in Figure 4-15. The spray cone angle seems to be independent of injection pressure under atmospheric conditions, in agreement with the results of Nouri et al (1999) and Fraidl et al (1998). The reason for the observed independency of spray angle on injection pressure is attributed to the constant flow angle at the nozzle exit, a consequence of the fact that changing the injection pressure does not significantly alter the ratio of the swirl to the axial velocity, although it affects the amplitude of each component.

It is also observed that, at the front part of the spray, a counter flow vortex is developing during this stage. This toroidal vortex is attributed to the catching up of the droplets injected early by the following ones injected later. At the end of injection, the very late injected droplets start to move away from the injector and the spray's conical shape diminishes and transforms into two counter flow vortices. These vortices grow bigger in a certain fashion where their centres are fixed in their locations, with a distance between them of 50 mm, as shown in Figure 4-16 for the case of the 50 bar injection pressure. Similar observations have been made by Yamauchi and Wakisaka (1996) and by Zhao et al. (1996) who reported later in their subsequent extended study (1999) that for a mono-disperse spray of 40 μm droplets this counter flow vortex does not exist. This could be attributed to the mechanism of vortex formation since in this case, the droplets sizes are relatively large compared to the typical droplet sizes of 20 μm of swirl injectors and their wider size distribution that includes many smaller droplets. The larger droplets survive longer without losing their momentum and it is more difficult for them to change their direction. This observation is also confirmed by

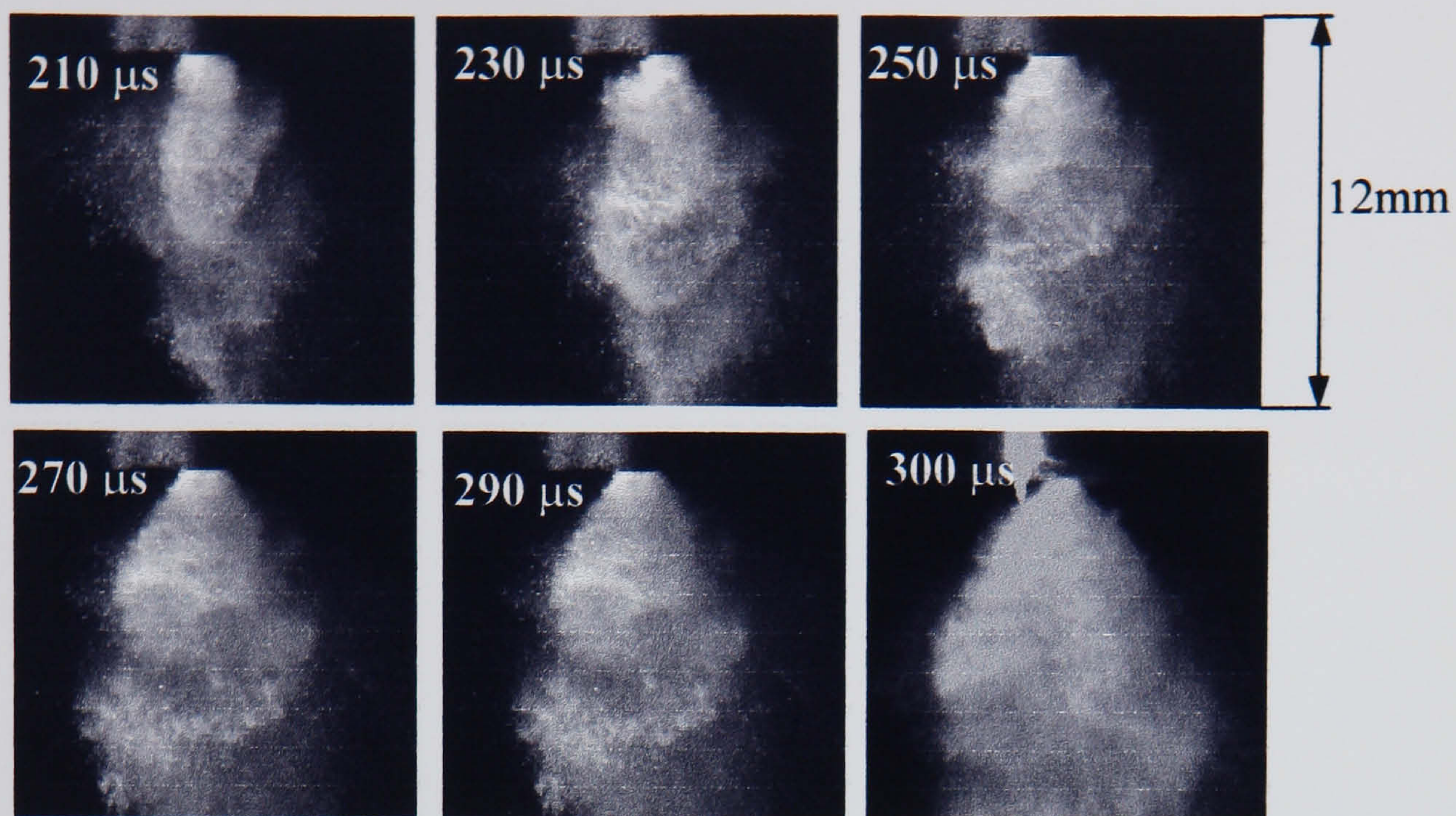


Figure 4-14 Spray images show the third stage of spray development at 70 bar injection pressure and different times from start of injection

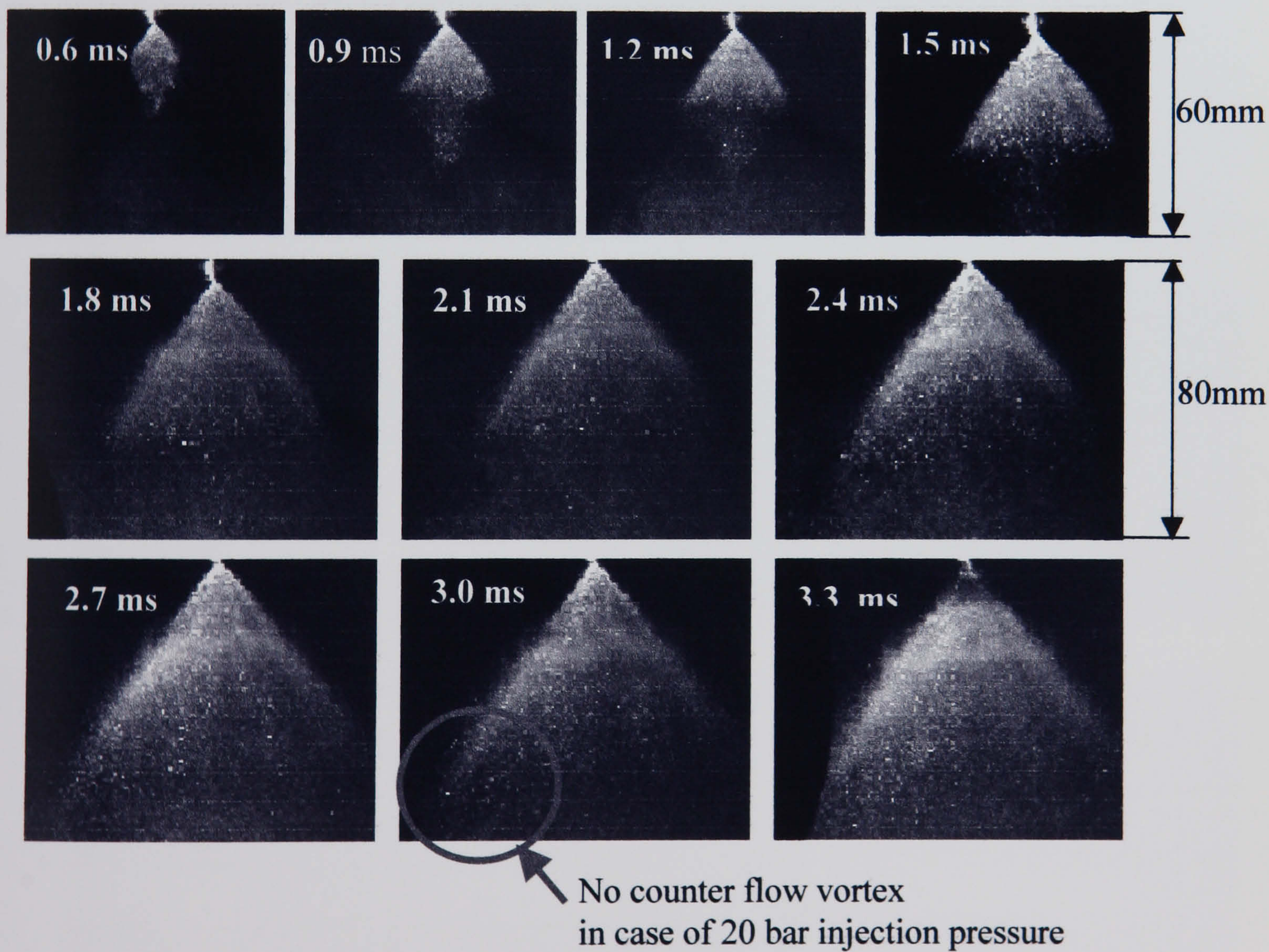


Figure 4-15 Spray images show the fourth stage of spray development at 20 bar injection pressure and different times from start of injection

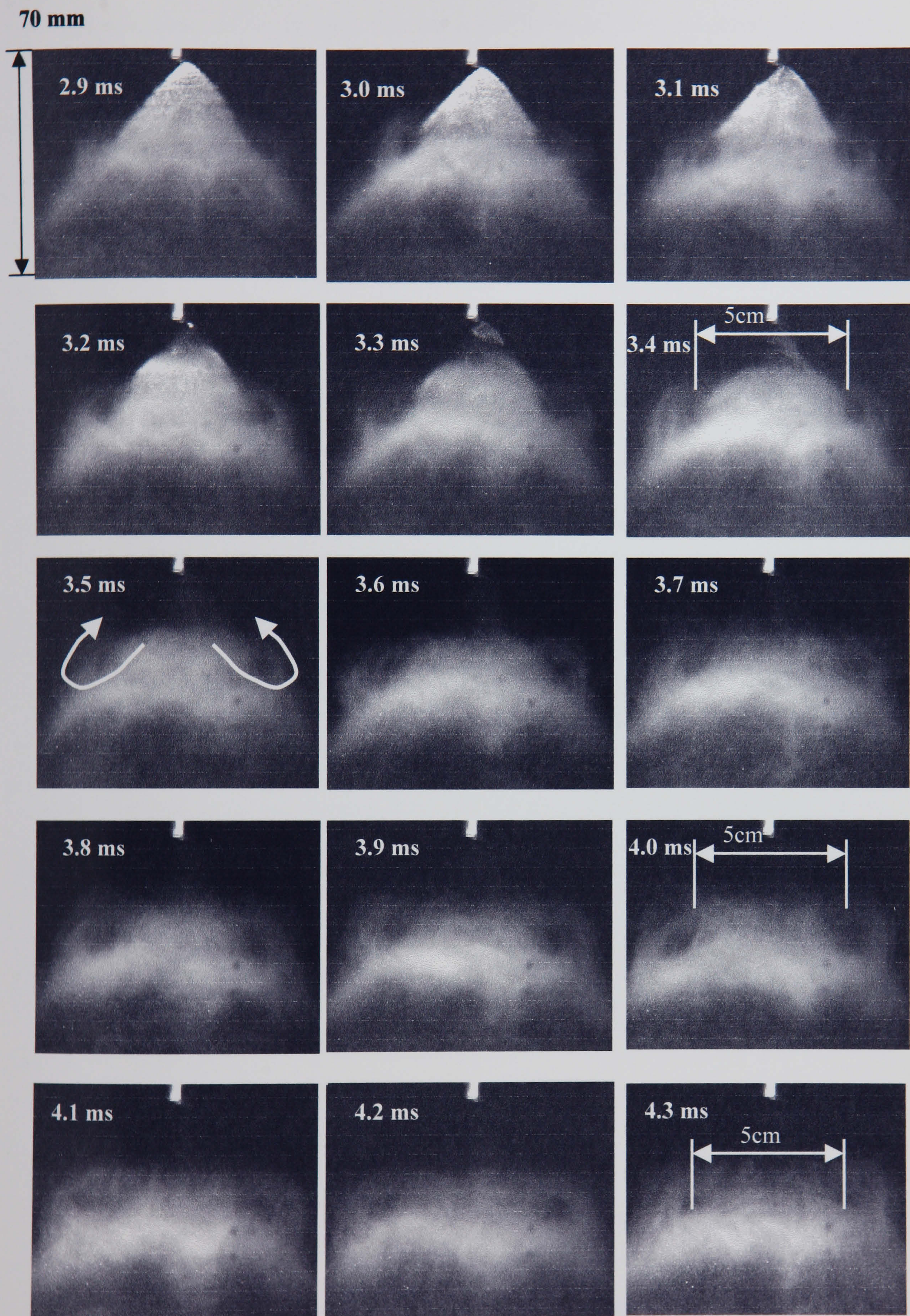


Figure 4-16 Spray images show the last stage of spray developing and after the end of injection, injection duration = 3ms, injection pressure = 50 bar

the images obtained for the spray injected at just 20 bar injection pressure where there is no sign of a spray counter flow vortex.

Similar stages of the spray have been identified by Shelby et al. (1998) although; both the second and third stages were not distinguished in their analysis. By using the PLIF technique, they reported a slight decrease in the spray angle when the spray reaches its full developed stage further downstream. Evers (1994) has reported another classification where the counter flow vortex is considered as a separate stage.

Since all these images have been taken under atmospheric conditions in still air, any momentum imparted to the air is transferred from the spray droplets momentum. However, the situation in engines is quite different, since the air has its own momentum and pattern, thus influencing the spray during its trajectory. Kashdan (2002) showed that the counter flow vortex diminishes when the spray is injected during the induction stroke whereas in another study by Kampanis (2003) it was noticed that the tumble air motion affects not only the head of the spray but also exerts some influence on the shape of the main body of the spray.

4.3.1.2 Spray penetration and angle

Figures 4-18a and 4-18b summarise the characteristics of the first three stages of the spray development by presenting the temporal variation of the spray tip penetration and the nozzle spray angle during their first 400 μs as a function of injection pressure. It is clear that both parameters increase faster at higher injection pressures as evidenced by the different slope of the line fitting the measurement points. The time required by the spray to reach its maximum angle increases as injection pressure decreases which is attributed to the time required for the film developed outside the nozzle to open up and reach its maximum divergence. The linear increase in the tip penetration proves that the tip of the spray is moving at almost constant velocity of 74.3, 64.3 and 45.7 m/s corresponding to 70, 50 and 20 bar injection pressures, respectively. It is also noticeable that; the rate of the increase in the spray penetration is not linear with injection pressure, but it is more significant at lower injection pressures.

Spray penetration is normally defined as the distance from the spray tip to the nozzle exit for example, Lefebvre (1989) and Byvel (1993). This definition is appropriate for symmetric diesel sprays but clearly inadequate for swirl-induced gasoline sprays where the spray structure changes drastically with time. As Figure

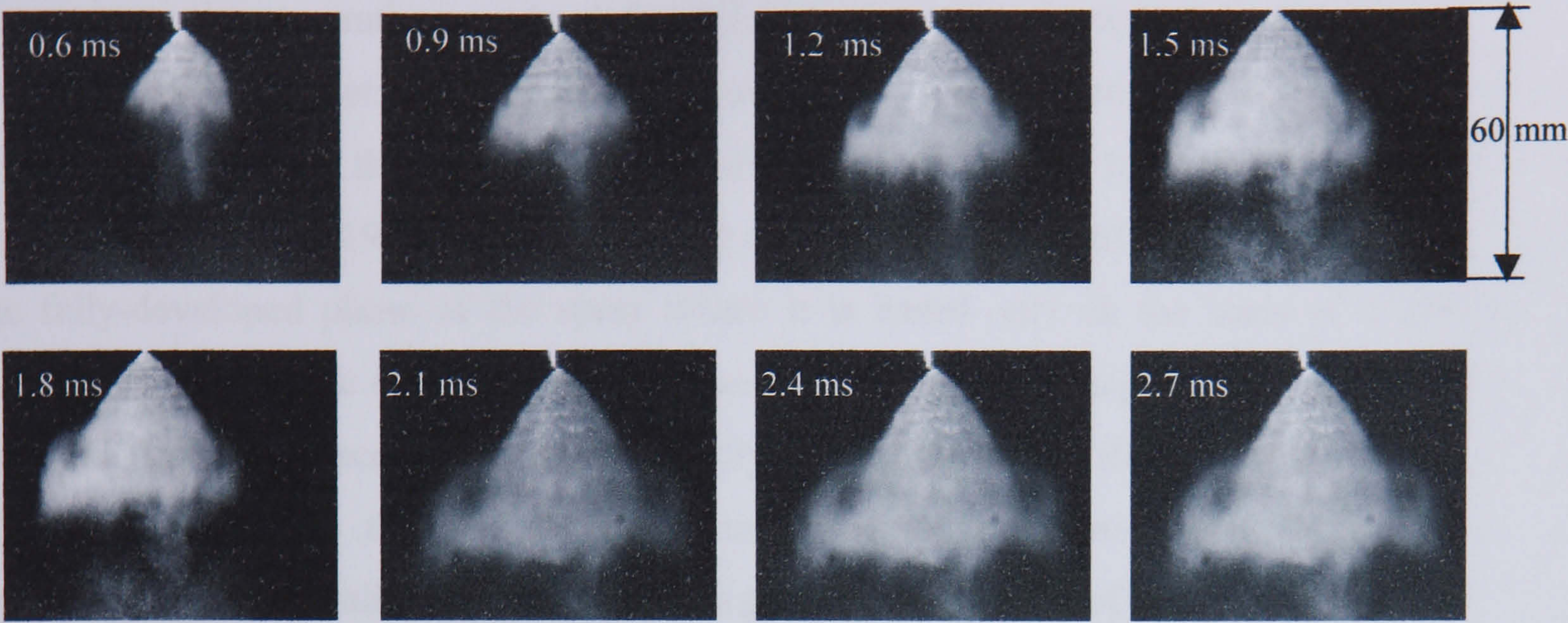
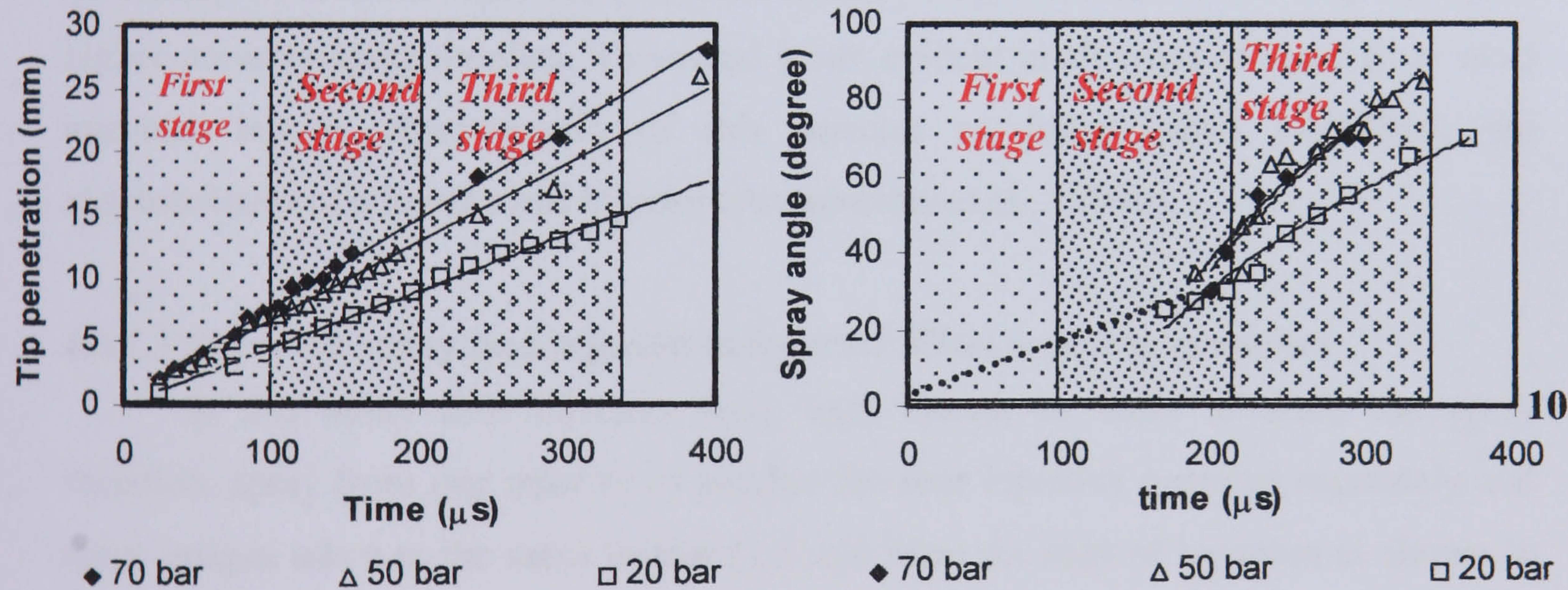


Figure 4-17 Spray images show the fourth stage of spray development at 70 bar injection pressure and different times from start of injection



a) Spray penetration during the early stages of spray development **b) Spray angle during the early stages of spray development**

Figure 4-18 Spray penetration and angle during the developing stages of the spray

4-19a shows, a complementary definition has been introduced to allow the spray penetration to be quantified during the whole injection period. During the first three stages spray tip penetration can be defined in the conventional way, since the spray is not fully-developed yet; however, at the fourth stage, the alternative definition of tip penetration based on the maximum spray width has been used to obtain the results presented in Figure 4-19b. This is dictated by the difficulty in identifying the spray tip at the fully-developed phase of the spray where it is based only on the basis of a few droplets moving in the central cone region and representing an insignificant fraction of the total fluid mass, according to the quantity of fuel present in the pre-spray. Thus, Figure 4-19b reveals that the expected trend of spray penetration increasing with injection pressure is valid only over a certain period, for this type of spray, as a result of the interdependence between injection pressure, size of droplets formed and droplet deceleration as a function of droplet diameter. Thus at the beginning the high pressure injection spray has more momentum and therefore it travels at higher velocity than the lower pressure injection spray. However, later on, the small droplets which are found more in the higher pressure case, decelerate much faster while the larger droplets associated with lower injection pressures sustain their momentum and may catch the higher-pressure spray droplets. This trend is not evident in all types of spray from swirl injectors but is characteristic of this injector as found when identifying the characteristics of a lower swirl injector (Arcoumanis et al., 1999b).

4.3.1.3 Spray-to-spray and injector-to-injector effect on spray geometry

In this study four injectors have been tested. In order to study the spray variation, spray from one injector to another the four injectors operated separately and spray images taken at the same instant (1.5 ms) from the start of injection as shown in Figure 4-20. It was found that the spray from one of these injectors has a very different spray angle, which implies that this injector has a different swirl angle. The other three injectors seem to have similar spray angle, although some variation was observed in the spray geometry between injectors. This highlights the need for accurate machining of the nozzle and swirler, especially the surface roughness of the nozzle as quoted by Tomoda et al. (1997). In order to examine how consistent and reliable the spray is in terms of its geometry, 20 images have been obtained and animated, revealing that the variation is negligible in terms of geometry, with minor variations present in the

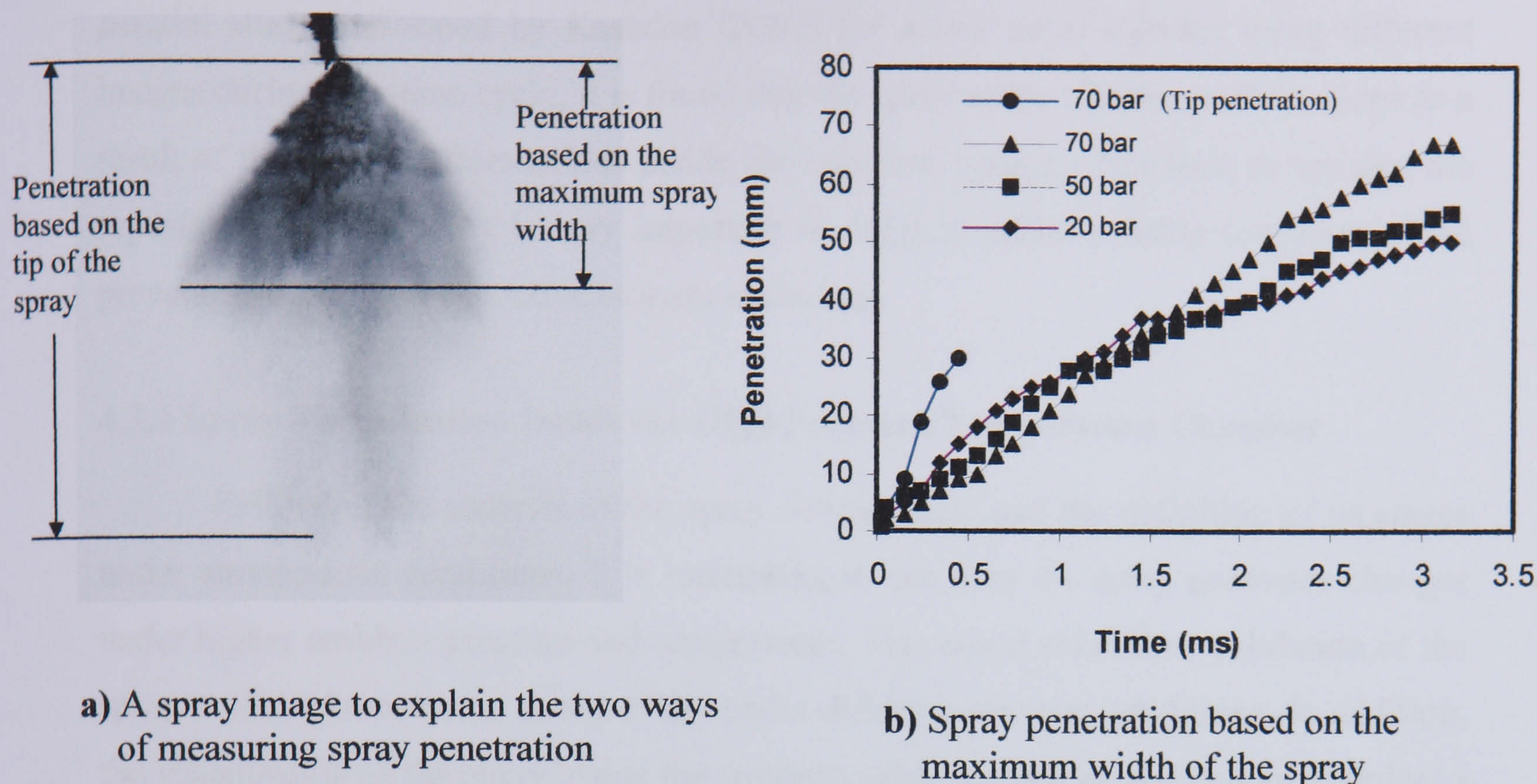


Figure 4-19 Spray penetration at 20, 50 and 70 bar injection pressure based on the maximum spray width at atmospheric ambient conditions

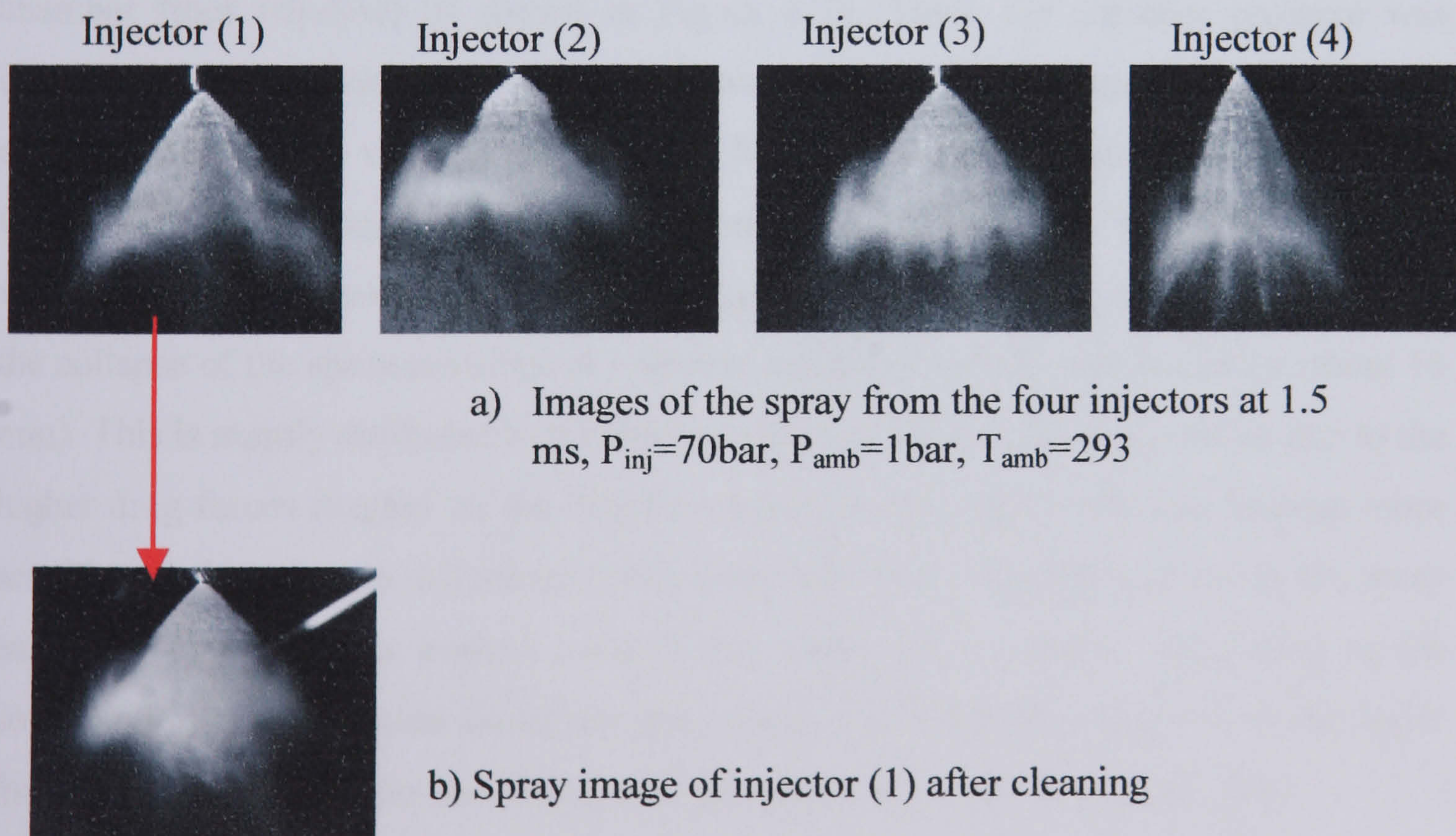


Figure 4-20 Spray geometry from four injectors and spray geometry change after cleaning one of these injectors

location of the layers in the spray which may change from one cycle to the next. In a parallel study performed by Kashdan (2002) for a low swirl injector using different images during the same cycle, it is found that the spray angle vibrates as it develops as a result of the pressure fluctuations inside the injection system. Needless to say that the repeatability of the spray is very important in order to achieve stable combustion and prevent misfiring and excessive exhaust emissions.

4.3.2 Spray Visualization Inside the High Pressure/Temperature Chamber

Following the analysis of the spray development and the definition of its stages under atmospheric conditions, it is interesting to see how the spray geometry changes under higher ambient pressure and temperature. This could also allow validation of the spray model (Arcoumanis et al., 1999) under different ambient conditions. In addition, the visualisation of the spray inside the constant volume chamber is essential in order to guide subsequent PDA measurements inside the high pressure chamber.

To achieve this purpose, the injector has been installed in one of the side windows of the constant volume chamber and spray images have been taken from the large front window at every 0.2 ms for an injection pressure of 70 bar. The first task was to visualise the spray under atmospheric chamber pressure and temperature (without the chamber front window) as shown in Figure 4-21. Then, the chamber pressure was increased to 4.5 bar and similar images have been obtained, Figure 4-22. Finally, the chamber temperature was increased to 500 K and a similar sequence of images taken, Figure 4-23. The images clearly show a dramatic reduction in the volume of the spray when the chamber pressure increases as compared to the atmospheric spray, reflected in the collapse of the spray envelope at a shorter distance from the nozzle exit (at about 10 mm). This is mainly attributed to the fast reduction in the droplets momentum due to the higher drag forces exerted on the droplet motion causing the droplets to become more sensitive to the pressure difference between the air located outside and inside the spray envelope; they are thus pushed towards the centre of the spray. According to the conservation of momentum during the early stage of droplet trajectory, where the major force is the drag force, the deceleration of the droplet takes the following form:

$$\frac{dU_d}{dt} = \frac{3\rho_a}{4\rho_d D_d} C_d (U_a - U_d) |U_a - U_d| \quad (4-1)$$

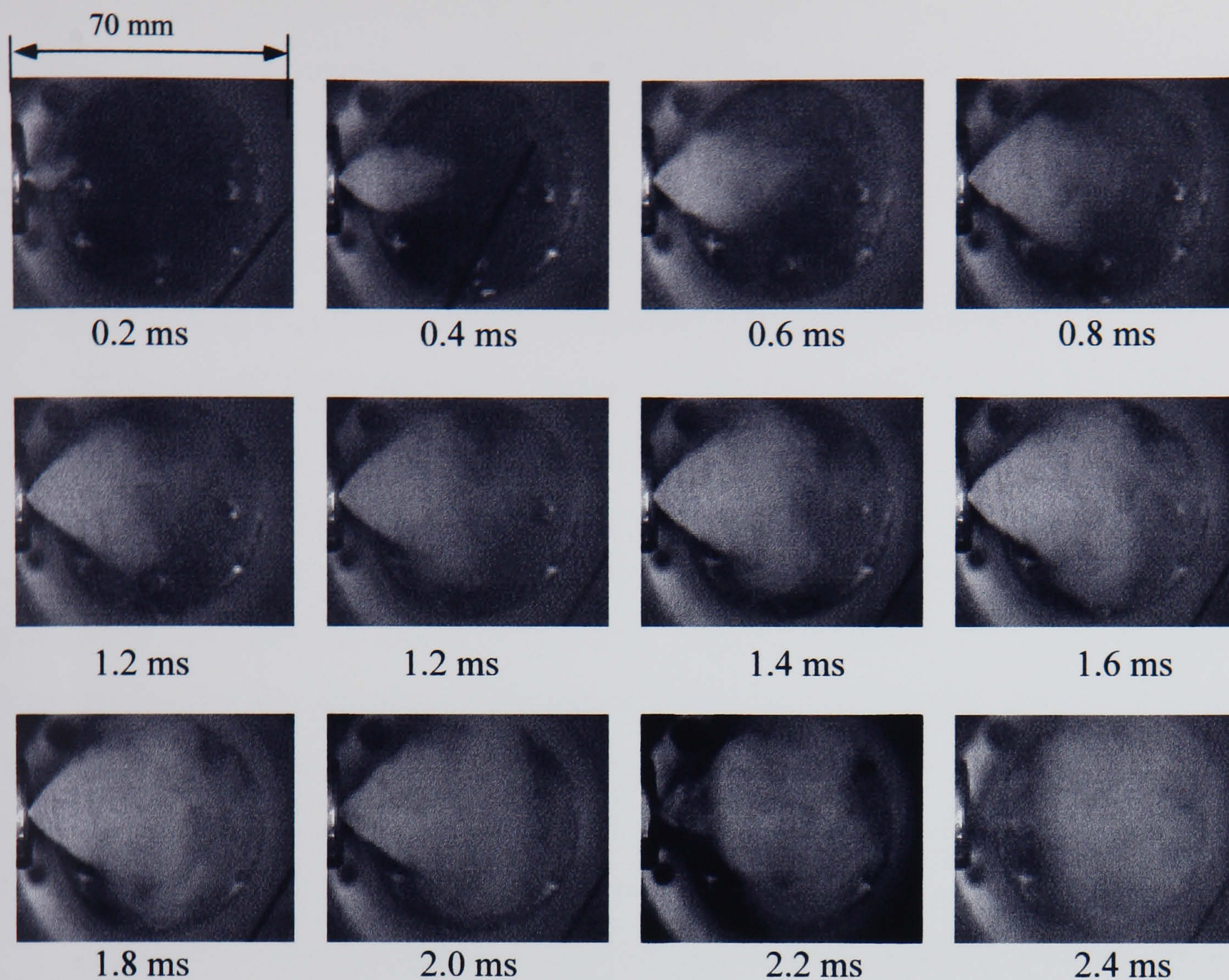


Figure 4-21 Spray images at different times inside the constant volume chamber under atmospheric pressure and temperature, $P_{inj}=70$ bar, $t_{inj}=2$ ms

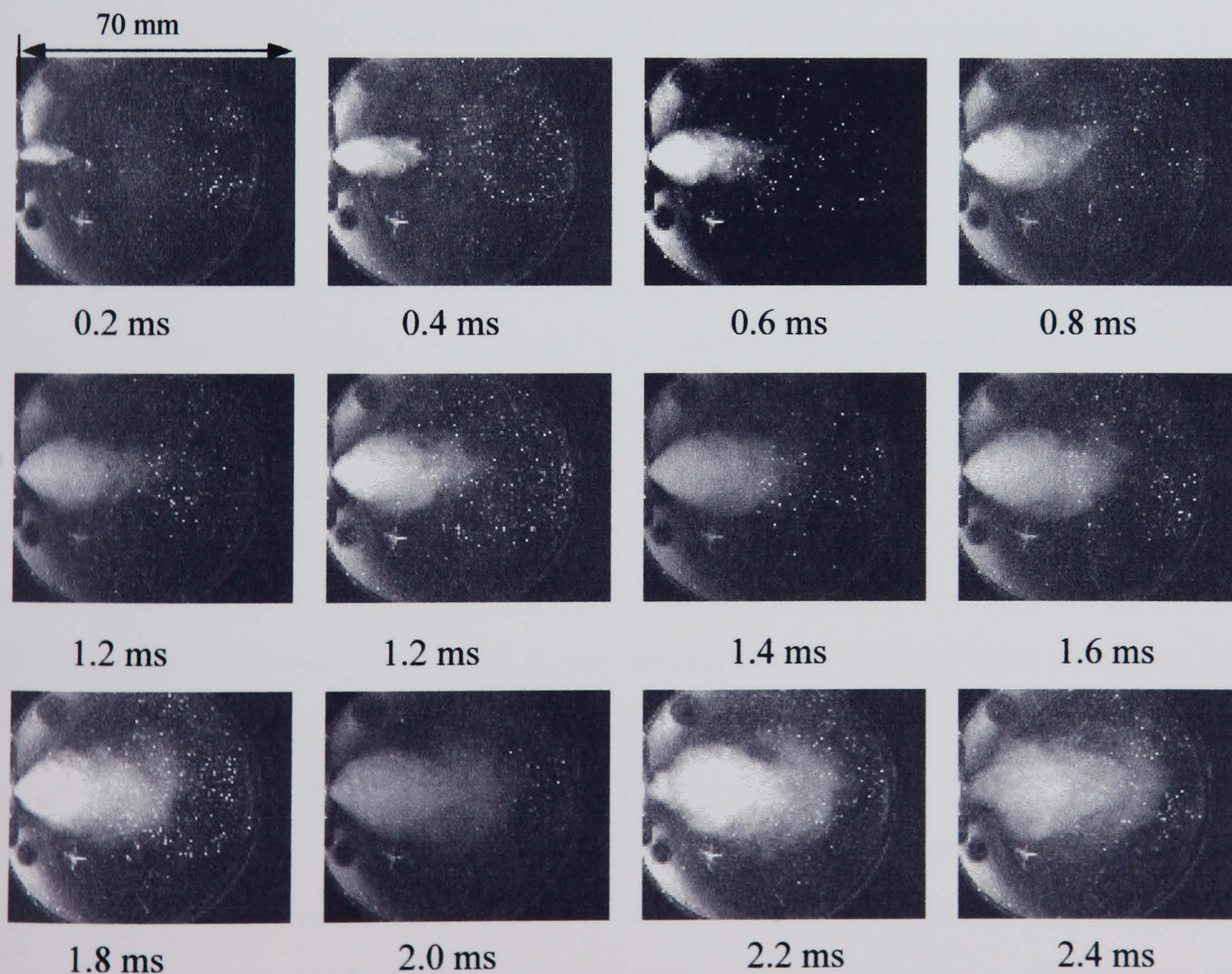


Figure 4-22 Spray images at different times inside the constant volume chamber under 4.5bar and 293 K ambient pressure and temperature, $P_{inj}=70$ bar, $t_{inj}=2$ ms

Although the above equation considers only the drag force acting on a single droplet neglecting the droplet-to-droplet interaction, it shows that droplets with larger diameter decelerate less than the smaller ones and that droplets decelerate in half the time when the density of the air is doubled. It should be noted that this equation does not consider empirically a drag coefficient which is a function of the droplet Reynolds number.

Figure 4-24 shows four spray images, taken at the same time of 1.5 ms and at the same injection pressure, for three chamber conditions. In order to identify more clearly the geometry of the spray for these various conditions, the contours of the spray images at four times of 0.8, 1.2, 1.6 and 2.2 ms have been calculated and superimposed in the same graph, presented in Figure 4-25. These images reveal that the initial spray angle does not change for the three different chamber conditions, and that the difference between one image and another with increasing time is the location of the collapse of the main body of the spray. As the time increases, the distance from the nozzle exit to the location where the spray collapses becomes larger

In order to characterise the difference in the spray geometry between the three operating conditions, contours have been drawn for spray images corresponding to the same time of 1.4 ms and superimposed in the same image (Figure 4-26). This figure shows that the nozzle spray angle is independent of the ambient conditions and therefore it depends only on the nozzle design, as reported earlier by Nouri et al. (1999). Although these results contradict those presented recently by Comer et al (2001) who claimed that the spray angle decreases linearly with ambient pressure, it is believed that the way and the location of measuring the spray angle are different in the two cases. Probably, these results may represent the average angle along the spray and not the nozzle spray angle. Figure 4-26 also shows a slight increase in the spray volume as the ambient temperature increases towards the end of its travel. This increase could be attributed to spray vaporisation with the vapour following the airflow and higher concentration close to the actual contour of the spray. Further evidence of the effect of air temperature on spray geometry is the appearance of the backflow involving small droplets and liquid vapour at the outer edge of the main body of the spray, which is attributed to the air-pattern around the spray.

Comparison of the spray penetration for the three cases reveals that there is about 12% decrease in spray penetration at 1.6 ms when the chamber pressure increases to 4.5 bar, as in cases (1) and (2); this comparison is based on the tip of the spray in the

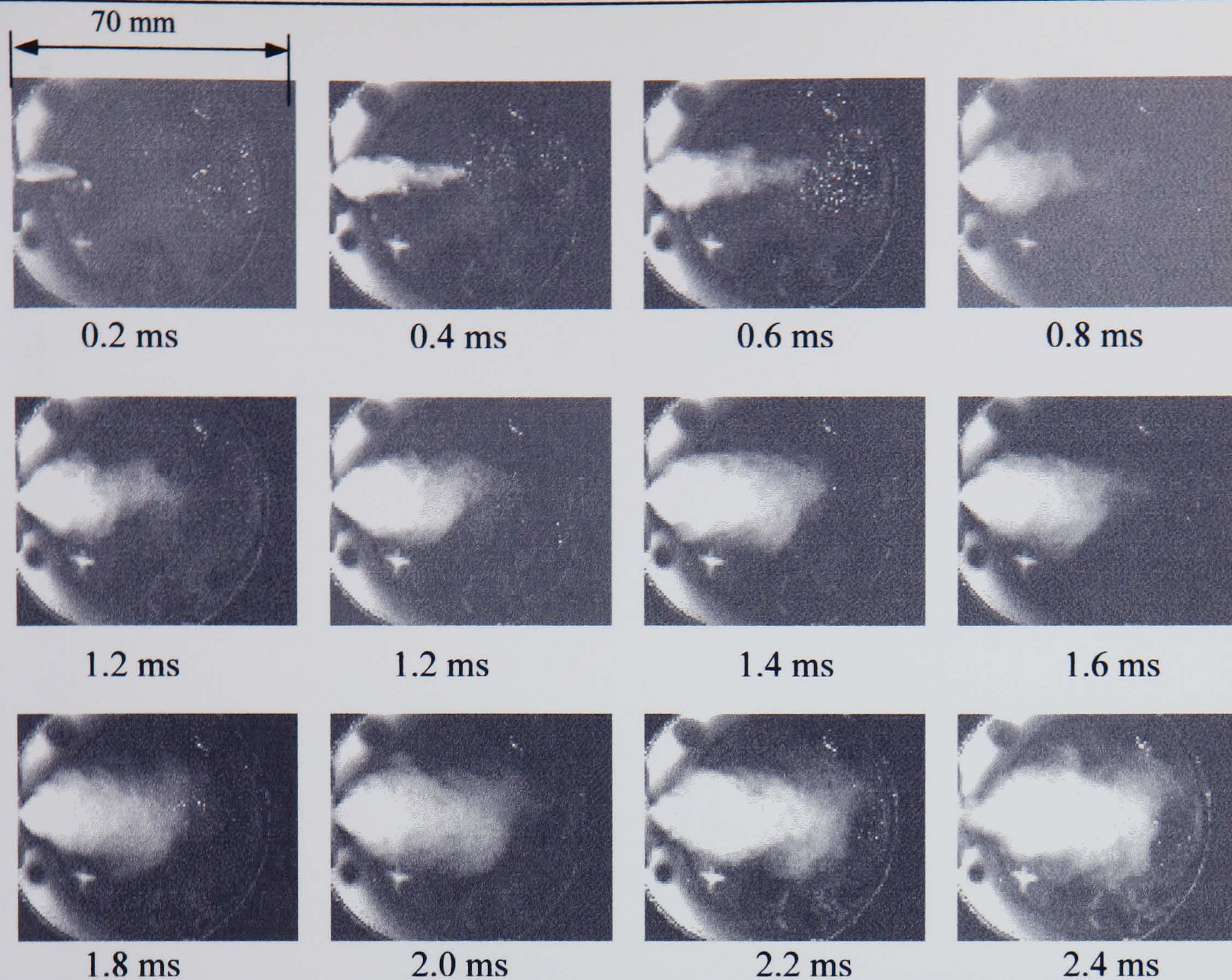
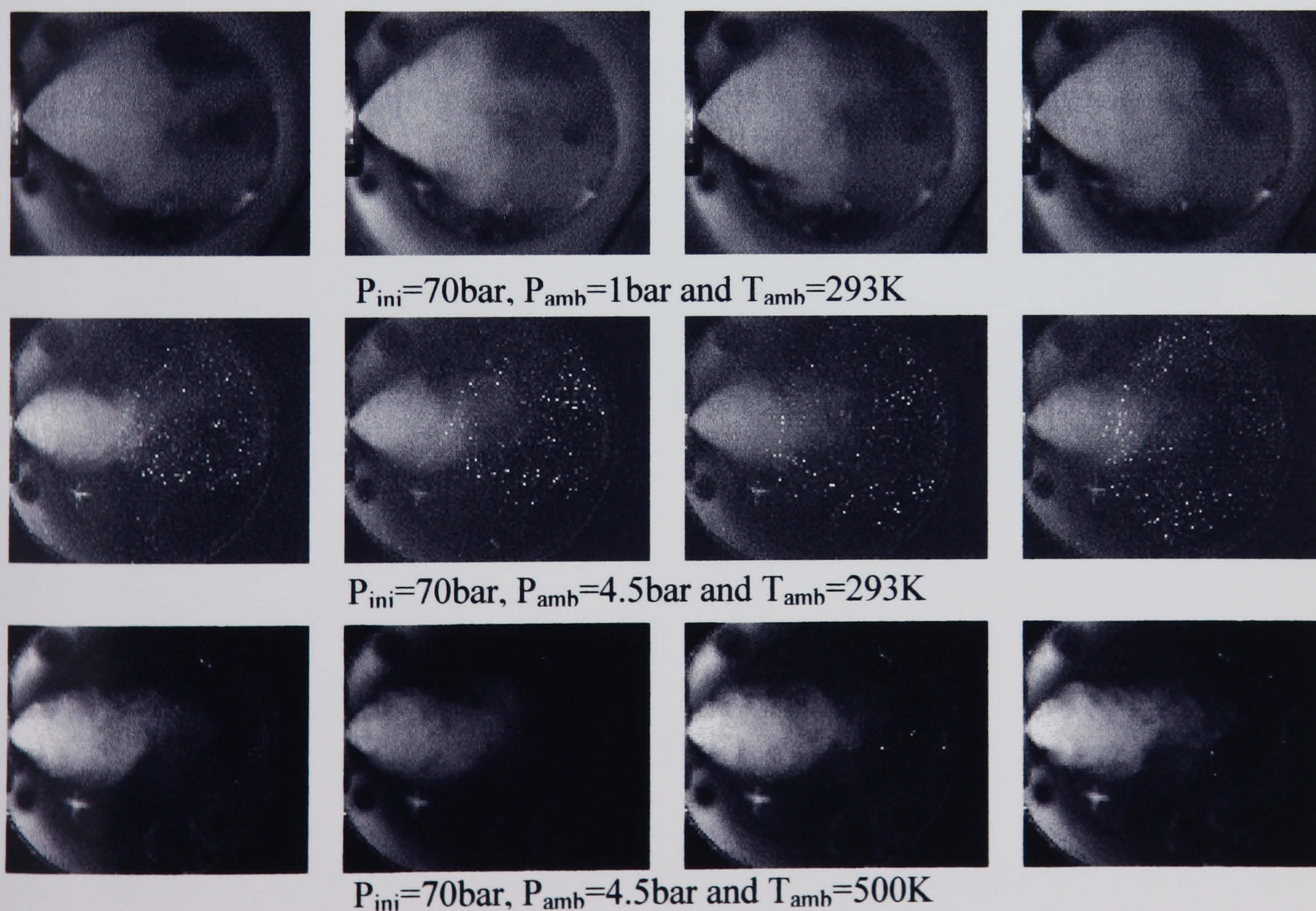


Figure 4-23 Spray images at different times inside the constant volume chamber under 4.5bar and 500K ambient pressure and temperature, $P_{inj}=70$ bar, $t_{inj}=2$ ms.



c) Images from the same injector at the same time 1.5 ms for the three conditions

Figure 4-24 Spray images show the effect of spray-to-spray variation and the effect of ambient conditions on spray geometry.

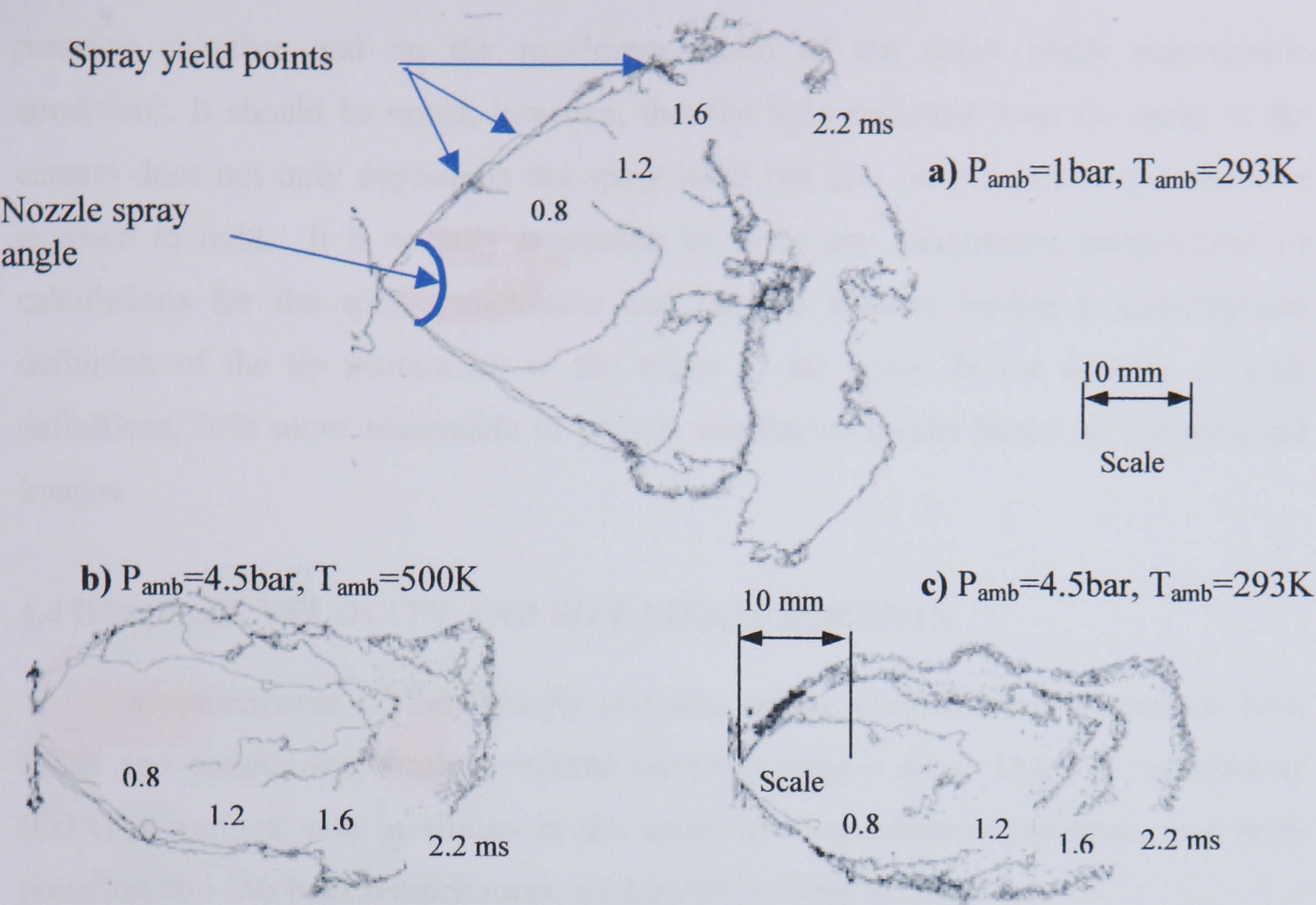


Figure 4-25 Spray images contours at different times inside the constant volume chamber under different ambient conditions, $P_{inj}=70\text{ bar}$, $t_{inj}=2\text{ms}$.

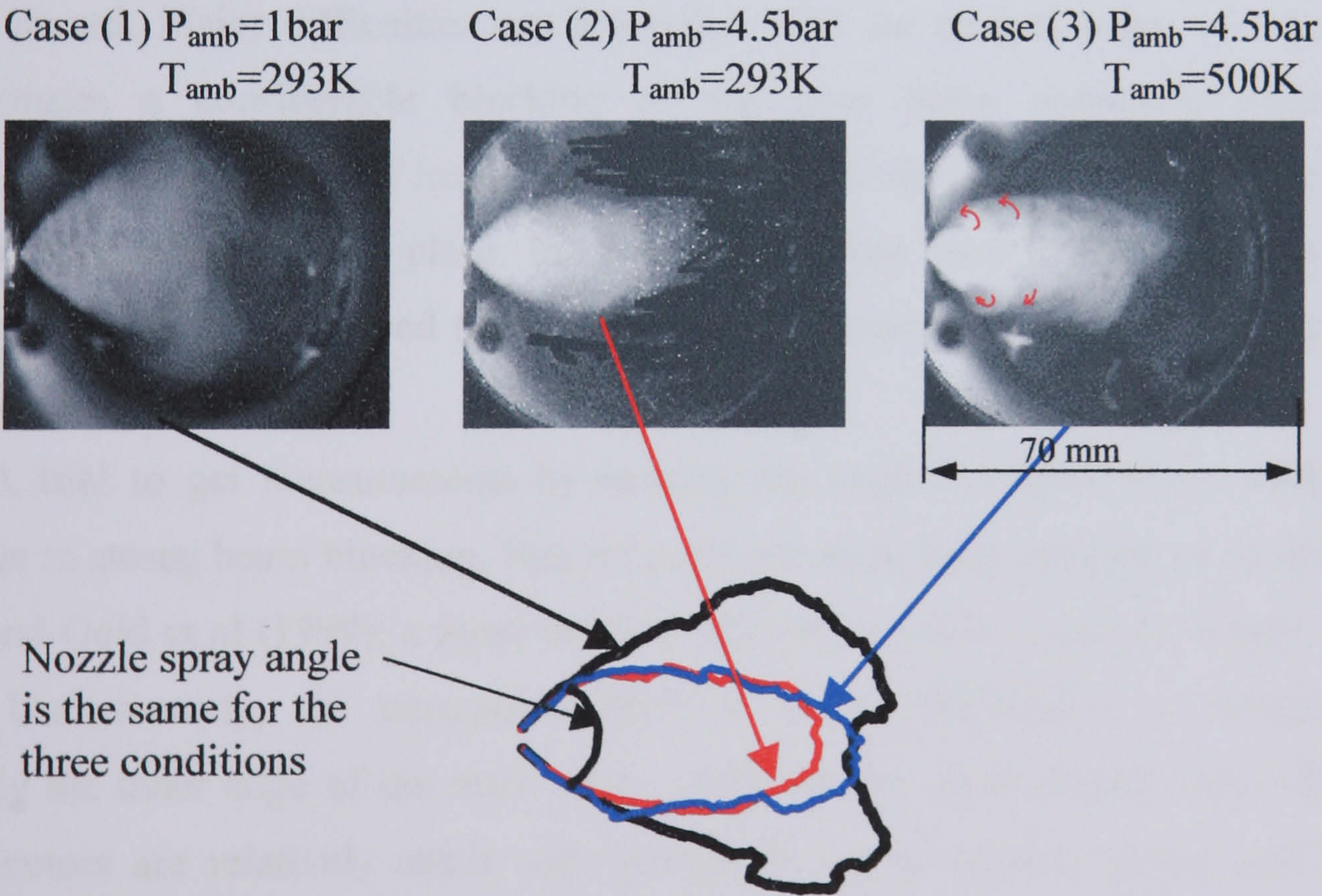


Figure 4-26 Spray images and contour under different ambient pressures and temperatures inside the constant volume chamber at 1.6ms , $P_{inj}=70\text{ bar}$, $t_{inj}=2\text{ms}$.

pressure chamber and on the maximum width of the spray under atmospheric conditions. It should be noted, however, that the light reflected from the spray to the camera does not only depend on the spray itself but also on the area of the chamber exposed to light. It is actually inaccurate to make any quantitative comparisons or calculations for the spray penetration and volume without having a unambiguous definition of the tip penetration or the edges of the spray. In the absence of such definitions, it is more reasonable to present qualitative results based on the obtained images.

4.4 DROPLET VELOCITY AND SIZE MEASUREMENTS

Measurements of the velocity and size of the droplets were performed both inside and outside the constant volume chamber using a phase Doppler anemometer (PDA) at various axial positions in the spray for two injection pressures, two back pressures and two back temperatures, all defined in Table 4-1.

4.4.1 Application of PDA to DISI Spray Measurements

PDA measurements in a gasoline direct-injection swirl injector are not straightforward. Major difficulties are associated with the conical shape of the spray which causes a considerable blocking of the laser beam especially when the measurement point is located inside the spray cone. In this case, a dramatic drop in measured data rate may take place. In order to minimise laser beam attenuation, the control volume has been moved from the centre of the spray upwards, until it reached the upper edge of the spray.

A trial to get measurements by moving the control volume horizontally has failed due to strong beam blocking. Similar problems have been reported by Nouri et al (2000) and Gold et al (1999); a more detailed discussion can be found in Wigley et al. (1998). Unfortunately, the attenuation problem caused difficulties in determining accurately the inner edge of the main spray. Although the spray droplet sizes of DISI swirl injectors are relatively small with associated higher surface tension and more spherical shape than the larger droplets leading to more accurate PDA droplet size measurements, there are still some difficulties in eliminating the source of noise existing in the lower range of the droplet size measurements. Besides these difficulties, there are other sources of error when using the PDA system, such as the multiple occupancy of

droplets in the control volume especially close to the nozzle exit. A partial solution to this problem is to use very small control volumes to reduce the ‘multiple occupancy’ issue, or small apertures (50 μm) to allow focusing on a fraction of the control volume. The non-sphericity of the liquid ligaments present in the spray is another parameter that influences the accuracy of the measurements, especially near the nozzle exit.

4.4.2 PDA Measurements Points

Simultaneous measurements of droplet velocity and size were conducted using a 1-D PDA system. The radial velocity was measured by rotating the injector at 90 degree in the atmospheric measurements something which was not possible inside the constant volume chamber where only axial velocity measurements were obtained. The number of measurement attempts used in this study was 25,000 under atmospheric conditions and 15,000 inside the constant-volume chamber over a time window of 4ms. Mean droplet velocities and diameters have been estimated at each measurement point within 20 time bins of 0.2 ms width. About 11 radial points have been measured across the spray moving from the spray axis to the spray edge at axial distances of 5,10,15,20 and 30mm axial distances from the nozzle along the spray axis; in all cases the injection duration was set at 2ms.

Because of the optical limitations of the chamber interior, axial velocity measurements under high pressure/temperature conditions were limited to 5, 10 and 15 mm axial distances from the nozzle exit. The measurements points throughout spray are shown in Figure 4-27 and summarised in Table 4-3.

4.4.3 PDA Results

In order to obtain a complete picture of the spray development, the temporal and spatial droplet diameter and velocity characteristics are described below separately and in sufficient detail to allow validation of CFD spray models.

4.4.3.1 Temporal Analysis

Typical results for the PDA system at different points within the spray, i.e. at different axial positions from the spray axis for a 70 bar injection pressure and atmospheric ambient conditions are shown in Figure 4-28. The scattered points represent individually measured droplet velocities and sizes during a time window of

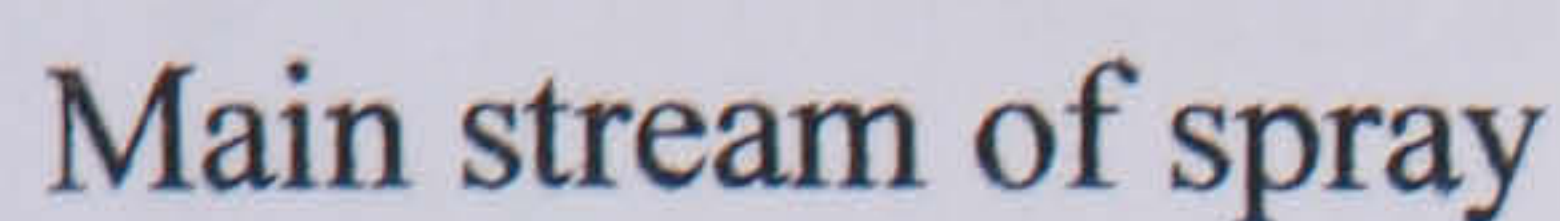


Figure 4-27 PDA measurement points superimposed on a spray image at 70 bar injection pressure and atmospheric ambient conditions

X (mm)	R (mm) for Case (1) $P_{inj}=70$ bar $P_{amb}=1$bar $T_{amb}=293$K	R (mm) for case (2) $P_{inj}=70$ bar $P_{amb}=4.5$bar $T_{amb}=293$K	R (mm) for case (2) $P_{inj}=70$ bar $P_{amb}=4.5$bar $T_{amb}=500$K	R (mm) for Case (4) $P_{inj}=20$ bar $P_{amb}=1$bar $T_{amb}=293$K
5	5,5.5,5.75,6	2,3,4,5,6	4,5	5,5.5,6,6.5
10	0,2,4,6,7.5,8.25 8.5,9,9.5,10,10.5	2,4,6,6.5,7, 7.5,8,9	2,4,6,7,8,9	0,3,6,8,9,10,11
15	0,6,9,,11,11.5,12, 13,14,15,15.5	0,3,6,7,8,9, 10	0,3,7,8,9	0,3,5,9,11,12, 13,14,15.16
20	0,3,6,9,12,14,15, 16,17,18,20			0,4,8,12,16, 18,20
30	0,5,10,15,17.5, 20,22.5,25,27.5,30			0,5,10,15,20, 25,27.5,30

Table 4-3 Locations of the PDA measurement points inside the spray for the four conditions

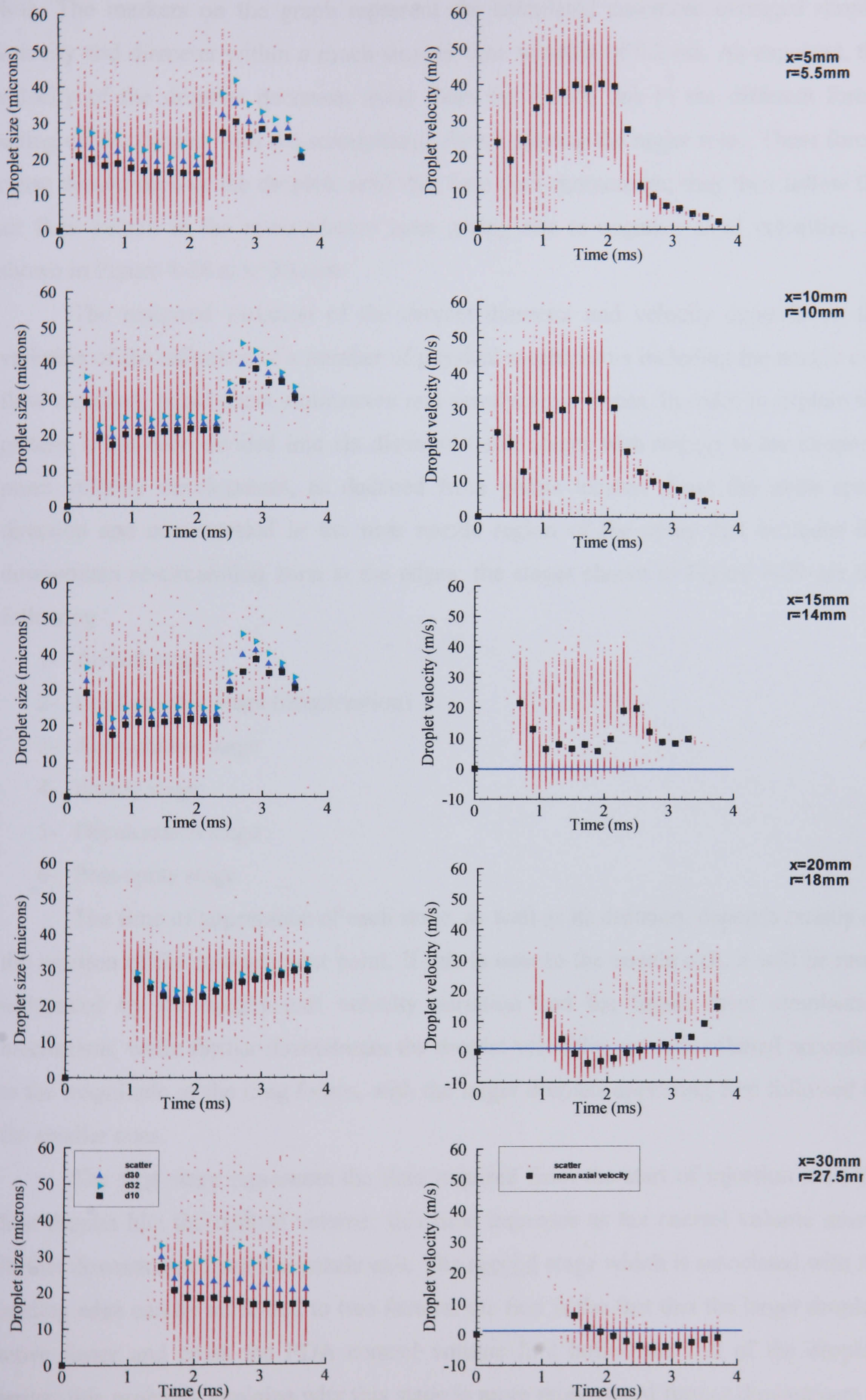


Figure 4-28 Temporal variation of droplet size and axial velocity at different axial distances within the main stream of spray

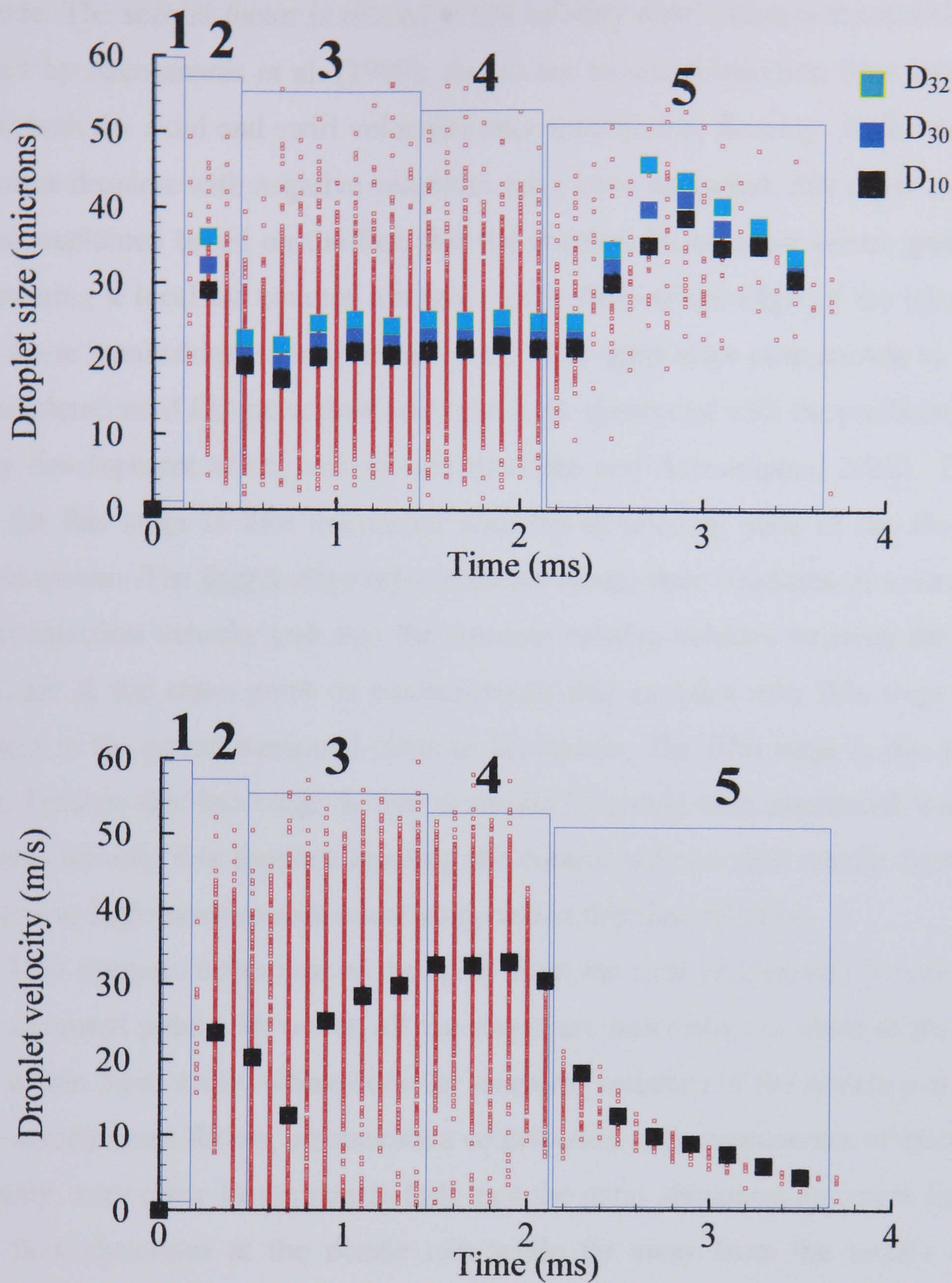
4ms. The markers on the graph represent the calculated ensemble averaged droplet velocity and diameter within a much smaller time window of 0.2 ms. As expected, the velocity of the droplets decreases away from the nozzle due to the different forces acting on the droplet with the aerodynamic forces playing the major role. These forces cause deceleration of the droplets until they lose their momentum; they then follow the air flow pattern in the re-circulation zone giving rise to negative axial velocities, as shown in Figure 4-28 at $x=30$ mm

The temporal variation of the droplet diameter and velocity depends on the variation of the influence of a number of physical mechanisms including the nozzle exit flow variation, liquid sheet atomisation and aerodynamic forces. In order to explain the pattern, it has been divided into six distinguishable stages with respect to the temporal mean velocity development, as deduced from points located along the main spray direction and concentrated in the near nozzle region of the spray that excludes the downstream re-circulation zone at the edges; the stages shown in Figure 4-29 are the following :

- 1- Arrival stage
- 2- Leading edge stage (deceleration)
- 3- Acceleration stage
- 4- Steady stage
- 5- Deceleration stage
- 6- Post-spray stage

The time of appearance of each stage, as well as its duration, depends mainly on the location of the measurement point. If this is near to the nozzle exit, it will be more influenced by the nozzle exit velocity variation and the liquid sheet atomisation mechanism, while further downstream the droplet velocities are redistributed according to the magnitude of the drag forces, with the larger droplets appearing first followed by the smaller ones.

The first stage represents the time required from the start of injection until the first droplet hits the control volume; this time increases as the control volume moves further downstream from the nozzle exit. The second stage which is associated with the leading edge can be attributed to two factors: the first is the fact that the larger droplets move faster and cross the PDA control volume first before the rest of the droplets arrive; this probably explains why this stage is more pronounced further downstream of



- 1- Arrival stage
- 2- Leading edge stage
- 3- Acceleration stage
- 4- Steady stage
- 5- Deceleration stage
- 6- Post-spray stage

Figure 4-29 The six distinguishable stages in the temporal variation of the ensemble average axial velocity of droplets and their corresponding average diameter within the main stream of the spray $x=10\text{mm}$, $r=10\text{mm}$, $P_{inj}=70\text{ bar}$, $t_{inj}=1.5\text{ ms}$, $P_{amb}=1\text{ bar}$

the nozzle. The second factor is related to the velocity distribution at the nozzle exit, as predicted by Arcoumanis et al. (1999), due to the transient injection flow rate and the fact that both the axial and swirl velocities take sometime to develop. At the end of this stage, some droplets with negative velocities have been measured; this negative velocity could be explained based on the fact that the droplets in the main stream push the air away creating a local air counter rotating vortex flow at the edge of the spray which pushes some small droplets into the air path. The third stage corresponds to the time when acceleration of the mean flow takes place, in agreement with the predictions of the velocity development at the nozzle exit (Gavises and Arcoumanis, 2002). The main reason for this stage is thus associated with the developing state of the flow in the injection system. The fourth stage represents the steady state condition as a result of the constant injection velocity and also the constant relative velocity between the droplets and the air at the same point of measurement; this explains why this stage is more prominent in the points measured close to the nozzle. The fifth stage is due to needle closing. Further, this last stage, is due to needle bouncing with associated low droplet velocities and only few droplets crossing the control volume after needle closure, thus giving rise to high measurement uncertainty within this time window.

This dynamic behaviour of the spray from the start of injection is noticeable at all the measured points. However, all the stages are more obvious close to the injector nozzle within 5mm away, where both the temporal variation of the nozzle exit velocity and the aerodynamic forces, are making a contribution to the appearance of these stages. Apparently, very close to the nozzle hole exit the spray dynamics are more influenced by the flow dynamics at the nozzle exit while far away from the nozzle exit, the aerodynamic forces play a major role in altering the spray dynamics.

Far away from the nozzle exit at $x=30$ mm, the droplets are exposed for longer time to aerodynamic forces which decelerate the smaller droplets faster than the larger ones; therefore large droplets arrive first at the control volume followed by the smaller ones. This redistribution of the droplets' velocity and size becomes more distinct as their distance from the injector nozzle increases, where the leading edge stage becomes dominant; accordingly the spray further away from nozzle exit seems to be continuously decelerating during the time of the measurements. These trends are presented in Appendices 4-A, 4-B, 4-C and 4-D which show the temporal variation of the ensemble-averaged velocity and diameter during the measurement time (4ms) at different axial

distances from the nozzle exit and radial distances from the spray centre line for four operating conditions.

Figure 4-29 also shows the droplet diameter scatter plot and the ensemble-averaged linear, volume and volume-surface diameters D_{10} , D_{30} and D_{32} calculated on the basis of the same time step of 0.2 ms. This graph reveals that the average droplet diameter remains almost constant throughout the most of previously mentioned stages except for the leading edge and post-spray stages; similar results have been reported by VanDerWege and Hochgreb (2000) and more recently confirmed by Allocca et al. (2001).

In order to reveal the effect of chamber pressure/temperature and the injection pressure on the spray characteristics, the measurement points have to be comparable. Since the geometry of the spray changes with varying chamber conditions, it is unrealistic to compare one measurement point at one condition with the same point at the same location but at different operating conditions. In order to overcome this problem and also minimise the quantity of data to be used to verify the spray mathematical model, all the measurement points across the spray at the same axial distance have been added together except the data collected inside the spray cone to avoid the pre-spray droplets, the entrained droplets and the uncertainty of measurements in this region. Therefore, only the droplets located within the main stream of the spray are counted. By averaging the diameter and velocity of these droplets during a time step of 0.5 ms, the spray can be represented by only one droplet at each time step and at each section (5, 10, 15, 20 and 30 mm) from the nozzle exit for the case of the atmospheric spray while for the spray in the chamber it is represented only by three droplets at 5, 10 and 15 mm. Since the arrival time of droplets to the spray sections are different due to different distances the droplets travel to reach the control volume, it is not possible to compare the results at different sections within a time step of 0.5 ms starting from the fuel injection; it seems more meaningful to start from the time the first droplet arrives at the control volume. Therefore, every section is considered to have its own starting time which is delayed from injection time by the arrival time of droplets.

Figure 4-30 shows the droplet size distribution of all droplets collected across the whole section at three axial distances of 5, 10 and 15 mm and for four operating conditions at four successive time steps of 0.5, 1, 1.5 and 2 ms. The graph at 20 bar injection pressure shows significantly larger droplet diameters compared to the 70 bar

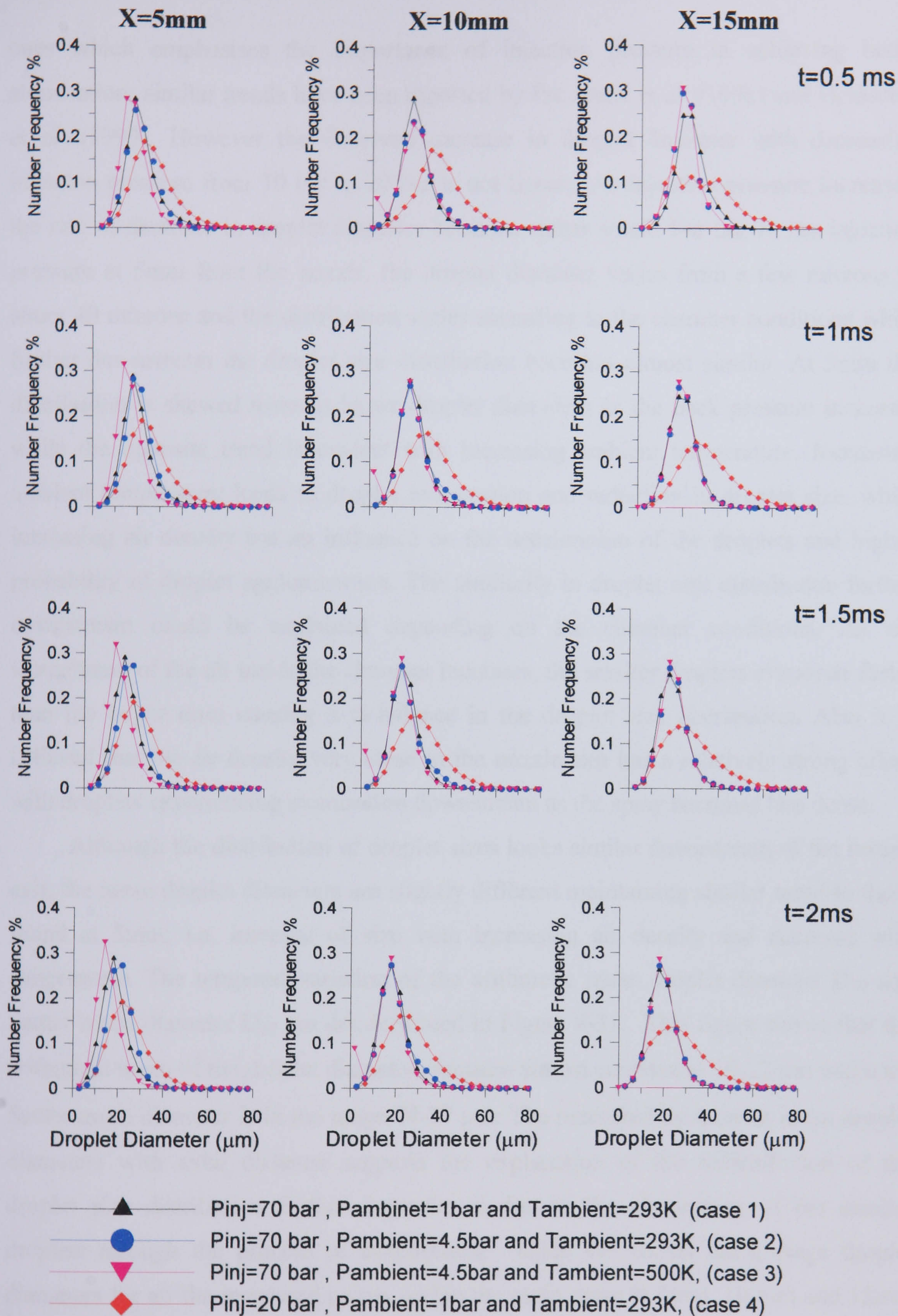


Figure 4-30 Droplet size distribution for the four operating conditions at three different axial distances within the main stream of the spray at different times

ones which emphasises the importance of injection pressure in achieving better atomisation; similar trends have been reported by Preussner et al. (1998) and Hentschel et al. (1999). However the observed increase in droplet diameter with decreasing injection pressure from 70 bar to 20 bar is not linear. As injection pressure increases, the rate of decrease in droplet diameter becomes rather small. For the 70 bar injection pressure at 5mm from the nozzle, the droplet diameter varies from a few microns to about 40 microns and the distribution varies according to the chamber conditions while further downstream the droplet size distribution becomes almost similar. At 5mm the distribution is skewed towards larger droplet diameters as the back pressure increases while the opposite trend is evident with increasing ambient temperature. Increasing ambient temperature leads to droplet evaporation and reduction in droplet size, while increasing air density has an influence on the deceleration of the droplets and higher probability of droplet agglomeration. The similarity in droplet size distribution further downstream could be explained depending on the chamber conditions. As the temperature of the air inside the chamber increases, the smaller droplets evaporate faster than the larger ones causing a re-balance in the droplet size distribution. Also it is believed that the air density very close to the nozzle exit has a relatively strong effect with droplets experiencing atomisation downstream as the spray becomes less dense.

Although the distribution of droplet sizes looks similar downstream of the nozzle exit, the mean droplet diameters are slightly different maintaining similar trend to those found at 5mm; i.e. increase of size with increasing air density and decrease with temperature. The temporal variation of the arithmetic mean droplet diameter D_{10} and Sauter mean diameter D_{32} are demonstrated in Figure 4-31. This figure shows that the arithmetic mean of the droplet diameters remains almost constant at 19-22 μm while the Sauter mean diameter is in the range 24-27 μm . The measured increase in mean droplet diameters with axial distance supports the explanation of the redistribution of the droplet size distribution further downstream due to the elimination of the smaller droplets through the process of evaporation. Table 4-4 shows the average droplet diameters for all the measured points within the main spray at 5mm, 10 mm and 15mm integrated over the whole injection duration 2 ms for four operating conditions. According to this table, it could be concluded that increasing the air density not only causes a collapse of the spray but also an increase in mean droplet diameter of the whole spray that could be attributed to droplets coalescing. The increase in air

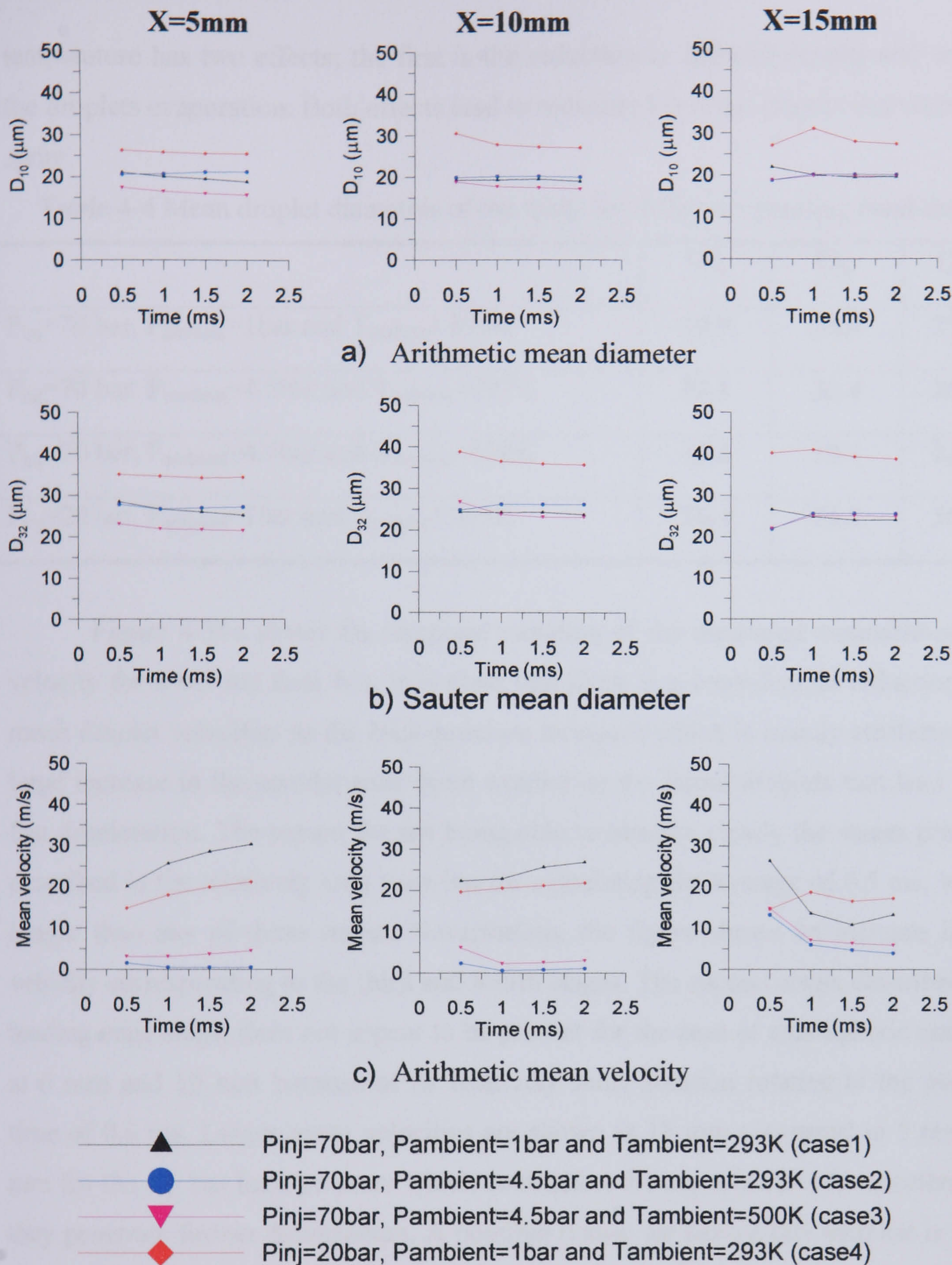


Figure 4-31 Temporal variation of mean droplet diameters and velocity for the four operating conditions within the main stream of the spray at three different axial distances

temperature has two effects; the first is the reduction in ambient density and second is the droplets evaporation. Both effects lead to reducing the mean droplet diameters of the spray.

Table 4-4 Mean droplet diameters of the spray for different operating conditions.

	D ₁₀	D ₃₀	D ₃₂
P _{inj} =70 bar, P _{ambient} =1bar and T _{ambient} =293K	19.9	22.4	25.4
P _{inj} =70 bar, P _{ambient} =4.5bar and T _{ambient} =293K	20.4	23.4	26.6
P _{inj} =70 bar, P _{ambient} =4.5bar and T _{ambient} =500K	16.2	19.4	22.9
P _{inj} =20 bar, P _{ambient} =1bar and T _{ambient} =293K	26.4	31.1	36.6

Figure 4-31c shows the temporal variation of the measured ensemble-averaged velocity for a 0.5 ms time bin. It is clear that there is a considerable reduction in the mean droplet velocities as the back-pressure increases which is mainly attributed to the large increase in the aerodynamic force exerted on the liquid droplets that lead to their fast deceleration. The reason for not being able to observe clearly the stages previously described is the relatively long time bin for calculating the average of 0.5 ms, which is longer than any of these stages. Nevertheless, the figure shows an increase in mean velocity corresponding to the third and fourth stages. The second stage, described as the leading edge stage, does not appear to be present for the case of atmospheric conditions at 5 mm and 10 mm because of its relatively short duration relative to the averaging time of 0.5 ms. Larger mean velocities are shown at 15 mm compared to 5 mm or 10 mm for the 4.5 bar back pressure which contradicts the expected droplet deceleration as they penetrate further downstream. A possible reason for having this increase is that the points measured at 15 mm are not within the main body of the spray since the spray starts to collapse at 10 mm, as has been shown previously in the images. Therefore the droplets measured at x=15 mm are expected to belong to the group of droplets injected at the inner edge of the main stream which have relatively higher axial velocities.

It is worth mentioning that for such low velocity droplets at the back pressure of 4.5 bar where the corresponding Weber number is relatively small, the probability of droplet agglomeration is high. Unfortunately, modelling this kind of dense spray at high back pressures is rather complicated since the commonly-used single-droplet analysis may give unrealistic results. This highlights the need for further studies of the drag

forces exerted on droplet clouds under high pressure conditions and the associated droplet-to-droplet interaction.

4.4.3.2 Spatial analysis

In order to identify how the mean droplet diameter varies across the spray, the mean diameter has been calculated at each radial location in the main spray and at different axial positions from the nozzle. Figure 4-32 reveals a trend for the larger droplets to be located at the outer edge of the spray envelope for the 70 bar injection pressure and for two times from injection of 0.3 ms and 1.5 ms. This observation is more evident in the near-nozzle part of the spray, upstream of the re-circulation zone; similar results are shown in Figure 4-33 for 20 bar pressure at 0.7 and 1.5 ms which confirm the same trend at all sections where there is no evidence of recirculation. This could be attributed to the fact that larger droplets maintain their momentum penetrating radially further, while smaller droplets lose their radial momentum more easily and decelerate in the central part of the spray. The trend of the larger droplets to be located at the outer edge is also evident inside the high pressure chamber and it becomes more pronounced there due to the higher contribution of the aerodynamic forces. Similar results have been reported by Gold et al. (1999). It may be useful to mention that large droplets have also been measured in the pre-spray which appears around the axis at the early period of spray development (for more details see appendix 4-A, 4-B, 4-C and 4-D).

Measurements at 20 bar injection pressure also show that the droplets located inside the spray cone are relatively small and are entrained by air due to the locally low pressure. However at 70 bar injection pressure it proved impossible to measure statistically a representative number of droplets, although the few droplets acquired had relatively small sizes. The reason for not extending the measurement points at 70 bar to the spray axis is the high uncertainty in calculating the mean diameter due to the limited number of droplets. This problem, as already explained, is related to the beam attenuation effect which becomes more pronounced at the 70 bar injection pressure, when more fuel is injected and droplet sizes are generally smaller.

The velocity vectors along the measurement points are plotted in Figure 4-34 for the 70 bar injection pressure; two time steps are chosen for presentation 0.5 ms and 1.5 ms. The figure shows that at the early stages, the droplets at the centre line have the highest velocity since they represent the pre-spray. These droplets are of some concern

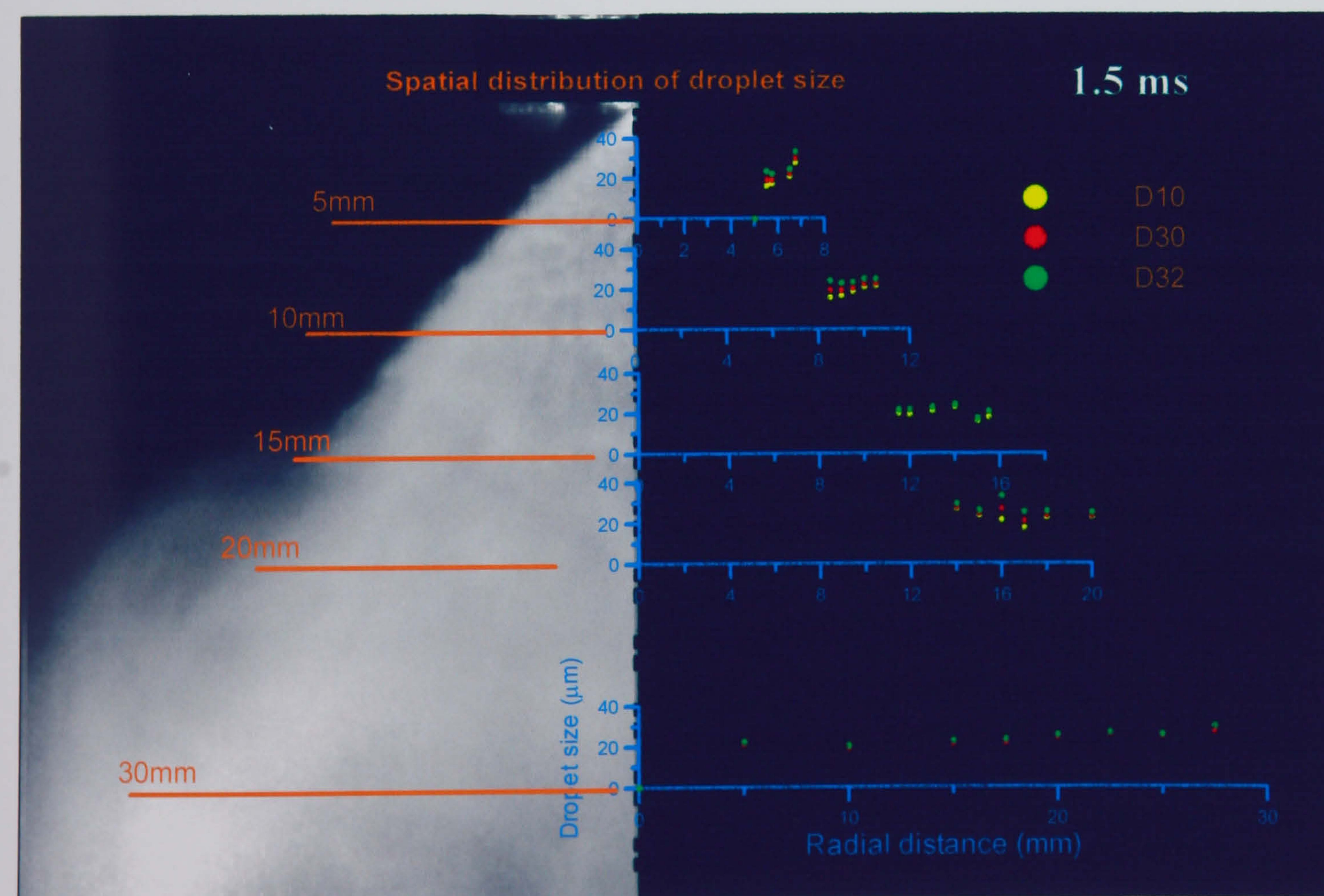
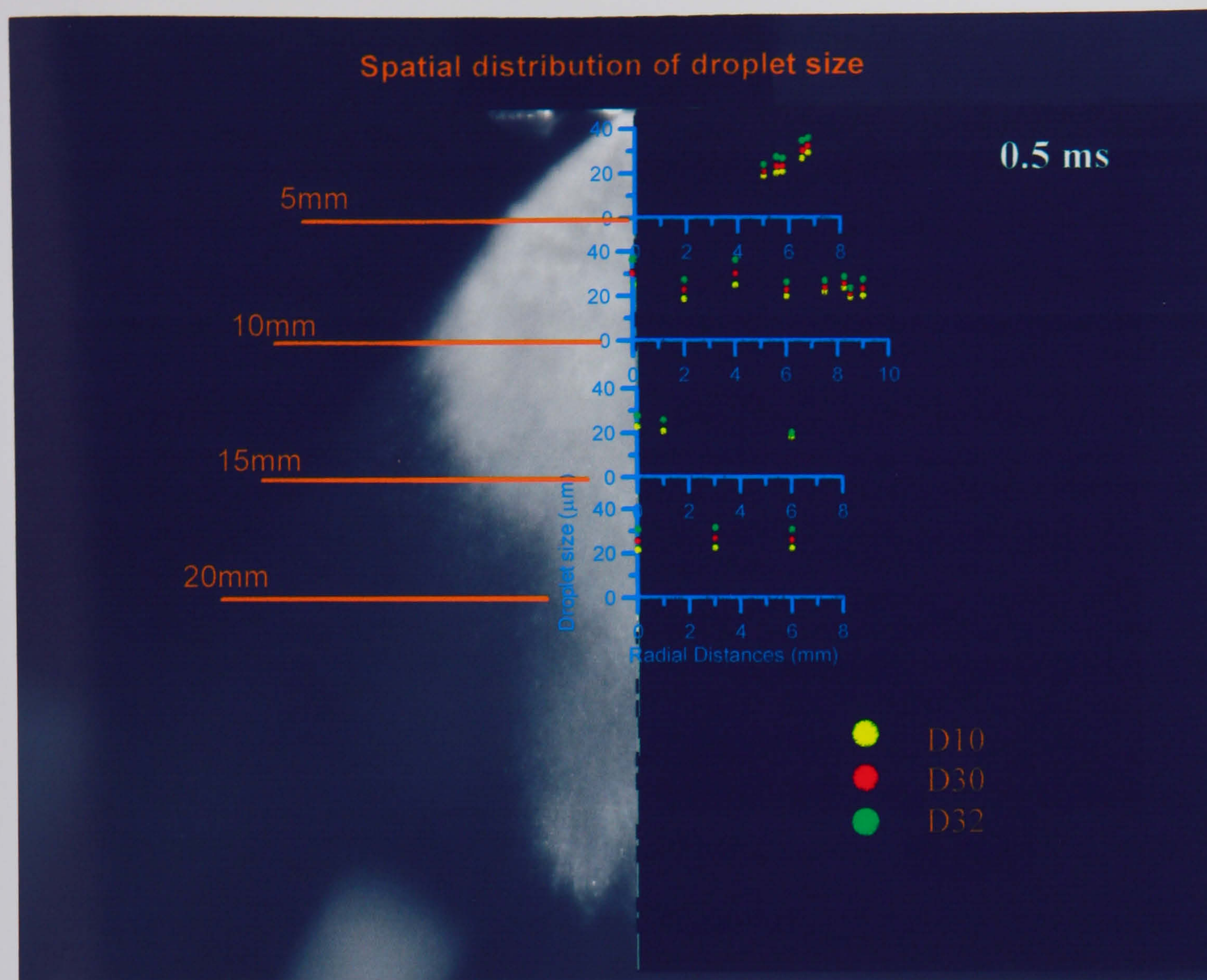


Figure 4-32 Spatial distribution of mean droplet diameters at 0.5 ms and 1.5 ms, $P_{inj}=70$ bar $P_{ambient}=1$ bar and $T_{ambient}=293\text{K}$.

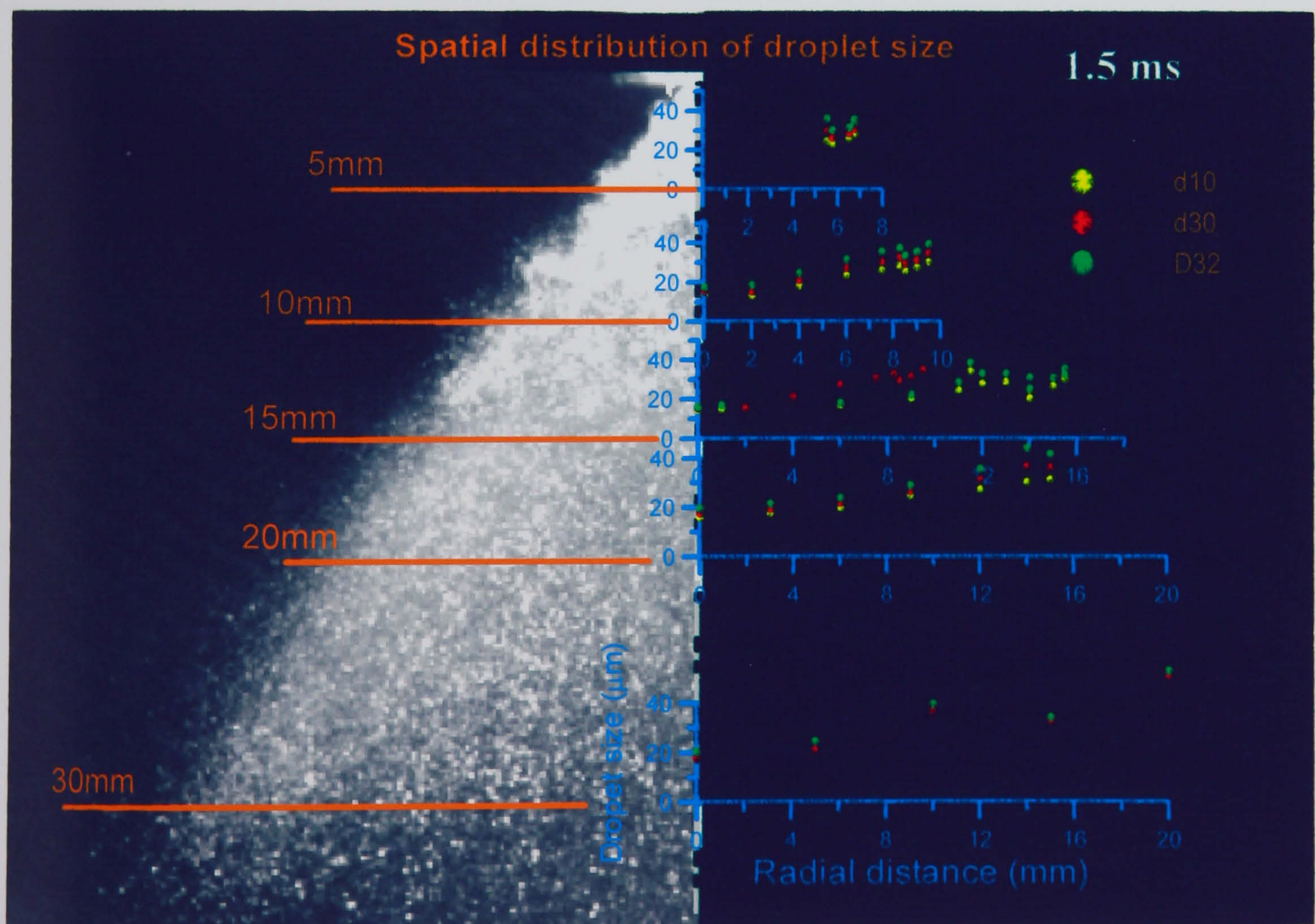
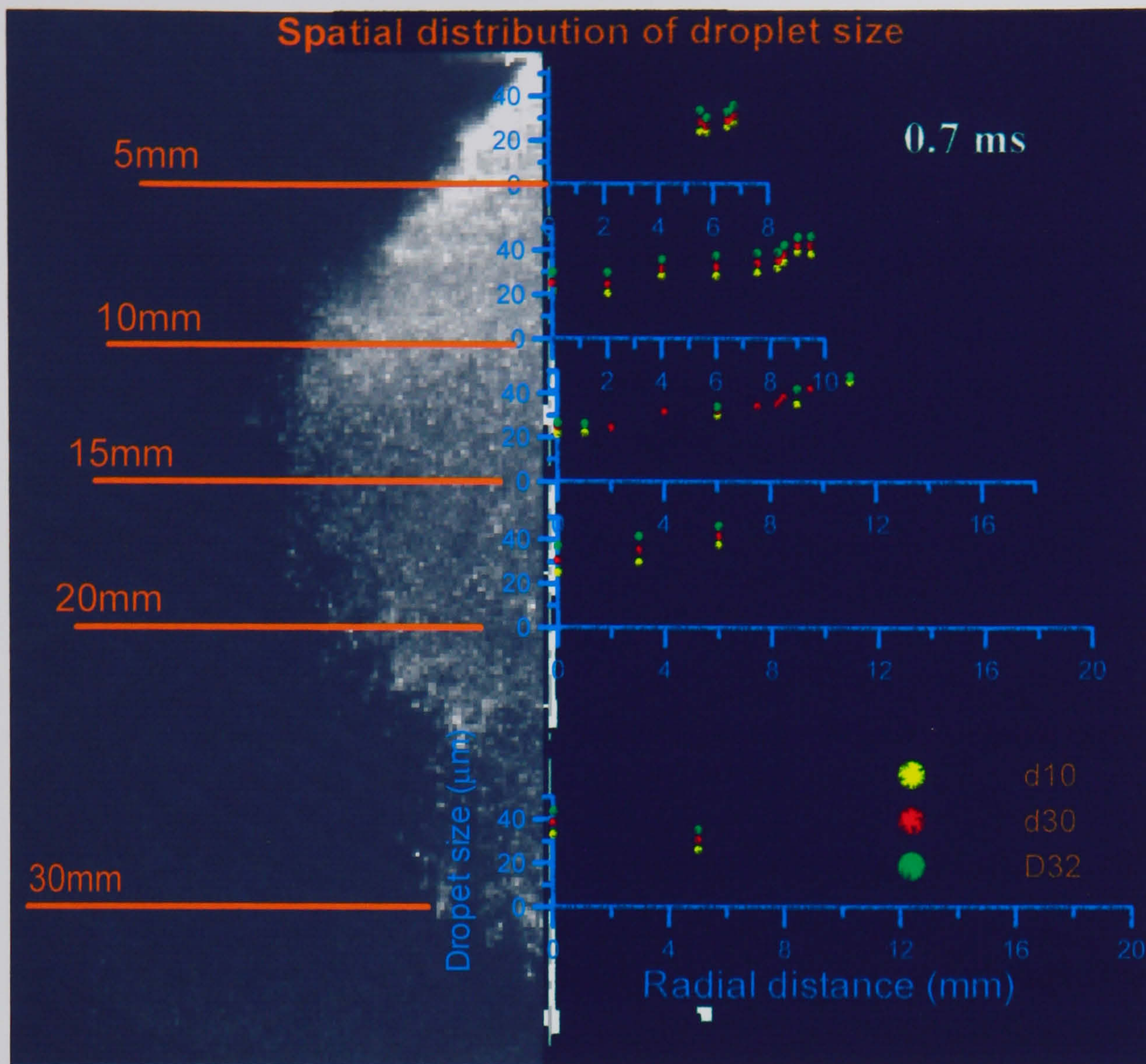


Figure 4-33 Spatial distribution of mean droplet size at 0.7 ms and 1.5 ms, $P_{\text{inj}}=20$ bar, $P_{\text{ambient}}=1$ bar and $T_{\text{ambient}}=293\text{K}$.

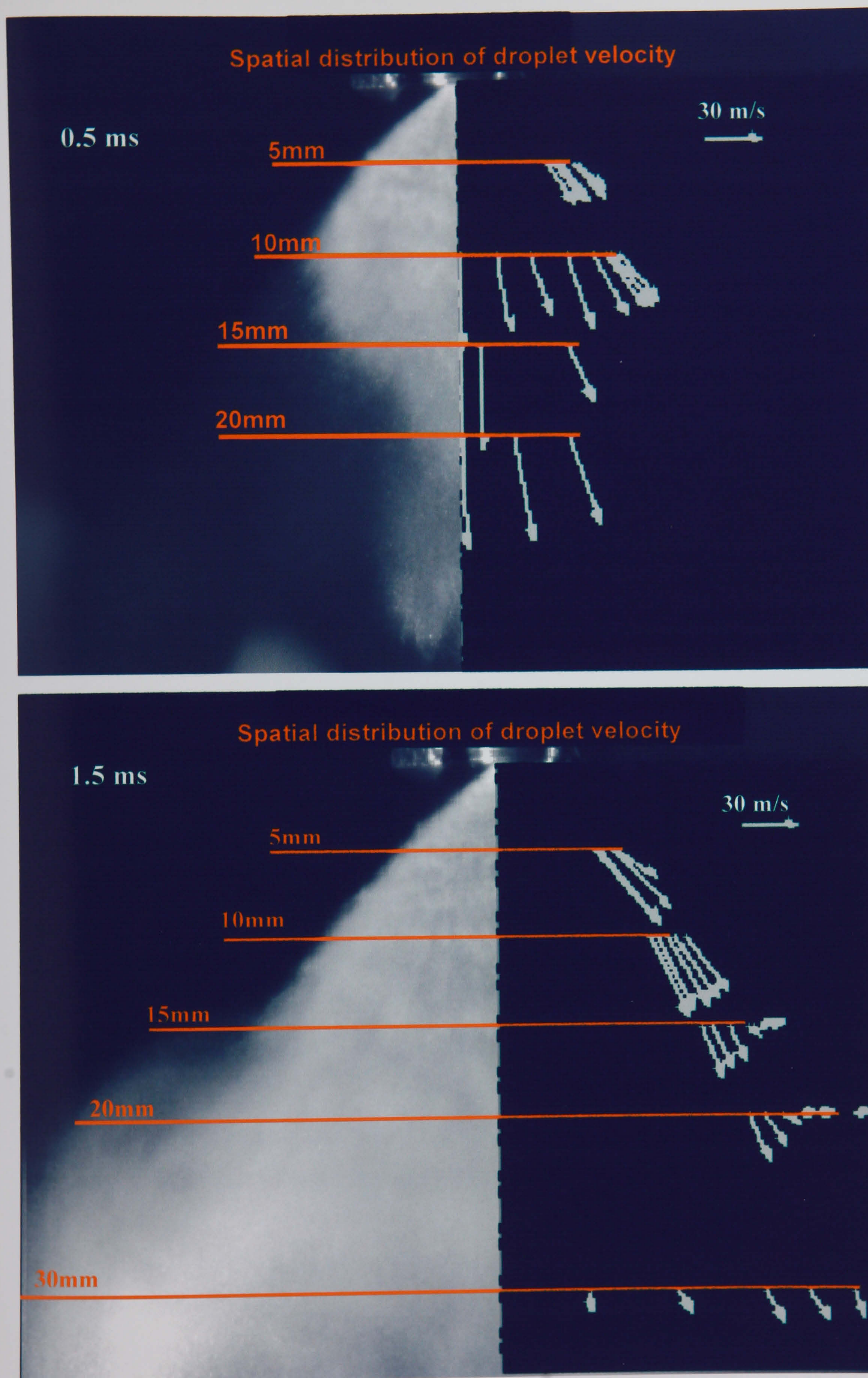


Figure 4-34 Spatial distribution of mean droplet velocity at 0.5 ms and 1.5 ms, $P_{inj}=70$ bar, $P_{ambient}=1$ bar and $T_{ambient}=293K$.

inside an engine since some of these droplets have a large size and impinge on the piston while the rest of the spray lags behind. It is believed that the droplets representing the pre-spray are mainly responsible for the unburned hydrocarbon emissions since large droplets need relatively long time to evaporate. As has been discussed before the pre-spray is created during the needle opening period and can represent a significant fraction poorly atomised fluid depending on the nozzle geometry.

For the case of fully developed spray (1.5 ms), the velocity vectors show that the mean droplet velocity decreases as droplets move towards the edge of the spray. Accordingly, droplets of larger size move towards the edge with relatively smaller mean velocity while smaller droplets possess higher velocity and are located in the central region. In addition, the velocity vectors show that the radial-to-axial velocity ratio increases with increasing radial distance. Further downstream, small droplets trapped inside the recirculation zone cause the mean droplet velocity to drop and a negative mean velocity has been recorded at some measurement points. The velocity vectors for the 20 bar injection pressure shown in Figure 4-35 show similar trends with measurements extended up to the central line. The measurements during the fully developed spray reveal that the velocity of the droplets located inside the cone are not only small in size but also have lower velocity. However, this is not the case during the spray developing phase when larger velocities are associated with droplets located closer to the spray centre line. Unfortunately at the near-nozzle position of 5mm the experimental uncertainty is high because the spray is dense and beam attenuation and blocking are significant which makes accurate measurements possible only at the outer edges of the spray.

4.5 CONCLUSIONS

Spray images have revealed the macroscopic structure of the spray in the form of distinct stages identified during testing under four operating conditions. In particular:

- Sprays exhibit four stages during its development following the injector opening delay time which has been estimated to be 0.3 ms:
 - (a) bulk of liquid jet located around the spray centreline
 - (b) asymmetric non hollow spray
 - (c) developing hollow cone spray
 - (d) developed hollow spray with counter flow vortex at the front of the spray

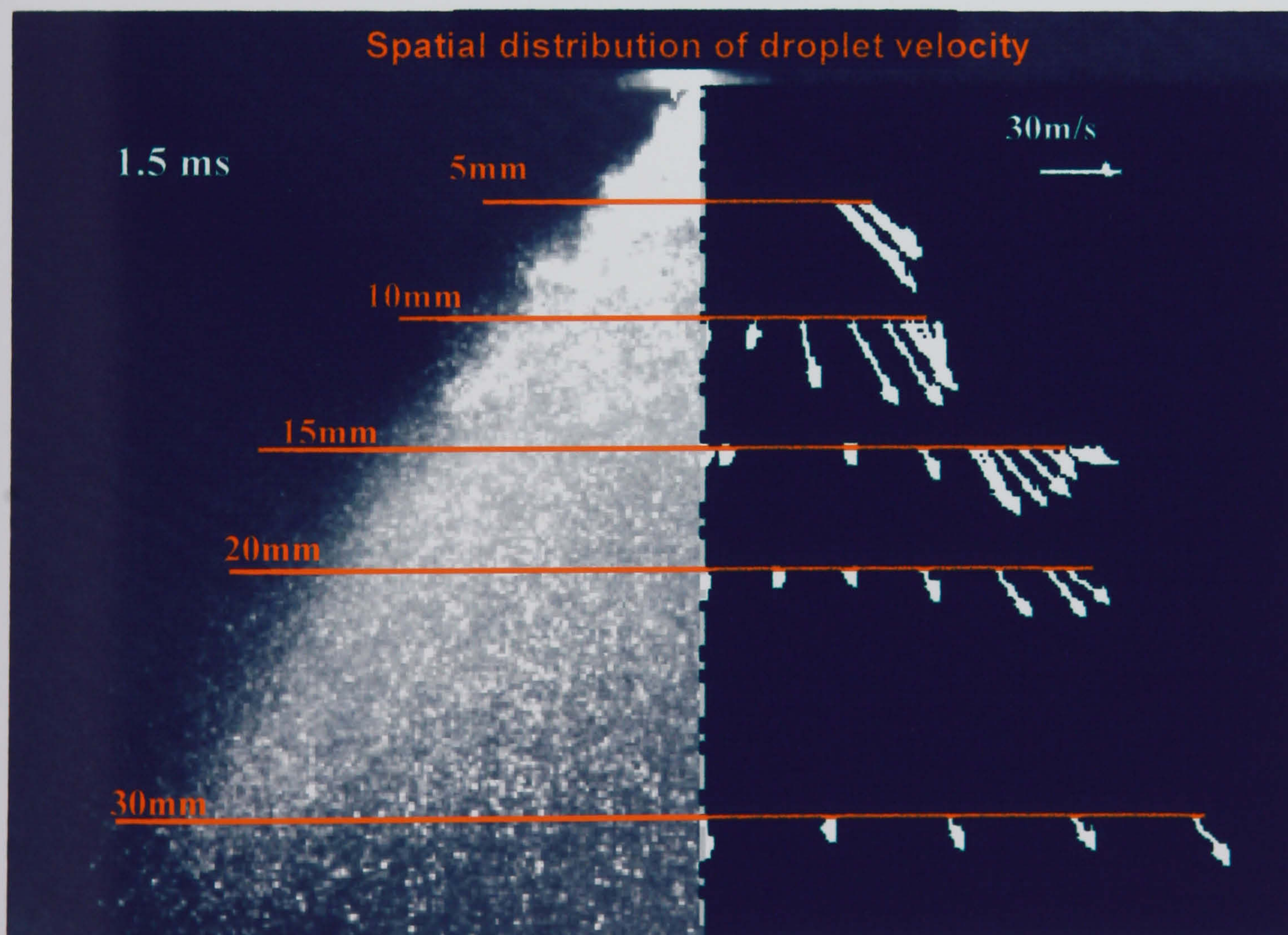
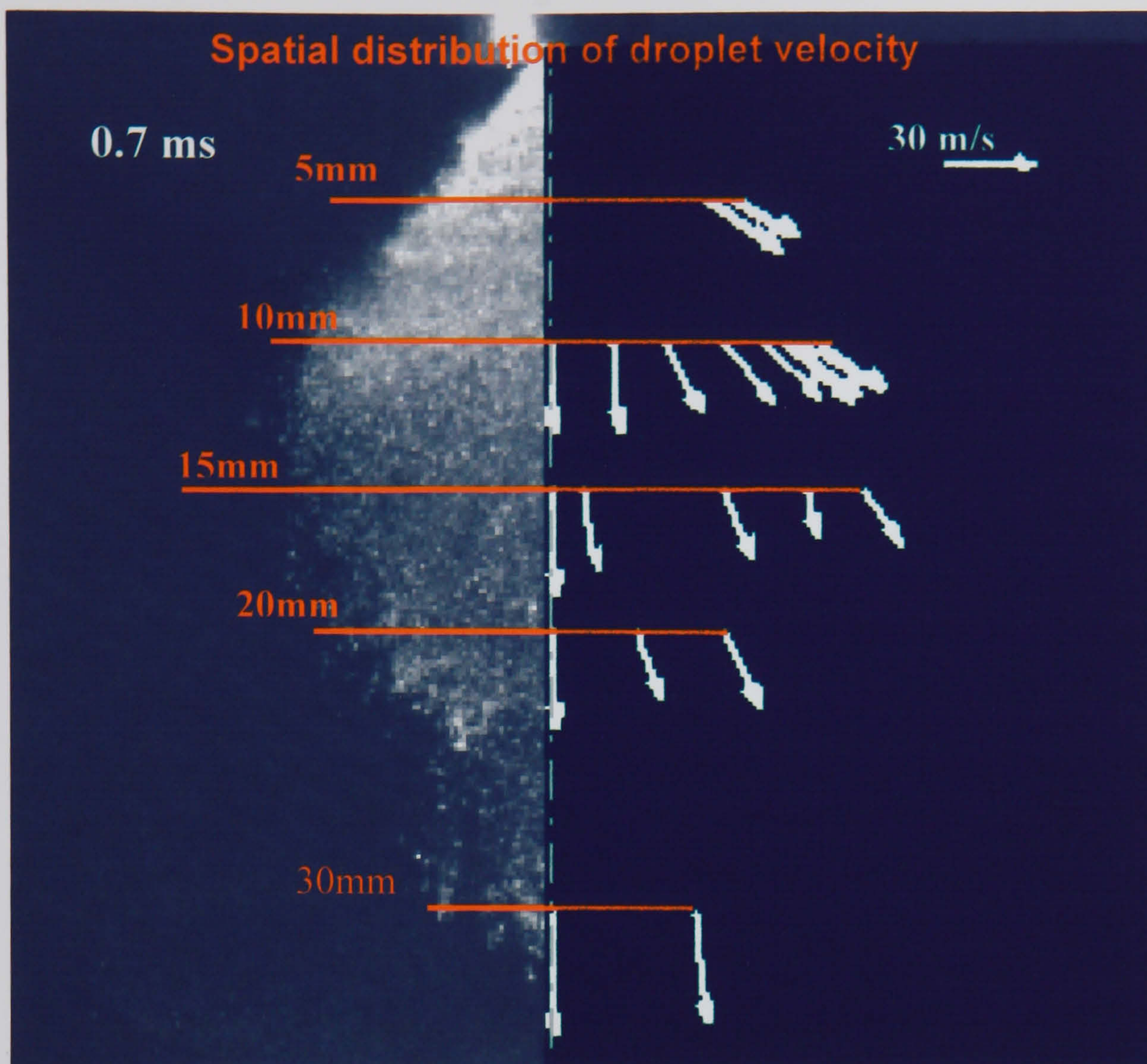


Figure 4-35 Spatial distribution of mean droplet velocity at 0.7 ms and 1.5 ms, $P_{inj}=20$ bar, $P_{ambient}=1$ bar and $T_{ambient}=293K$.

- The first, second and third stages occur within 0.3 ms which is the time corresponding to the needle opening and are responsible of the pre-spray
- Images reveal poor atomisation at the end of the second stage
- Two development methods of calculating spray penetration are explained; one based on the spray tip and the other based on the maximum spray width.
- Spray angle increases during the third stage of spray development.
- The geometry of the spray seems to remain unchanged from one injection to another, although the spray characteristics from similar injectors may be different due to small variations in nozzle machining and the sensitivity of spray structure to the formation of deposits inside the nozzle.
- Spray images inside the constant volume chamber revealed that increasing air density leads to collapse of the spray within a short distance from the injector nozzle resulting in significant reduction in spray volume despite the nozzle spray angle remaining unaffected.
- The effect of increasing air temperature is an apparent increase in spray volume due to the decrease in air density and the presence of a larger number of small droplets that follow the air flow pattern more closely.

Detailed PDA measurements of the droplet characteristics in the spray generated by a pressure swirl atomizer revealed the following aspects of the microscopic structure of the spray:

- The temporal variation of the mean droplet velocities within the main spray apart from the recirculation zone can be divided into six stages:
 - (a) Arrival stage
 - (b) Leading edge stage (deceleration)
 - (c) Acceleration stage
 - (d) Steady stage
 - (e) Deceleration stage
 - (f) Post-spray stage

The appearance of each stage depends on the distance from the nozzle exit to the point of measurement as a result of the different contribution of the aerodynamic force in shaping the initial temporal profile of the mean velocity at the nozzle exit.

- The temporal variation of the mean droplet sizes reveals that the average droplet size remains almost constant throughout the various spray stages with the exception of the leading edge and the post-spray stages
- A rather symmetric droplet size distribution resembling a log normal distribution, slightly skewed towards smaller droplets was identified in the main spray
- Increasing the ambient pressure to 4.5 bar leads to an increase in mean droplet size and significant decrease in mean velocity when compared with measurements at the same axial distance under atmospheric pressure; this highlights the increased probability of droplet agglomeration.
- The spatial droplet size distribution revealed that larger droplets are preferentially located at the outer edges of the spray but are more prevalent in the pre-spray
- The spatial distribution also reveals that during the early stage, the droplets located at the centre of the spray have the highest velocities and relatively larger sizes, representing the pre-spray. On the contrary, after the spray is developed, droplets located in the centre are small with relatively low velocities.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

This is the last chapter of the present thesis and is divided into two sections. The first one summarizes the most important conclusions of the project while the second one includes some recommendations for future work.

6.1 CONCLUSIONS

During this investigation, the flow inside a transparent nozzle extension of the production nozzle of a DISI pressure swirl atomiser has been visualised in order to identify the internal flow and the exiting flow conditions under different operating conditions. After identifying these conditions, two sub-models have been developed to link the flow at the nozzle exit with the atomised spray which consider the forces acting on the liquid film and make use of the linear instability analysis. In addition, the global spray characteristics have been identified by image analysis and quantified locally by a 1-D PDA system.

With the obtained quantitative data for the liquid flow conditions at the nozzle exit and the spray characteristics in addition to the computer two sub-models, it was possible to develop and validate an in-house complete CFD model that includes a 1-D model for the fuel injection equipment, a 3-D model for the flow inside the injector up to the nozzle exit, a 2-D-two-phase model for the liquid film inside the swirl injector nozzle, the developed film divergent model, the developed atomisation model and a conventional spray sub-model. Details of the complete CFD model that includes the two sub-models introduced in this study have been presented by Gavaises and Arcoumanis (2001).

Besides the use of the quantitative data, general conclusions have been drawn that could be summarised as follows:

- Visualising the flow inside the transparent nozzle revealed the formation of a liquid film on the wall with trapped central air inside the nozzle which occurs soon after the needle reaches its full lift. The mean thickness of the film ($\sim 90\ \mu\text{m}$) was nearly independent of the injection pressure within the practical range used in DISI engines and it remains constant after its formation until the end of the fuel injection process.
- Increasing the nozzle length was found to lead to an increase in the mean film thickness and to a relatively more significant reduction in the swirl than in the axial velocities, resulting in a smaller spray angle. However, the quantity of the pre-spray which is poorly atomised and could lead to hydrocarbon emissions, was not as significant as in the case of shorter nozzle hole lengths.
- Since spray images showed that the hollow structure of the spray could be formed at very low injection pressures (3 bar) after the needle reaches its full lift, the reason for the formation of the pre-spray is the presence of pressure fluctuations which disrupt the formation of the film. Accordingly, it is crucial for designers to optimise the duration of needle opening and avoid any edges that may cause disturbance to the film inside the nozzle in order to minimise the amount of the pre-spray.
- Images of high magnification for the flow near the nozzle exit revealed that there are three-dimensional waves on the liquid film surface after injection from the nozzle exit whose magnitudes increase downstream of the nozzle. These waves cause two modes of droplet formation, through ligaments in the direction of injection and through wave crest stripping perpendicular to the liquid film; these observations support the use of the linear instability analysis in representing the atomisation process and predicting the mean droplet diameter soon after the film break-up.
- Flow images revealed the need to use three angles to characterise the flow close to the nozzle exit; the flow angle, the film divergent angle and the nozzle spray angle. The relationship between these three angles has been identified during the fully developed stage through a sub-model based on analyses of the forces acting on the liquid film during its penetration outside the nozzle and the geometric movement of a single droplet after its separation from the liquid film. It has been

shown that the nozzle spray angle is equal to the nozzle flow angle when the flow is fully atomised at the nozzle exit.

- The film divergence sub-model showed that injection velocity does not affect the spray angle within the operation range of DISI injectors, in agreement with observation of the spray images. Similarly the fuel density and surface tension are also found to have a very small effect on the film divergence. However, any small differences between the air pressure outside and inside the conical film may lead to the collapse of the film within a shorter distance.
- An approach has been developed to estimate the break-up length if the nozzle spray angle is known by considering the film divergence and droplet trajectory after its separation from the liquid film.
- An atomisation sub-model has been introduced based on the linear instability analysis for the purpose of estimating the mean droplet diameter of the fuel after the film break-up. This model has been linked to a Lagrangian spray model and validated with the spray droplet size measurements downstream of the nozzle exit using the PDA system for different operating conditions.
- Spray images showed that the spray development consists of four stages following the injector opening delay time estimated to be 0.3 ms; The first, second and third stages occur during the needle opening time when high pressure fluctuations at the nozzle exit occur.
- Two ways of calculating the spray penetration have been presented; one based on the tip of the spray and the other based on the maximum spray width.
- Spray images inside a constant volume chamber revealed that increasing air pressure leads to a collapse of the spray within a short distance from the injector nozzle resulting in significant reduction in the spray volume despite the nozzle spray angle remaining unaffected. Increasing air temperature increases the spray volume due to the decrease in air density and the presence of a larger number of small droplets that follow the air flow pattern more closely.
- PDA measurements revealed that the temporal variation of the mean droplet velocity within the main stream of the spray, apart from the recirculation zone, exhibits six stages: arrival, leading edge (deceleration), acceleration, steady, deceleration and post-spray; the appearance of each stage depends on the distance from nozzle exit to the point of measurement.

- The temporal variation of the mean droplet diameter revealed that the average droplet size remains almost constant throughout the previously mentioned stages except for the leading edge and the post-spray stage.
- The spatial distribution also revealed that during the early stage, the droplets located at the centre of the spray have the highest velocities and relatively larger sizes, representing the pre-spray. On the contrary, after the spray is developed, droplets located in the centre are the smallest with relatively low velocities while the droplets located at the outer edge of the spray are generally larger.

5.2 RECOMMENDATIONS FOR FUTURE WORK

The work presented in this thesis could be extended based on the drawn conclusions in order to identify all the parameters controlling the fuel atomisation and dispersions as well as the spray characteristics. It is also an aim to understand how exactly the atomisation process proceeds. The following points are topics related to this study which require further investigations.

- It has been concluded that the disruption of the film inside the nozzle during the needle opening period due to the associated flow dynamics is responsible for the formation of the pre-spray. It could be interesting to do some additional work by changing the chamfer radius of the nozzle to allow the flow to enter into the nozzle smoothly, so that the film could be formed in a shorter period of time or to reduce the needle opening time and examine damping of the associated pressure fluctuations inside the nozzle, thus reducing the quantity of the pre-spray.
- Since the visualisation of the film thickness inside a real size injector proved to be successful it could be useful to extend this study to identify the film thickness as a function of fuel viscosity and density, nozzle temperature and the ratio between the swirl and axial velocity. It could also be useful to identify the spray mean droplet diameter as a function of different film thickness.
- Although a sub-model has been introduced for the film divergence and droplet divergence near the nozzle exit, it could be improved by taking into account all the forces applied on the droplet to improve the prediction of the droplet

trajectory and its radial divergence, thus improving the prediction of the break-up length.

- Some effort has been put to measure the surface wavelength that has been observed at the nozzle exit but it was unsuccessful because of the small scale of the waves; it could be worth to making a large scale model of this injector and use the image analysis technique with FFT to try to obtain the dominant wavelength which could be very useful in developing the linear instability analysis and the break-up model.
- The hollow cone structure of the spray from the swirl injector has been found collapsing after a short distance when the spray was injected against pressurised air. It could be useful to identify which parameters are mainly responsible for this collapse; for example, if the pressure inside and outside the spray envelope is controlled and the effect of pressure difference is revealed. This could lead to a combustion chamber design equipped with a swirl injector that can keep the same hollow spray structure under different conditions and, therefore, could be more applicable to a close spacing DISI combustion configuration.
- It could also be useful to visualise the flow inside the nozzle when the fuel injected into high pressure/temperature similar to the real engine conditions during part load operation.
- Characterising the flow inside the nozzle and near-nozzle exit during the opening time of the needle could be a full topic and extension to this study.

Appendix 1-A

Liquid film thickness estimation according to the analysis published by Giffen and Muraszew (1953)

The thickness of the liquid film formed at the nozzle exit is directly related to the area of the air core. To find the relationship between the atomizer dimensions and the size of the air core, the flow is assumed frictionless by Giffen and Muraszew (1953) and lefebvre (1989). The liquid flow pattern inside the nozzle can be considered has a spiral motion on a free vortex. Since the angular momentum for a frictionless flow is conservative, the following relationship between the tangential velocity w and radius r could be conducted.

$$wr = \sum VR \cos(\theta_i) / n \quad (1)$$

where V is the inlet velocity, θ is the cylindrical hole inclination angle, n is the number of holes which equal to 4 and R is the radius as shown in Fig (1) since the inlet velocity, the radius and the initial inclination angle are the same for the four holes, then the above equation could be simplified as

$$wr = VR \cos(\theta)$$

The value of V can be calculated from the mass flow rate of the fuel as follows

$$V = \frac{\dot{m}}{\rho_l A_p} \quad (2)$$

where A_p is the summation of the cross sectional area of the 4 holes.

Applying Bernoulli's equation yields:

$$P = p + 0.5\rho_l u^2 + 0.5\rho_l w^2 = \text{constant}$$

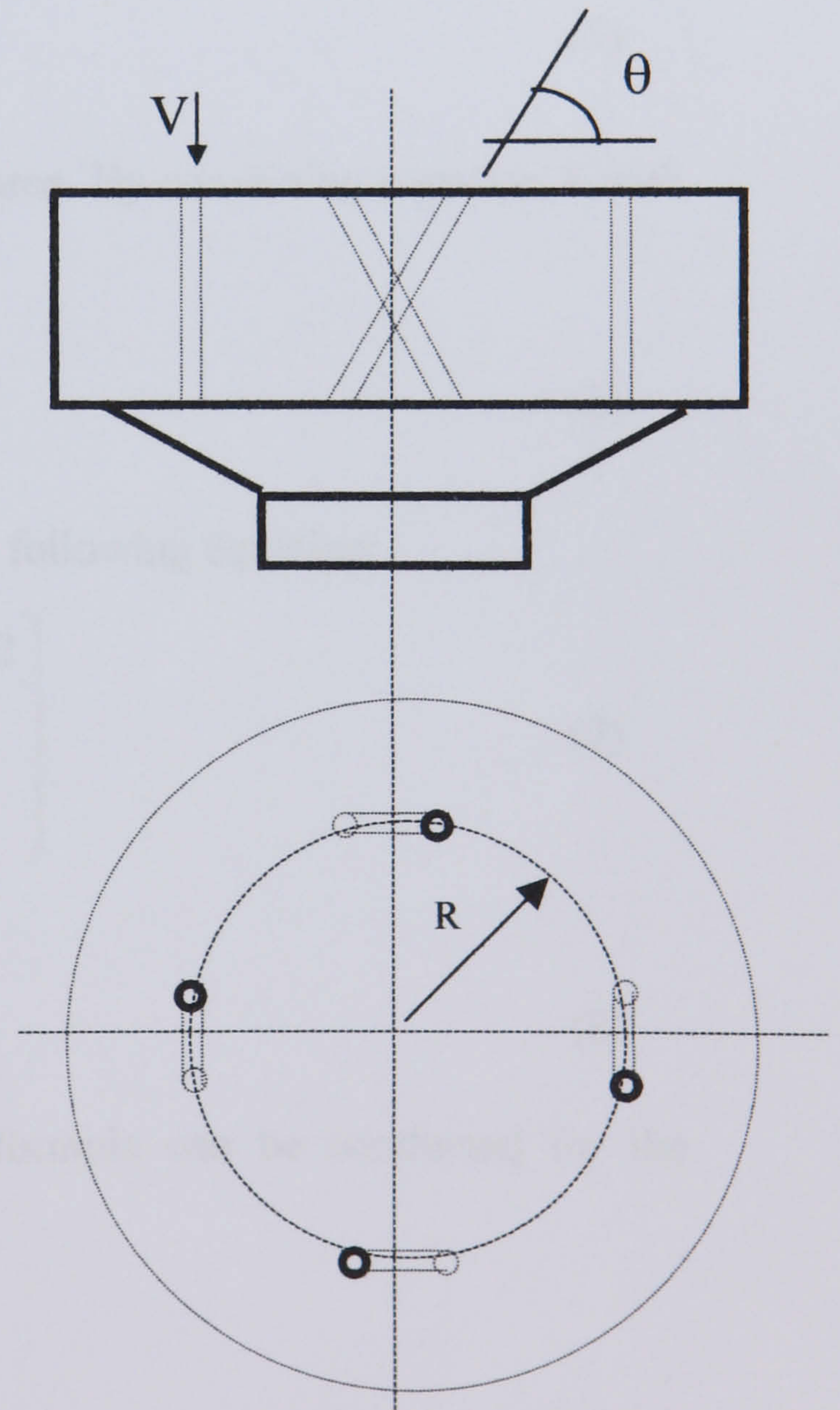


Figure showing how the swirl motion is formed through the four tangential holes inside the swirler that is located above the nozzle.

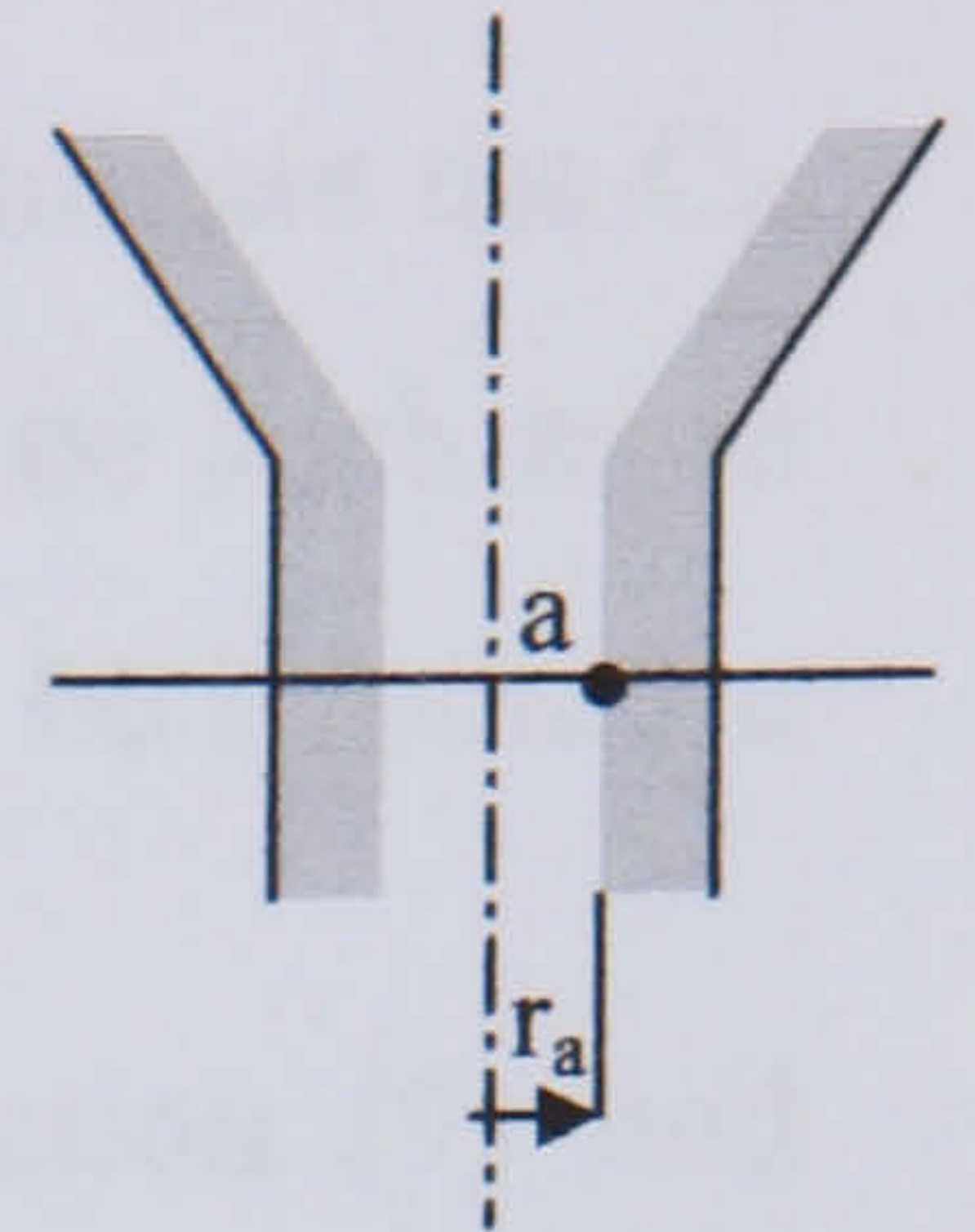
where P is the total pressure, and p is the static pressure

Assuming the flow is steady and fluid is non viscous fluid, the axial velocity u is uniform and constant for all values of r . Accordingly, for a free vortex alone:

$$p + 0.5\rho_l u^2 = \text{constant}$$

At the air core (point a), static pressure is equal to the back pressure which is equal to the atmospheric pressure, i.e. $p = 0$ and therefore;

$$P = 0.5 \rho_l (u_a^2 + w_a^2) \quad (3)$$



Since the axial velocity is constant across the whole section then, $u_a = u$ and accordingly:

$$P = 0.5 \rho_l (u^2 + w_a^2) \quad (4)$$

Form the conservation of mass at the nozzle outlet:

$$u = \frac{\dot{m}}{\rho_l (A_o - A_a)} \quad (5)$$

where A_o is the nozzle area and A_a is the air core area. By combining equation 5 with equations 1 and 2 yields:

$$w_a = \frac{\dot{m} R \cos(\theta)}{\rho_l A_p r_a} \quad (6)$$

Substituting for u and w_a into equation 4 leads to the following equation:

$$P = 0.5 \rho_l \left[\left(\frac{\dot{m} R \cos(\theta)}{\rho_l A_p r_a} \right)^2 + \left(\frac{\dot{m}}{\rho_l (A_o - A_a)} \right)^2 \right] \quad (7)$$

by taking into consideration that :

$$\dot{m} = C_D A_o \sqrt{2 \rho_l P} \quad (8)$$

and combining equation 7 and 8 the following formula can be conducted for the discharge coefficient C_D :

$$\frac{1}{C_D^2} = \frac{1}{K_1^2 X} + \frac{1}{(1-X)^2} \quad (9)$$

where $X = A_a / A_o$ and $K_1 = A_p / \pi r_o R \cos(\theta)$

The above equation provides a relationship between the atomizer dimensions, the size of the air core and the discharge coefficient of the nozzle. To eliminate the C_D from equation (9) it is assumed that the value of air core size is always be such as to give maximum flow. The value of C_D , expressed as a function of X will be maximum

when $\frac{dC_D}{dX} = 0$. By calculating the first derivation $\frac{dC_D}{dX}$ from equation (9) and

consider it equal to zero, the following equation could be deduced:

$$2k_1^2 X^2 = (1 - X)^3 \quad (10)$$

Substituting by the value of $k_1 = 2A_p / \pi R d_o \cos(\theta)$ into equation 10 results in a relationship between injector dimensions and expected air core diameter or film thickness.

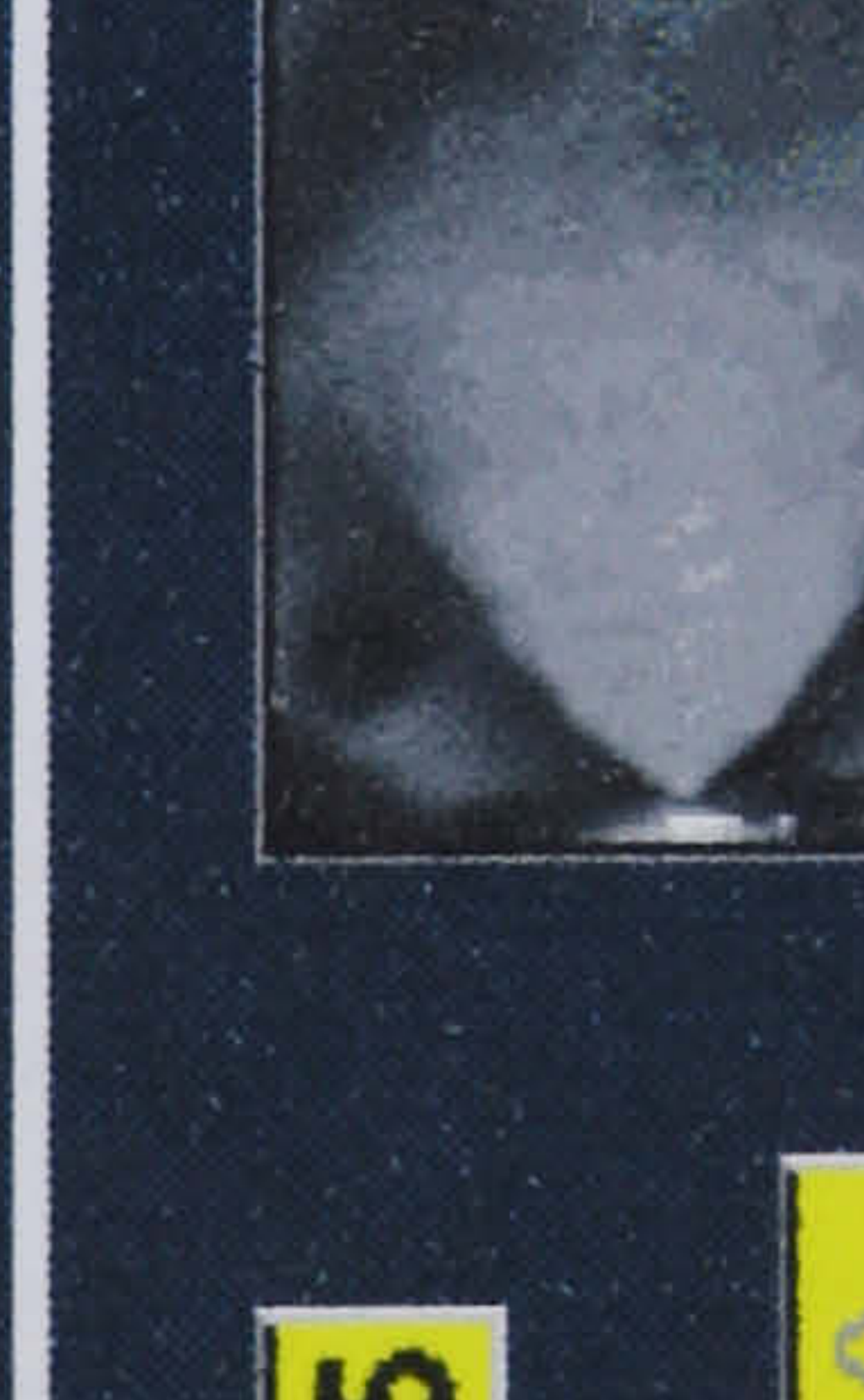
$$\left(\frac{A_p}{2R d_o \cos(\theta)} \right)^2 = \frac{\pi^2 (1 - X)^3}{32 X^2} \quad (11)$$

From equation (9) the value of X can be obtained as a function of the nozzle geometry, consequently, the corresponding value of liquid film thickness t can readily obtained as follows:

$$X = \frac{(d_o - 2t)^2}{d_o^2} \quad (12)$$

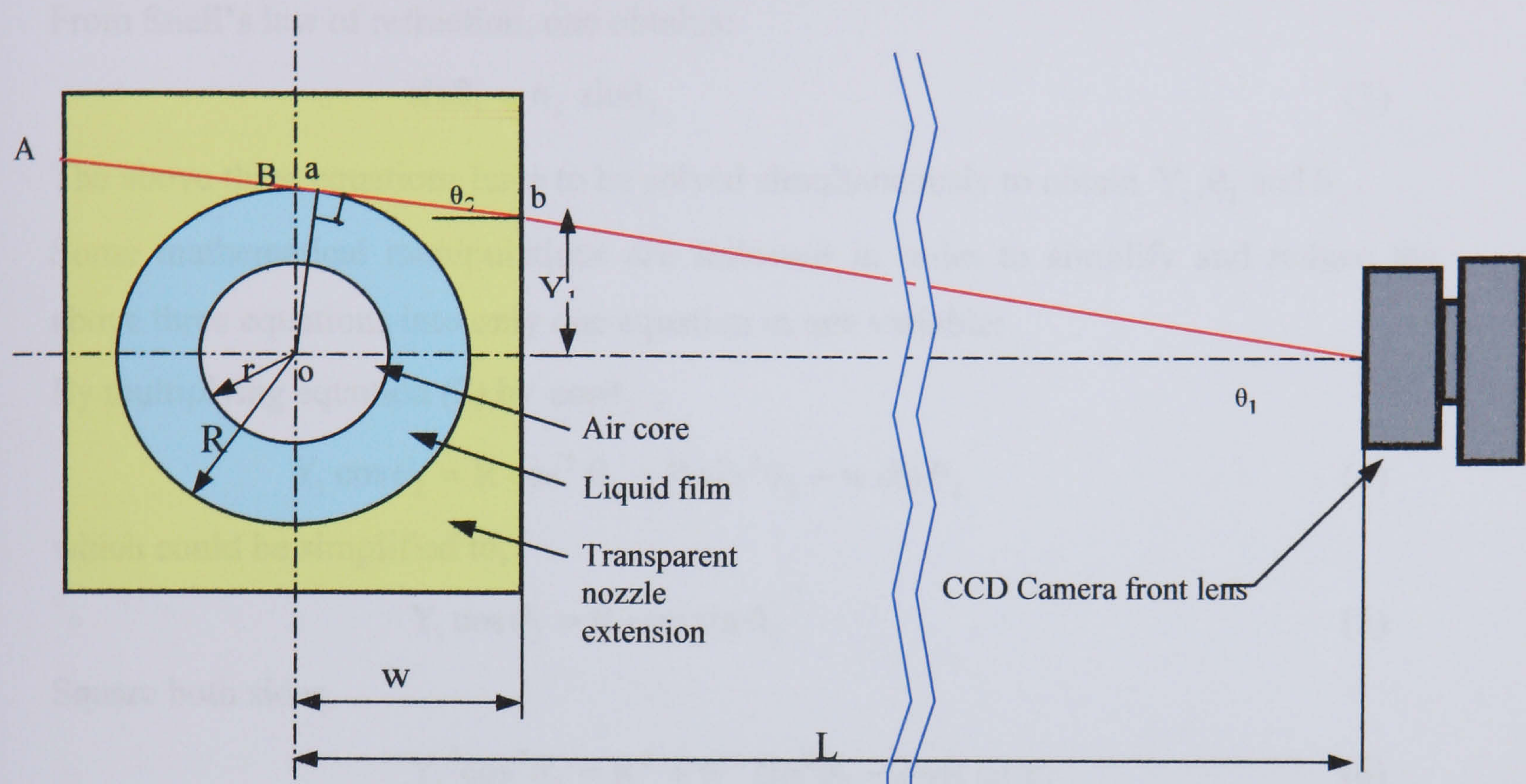
Appendix 2-A

Graphic user interface panel to control the injection system and the instrumentation

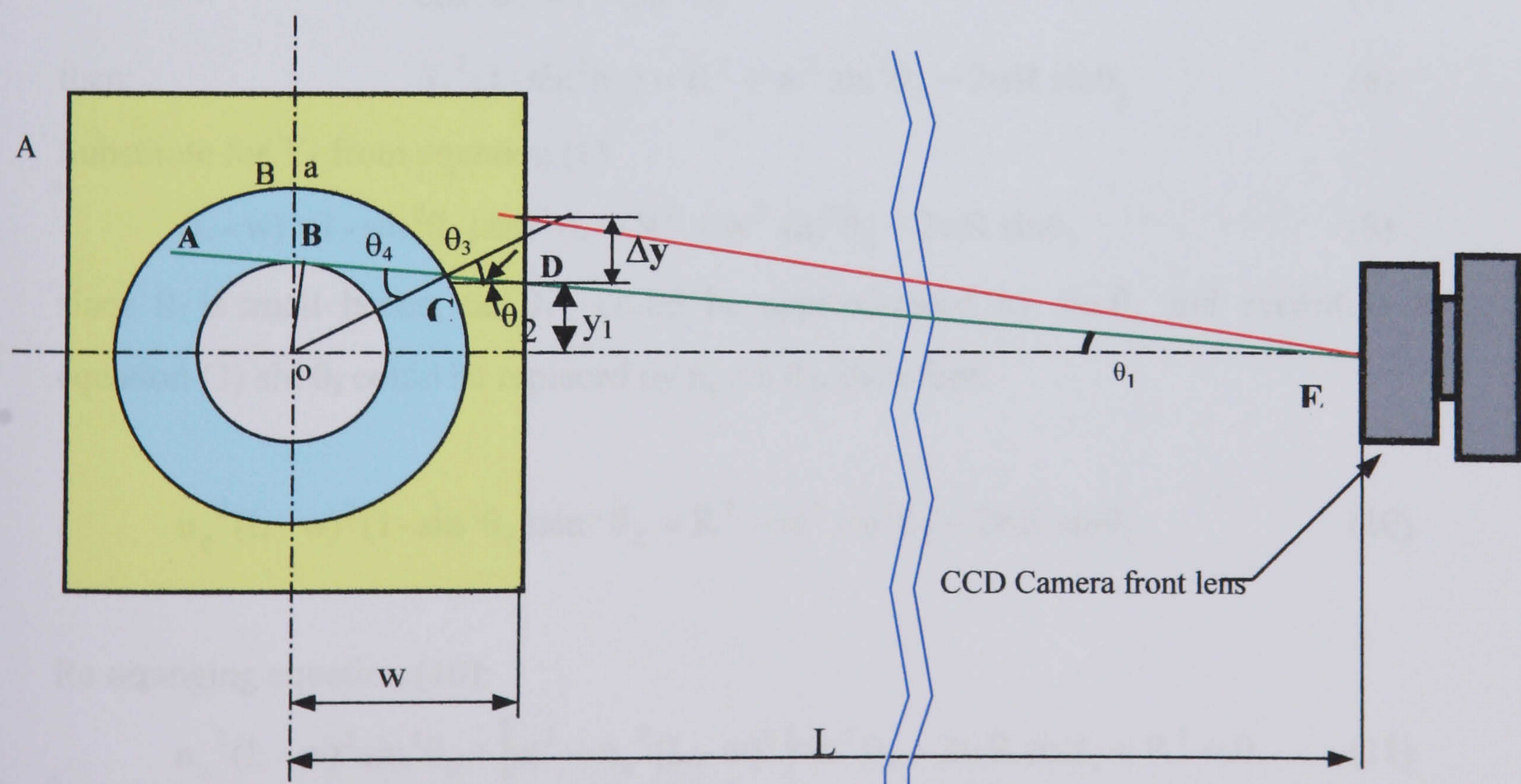
Injection Signal		Triggering Signals	
☐ Continuous	☒ Discrete	☒ Multi Trigger	☐ One Trigger
No. of inj.	On time	After how many Cycles triggering occurs	
1000	1000		2
		Triggering signal duration (us)	100
		Triggering time from the start of injection in (us)	1000
		Triggering time from the first triggering signal (us)	1500
		Triggering time from the second triggering signal (us)	100
☒ Manual Operation ☐ Engine Simulation			
Injection Time in (ms)	1.5		
Engine (rpm)	700		
Injection Frequency Hz	6		
EXIT		STOP	
		RUN	

Appendix 2-B

Calculating the error in measuring the film thickness from the images arises from light refraction due to the curvature of the interphases and the difference in the refractive indices of the white spirit, Perspex and air



a) Figure 1



b) Figure 2

From Figure 1, the following relationship can be deduced:

$$Y_1 = (L - w) \tan \theta_1 \quad (1)$$

$$Y_1 = R \cos \theta_2 - (w - R \sin \theta_2) \quad (2)$$

From Snell's law of refraction, one obtains:

$$\sin \theta_1 = n_g \sin \theta_2 \quad (3)$$

The above three equations have to be solved simultaneously to obtain Y_1, θ_1 and θ_2 .

Some mathematical manipulations are followed in order to simplify and reduce the above three equations into only one equation in one variable:

By multiplying equation (2) by $\cos \theta_2$;

$$Y_1 \cos \theta_2 = R \cos^2 \theta_2 + R \sin^2 \theta_2 - w \sin \theta_2 \quad (4)$$

which could be simplified to,

$$Y_1 \cos \theta_2 = R - w \sin \theta_2 \quad (5)$$

Square both sides,

$$Y_1^2 \cos^2 \theta_2 = R^2 + w^2 \sin^2 \theta_2 - 2wR \sin \theta_2 \quad (6)$$

since;

$$\cos^2 \theta_2 = 1 - \sin^2 \theta_2 \quad (7)$$

then;

$$Y_1^2 (1 - \sin^2 \theta_2) = R^2 + w^2 \sin^2 \theta_2 - 2wR \sin \theta_2 \quad (8)$$

Substitute for Y_1 from equation (1)

$$(L - w)^2 (1 - \sin^2 \theta_2) \tan^2 \theta_1 = R^2 + w^2 \sin^2 \theta_2 - 2wR \sin \theta_2 \quad (9)$$

since θ_1 is small hence, $\tan \theta_1$ could be approximated by $\sin \theta_1$ and according to equation (3) $\sin \theta_1$ could be replaced by $n_g \sin \theta_2$, therefore:

$$n_g^2 (L - w)^2 (1 - \sin^2 \theta_2) \sin^2 \theta_2 = R^2 + w^2 \sin^2 \theta_2 - 2wR \sin \theta_2 \quad (10)$$

Re-arranging equation (10):

$$n_g^2 (L - w)^2 \sin^4 \theta_2 + [w^2 - n_g^2 (L - w)^2] \sin^2 \theta_2 - 2wR \sin \theta_2 + R^2 = 0 \quad (11)$$

The above equation could be solved analytically using Matlab to obtain 4 roots for $\sin \theta_2$. The root whose value between -1 and 1 is considered as the solution.

Having transferred the edge of the nozzle hole to the front surface of the transparent nozzle, we can proceed to transfer n points from the edge of the film inside the hole to its corresponding points on the same surface. In order to transfer a point spaced a distance from the nozzle diameter, which represents the actual film thickness, let us follow the general path of the light denoted ABCDE, which ended at the centre of the front lens of the camera E, as shown in Figure 2.

The following mathematical manipulations represent how this transfer could be done for a light ray spaced by a distance Δy from point b previously shown in Figure 1, therefore:

$$y_1 = Y_1 - \Delta y \quad (12)$$

From the geometry,

$$\tan\theta_1 = \frac{y_1}{L - w} \quad (13)$$

To obtain θ_2 Snell's law of refraction could be applied:

$$\sin\theta_1 = n_g \sin\theta_2 \quad (14)$$

In order to determine the intersection of line CD with the nozzle hole, the equations of the line CD and the hole circle have been written and solved together to obtain the intersection point (x_i, y_i) as follows

$$\text{Equation of the line CD: } y = -\tan\theta_2 x + (y_1 + w\tan\theta_2) \quad (15)$$

$$\text{Equation of the circle: } y^2 + x^2 = R^2 \quad (16)$$

Having identified the intersection point between the nozzle hole and the light ray in the Perspex (x_i, y_i) , the incident and refracted angles of the light at the inter-phase surface between Perspex and liquid film could then be determined from the following equations:

$$\theta_3 = \tan^{-1}\left(\frac{y_i}{x_i}\right) + \theta_2 \quad (17)$$

$$\theta_4 = \sin^{-1}\left(\frac{n_g}{n_l} \sin\theta_3\right) \quad (18)$$

The equation of the line ABC passes through the liquid film could then be written in the following form:

$$y = -\tan(\theta_4 - \theta_3 + \theta_2)x + y_i + x_i \tan(\theta_4 - \theta_3 + \theta_2) \quad (19)$$

The equation of the line (oB) drawn from the centreline of the hole (o) and perpendicular to the line ABC could be written in the following form.

$$y = \frac{1}{\tan(\theta_4 - \theta_3 + \theta_2)} x \quad (20)$$

The coordinate of the intersection point (B) between the line OB and the line AB could be determined by solving equation 19 and 20 together :

$$\frac{1}{\tan(\theta_4 - \theta_3 + \theta_2)} x_B = -\tan(\theta_4 - \theta_3 + \theta_2)x_B + y_i + x_i \tan(\theta_4 - \theta_3 + \theta_2) \quad (21)$$

Having identified the intersection point (x_B, y_B) which located at inter-phase between air and liquid film, the radius of the air core could then be calculated from the following equation:

$$R_a = \sqrt{x_B^2 + y_B^2} \quad (22)$$

The actual dimensionless film thickness t_{act} could then be identified:

$$t_{act} = (R - R_a)/R \quad (23)$$

The dimensionless film thickness t as measured from the images:

$$t = \Delta y/Y \quad (24)$$

Finally the error could be calculated from the image thickness and the actual thickness from the following equation,

$$\text{Error \%} = 100 \times (t - t_{act}) / t_{act} \quad (25)$$

APPENDIX 3-A

Atomisation modelling using the linear instability analysis theory

Figure 1 shows schematically the annular liquid sheet assumed to exit from the injection hole. The film exits with an axial velocity defined as W , while the axial air velocity at its inner and external sides are defined as W_{ai} and W_{ao} , respectively. Employing the assumptions described in the previous paragraph, the continuity and the momentum equations can be written in the form:

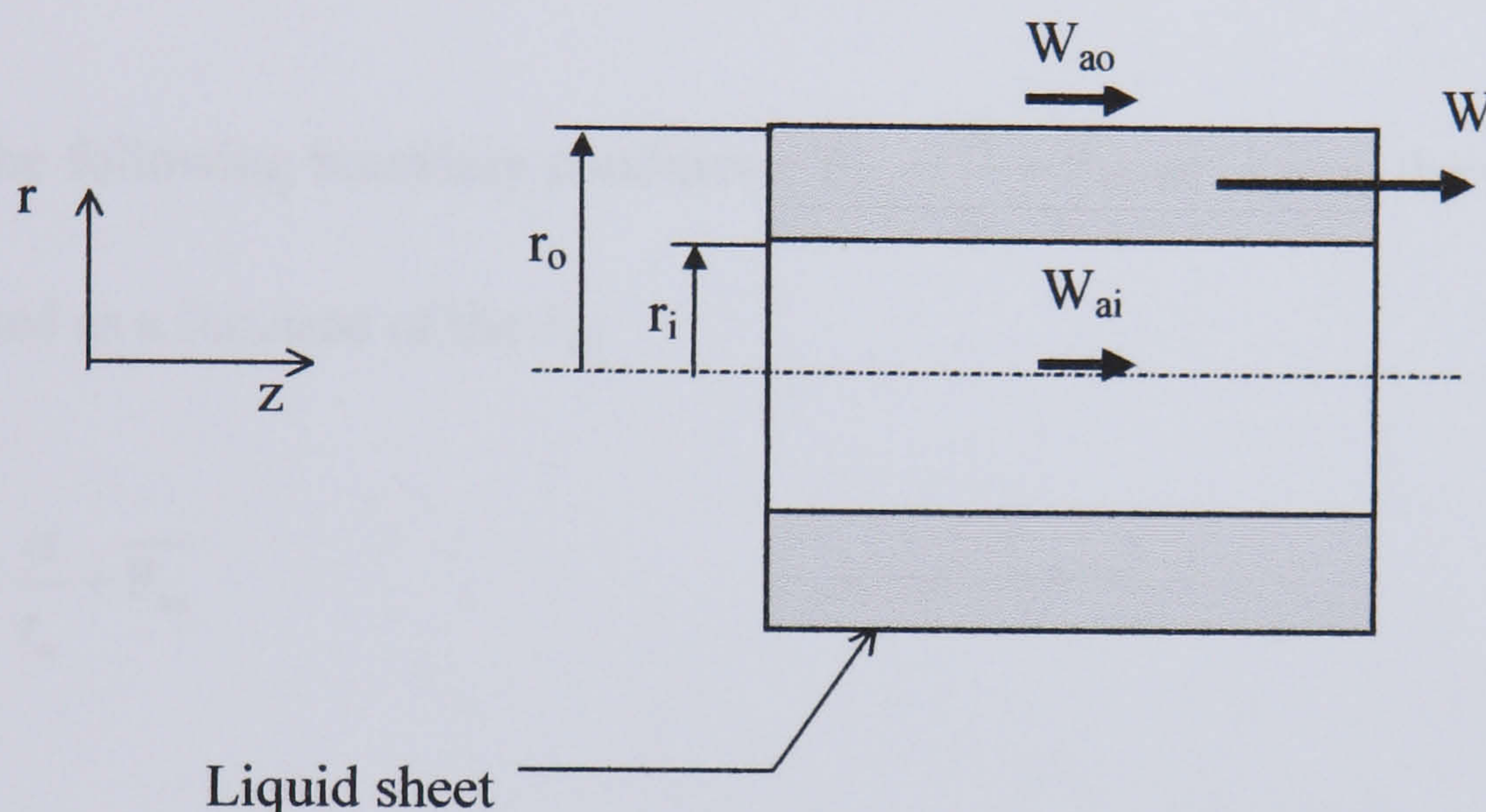


Figure 1 Schematic representation of the annular liquid sheet exiting from the injection hole and definition of the notation for the dimensions and the air velocities inside and outside the liquid sheet

$$\frac{\partial V_j}{\partial t} + (V_j \cdot \nabla) V_j = -\frac{1}{\rho} \nabla P_j \quad (1)$$

$$\nabla \cdot V_j = 0 \quad (2)$$

The above equations are valid and apply both for the air inside and outside the annular liquid sheet, as well as for the liquid sheet itself. Let $\bar{U}, \bar{V}, \bar{W}$ denote the relative to the gas liquid velocity components in the radial (r), circumferential (θ) and axial (z) directions, respectively. It is assumed that the mean flow of the liquid is undisturbed, while the following flow conditions are also adopted:

$$\bar{U} = 0, \quad \bar{V} = \frac{A_1}{r}, \quad \bar{W} = W_1 \quad r_i \leq r \leq r_o \quad (3)$$

where, A_1 and W_1 are constants. Substituting (3) into (1) and (2), the pressure field can be analytically estimated as:

$$\bar{P}_1 = -\frac{1}{2}\rho A_1^2 / r^2 + C \quad (4)$$

Considering the following boundary conditions $\bar{P}_{1o} = \frac{\sigma}{r_o} + \bar{P}_{ao}$ at $r = r_o$ the constant C

can be estimated as a function of the P_o :

$$C = \frac{1}{2}\rho \frac{A_1^2}{r_o^2} + \frac{\sigma}{r_o} + \bar{P}_{ao} \quad (5)$$

Substituting this constant into equation (4), it yields:

$$\bar{P}_1(r) = -\frac{1}{2}\rho \frac{A_1^2}{r^2} + \frac{1}{2}\rho \frac{A_1^2}{r_o^2} + \frac{\sigma}{r_o} + \bar{P}_{ao} \quad (6)$$

It is further assumed that the velocity and pressure fields can be approximated as the sum of a mean and a fluctuating part, which can be expressed in the form:

$$U_1 = \bar{U}_1 + u_1, \quad V_1 = \bar{V}_1 + v_1, \quad W_1 = \bar{W}_1 + w_1, \quad P_1 = \bar{P}_1 + p_1 \quad (7)$$

where $\bar{U}_1, \bar{V}_1, \bar{W}_1, \bar{P}_1$ represent the solution for the undisturbed flow. Substituting (7) into the continuity and momentum equations results:

$$\frac{1}{r} \frac{\partial}{\partial r} r(\bar{U}_1 + u_1) + \frac{1}{r} \frac{\partial}{\partial \theta} (\bar{V}_1 + v_1) + \frac{\partial}{\partial z} (\bar{W}_1 + w_1) = 0 \quad (8)$$

$$\begin{aligned}
 & \frac{\partial (\bar{U}_1 + u_1)}{\partial t} + (\bar{U}_1 + u_1) \frac{\partial (\bar{U}_1 + u_1)}{\partial r} + \frac{(\bar{V}_1 + v_1)}{r} \frac{\partial (\bar{U}_1 + u_1)}{\partial \theta} \\
 & \quad + (\bar{W}_1 + w_1) \frac{\partial (\bar{U}_1 + u_1)}{\partial z} - \frac{(\bar{V}_1 + v_1)^2}{r} = -\frac{1}{\rho_1} \frac{\partial (\bar{P}_1 + p_1)}{\partial r} \\
 & \frac{\partial (\bar{V}_1 + v_1)}{\partial t} + (\bar{U}_1 + u_1) \frac{\partial (\bar{V}_1 + v_1)}{\partial r} + \frac{(\bar{V}_1 + v_1)}{r} \frac{\partial (\bar{V}_1 + v_1)}{\partial \theta} + \\
 & \quad (\bar{W}_1 + w_1) \frac{\partial (\bar{V}_1 + v_1)}{\partial z} + \frac{(\bar{U}_1 + u_1)(\bar{V}_1 + v_1)}{r} = -\frac{1}{\rho_1} \frac{\partial (\bar{P}_1 + p_1)}{r \partial \theta} \\
 & \frac{\partial (\bar{W}_1 + w_1)}{\partial t} + (\bar{U}_1 + u_1) \frac{\partial (\bar{W}_1 + w_1)}{\partial r} + \frac{(\bar{V}_1 + v_1)}{r} \frac{\partial (\bar{W}_1 + w_1)}{\partial \theta} + (\bar{W}_1 + w_1) \frac{\partial (\bar{W}_1 + w_1)}{\partial z} = \\
 & \quad -\frac{1}{\rho_1} \frac{\partial (\bar{P}_1 + p_1)}{\partial r}
 \end{aligned} \tag{9}$$

Since $\bar{U}_1 = 0$, $\bar{V}_1 = \frac{A_1}{r}$, $\bar{W}_1 = W_1$ and $\bar{P}_1 = -\frac{1}{2}\rho \frac{A_1^2}{r^2} + C$, then, the following conditions result:

$$\frac{\partial \bar{P}_1}{\partial r} = \rho \frac{A_1^2}{r^3}, \quad \frac{\partial \bar{P}_1}{\partial \theta} = 0, \quad \frac{\partial \bar{P}_1}{\partial z} = 0, \quad \frac{\partial \bar{V}_1}{\partial r} = -\frac{A_1}{r^2} \text{ which can be substituted into}$$

equations (8) and (9) in order to obtain the final form of the continuity and momentum equations. By neglecting the second order terms of the small perturbations, it results:

$$\begin{aligned}
 & \frac{1}{r} \frac{\partial (ru_1)}{\partial r} + \frac{1}{r} \frac{\partial v_1}{\partial \theta} + \frac{\partial w_1}{\partial z} = 0 \\
 & \frac{\partial u_1}{\partial t} + \frac{A_1}{r^2} \frac{\partial u_1}{\partial \theta} + W_1 \frac{\partial u_1}{\partial z} - \frac{2A_1}{r^2} v_1 = -\frac{1}{\rho_1} \frac{\partial p_1}{\partial r} \\
 & \frac{\partial v_1}{\partial t} + \frac{A_1}{r^2} \frac{\partial v_1}{\partial \theta} + W_1 \frac{\partial v_1}{\partial z} = -\frac{1}{\rho_1 r} \frac{\partial p_1}{\partial \theta}
 \end{aligned} \tag{10}$$

$$\frac{\partial w_1}{\partial t} + \frac{A_1}{r^2} \frac{\partial w_1}{\partial \theta} + W_1 \frac{\partial w_1}{\partial z} = -\frac{1}{\rho_1} \frac{\partial p_1}{\partial z}$$

Similar equations can be written for the air outside and inside the liquid sheet by substituting as follows:

$$\begin{aligned} \rho_{ao} &= \rho, \quad W_{ao} = W_o \quad \text{and} \quad A_{ao} = A \quad \text{for the air outside the liquid sheet, and} \\ \rho_{ai} &= \rho, \quad W_{ai} = W_o \quad \text{and} \quad A_{ai} = A \quad \text{for the air inside the liquid sheet.} \end{aligned}$$

Equations (10) are explicit in r , and the solution can be assumed to have the following form:

$$\begin{aligned} u_1 &= u_1^*(r) \exp(-i\omega t + in\theta + ikz) \\ v_1 &= v_1^*(r) \exp(-i\omega t + in\theta + ikz) \\ w_1 &= w_1^*(r) \exp(-i\omega t + in\theta + ikz) \\ p_1 &= p_1^*(r) \exp(-i\omega t + in\theta + ikz) \end{aligned} \tag{11}$$

In the above equations (11), n is an integer and k is a real number; ω is a complex number representing the growth rate of the waves; instability occurs when its imaginary part is positive. In this case, the amplitude of the perturbation will increase exponentially with time. Note that there are two similar sets of equations for the air outside and inside the liquid sheet. Substituting equations (11) in the continuity and the momentum equations (10), the following equations can be deduced:

$$\begin{aligned} \frac{1}{r} \frac{d}{dr} (r u_1^*) + i \frac{n}{r} v_1^* + i k w_1^* &= 0 \\ (-i\omega + \frac{inA_1}{r^2} + i k W_1) u_1^* - \frac{2A_1}{r^2} v_1^* &= -\frac{1}{\rho_1} \frac{dp_1^*}{dr} \\ (-i\omega + \frac{inA_1}{r^2} + i k W_1) v_1^* &= -\frac{1}{\rho_1} \frac{in}{r} \frac{dp_1^*}{dr} \end{aligned}$$

$$(-i\omega + \frac{inA_1}{r^2} + ikW_1)w_1^* = -\frac{1}{\rho_1}ik \frac{dp_1^*}{dr} \quad (12)$$

When eliminating p^* from equations (12) and after some simple mathematical manipulations, the following total differential equation can be deduced:

$$\frac{d^2 w_1^*}{dr^2} + \frac{1}{r} \frac{dw_1^*}{dr} - (k^2 + \frac{n^2}{r^2})w_1^* = 0 \quad (13)$$

The solution of this differential equation is a combination of Bessel functions of the first and second kind of order n . Using the equations (11), (12) and (13) the solution has the following form:

for $r_i \leq r \leq r_o$

$$\begin{aligned} u_1^* &= -iB_1 \frac{d(I_n(kr))}{d(kr)} - iC_1 \frac{d(K_n(kr))}{d(kr)} \\ v_1^* &= \frac{n}{kr} B_1 I_n(kr) + \frac{n}{kr} C_1 K_n(kr) \\ w_1^* &= B_1 I_n(kr) + C_1 K_n(kr) \\ p_1^* &= \frac{\rho_1}{k} (\omega - \frac{nA_1}{r^2} - kW_1)(B_1 I_n(kr) + C_1 K_n(kr)) \end{aligned} \quad (14)$$

where B_1 and C_1 are constants which can be estimated from the boundary conditions. A similar set of equations can be written for the air surrounding the cylinder and another set for the air inside the cylinder as follows. For the air inside the cylinder: $r \leq r_i$

$$\begin{aligned} u_{ai}^* &= -iB_{ai} \frac{d(I_n(kr))}{d(kr)} - iC_{ai} \frac{d(K_n(kr))}{d(kr)} \\ v_{ai}^* &= \frac{n}{kr} B_{ai} I_n(kr) + \frac{n}{kr} C_{ai} K_n(kr) \\ w_{ai}^* &= B_{ai} I_n(kr) + C_{ai} K_n(kr) \\ p_{ai}^* &= \frac{\rho_{ai}}{k} (\omega - \frac{nA_{ai}}{r^2} - kW_{ai})(B_{ai} I_n(kr) + C_{ai} K_n(kr)) \end{aligned} \quad (15)$$

For the air outside the cylinder: $r \geq r_o$

$$\begin{aligned}
 u_{ao}^* &= -iB_{ao} \frac{d(I_n(kr))}{d(kr)} - iC_{ao} \frac{d(K_n(kr))}{d(kr)} \\
 v_{ao}^* &= \frac{n}{kr} B_{ao} I_n(kr) + \frac{n}{kr} C_{ao} K_n(kr) \\
 w_{ao}^* &= B_{ao} I_n(kr) + C_{ao} K_n(kr) \\
 p_{ao}^* &= \frac{\rho_{ao}}{k} \left(\omega - \frac{nA_{ao}}{r^2} - kW_{ao} \right) (B_{ao} I_n(kr) + C_{ao} K_n(kr))
 \end{aligned} \tag{16}$$

Since the normal stress is continuous over the interfaces, the pressure jumps due to capillary forces at the interface. For the liquid surface outside the cylinder, i.e. at $r = r_o + \delta$

$$P_l = \sigma \xi + P_o \tag{17}$$

where ξ is the mean curvature of the disturbed boundary and it can take the following form:

$$\xi = \frac{1}{r_o} - \frac{\delta}{r_o^2} - \frac{1}{r_o^2} \frac{\partial^2 \delta}{\partial \theta^2} - \frac{\partial^2 \delta}{\partial z^2}$$

Assuming that δ has the same expression as u^*, v^*, w^* and p^* , i.e. that it can be written in the form:

$$\delta = \delta^* \exp(-i\omega t + in\theta + ikz) \tag{18}$$

$$\xi = \frac{1}{r_o} - \left(\frac{1}{r_o^2} - \frac{n}{r_o^2} - k^2 \right) \delta^* \exp(-i\omega t + in\theta + ikz) \tag{19}$$

Considering the Taylor expansion at $r = r_o$ and neglecting the second order terms, one can get the following expressions for the pressure at $r_o + \delta_o$:

$$P_1 = P_{lo} + \left. \frac{\partial P_1}{\partial r} \right|_{r=r_o} \delta_o,$$

$$P_a = P_{ao} + \left. \frac{\partial P}{\partial r} \right|_{r=r_o} \delta_o$$

Substitute the above expressions together with equations (14) and (16) into equation (17), and considering that $P_1 = \bar{P}_1 + p_1$ and $\bar{P}_{lo} = \frac{\sigma}{r_o} + \bar{P}_{ao}$, the following equation can be deduced:

$$\rho_1 \frac{A_1^2}{r_o^3} \delta_o^* + p_{lo}^* = \sigma \left(-\frac{1}{r_o^2} + \frac{n^2}{r_o^2} + k^2 \right) \delta_o^* + p_{ao}^* + \rho_{ao} \frac{A_{ao}^2}{r_o^3} \delta_o^* \quad (20)$$

Similar equations can be deduced for the inner surface of the cylinder and can be written in the following form:

$$\rho_1 \frac{A_1^2}{r_i^3} \delta_i^* + p_{li}^* = -\sigma \left(-\frac{1}{r_i^2} + \frac{n^2}{r_i^2} + k^2 \right) \delta_i^* + p_{ai}^* + \rho_{ai} \frac{A_{ai}^2}{r_i^3} \delta_i^* \quad (21)$$

From equations (14), the following equations can be obtained:

$$p_{lo}^* = \frac{\rho_1}{k} \left(\omega - \frac{nA_1}{r_o^2} - kW_1 \right) (B_1 I_n(kr_o) + C_1 K_n(kr_o)) \quad (22)$$

$$p_{li}^* = \frac{\rho_1}{k} \left(\omega - \frac{nA_1}{r_i^2} - kW_o \right) (B_1 I_n(kr_i) + C_1 K_n(kr_i)) \quad (23)$$

Similar equations can be written for the air, outside and inside the cylinder as shown in equations (22) and (23). From the first equation in (14), the constant B for the outside surface, is given by:

$$B_1 = \frac{i u_{lo}^*}{I'_n(kr_o)} - \frac{C_1 K'_n(kr_o)}{I'_n(kr_o)} \quad (24)$$

From the first equation in (16), the constant C₁ for the inside surface, is given by:

$$C_1 = \frac{i u_{li}^*}{K'_n(kr_i)} - \frac{B_1 I'_n(kr_i)}{K'_n(kr_i)} \quad (25)$$

Substituting from equation (24) into equation (25) yields to:

$$C_1 = \frac{i u_{li}^*}{K'_n(kr_i)} - \frac{i u_{lo}^*}{I'_n(kr_o)} \frac{I'_n(kr_i)}{K'_n(kr_i)} + \frac{C K'_n(kr_o)}{I'_n(kr_o)} \frac{I'_n(kr_i)}{K'_n(kr_i)} \quad (26)$$

By defining the variables I_R and K_R as

$$\frac{I'_n(kr_o)}{I'_n(kr_i)} = I_R \quad \text{and} \quad \frac{K'_n(kr_o)}{K'_n(kr_i)} = K_R$$

equation (26) takes the following form:

$$C_1 = \frac{i u_{li}^* I_R - i u_{lo}^*}{K'_n(kr_i)(I_R - K_R)} \quad (27)$$

$$B_1 = \frac{i u_{lo}^*}{I'_n(kr_o)} - \frac{i u_{li}^* I_R K_R}{I'_n(kr_o)(I_R - K_R)} + \frac{i u_{lo}^* K_R}{I'_n(kr_o)(I_R - K_R)} \quad (28)$$

For the air inside the cylinder it can be realised that

$$C_{ai} = 0 \quad (29)$$

since $K_n(kr)$ has a singularity at $r = 0$. Similarly,

$$B_{ai} = \frac{i u_{ai}^*}{I'_n(kr_i)} \quad (30)$$

For the air outside the cylinder, since $I_n(kr)$ has a singularity at $r = \infty$, then

$$B_{ao} = 0 \quad (31)$$

and

$$C_{ao} = \frac{i u_{ao}^*}{K'_n(kr_o)} \quad (32)$$

Thus B_l , C_l , B_{ai} , C_{ai} , B_{ao} and C_{ao} are defined as functions of u_{lo}^* , u_{li}^* , u_{ao}^* , u_{ai}^* . These variables have the following expression:

$$u_{lo} = \left(\frac{\partial}{\partial t} + \frac{A_1}{r^2} \frac{\partial}{\partial \theta} + W_1 \frac{\partial}{\partial z} \right) \delta_o \quad \text{at } r = r_o \quad (33)$$

and

$$u_{li} = \left(\frac{\partial}{\partial t} + \frac{A_1}{r^2} \frac{\partial}{\partial \theta} + W_1 \frac{\partial}{\partial z} \right) \delta_i \quad \text{at } r = r_i \quad (34)$$

Since δ_o and δ_i in the liquid are equal to those in air, then

$$u_{ao} = \left(\frac{\partial}{\partial t} + \frac{A_{ai}}{r^2} \frac{\partial}{\partial \theta} + W_{ao} \frac{\partial}{\partial z} \right) \delta_o \quad \text{at } r = r_o \quad (35)$$

$$u_{ai} = \left(\frac{\partial}{\partial t} + \frac{A_{ai}}{r^2} \frac{\partial}{\partial \theta} + W_{ai} \frac{\partial}{\partial z} \right) \delta_i \quad \text{at } r = r_i \quad (36)$$

$$\text{Since} \quad \delta_o = \delta_o^* \exp(-i\omega t + in\theta + ikz) \quad (37)$$

$$\text{and} \quad \delta_i = \delta_i^* \exp(-i\omega t + in\theta + ikz) \quad (38)$$

From equations (11) and (33) to (38) the following is obtained:

$$\begin{aligned} u_{lo}^* &= -iH_1 \delta_o^* \\ u_{li}^* &= -iH_3 \delta_i^* \\ u_{ao}^* &= -iH_9 \delta_o^* \\ u_{ai}^* &= -iH_{10} \delta_i^* \end{aligned} \quad (39)$$

Substituting equations (27) and (28) into equations (22) and (23), the following expressions for the pressure can be obtained:

$$p_{lo}^* = \frac{\rho_l}{k} H_1 \left[H_4 H_1 I_n(kr_o) \delta_o^* - H_5 H_3 I_n(kr_o) \delta_i^* + H_7 H_3 K_n(kr_o) \delta_i^* - H_8 H_1 K_n(kr_o) \delta_o^* \right] \quad (40)$$

$$p_{li}^* = \frac{\rho_l}{k} H_3 \left[H_4 H_1 I_n(kr_i) \delta_o^* - H_5 H_3 I_n(kr_i) \delta_i^* + H_7 H_3 K_n(kr_i) \delta_i^* - H_8 H_1 K_n(kr_i) \delta_o^* \right] \quad (41)$$

Similarly, considering equations (29) to (32) for air inside and the outside of the cylinder, then

$$p_o^* = \frac{\rho_{ao}}{k} H_9^2 \frac{K_n(kr_o)}{K_n'(kr_o)} \quad (42)$$

$$p_i^* = \frac{\rho_{ai}}{k} H_{10}^2 \frac{I_n(kr_i)}{I_n'(kr_i)} \quad (43)$$

where

$$H_1 = \omega - \frac{nA_1}{r_o^2} - W_1 k \quad (44)$$

$$H_3 = \omega - \frac{nA_1}{r_i^2} - W_1 k \quad (45)$$

$$H_4 = \left(\frac{1}{I_n'(kr_o)} + \frac{K_R}{H_6 * I_n'(kr_o)} \right) \quad (46)$$

$$H_5 = K_R * \frac{I_R}{H_6 * I_n'(kr_o)} \quad (47)$$

$$H_6 = I_R - K_R \quad (48)$$

$$H_7 = \frac{I_R}{H_6 * K_n'(kr_i)} \quad (49)$$

$$H_8 = \frac{1}{H_6 * K_n'(kr_i)} \quad (50)$$

$$H_9 = \omega - \frac{nA_{ao}}{r_o^2} - W_{ao} k \quad (51)$$

$$H_{10} = \omega - \frac{nA_{ai}}{r_i^2} - W_{ai} k \quad (52)$$

$$H_{11} = \sigma \left(\frac{-1}{r_o^2} + \frac{n^2}{r_o^2} + k^2 \right) \quad (53)$$

$$H_{12} = -\sigma \left(\frac{-1}{r_i^2} + \frac{n^2}{r_i^2} + k^2 \right) \quad (54)$$

Substituting equations (40) to (43) into equations (20) and (21) the following the be obtained:

$$A_1 \delta_o^{\bullet} + B_1 \delta_i^{\bullet} = 0 \quad (55)$$

$$A_2 \delta_o^{\bullet} + B_2 \delta_i^{\bullet} = 0 \quad (56)$$

Finally, by solving these two equations, one yields to the **characteristic equation**:

$$B_2 A_1 - B_1 A_2 = 0 \quad (57)$$

This equation can be rearranged to be in the following form:

$$a_4 \omega^4 + a_3 \omega^3 + a_2 \omega^2 + a_1 \omega + a_0 = 0 \quad (58)$$

The above equation represents the final characteristic equation used to calculate the growth rate ω . The coefficients of this equation are function of the wavelength, the nozzle exit conditions, the axial and the swirl air velocity and the physical properties of the air and the liquid sheet. Their values can be calculated from the following set of equations:

$$a_4 = g_2 \times g_8 + g_2 \times g_9 + g_3 \times g_8 + g_3 \times g_9 - g_4 \times g_6 \quad (59)$$

$$a_3 = g_2 \times g_8 \times p_1 + g_2 \times g_9 \times p_9 + g_3 \times g_8 \times p_{13} + g_3 \times g_9 \times p_5 - g_4 \times g_6 \times p_1 \quad (60)$$

$$a_2 = g_1 \times g_8 + g_1 \times g_9 + g_2 \times g_7 + g_2 \times g_8 \times p_2 - g_2 \times h_{12} + g_2 \times g_9 \times p_{10} - g_8 \times h_{11} \\ + g_3 \times g_7 + g_3 \times g_8 \times p_{14} - g_3 \times h_{12} + g_3 \times g_9 \times p_6 - g_4 \times g_6 \times p_2 - g_9 \times h_{11} \quad (61)$$

$$a_1 = g_1 \times g_8 \times f_3 + g_1 \times g_9 \times f_5 + g_2 \times g_7 \times f_1 + g_2 \times g_8 \times p_3 - g_2 \times h_{12} \times f_1 + g_2 \times g_9 \times p_{11} \\ - g_8 \times h_{11} \times f_3 + g_3 \times g_7 \times f_7 + g_3 \times g_8 \times p_{15} - g_3 \times h_{12} \times f_7 + g_3 \times g_9 \times p_7 - g_4 \times g_6 \times p_3 - \\ g_9 \times h_{11} \times f_5 \quad (62)$$

$$a_0 = g_1 \times g_7 + g_1 \times g_8 \times f_4 - g_1 \times h_{12} + g_1 \times g_9 \times f_6 + g_2 \times g_7 \times f_2 + g_2 \times g_8 \times p_4 \\ - g_2 \times h_{12} \times f_2 + g_2 \times g_9 \times p_{12} - g_7 \times h_{11} - g_8 \times h_{11} \times f_4 + h_{11} \times h_{12} + g_3 \times g_7 \times f_8 \\ + g_3 \times g_8 \times p_{16} - g_3 \times h_{12} \times f_8 + g_3 \times g_9 \times p_8 - g_4 \times g_6 \times p_4 - g_9 \times h_{11} \times f_6 \quad (63)$$

$$p_1 = f_3 + f_1 \quad (64)$$

$$p_2 = f_4 + f_1 \times f_3 + f_2 \quad (65)$$

$$p_3 = f_1 \times f_4 + f_2 \times f_3 \quad (66)$$

$$p4 = f2 \times f4 \quad (67)$$

$$p5 = f7 + f5 \quad (68)$$

$$p6 = f8 + f5 \times f7 + f6 \quad (69)$$

$$p7 = f5 \times f8 + f6 \times f7 \quad (70)$$

$$p8 = f6 \times f8 \quad (71)$$

$$p9 = f5 + f1 \quad (72)$$

$$p10 = f6 + f1 \times f5 + f2 \quad (73)$$

$$p11 = f1 \times f6 + f2 \times f5 \quad (74)$$

$$p12 = f2 \times f6 \quad (75)$$

$$p13 = f3 \times f7 \quad (76)$$

$$p14 = f4 + f7 \times f3 + f8 \quad (77)$$

$$p15 = f7 \times f4 + f8 \times f3 \quad (78)$$

$$p16 = f8 \times f4 \quad (79)$$

$$f1 = \frac{-2 * n * A}{r_o^2} - 2W_l k \quad (80)$$

$$f2 = \frac{n^2 * A^2}{r_o^4} + \frac{2nA_l}{r_o^2} W_l k + W_l^2 k^2 \quad (81)$$

$$f3 = \frac{-2 * n * A}{r_i^2} - 2W_l k \quad (82)$$

$$f4 = \frac{n^2 * A_l^2}{r_i^4} + \frac{2nA_l}{r_i^2} W_l k + W_l^2 k^2 \quad (83)$$

$$f5 = \frac{-2 * n * A_{ai}}{r_i^2} - 2 * k * W_{ai} \quad (84)$$

$$f6 = \frac{n^2 * A_{ai}^2}{r_i^4} + \frac{2 * n * k * A_{ai} * W_{ai}}{r_i^2} + W_{ai}^2 * k^2 \quad (85)$$

$$f7 = \frac{-2 * n * A_{ao}}{r_o^2} - 2 * W_{ao} * k \quad (86)$$

$$f8 = \frac{n^2 * A_{ao}^2}{r_o^4} + \frac{2 * n * k * A_{ao} * W_{ao}}{r_o^2} + W_{ao}^2 * k^2 \quad (87)$$

$$g1 = \frac{\rho_l * A^2}{r_o^3} - \frac{\rho_{ao} * A_{ao}^2}{r_o^3} \quad (88)$$

$$g2 = \frac{\rho_l}{k} (H_4 * I_n(kr_o) - H_8 * K_n(kr_o)) \quad (89)$$

$$g3 = -\frac{\rho_{ao}}{k} * \frac{K_n(kr_o)}{K_n'(kr_o)} \quad (90)$$

$$g4 = \frac{\rho_l}{k} (H_4 * I_n(kr_i) - H_8 * K_n(kr_i)) \quad (91)$$

$$g6 = \frac{\rho_l}{k} (H_7 * K_n(kr_o) - H_5 * I_n(kr_o)) \quad (92)$$

$$g7 = \frac{\rho_l * A_l^2}{r_i^3} - \frac{\rho_{ai} * A_{ai}^2}{r_i^3} \quad (93)$$

$$g8 = \frac{\rho_l}{k} (H_7 * K_n(kr_i) - H_5 * I_n(kr_i)) \quad (94)$$

$$g9 = -\frac{\rho_{ai}}{k} * \frac{I_n(kr_i)}{I_n'(kr_i)} \quad (95)$$

$$K_R = \frac{K_n'(kr_o)}{K_n'(kr_i)} \quad (96)$$

$$I_R = \frac{I_n'(kr_o)}{I_n'(kr_i)} \quad (97)$$

The roots of the characteristic equation (58) can be determined analytically from the following equations:

$$\begin{aligned} \omega_1 &= -A - B - C \\ \omega_2 &= -A - B + C \\ \omega_3 &= -A + B - C \\ \omega_4 &= -A + B + C \end{aligned} \quad (98)$$

where:

$$A = \frac{a_3}{4 * a_4} \quad (99)$$

$$B = \frac{1}{2} \sqrt{D_6} \quad (100)$$

$$C = \frac{1}{2} \sqrt{\left(\frac{a_3^2}{2a_4^2} - \frac{4a_2}{3a_4} - D_4 - D_5 - \frac{D_7}{4\sqrt{D_6}} \right)} \quad (101)$$

$$D_1 = a_2^2 - 3 \times a_3 \times a_1 + 12a_4 \times a_0 \quad (102)$$

$$D_2 = 2 \times a_2^3 - 9a_3 \times a_2 \times a_1 + 27 \times a_4 \times a_1 + 27 \times a_3^2 \times a_0 - 72 \times a_4 \times a_2 \times a_0 \quad (103)$$

$$D_3 = -4 \times D_1^3 + D_2^2 \quad (104)$$

$$D_4 = \frac{2^{\frac{1}{3}} D_1}{3 * a_4 * (D_2 + \sqrt{D_3})^{\frac{1}{3}}} \quad (105)$$

$$D_5 = \frac{(D_2 + \sqrt{D_3})^{1/3}}{3 * 2^{\frac{1}{3}} * a_4} \quad (106)$$

$$D_6 = \frac{a_3^2}{4a_4^2} - \frac{2a_2}{3a_4} + D_4 + D_5 \quad (107)$$

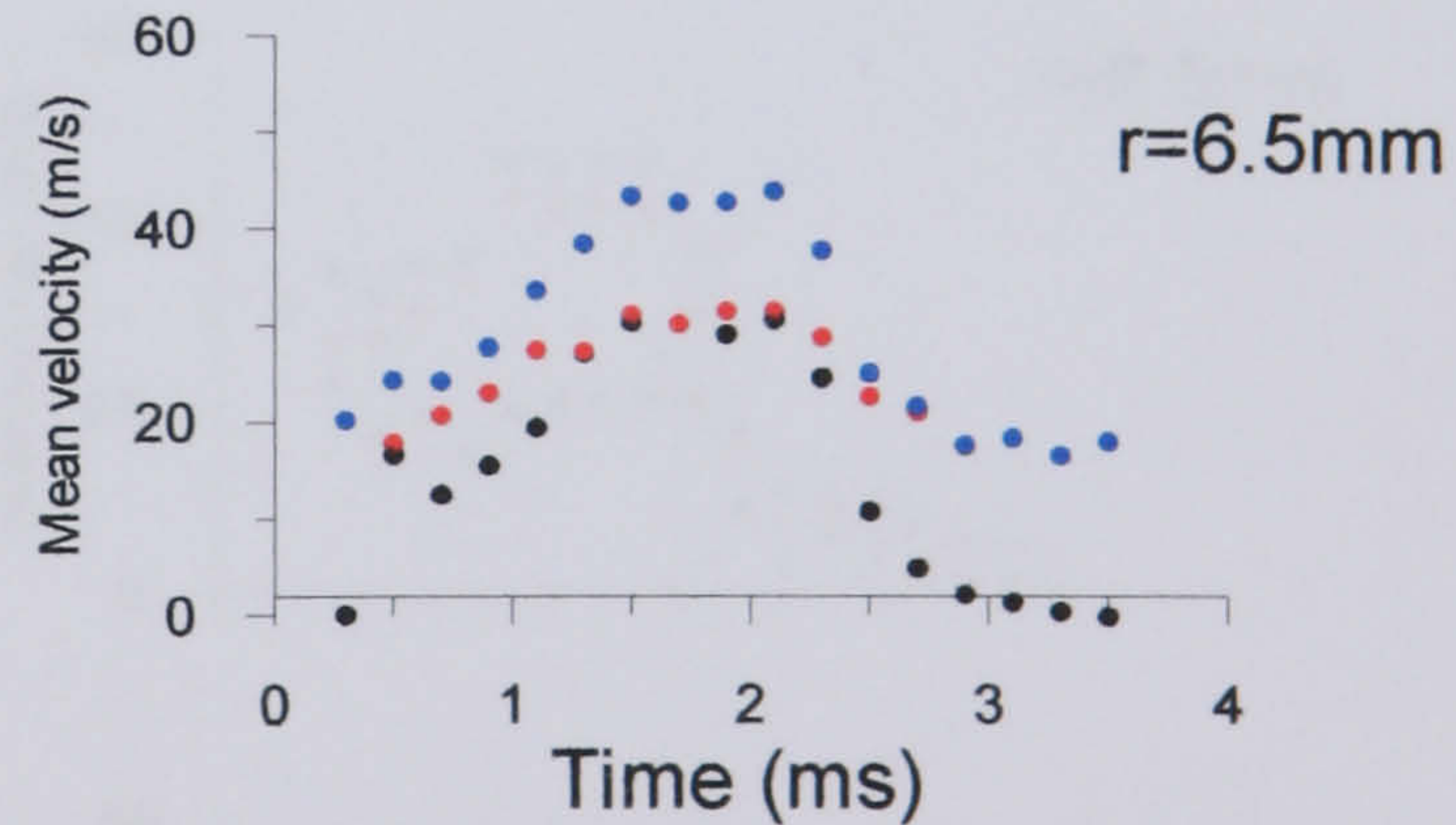
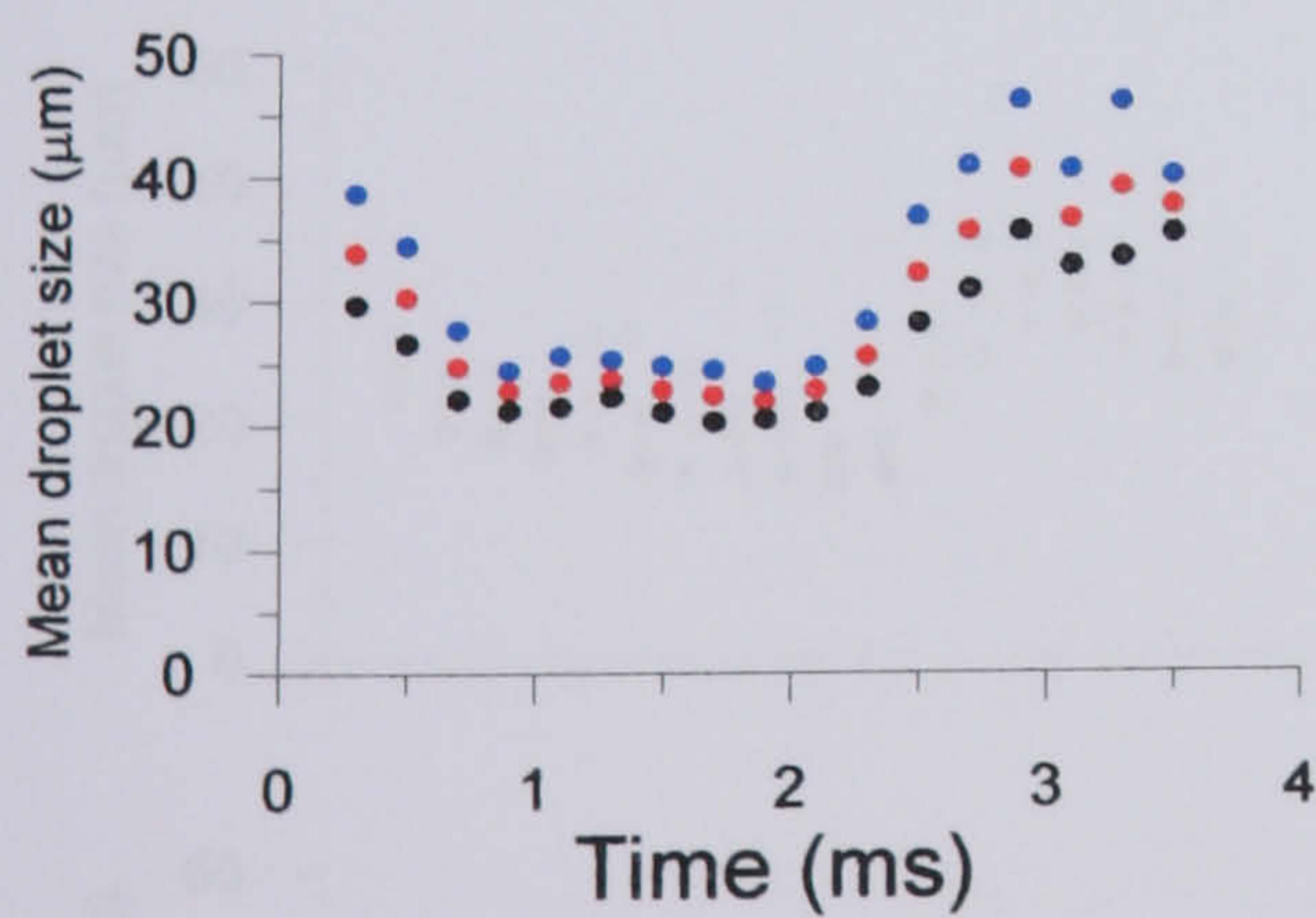
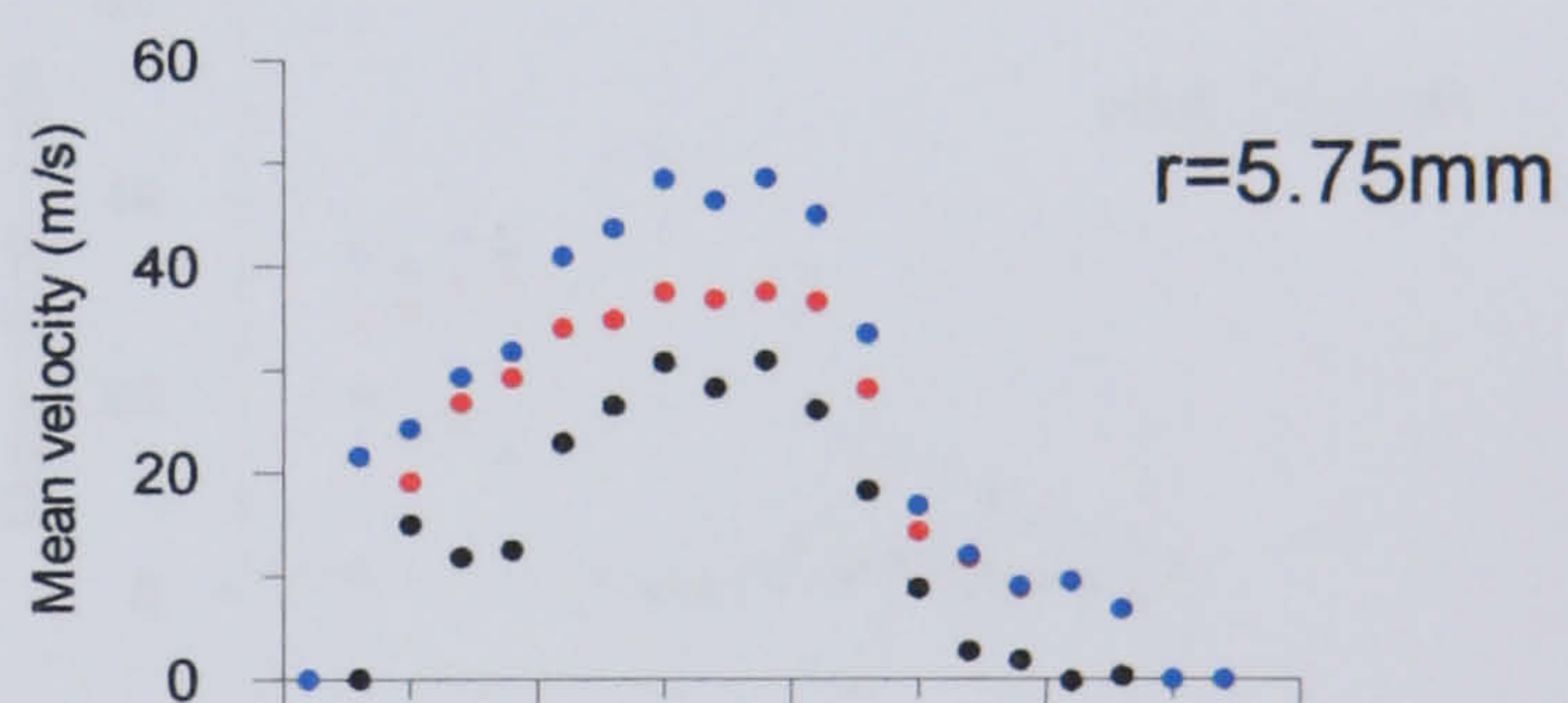
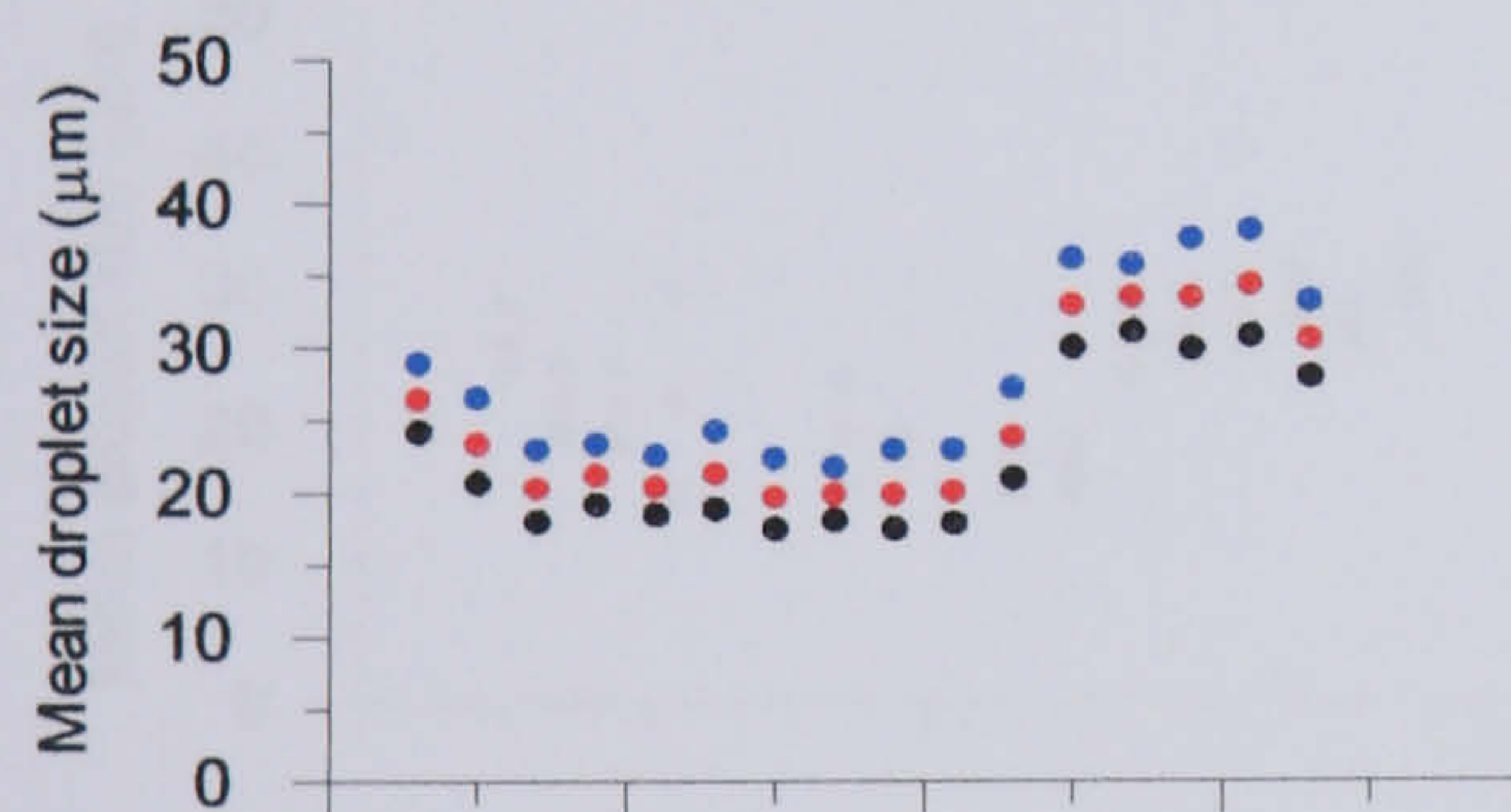
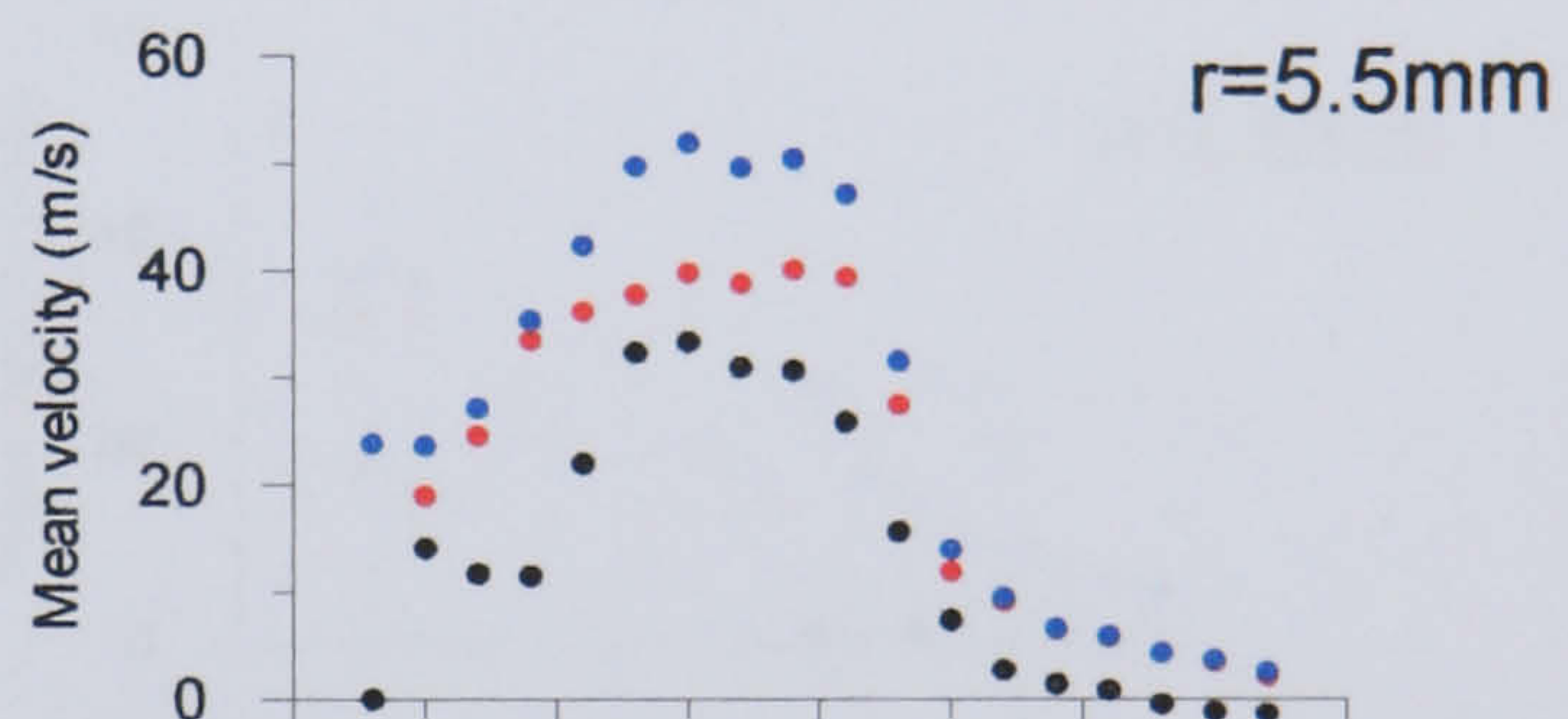
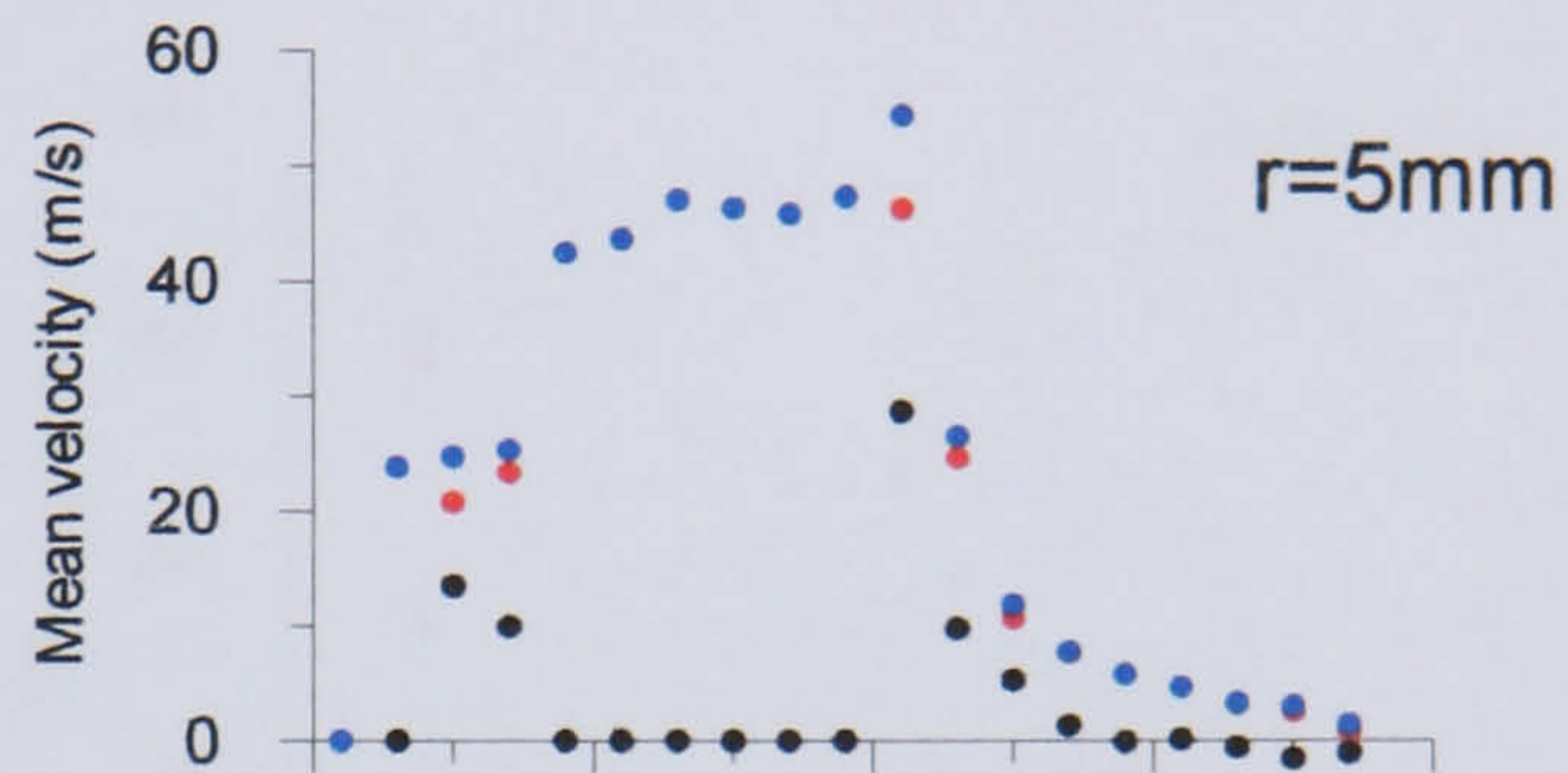
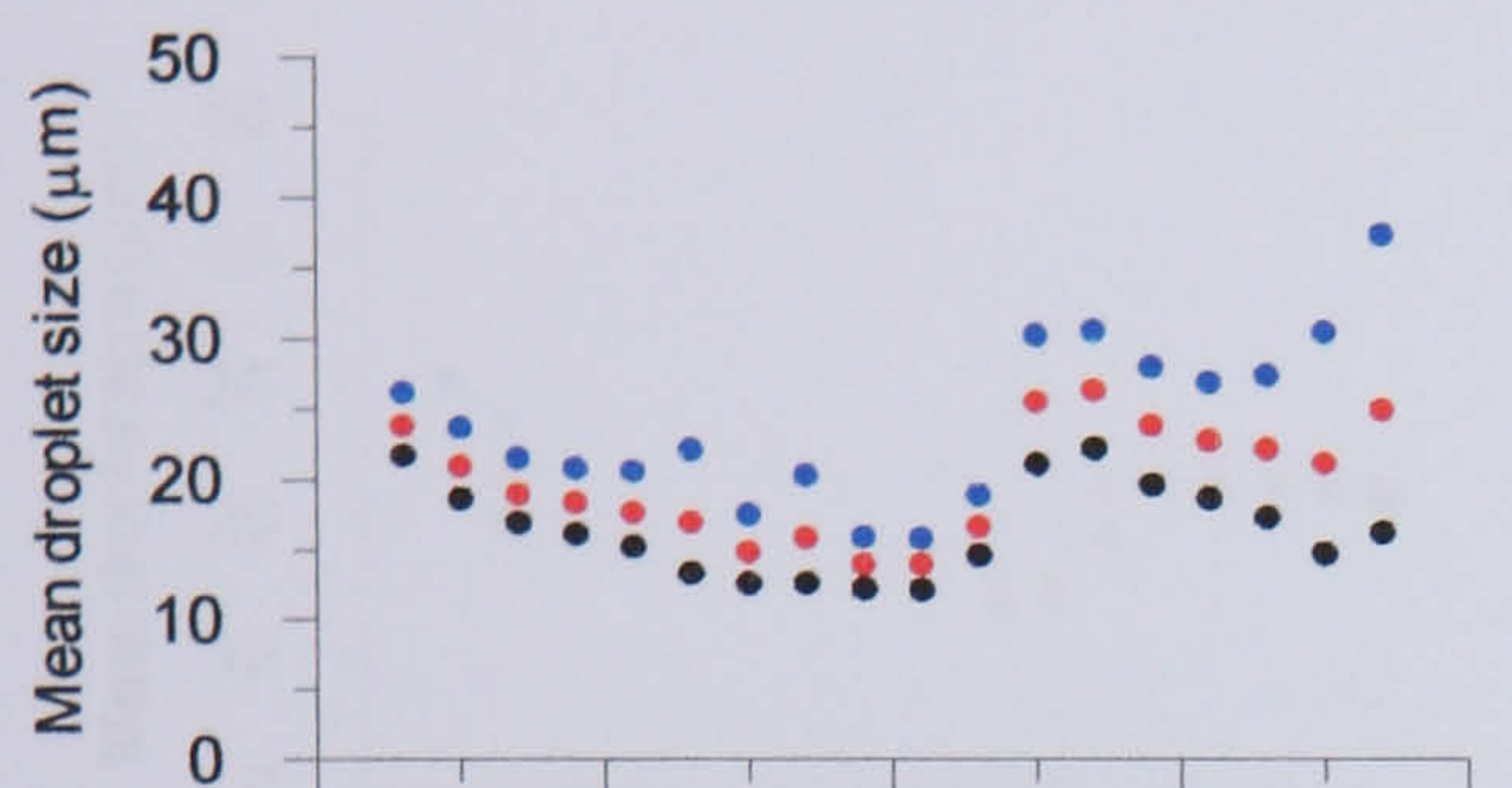
$$D_7 = -\frac{a_3^3}{a_4^3} + \frac{4a_3 a_2}{a_4^2} - \frac{8a_1}{a_4} \quad (108)$$

Out of these four roots, only the one with the maximum imaginary part is considered as the valid growth rate. The growth rate has to be calculated for all possible wave numbers in order to find its maximum value. In turn, this is considered as the **optimum growth rate** and its corresponding wave number as the **optimum wave number**.

Appendix 4-A

Temporal variation of mean droplet diameters and velocities
at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and different locations

$X = 5 \text{ mm}$

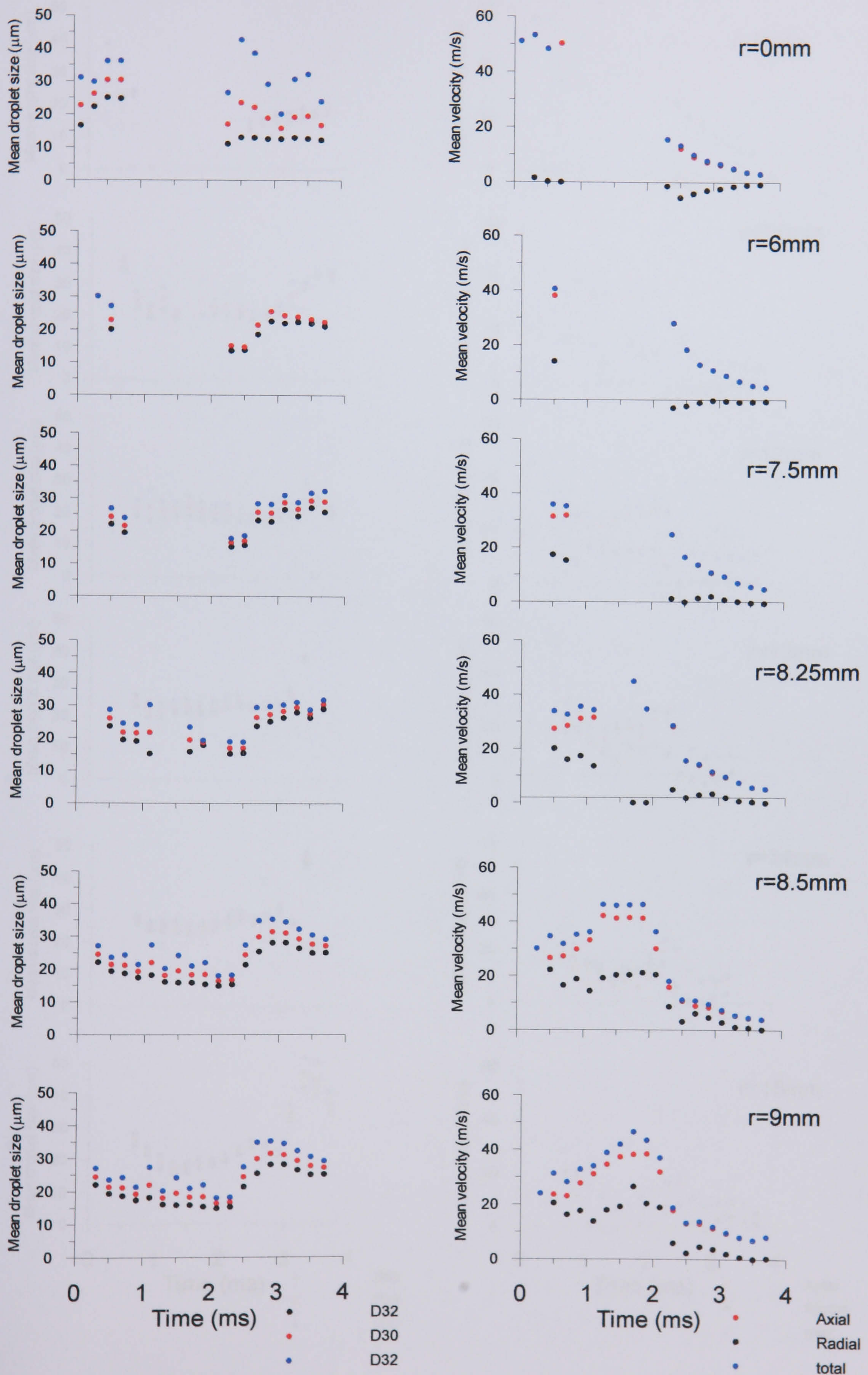


• D32
• D30
• D32

• Axial
• Radial
• total

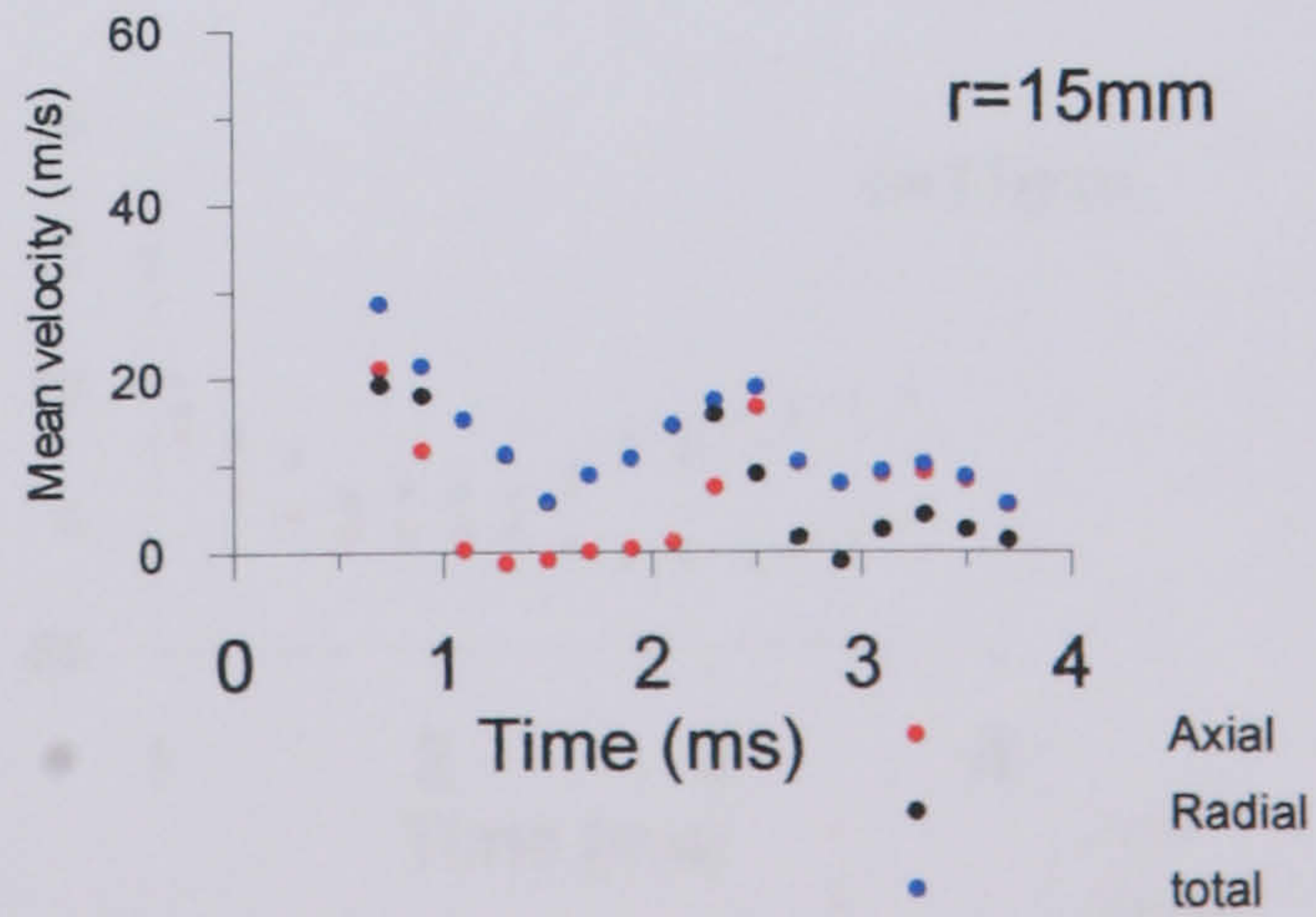
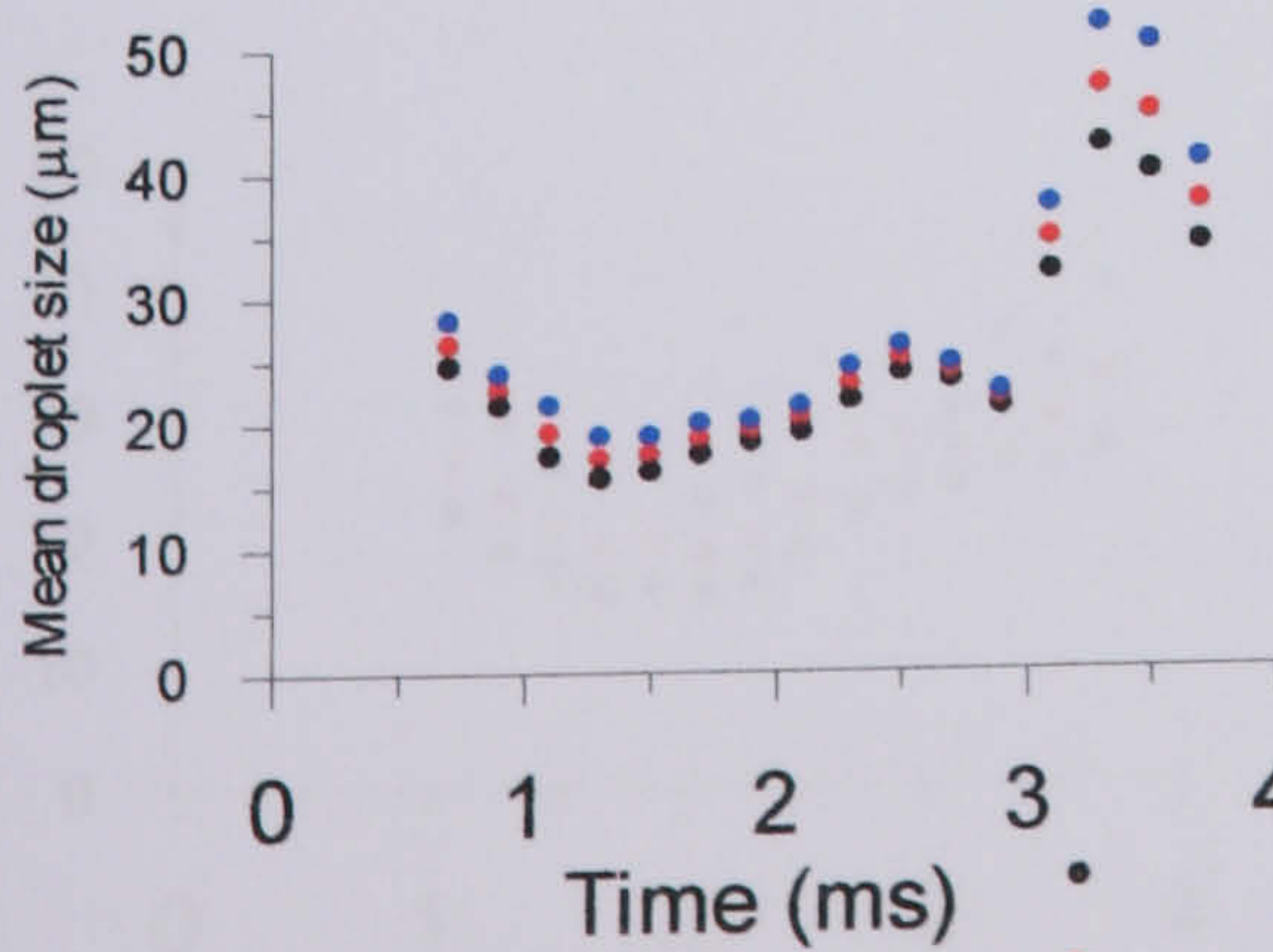
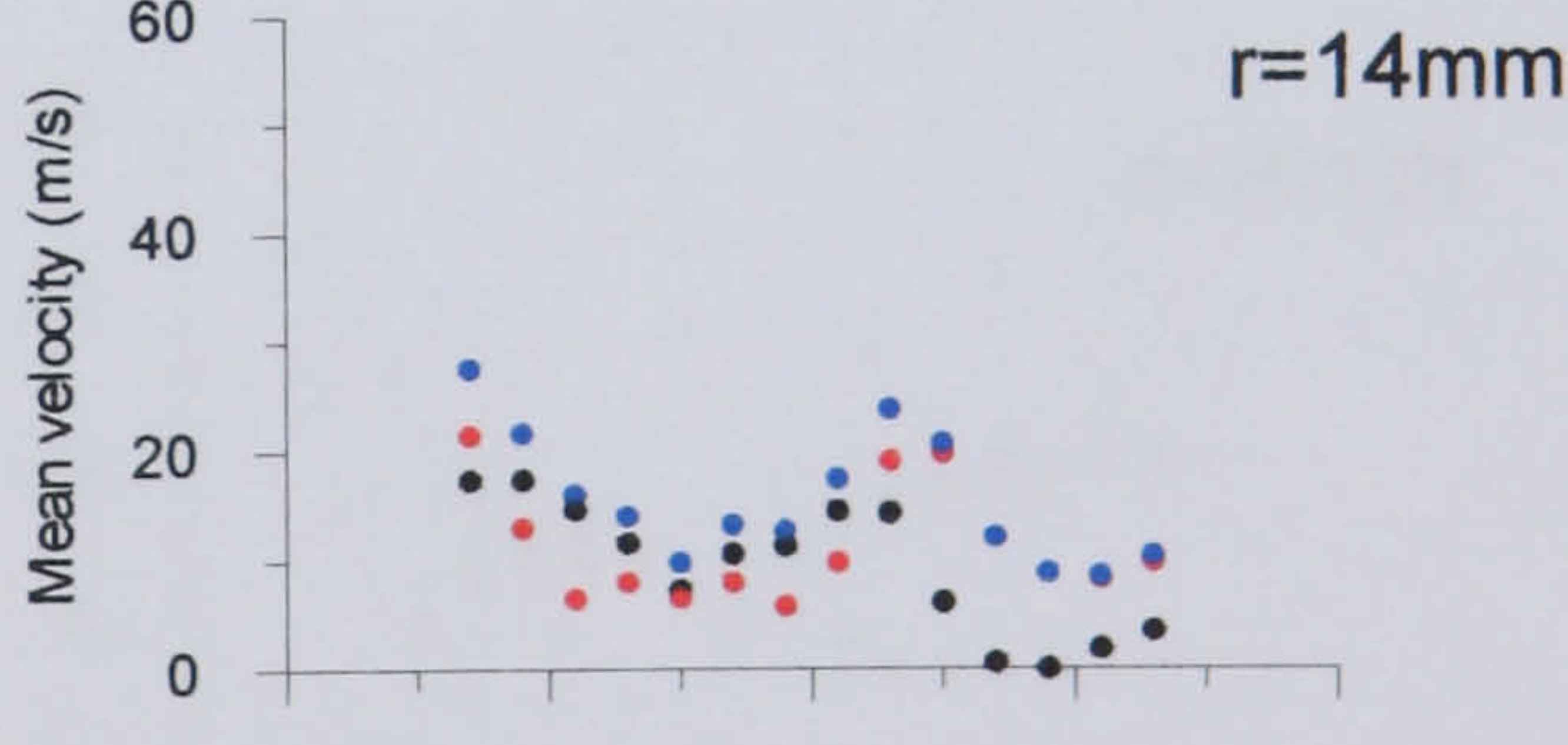
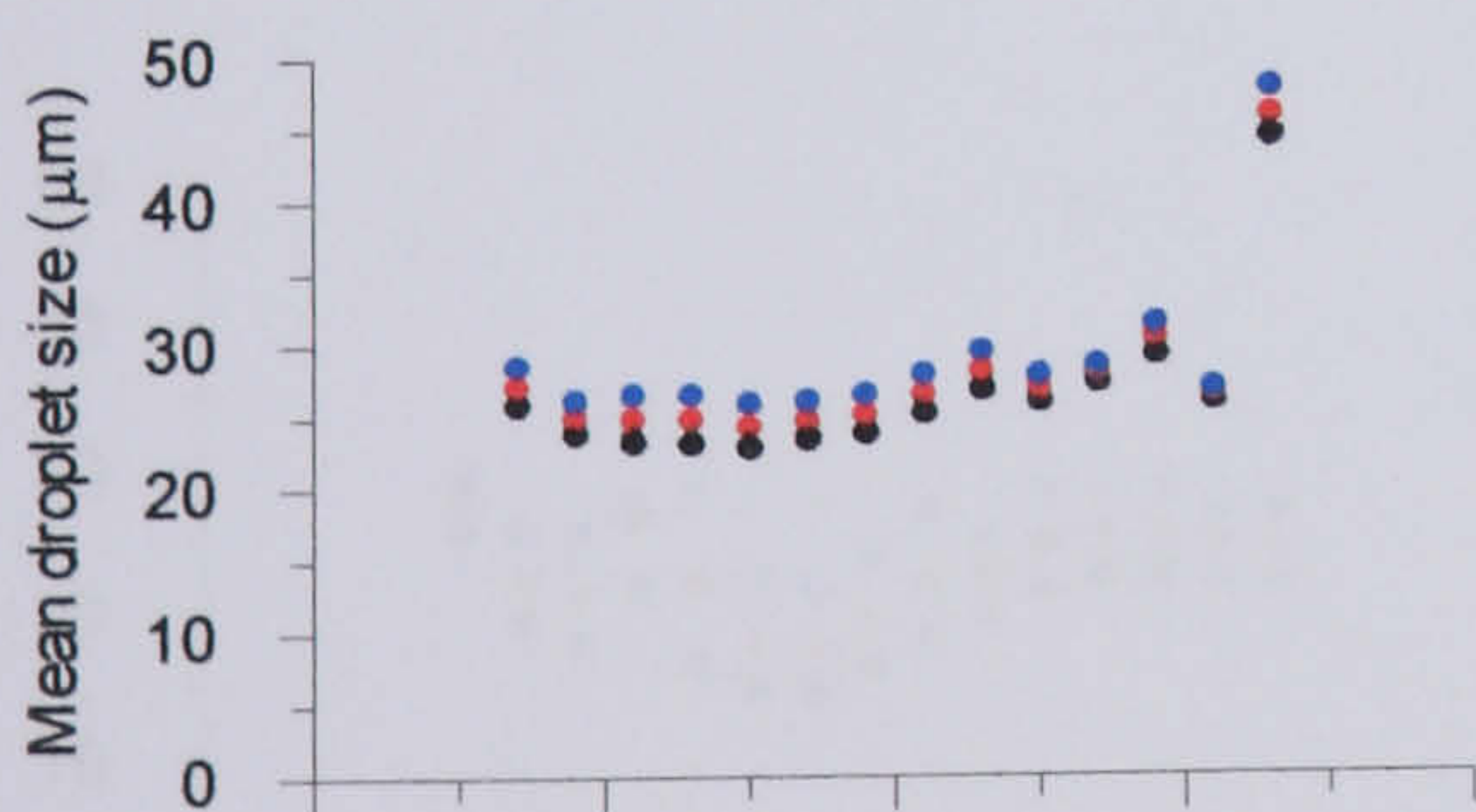
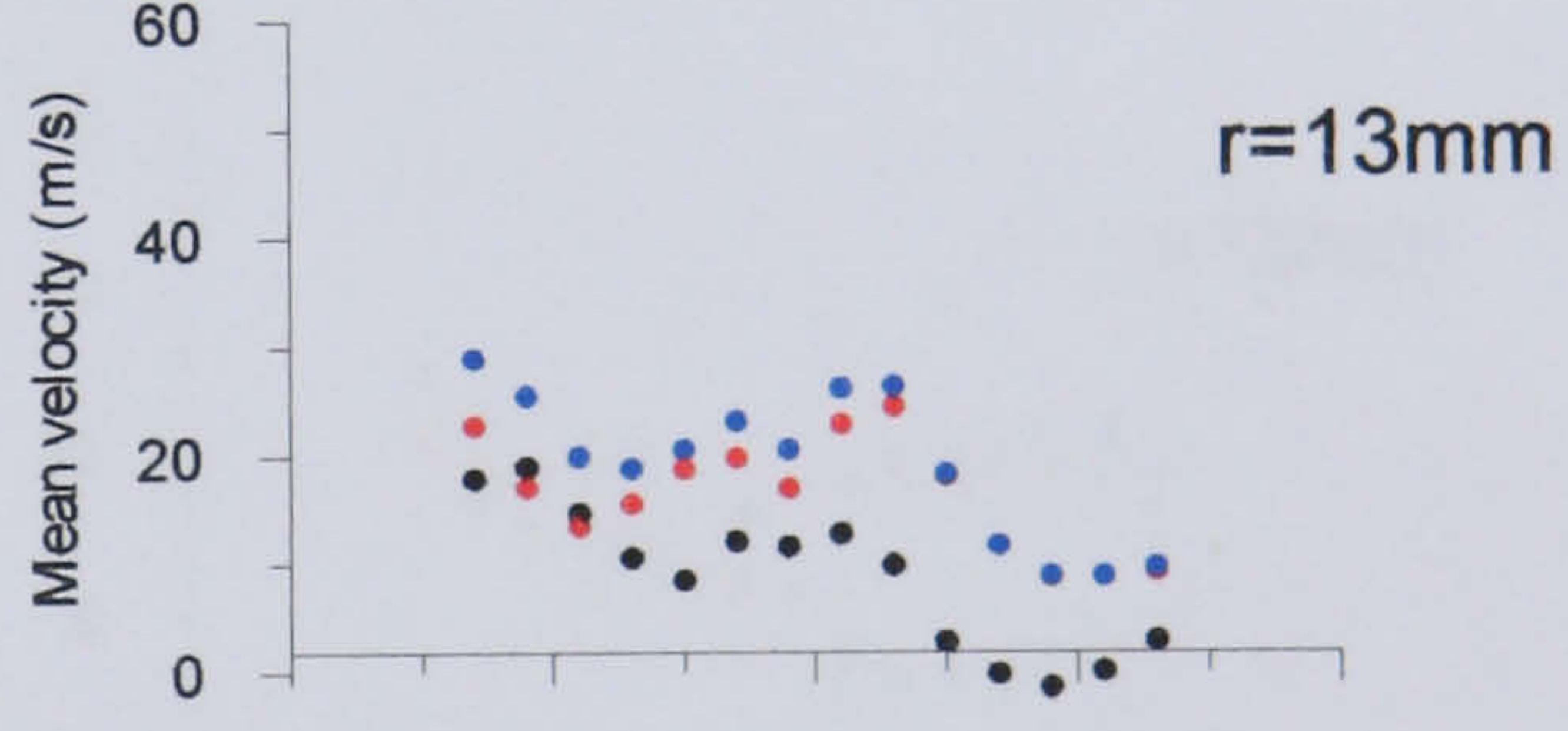
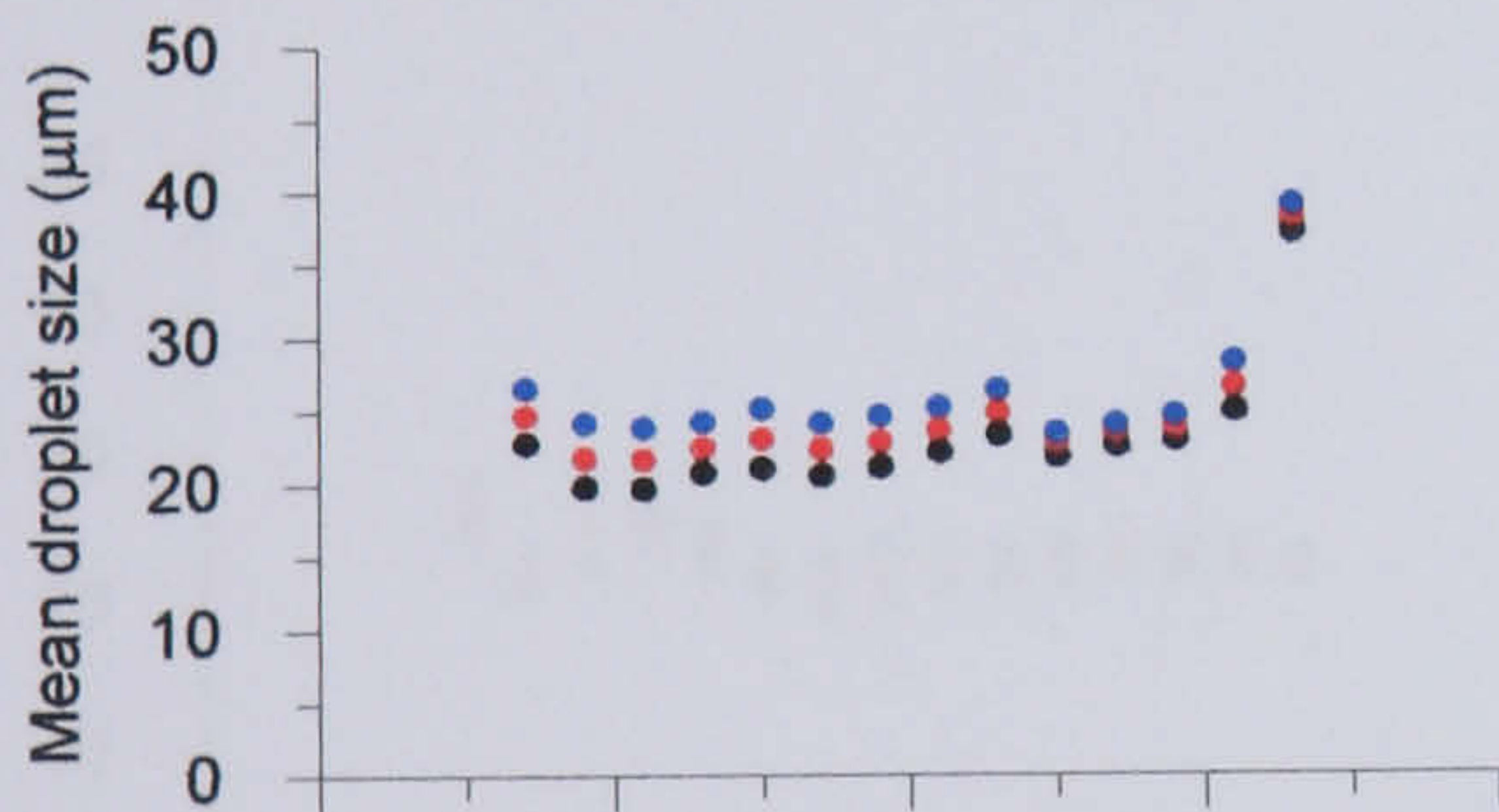
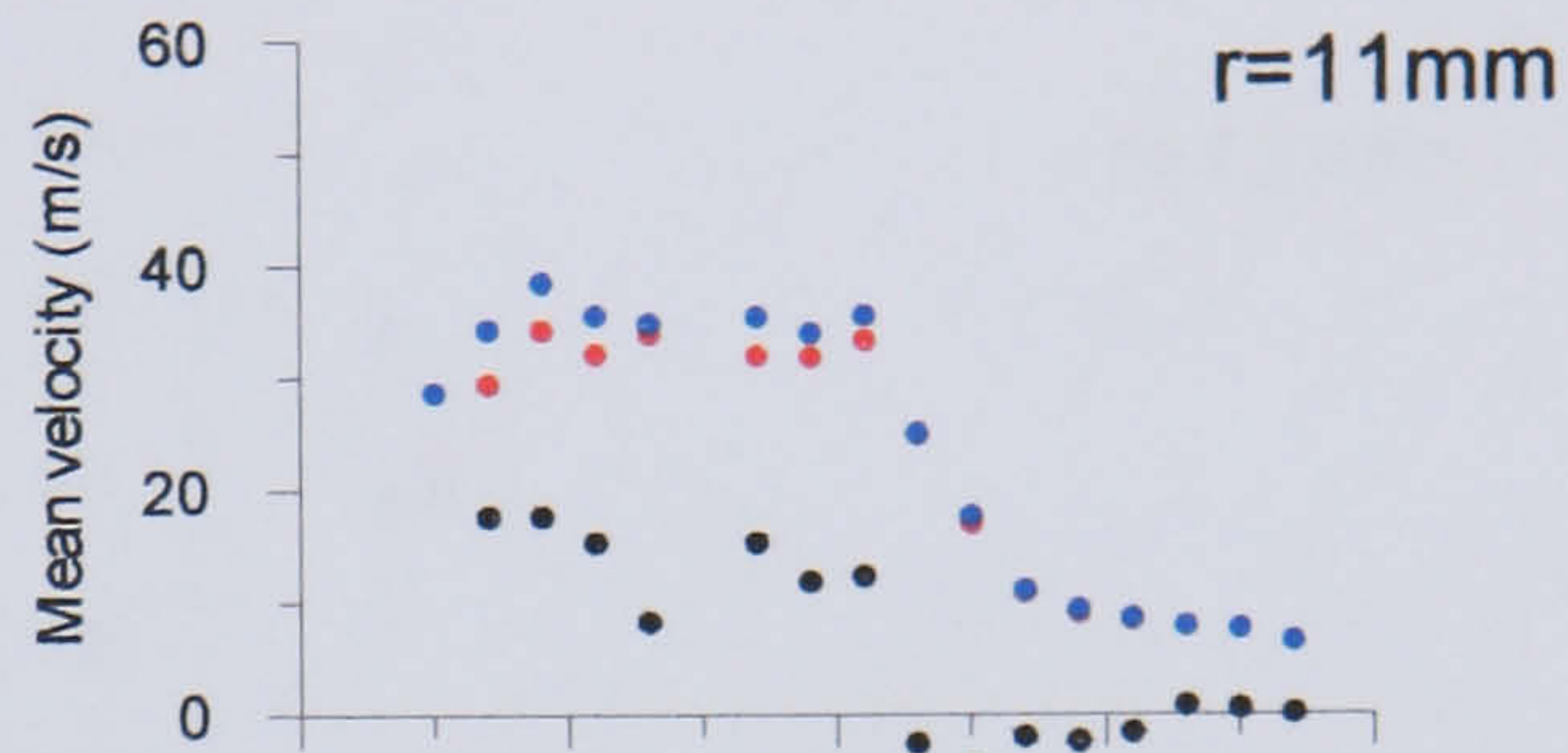
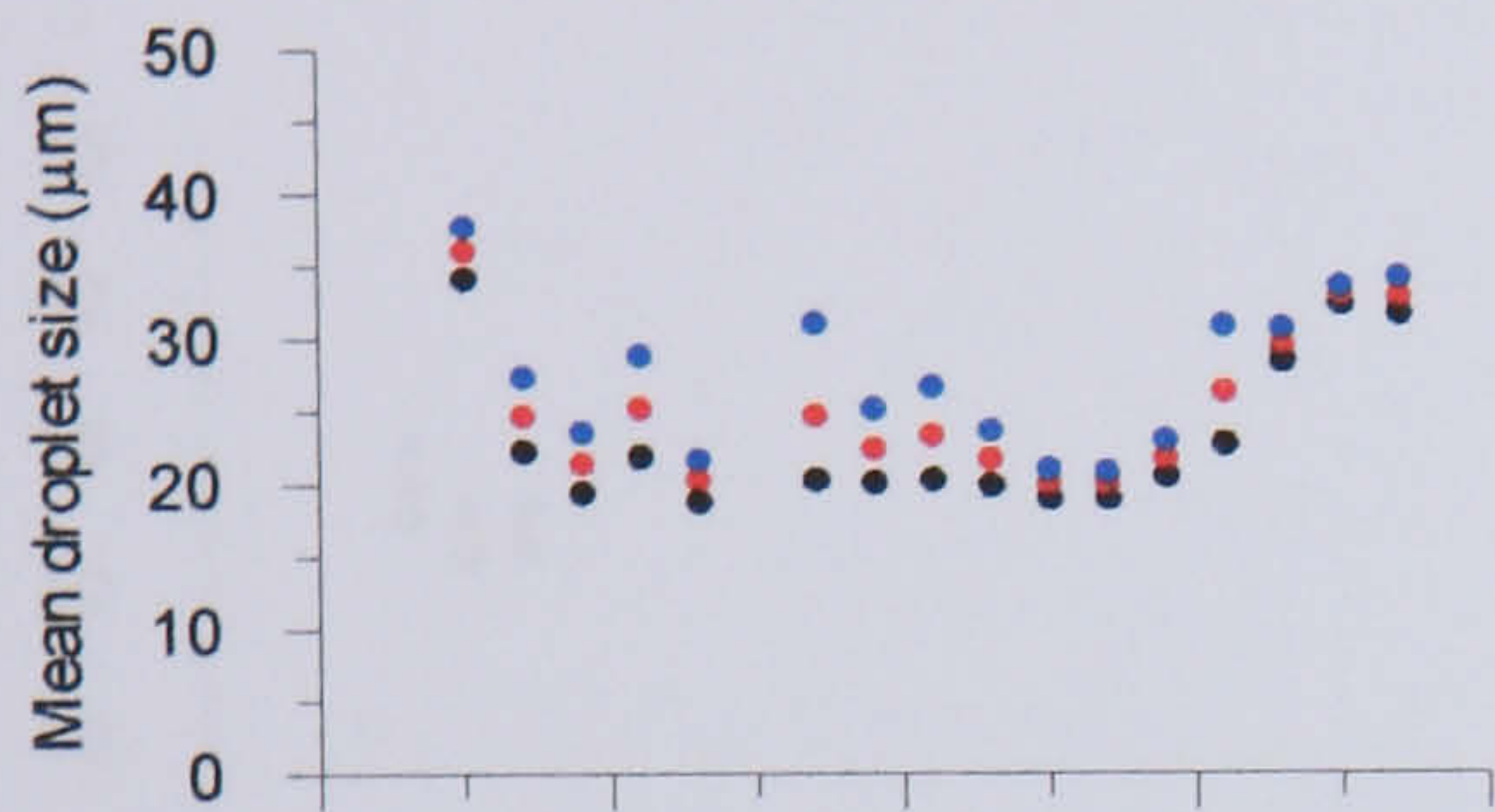
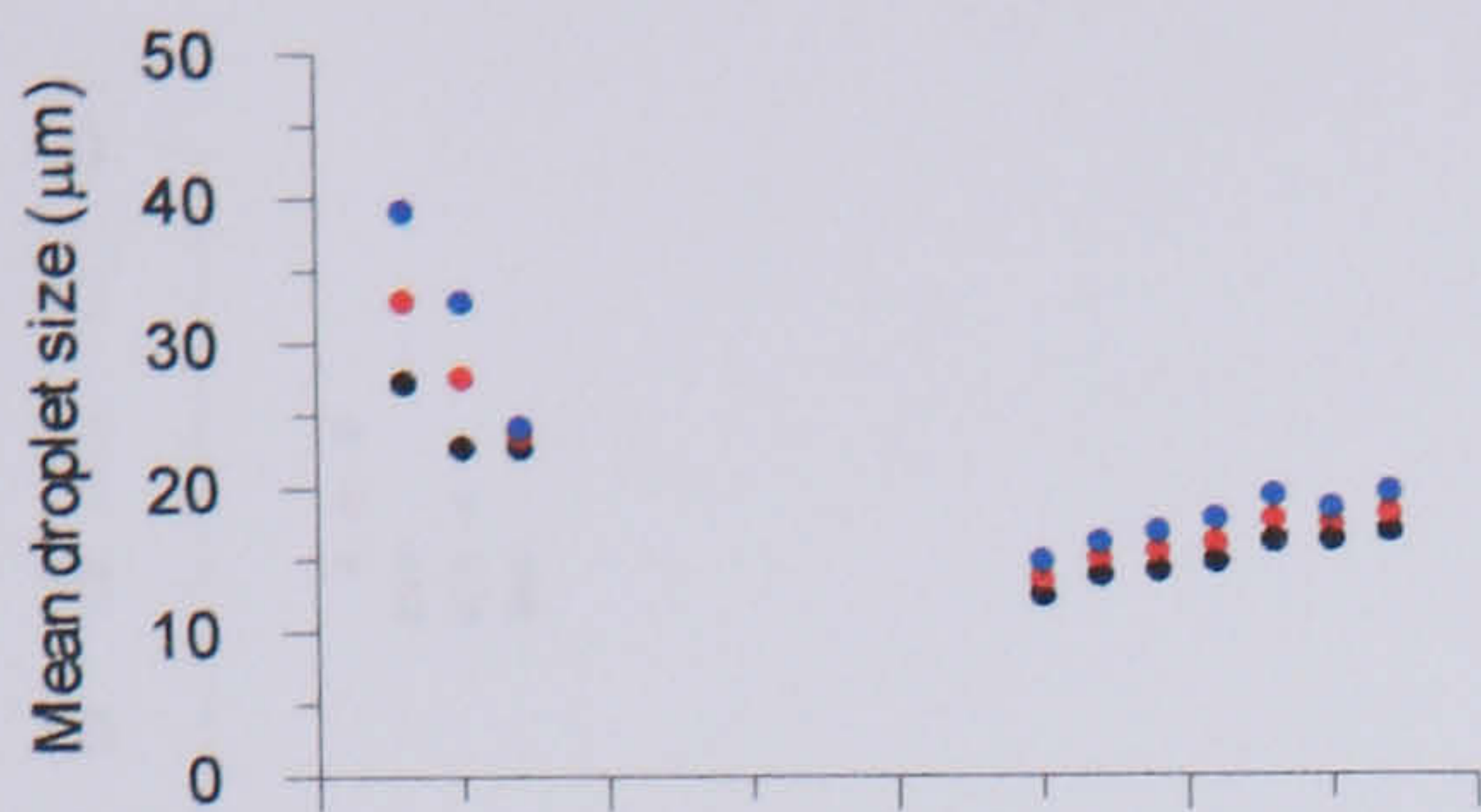
Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=10\text{ mm}$

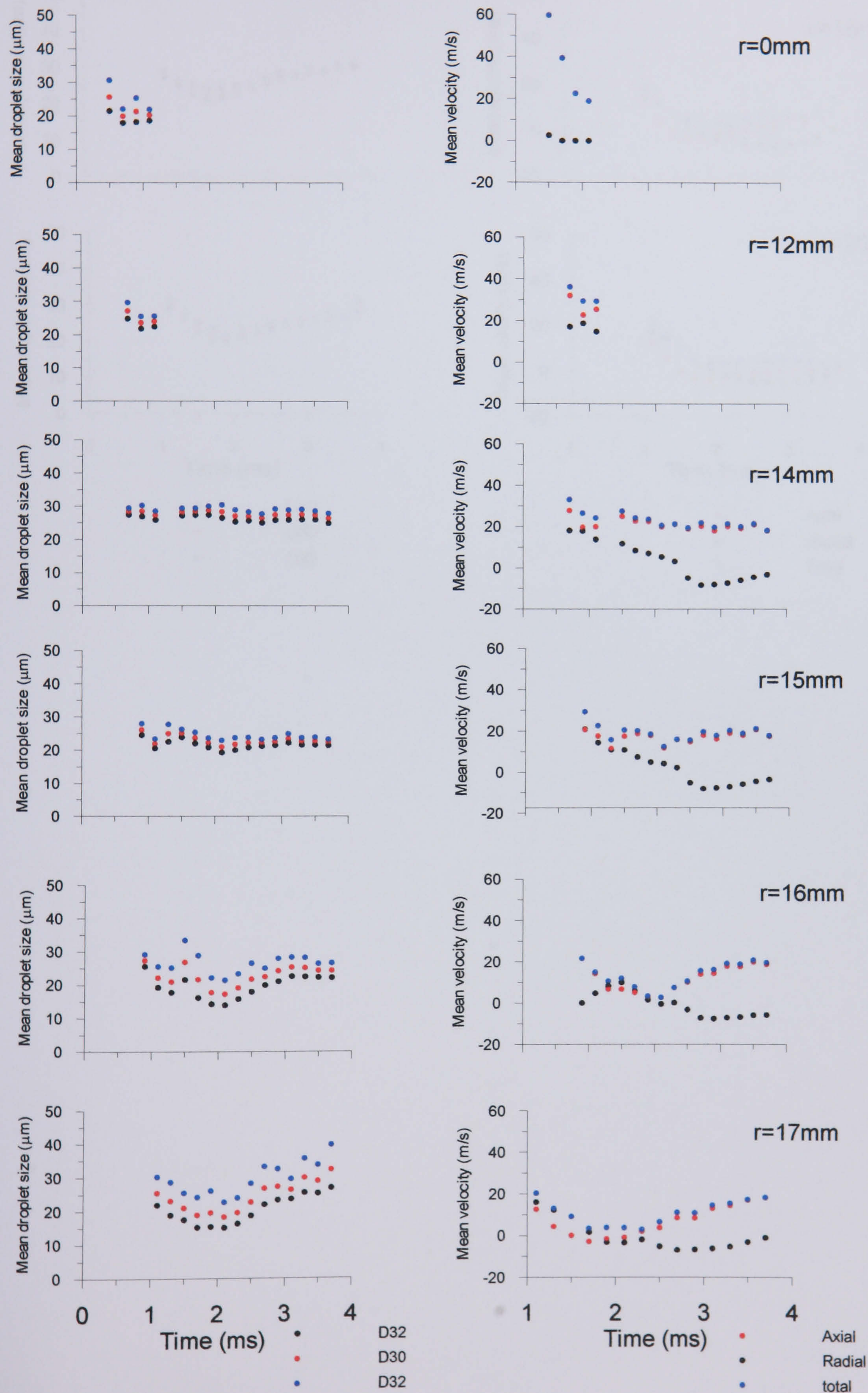


Temporal variation of mean droplet diameters and velocities

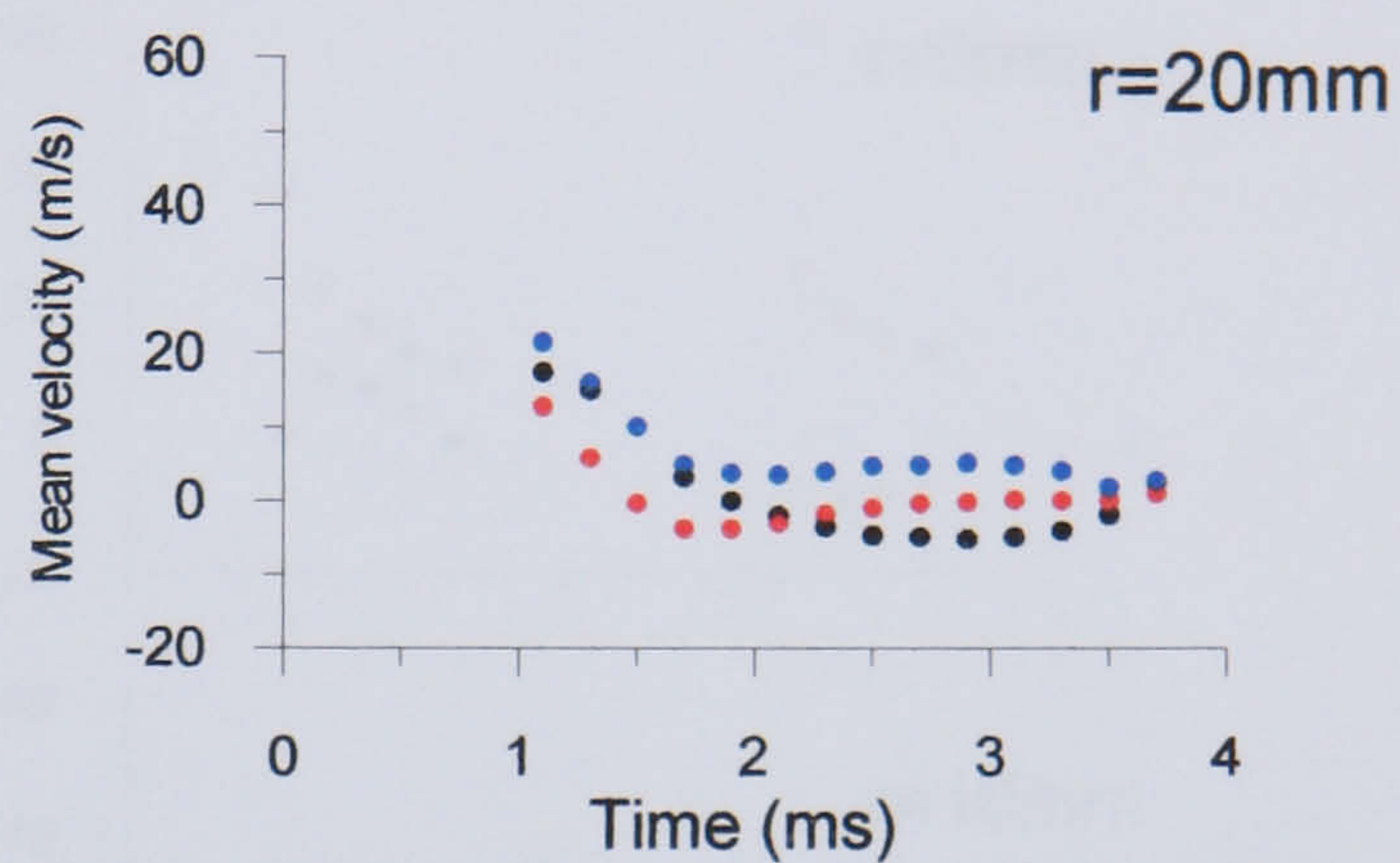
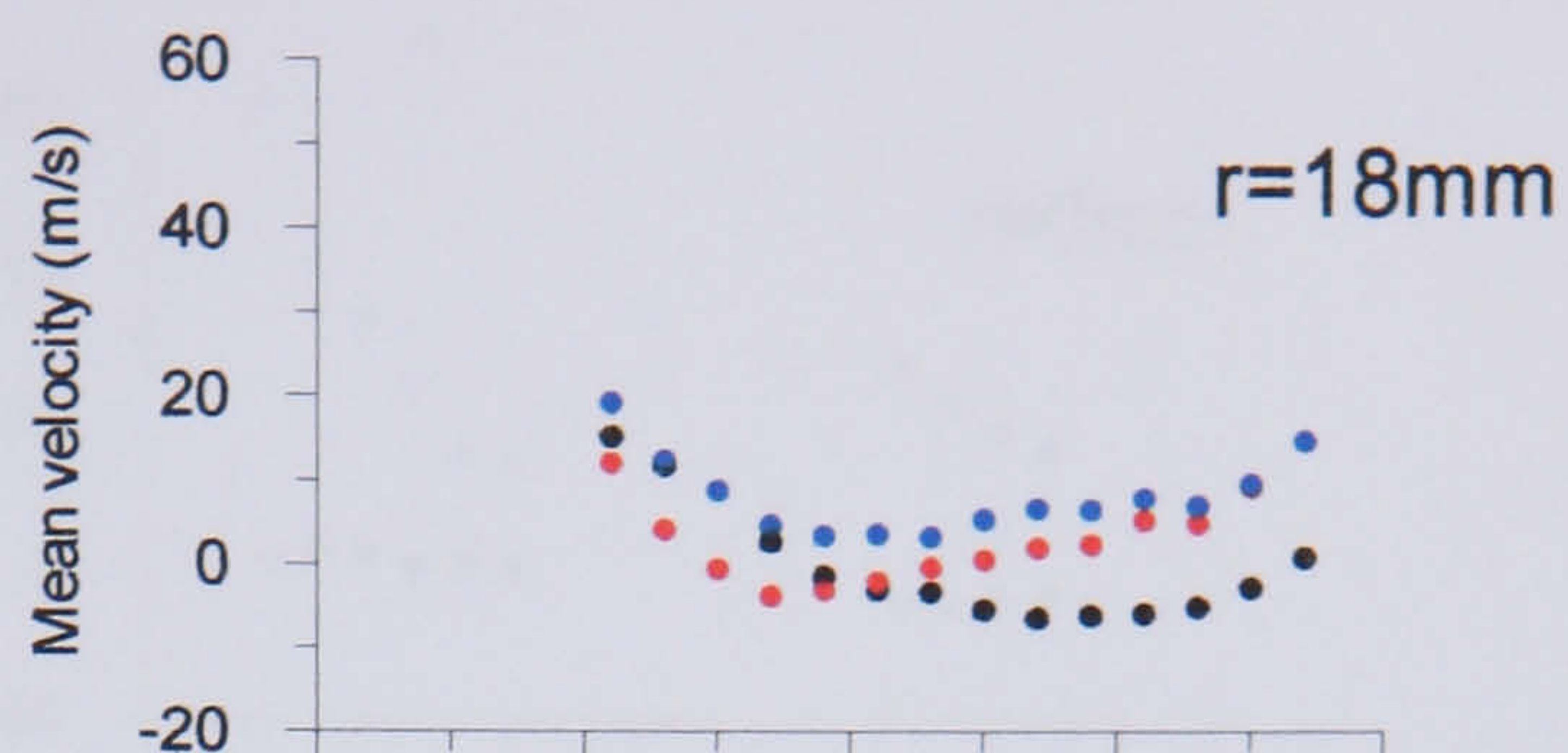
at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ at $X=15\text{mm}$



Temporal variation of mean droplet diameters and velocities at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=20\text{mm}$

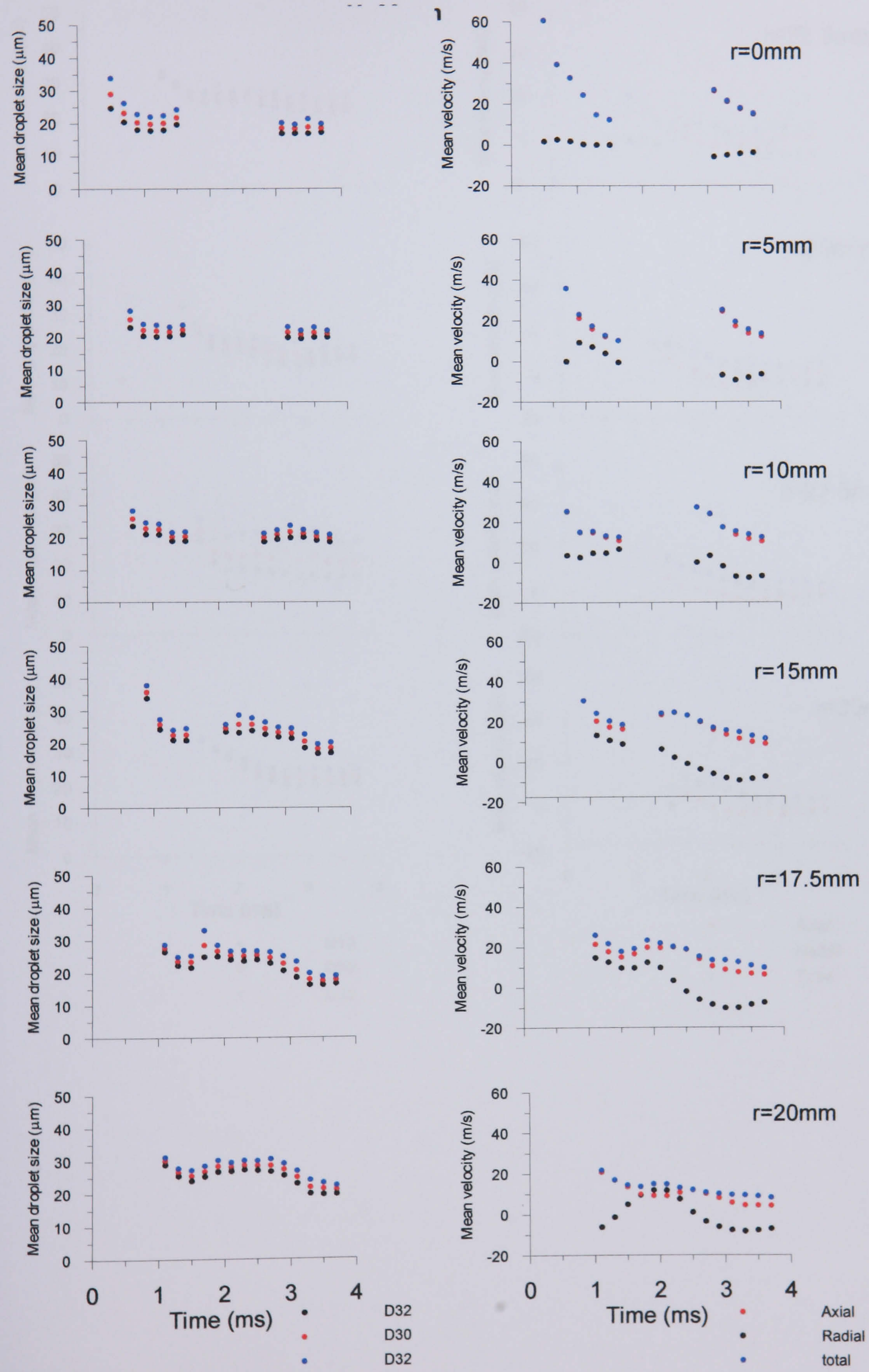


at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=20\text{mm}$ (continue)

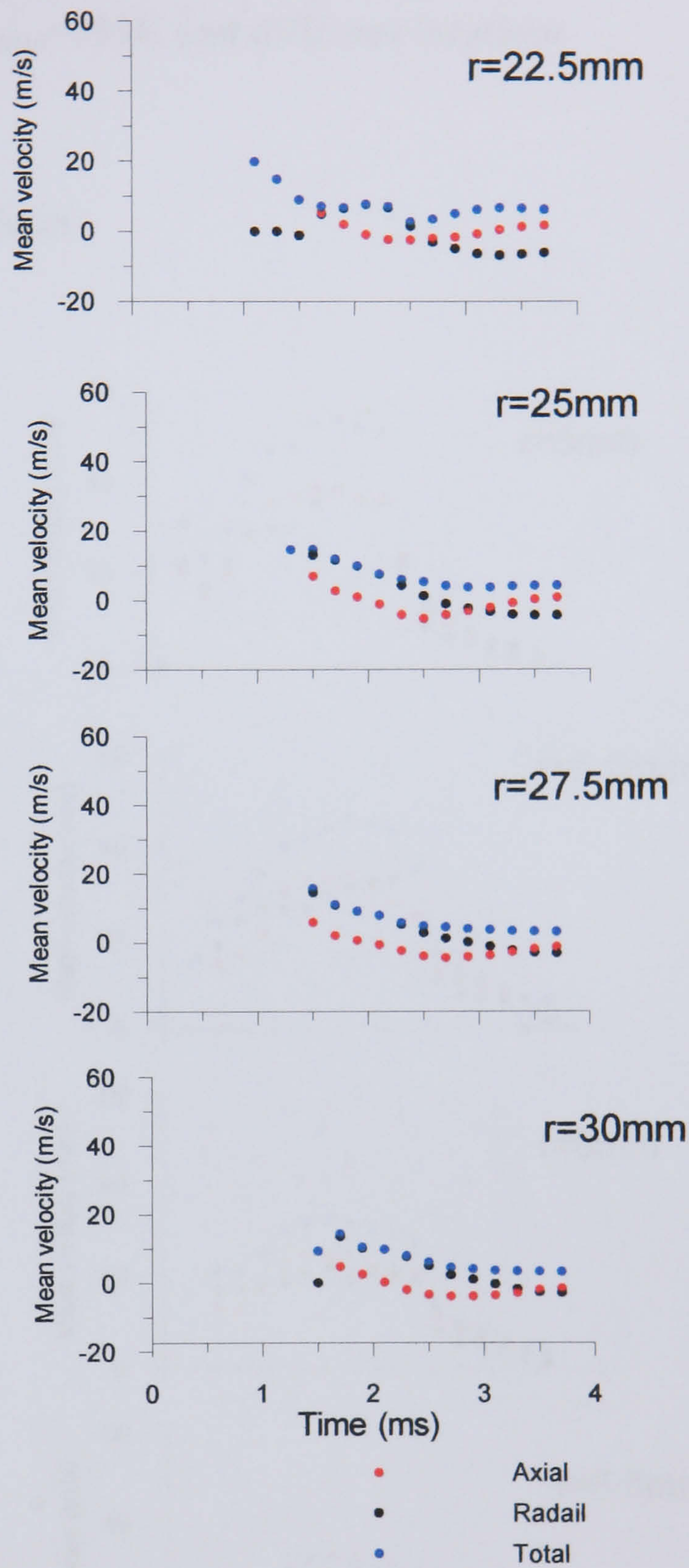
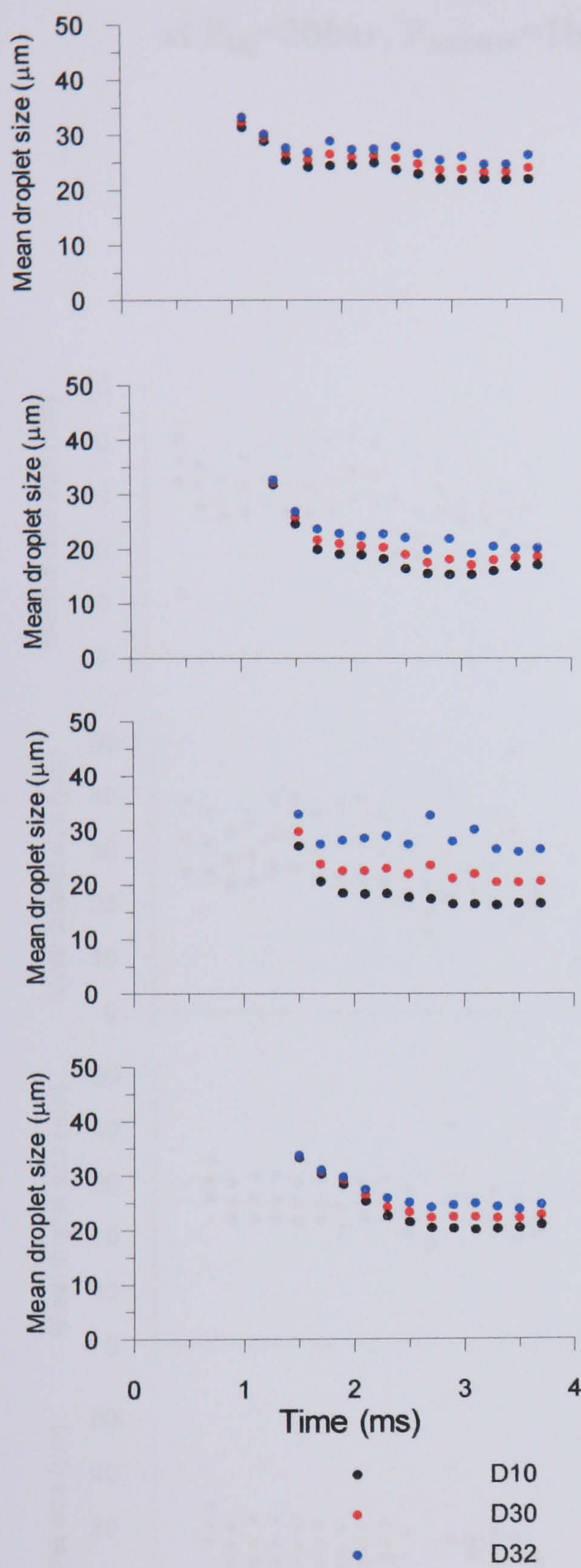


- Axial
- Radial
- Total

Temporal variation of mean droplet diameters and velocities
at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=30\text{mm}$



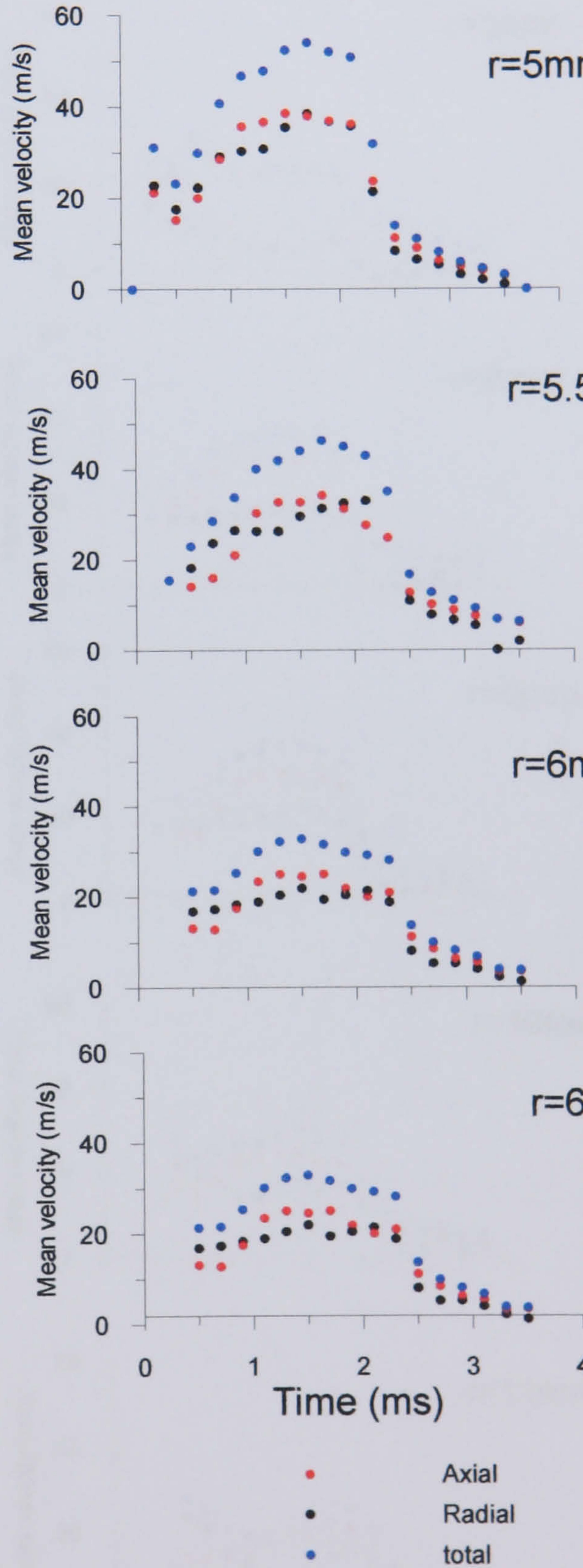
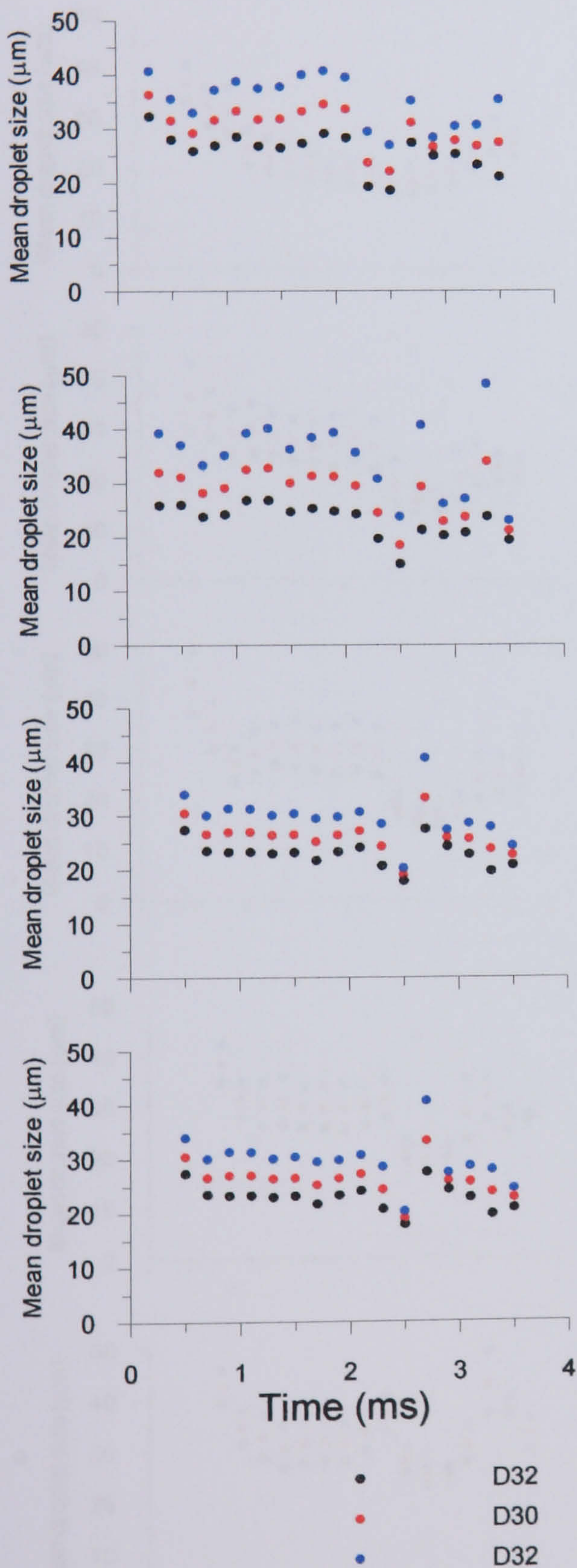
Temporal variation of mean droplet diameters and velocities
 at $P_{inj}=70\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=30\text{mm}$ (continue)



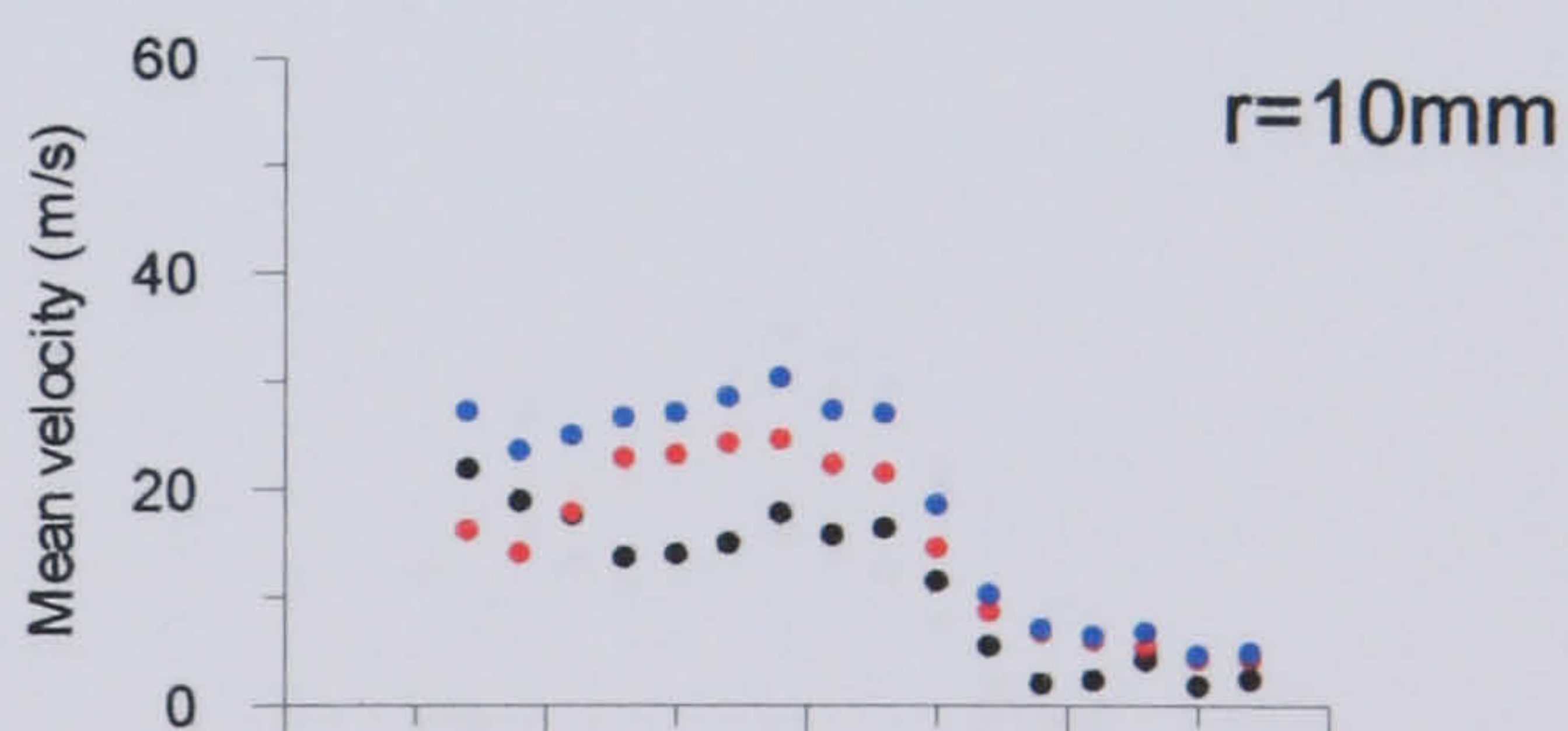
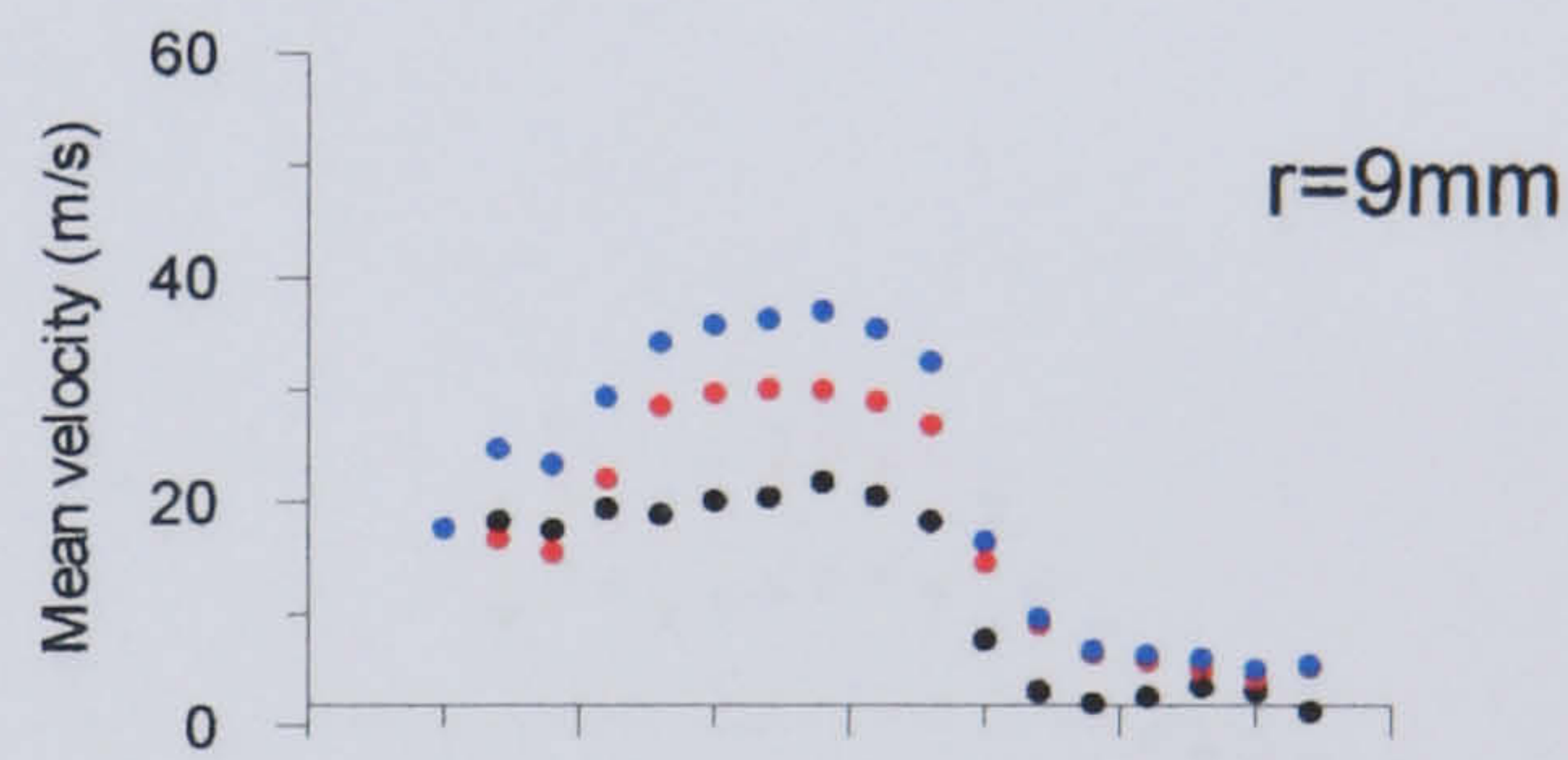
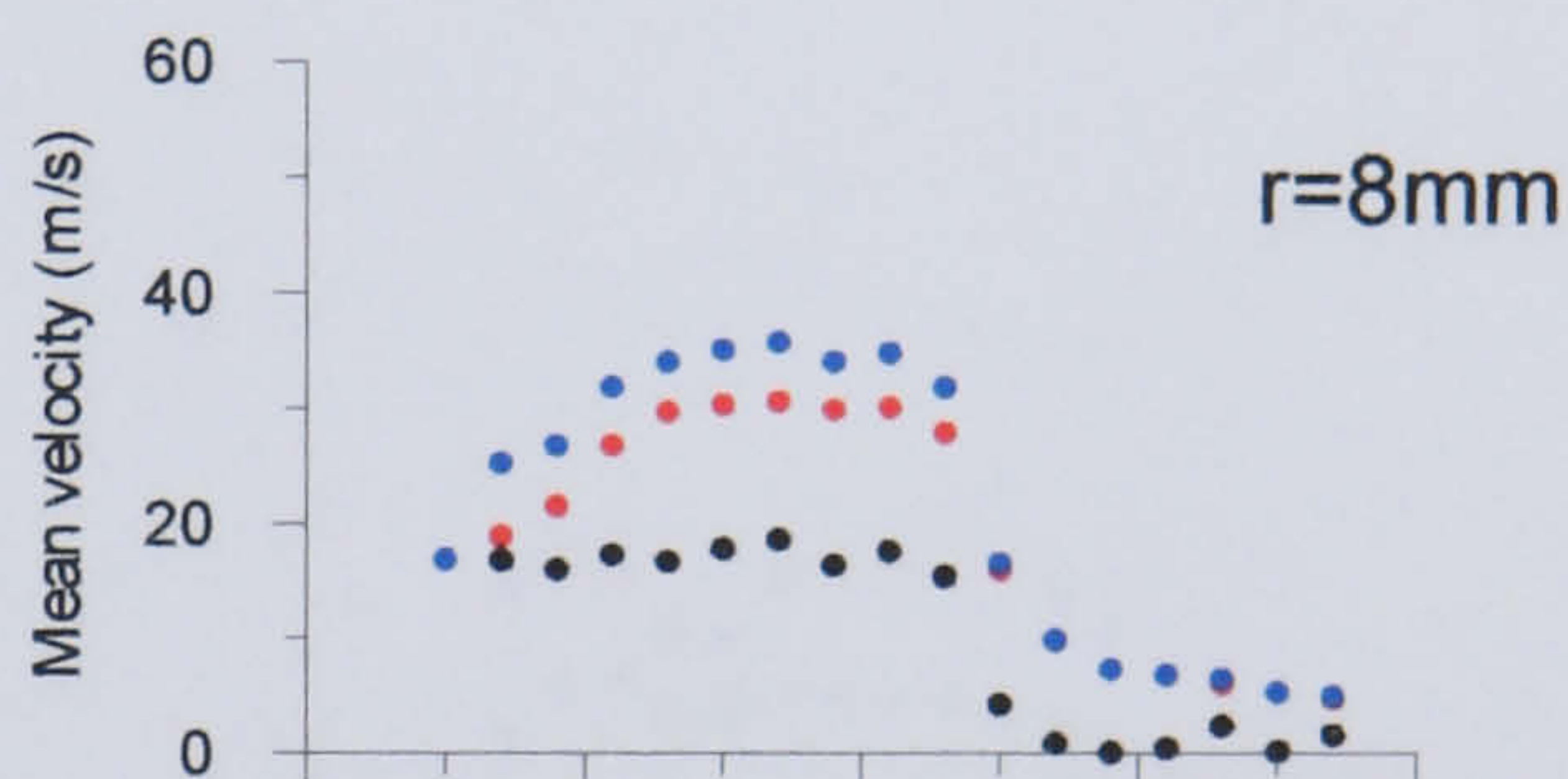
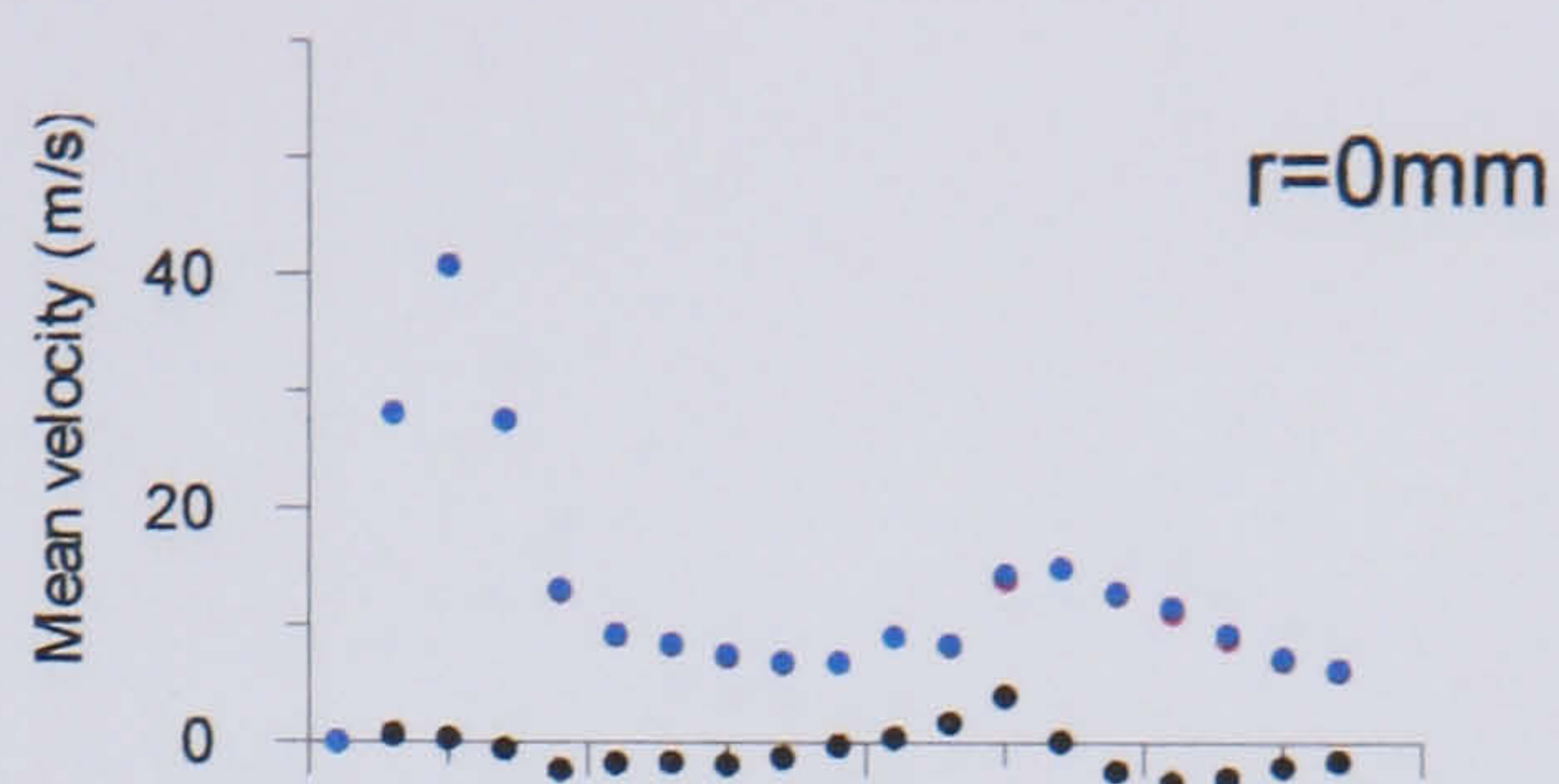
Appendix 4-B

Temporal variation of mean droplet diameters and velocities
at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and different locations

X=5mm



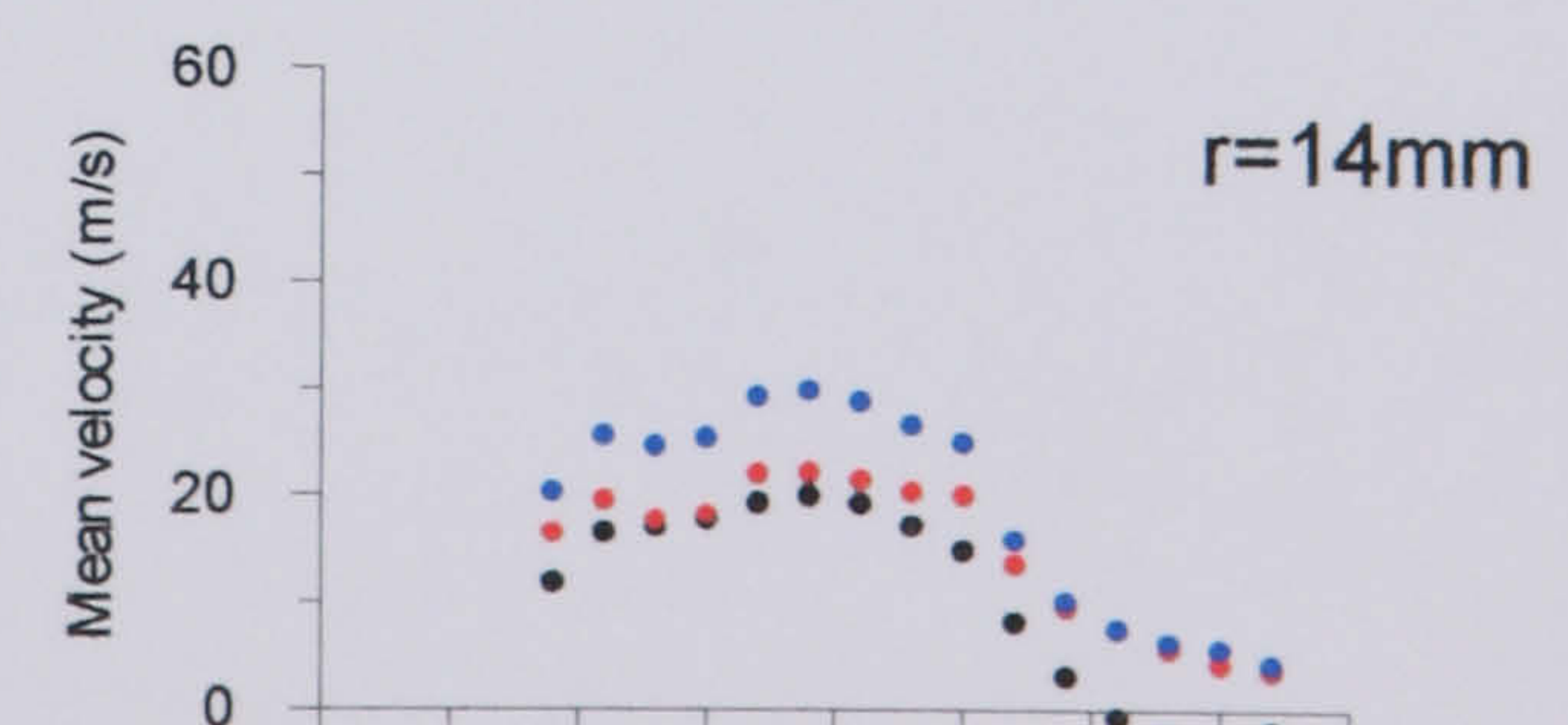
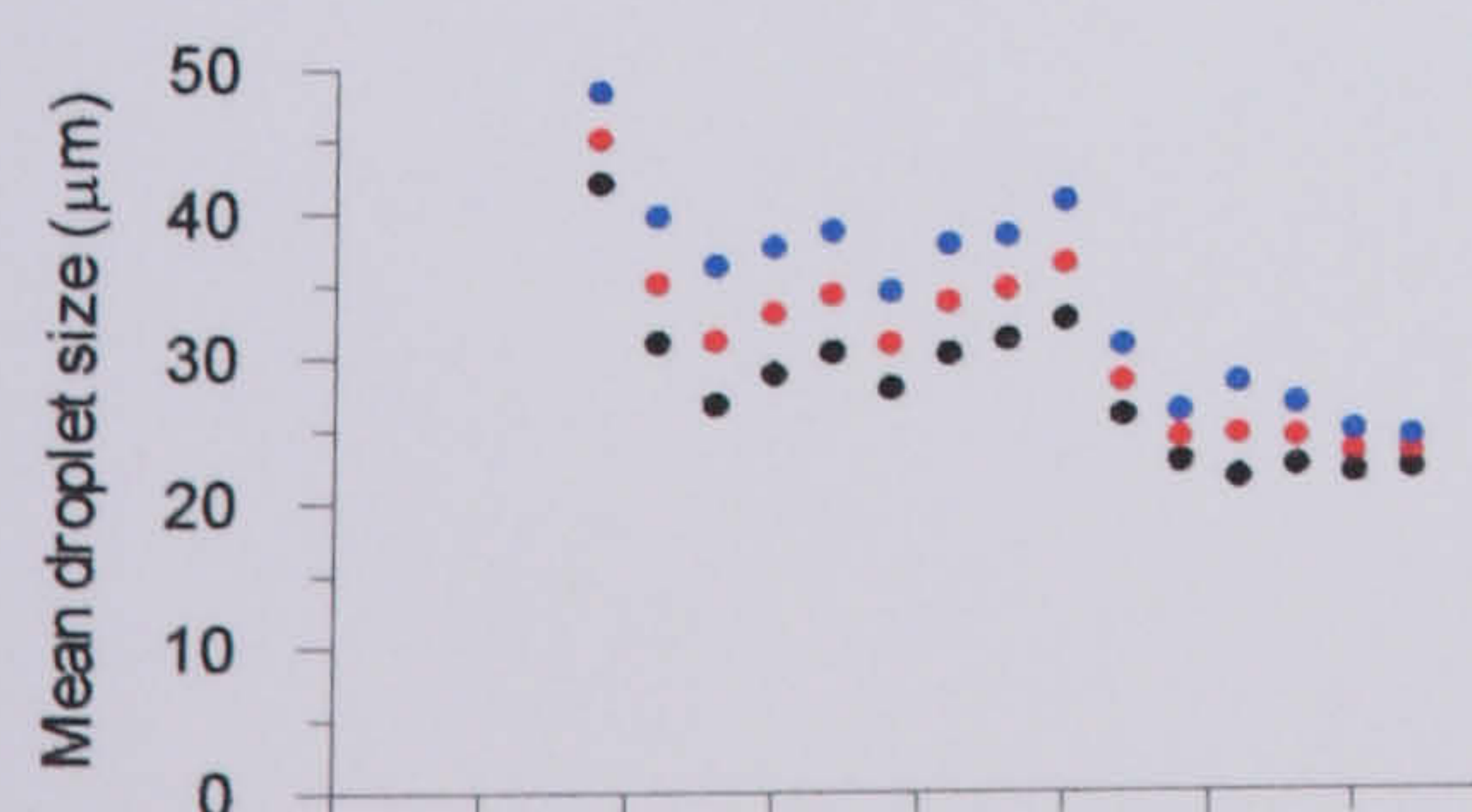
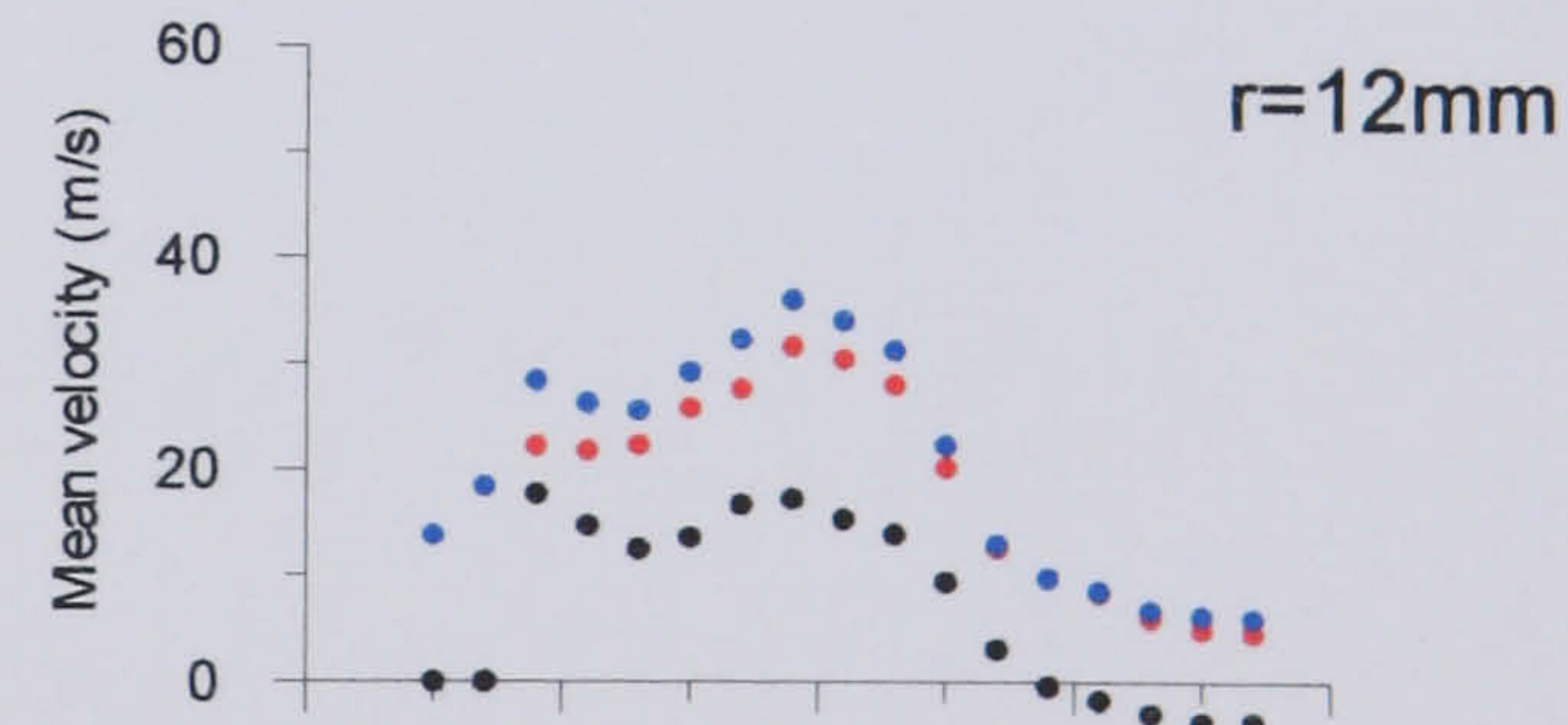
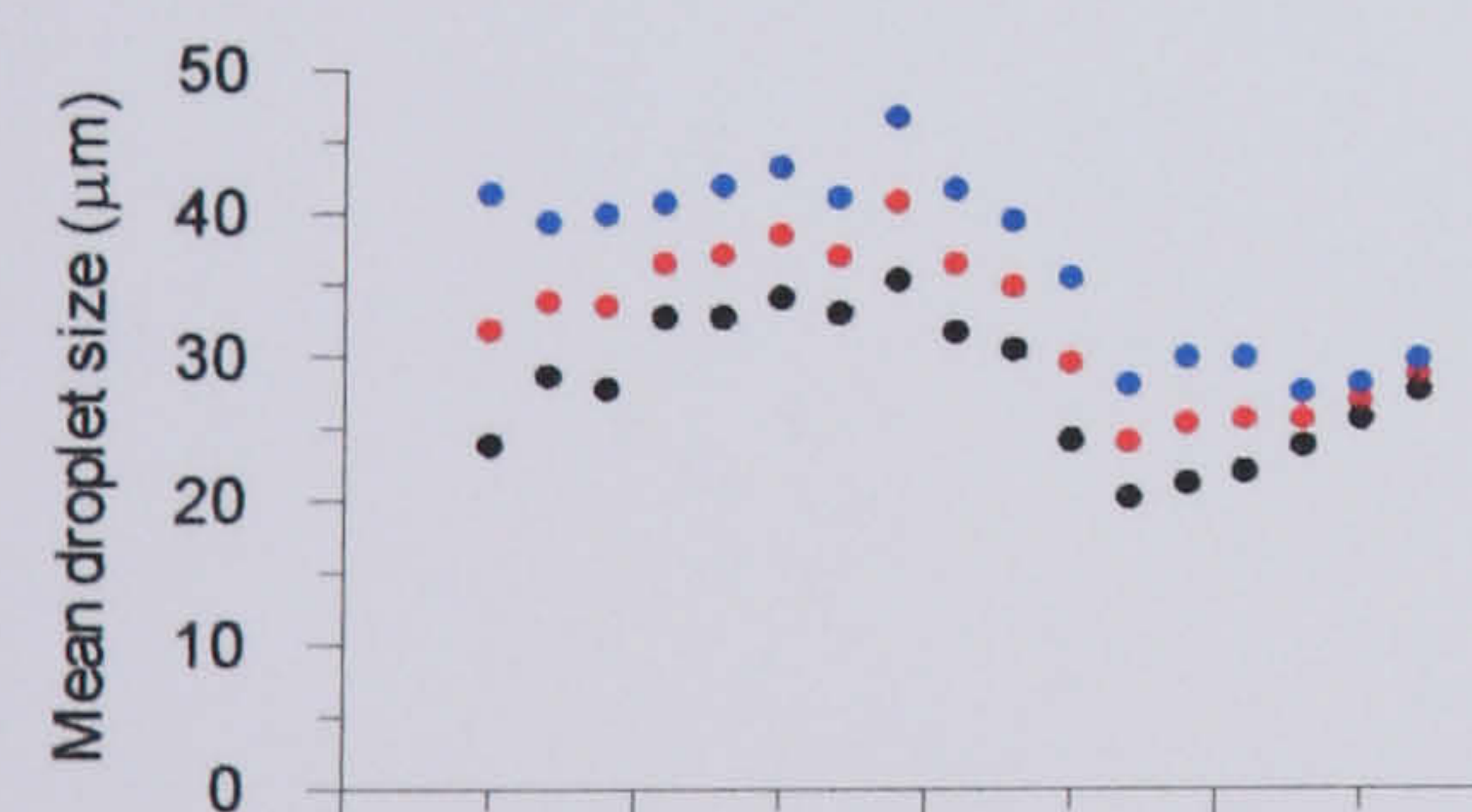
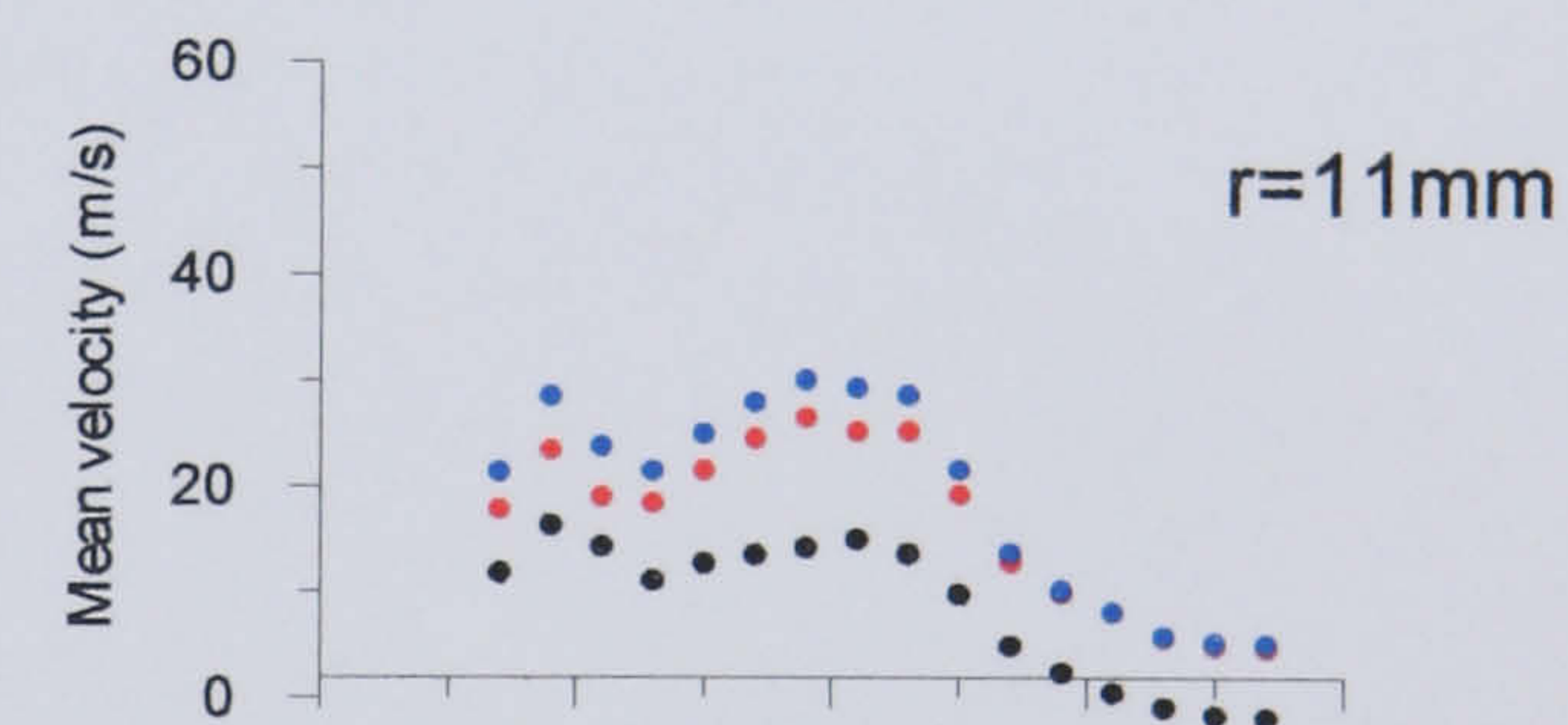
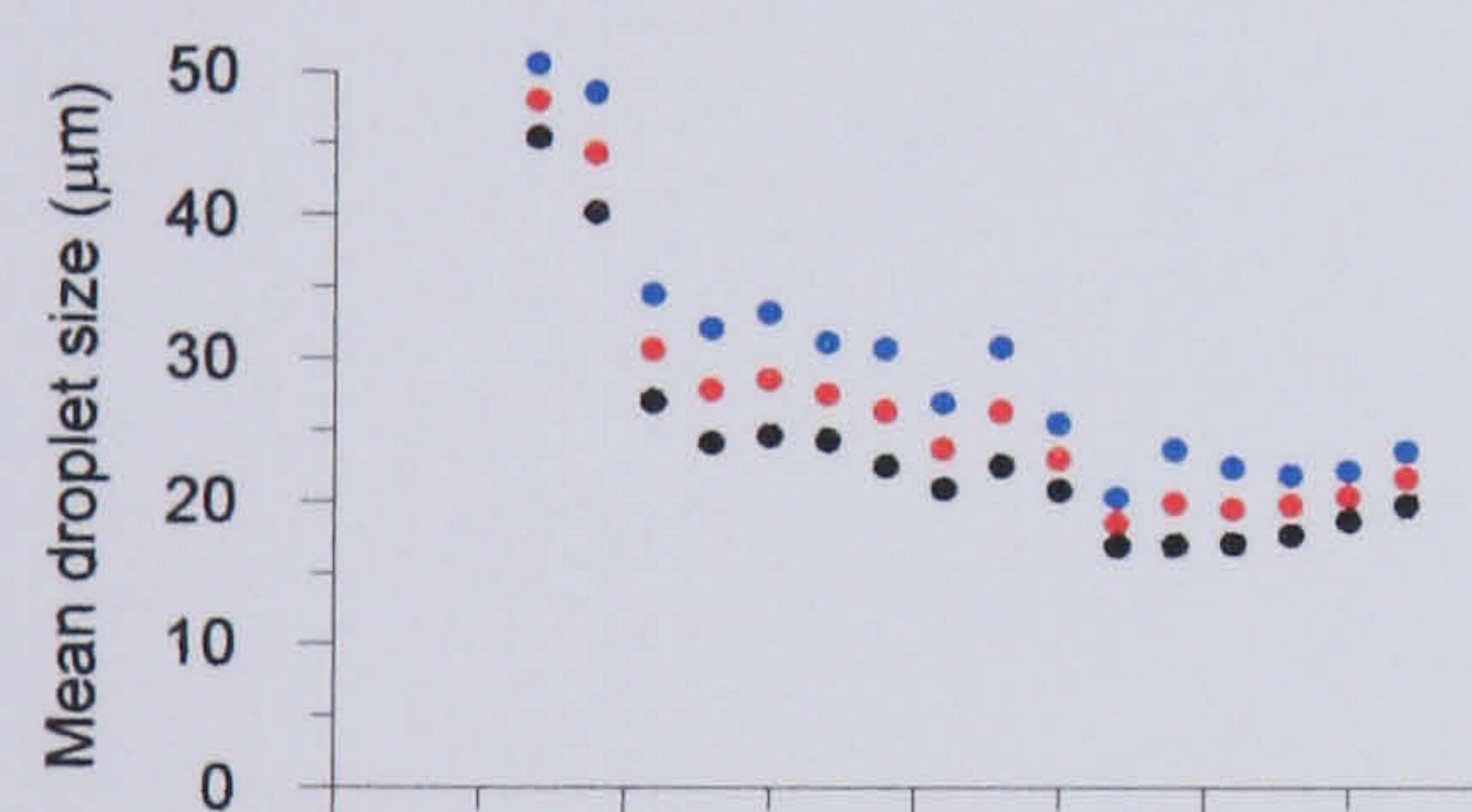
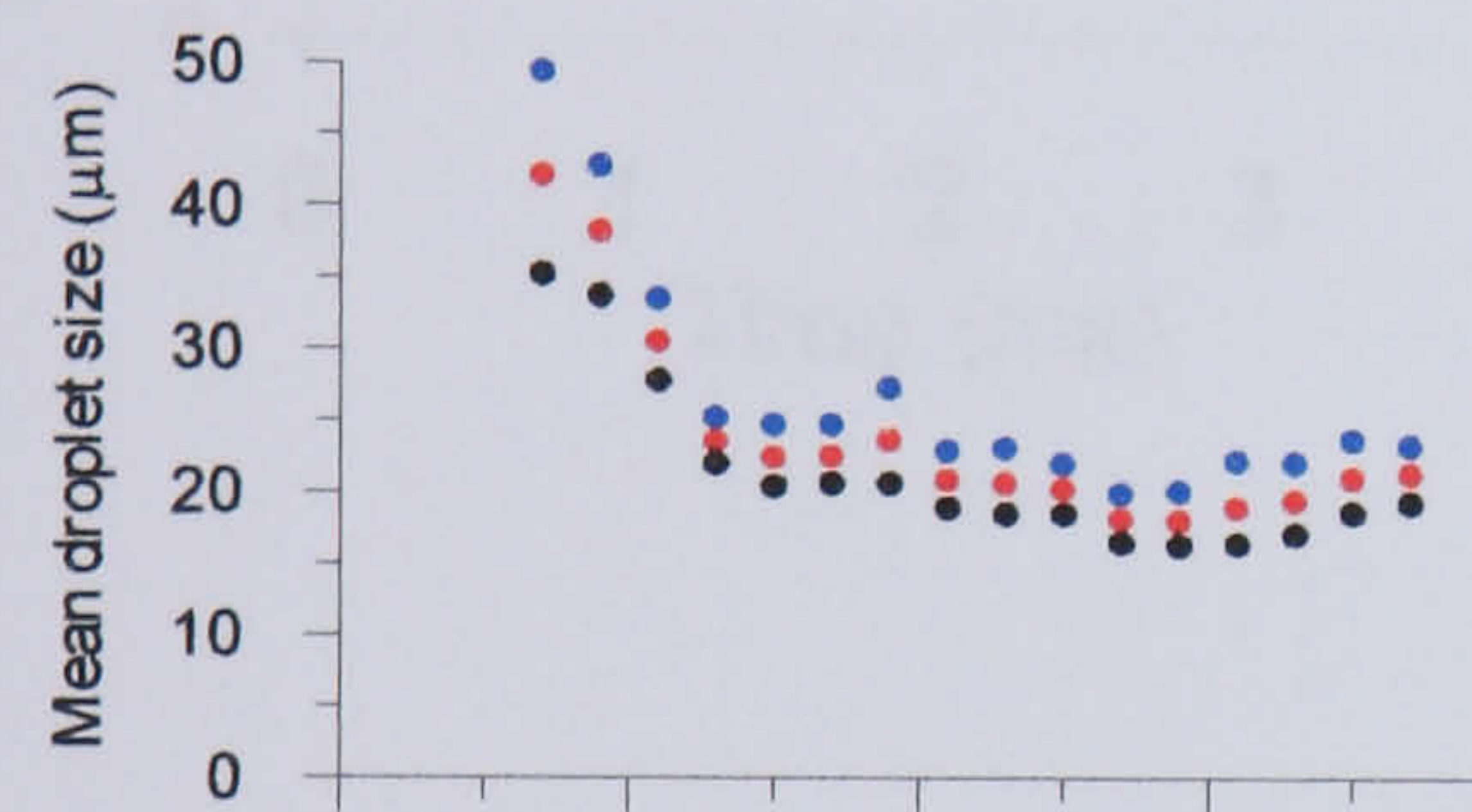
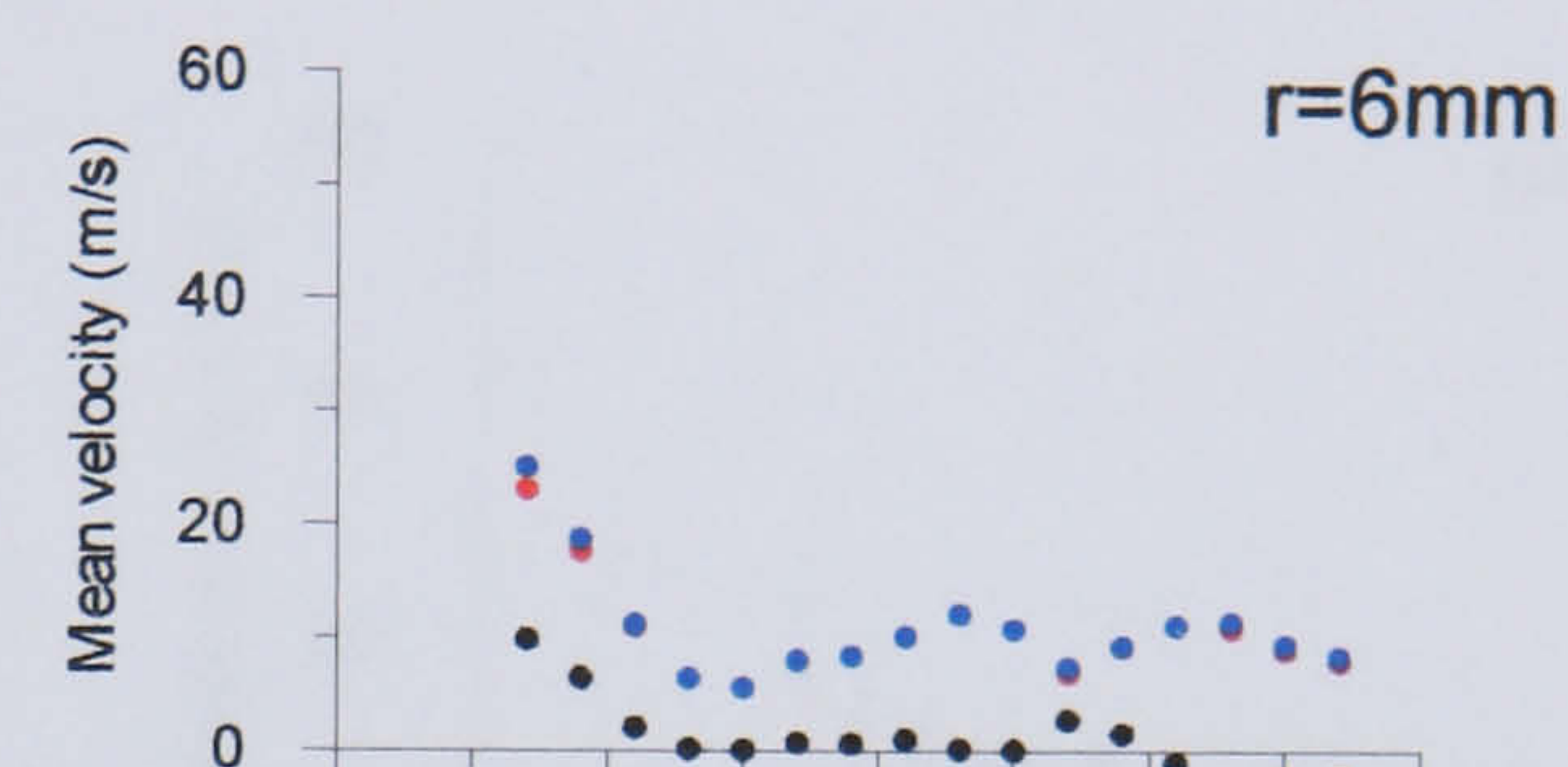
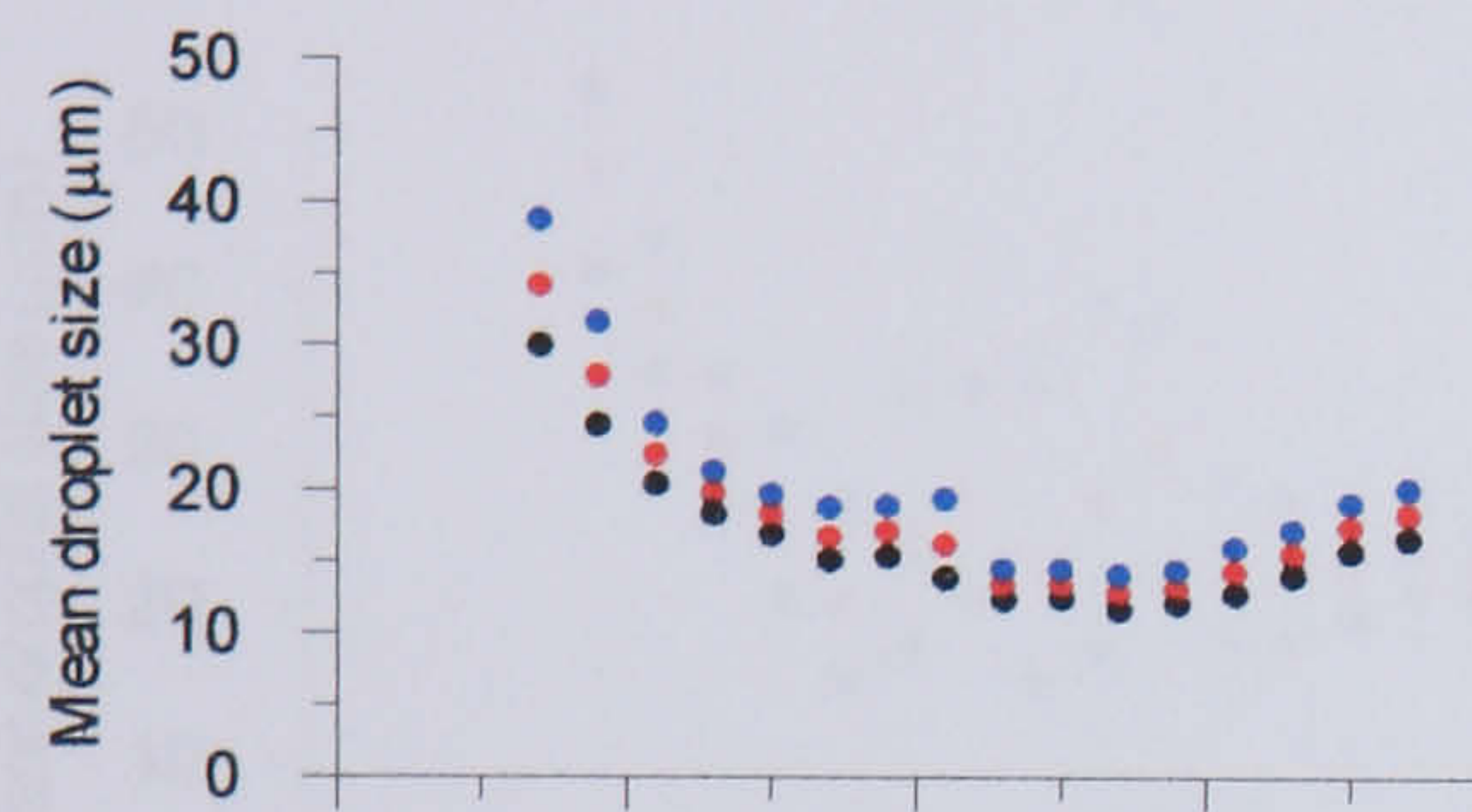
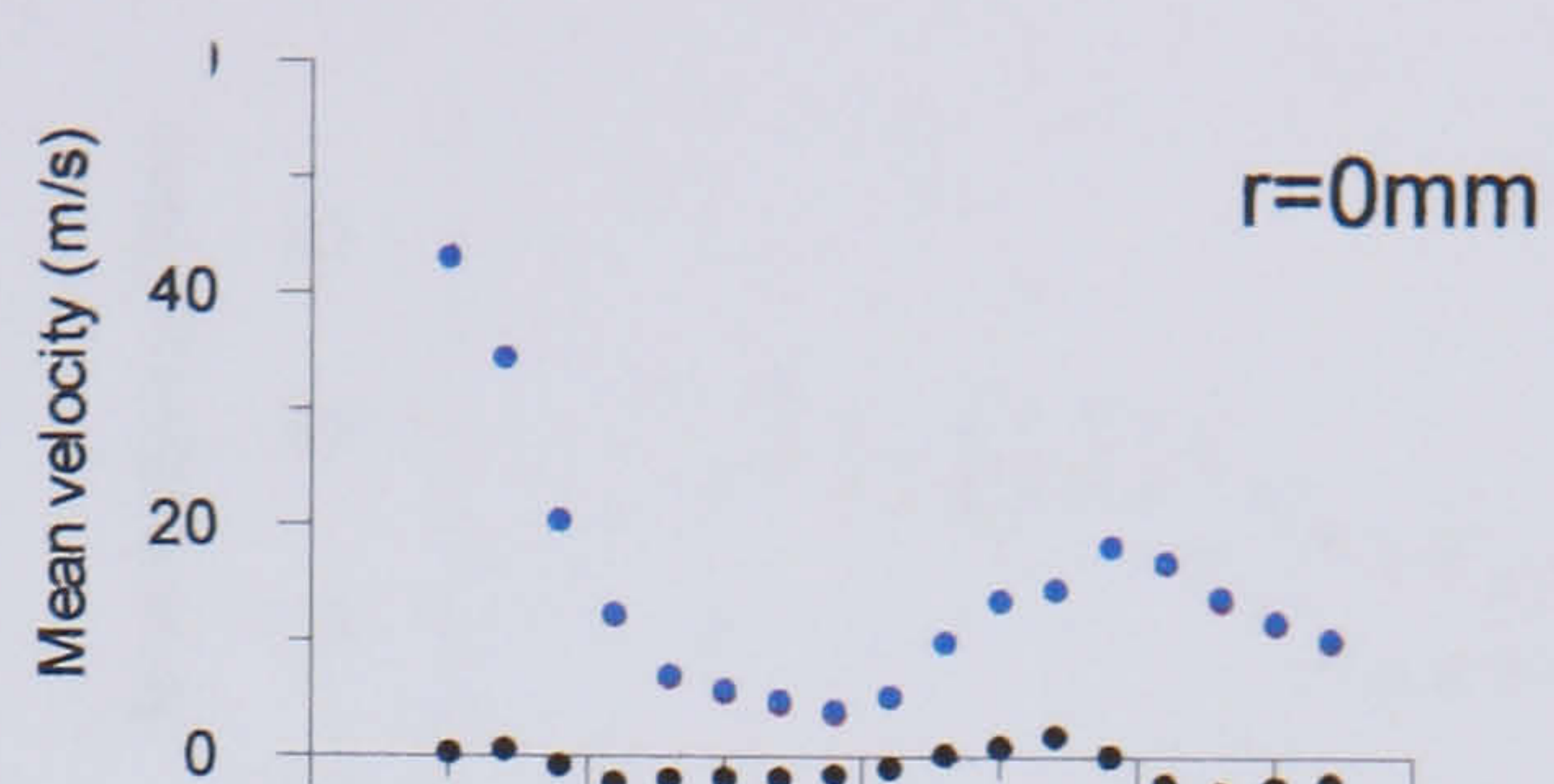
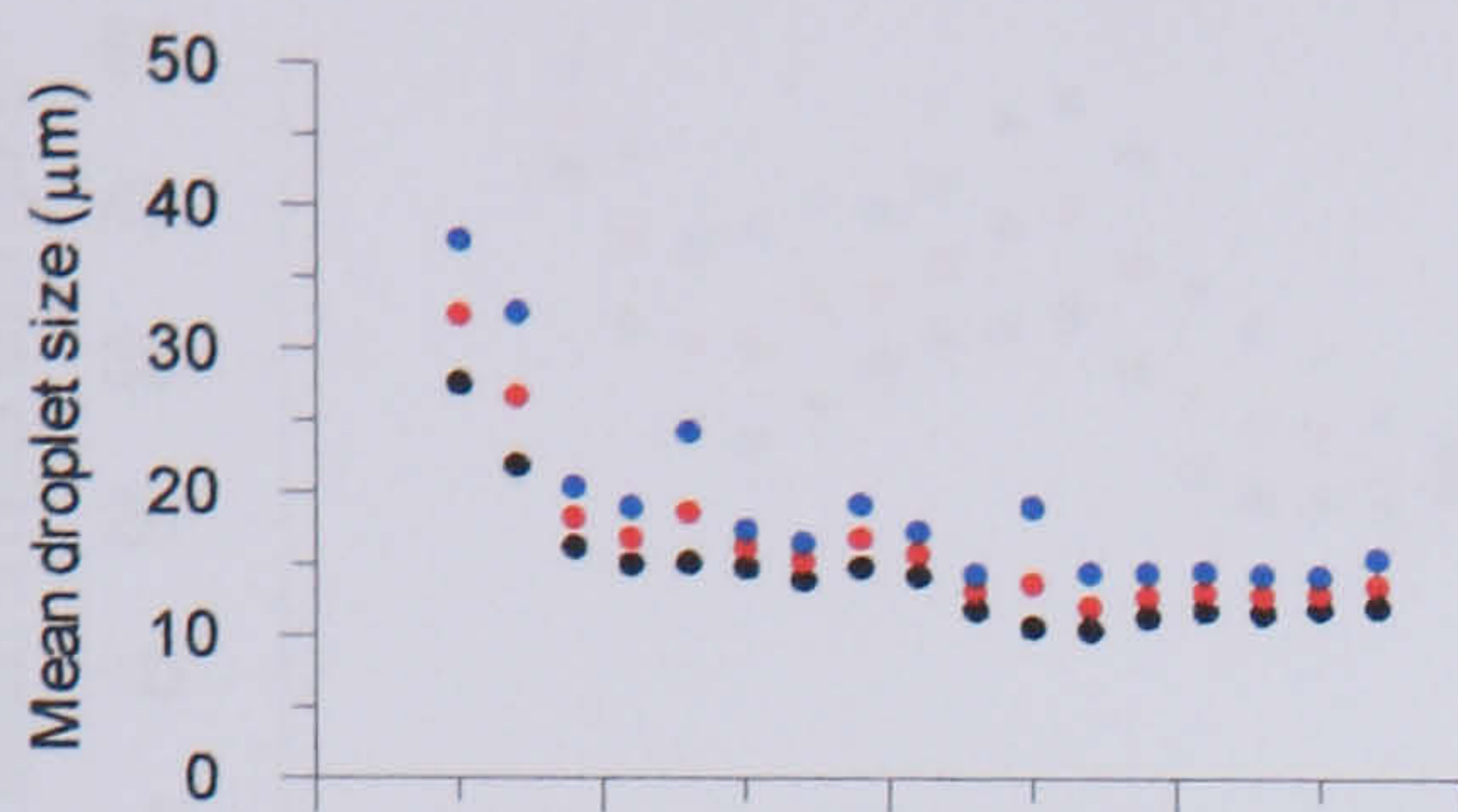
at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=10\text{mm}$



Time (ms) • Axial
 • Radial
 • total

Temporal variation of mean droplet diameters and velocities

at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=15\text{ mm}$



Time (ms)

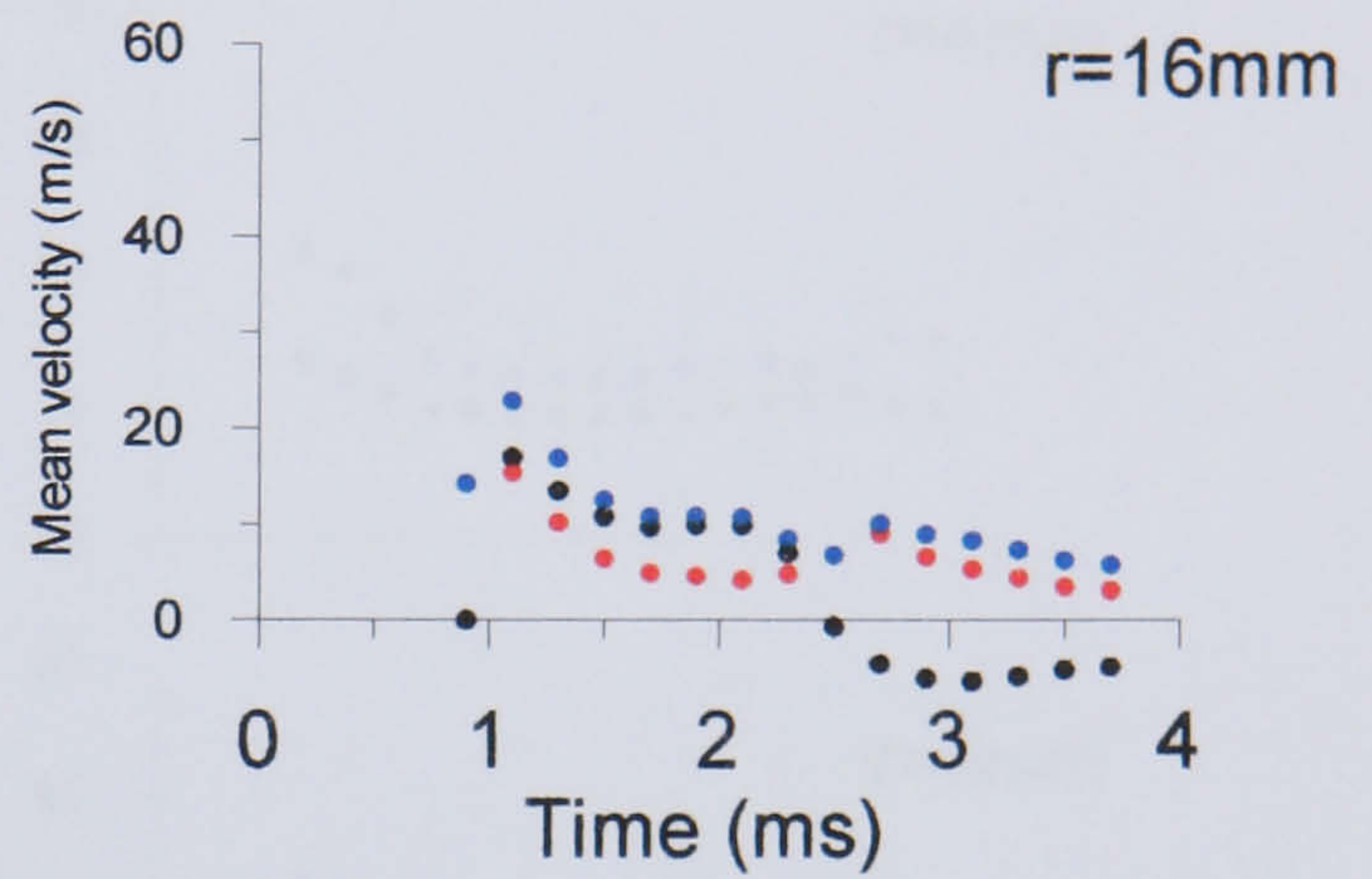
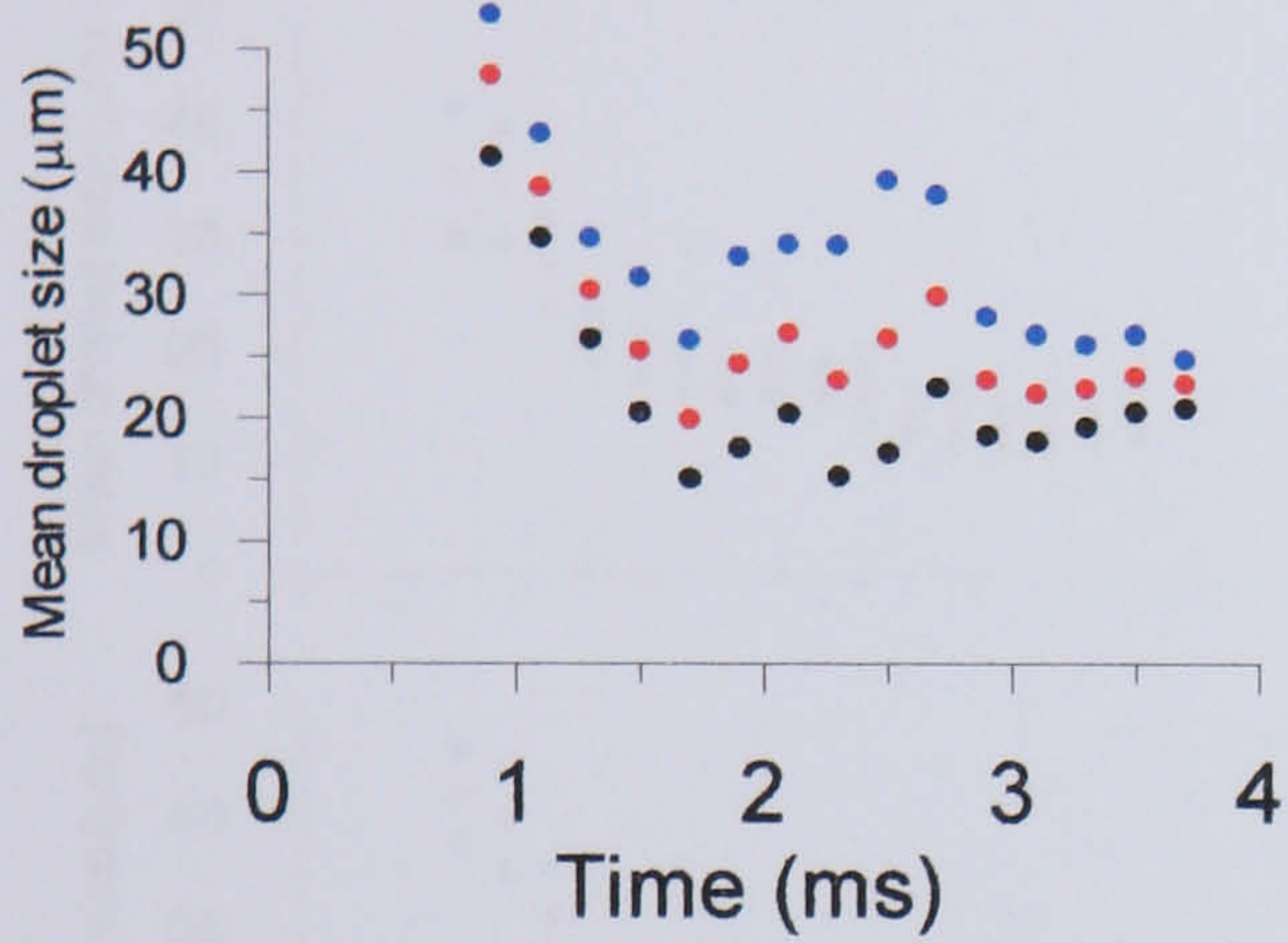
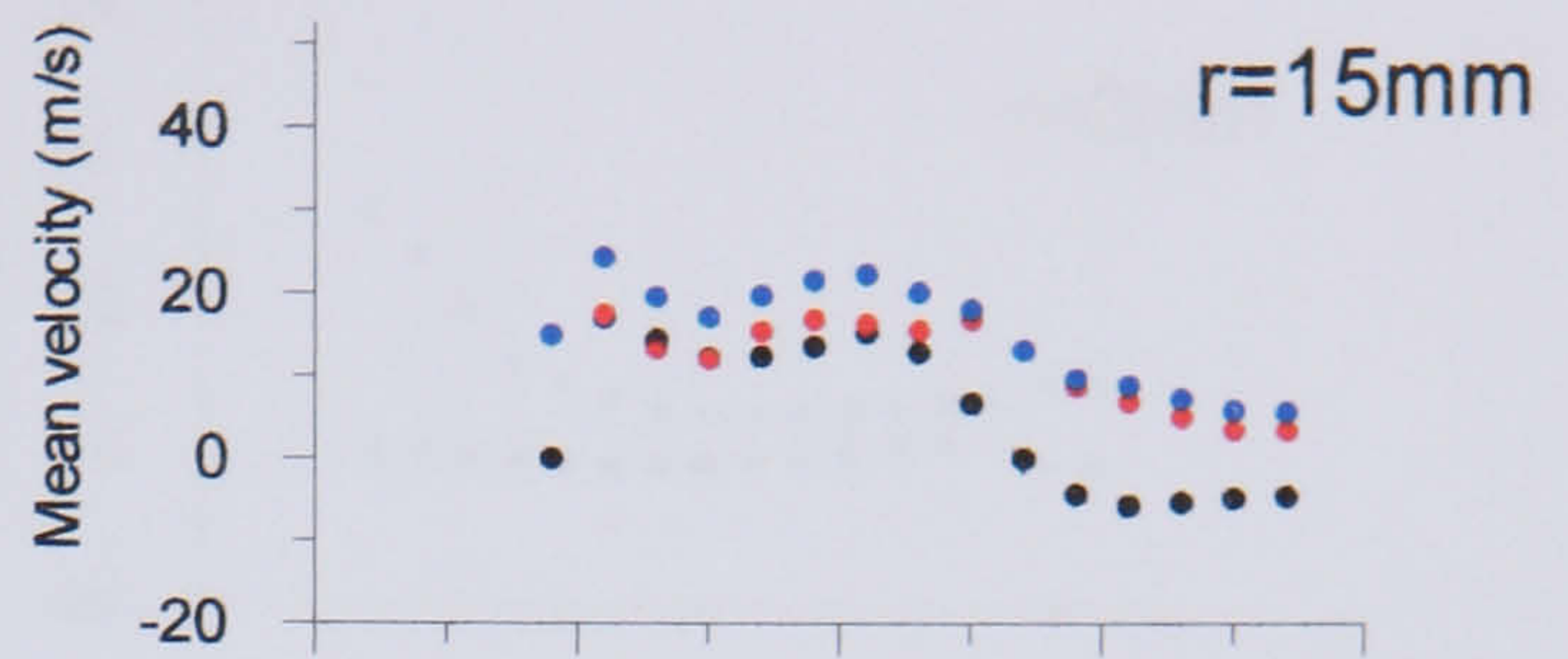
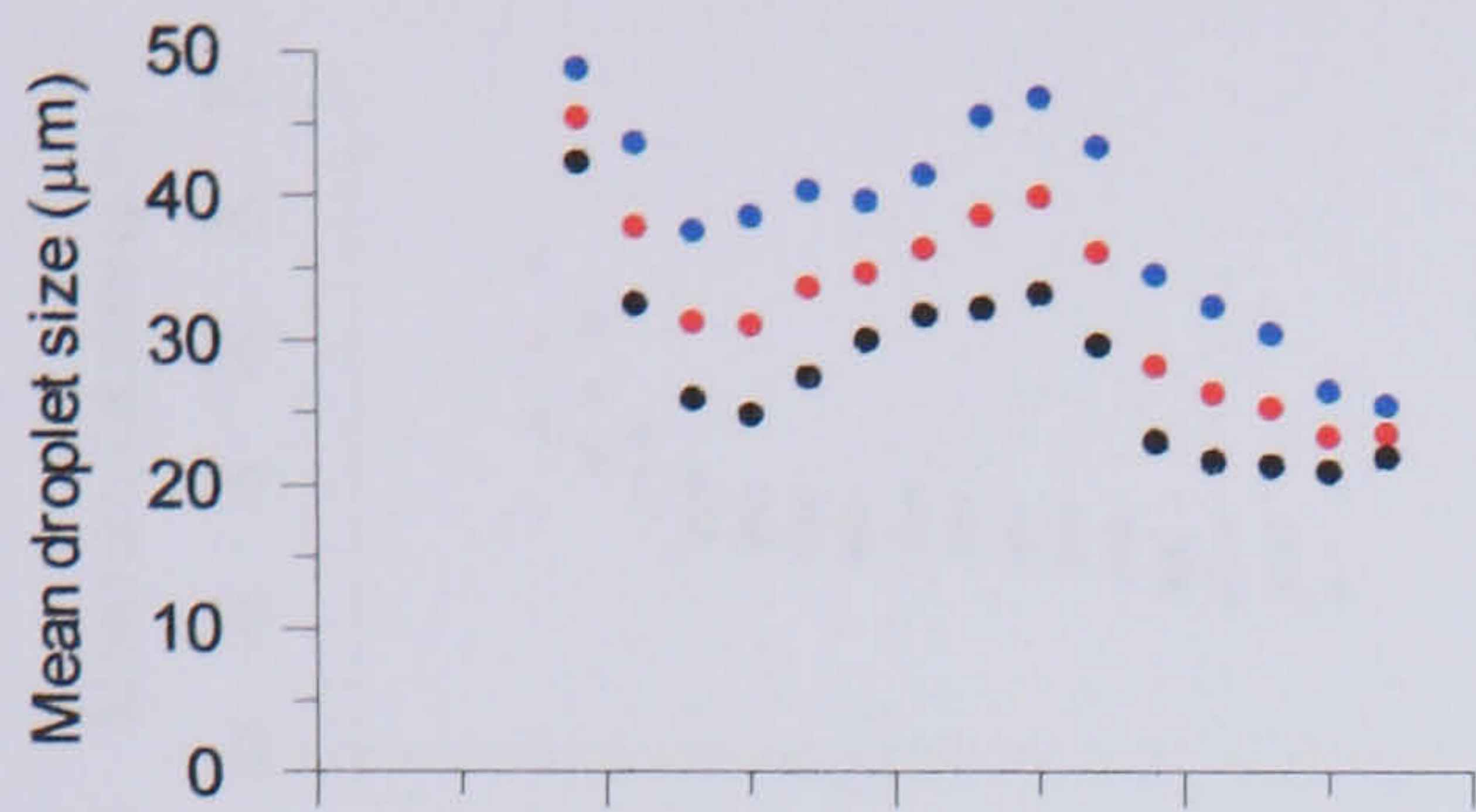
• D32
• D30
• D32

Time (ms)

• Axial
• Radial
• total

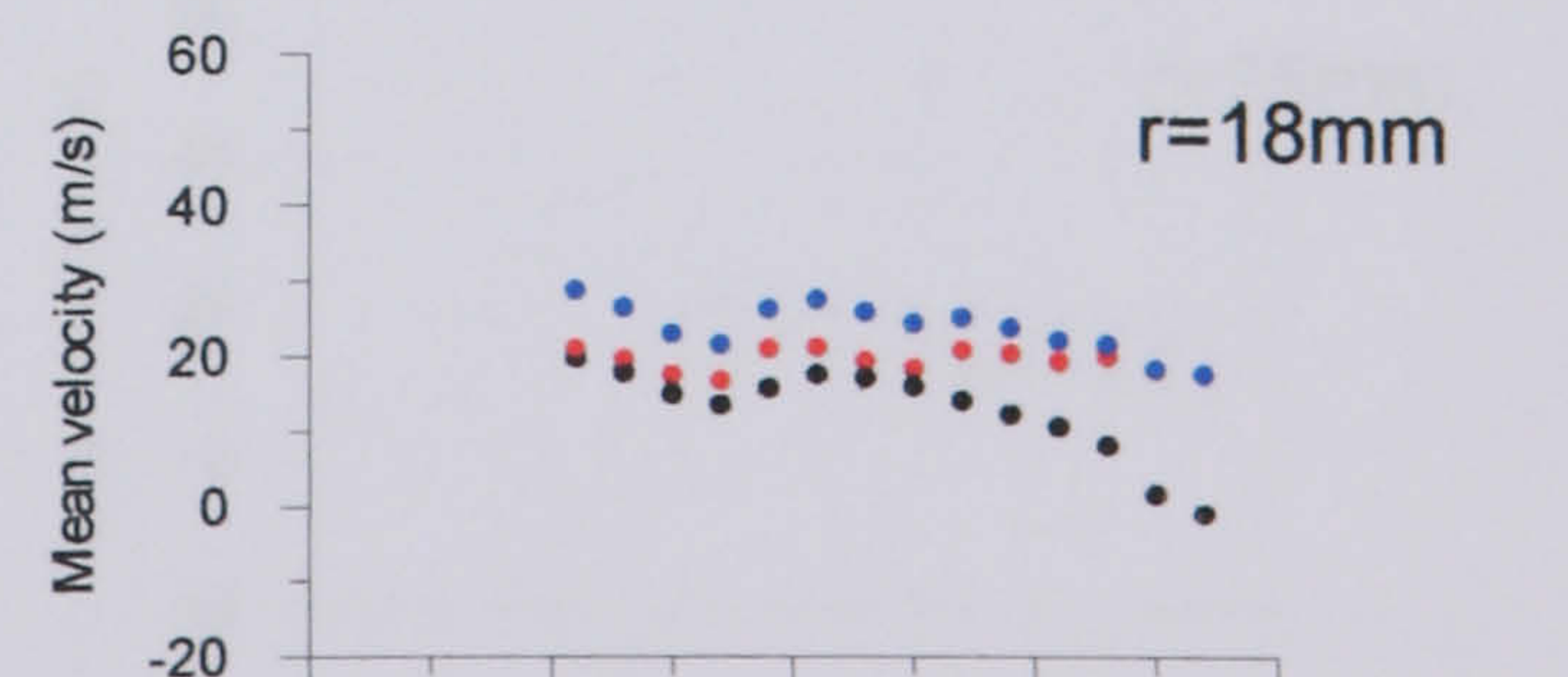
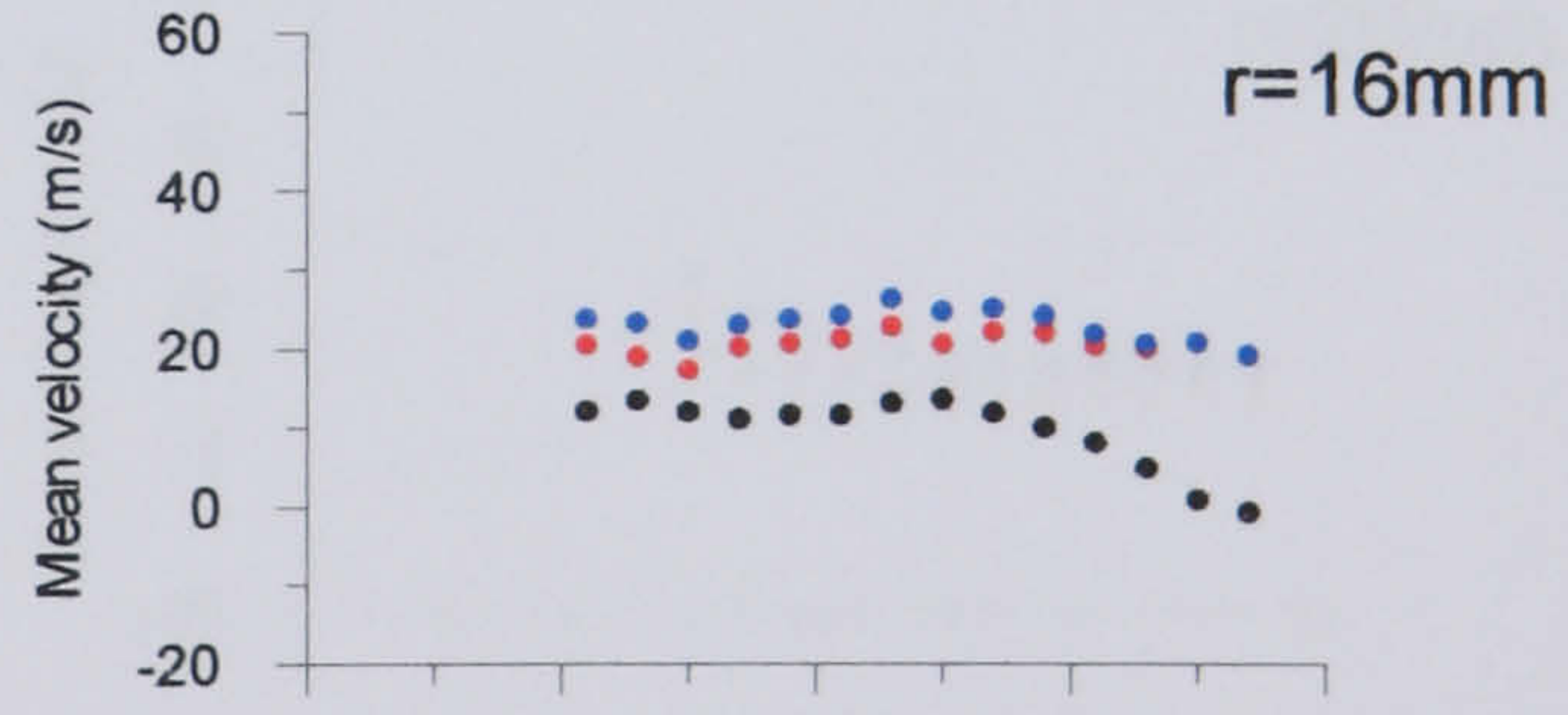
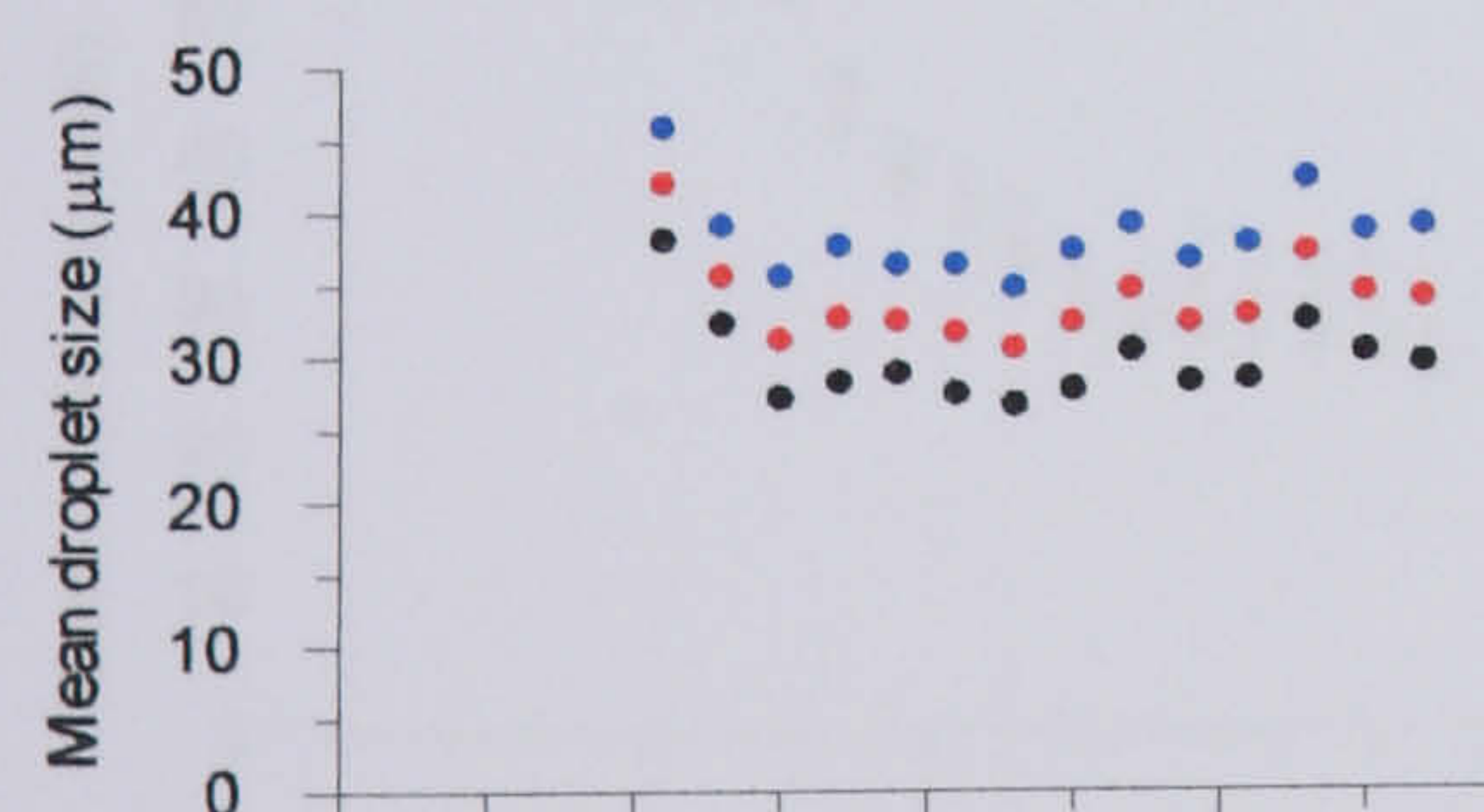
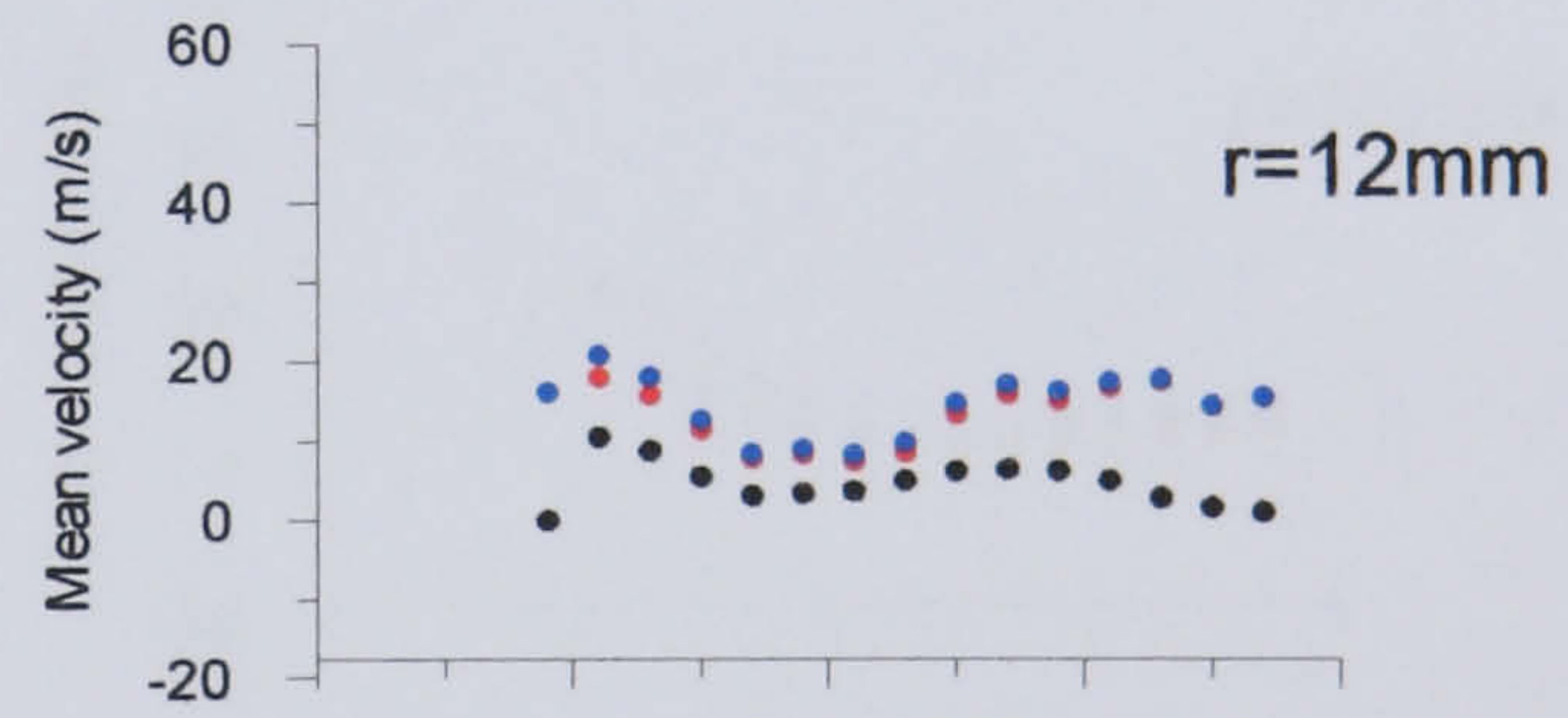
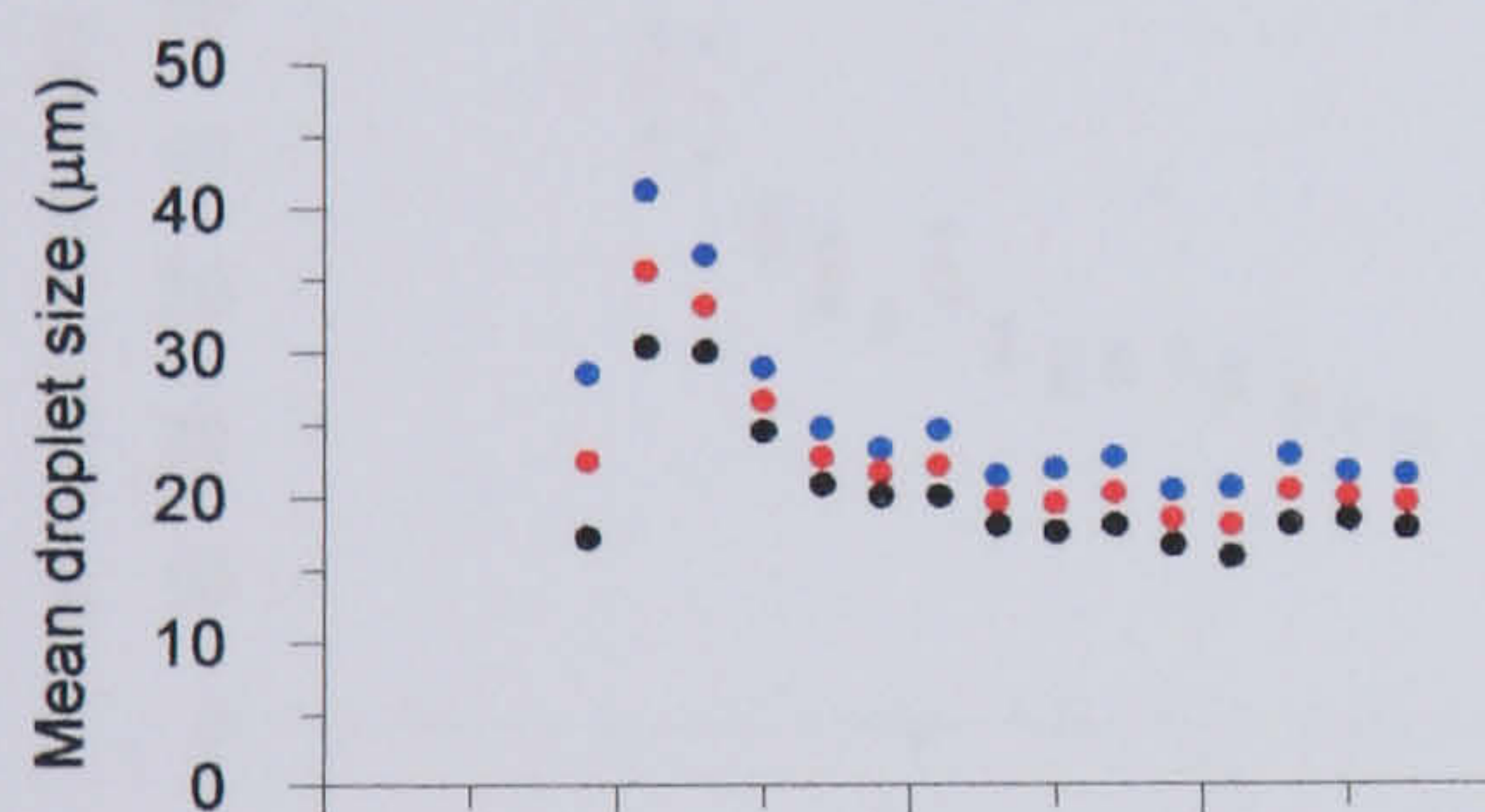
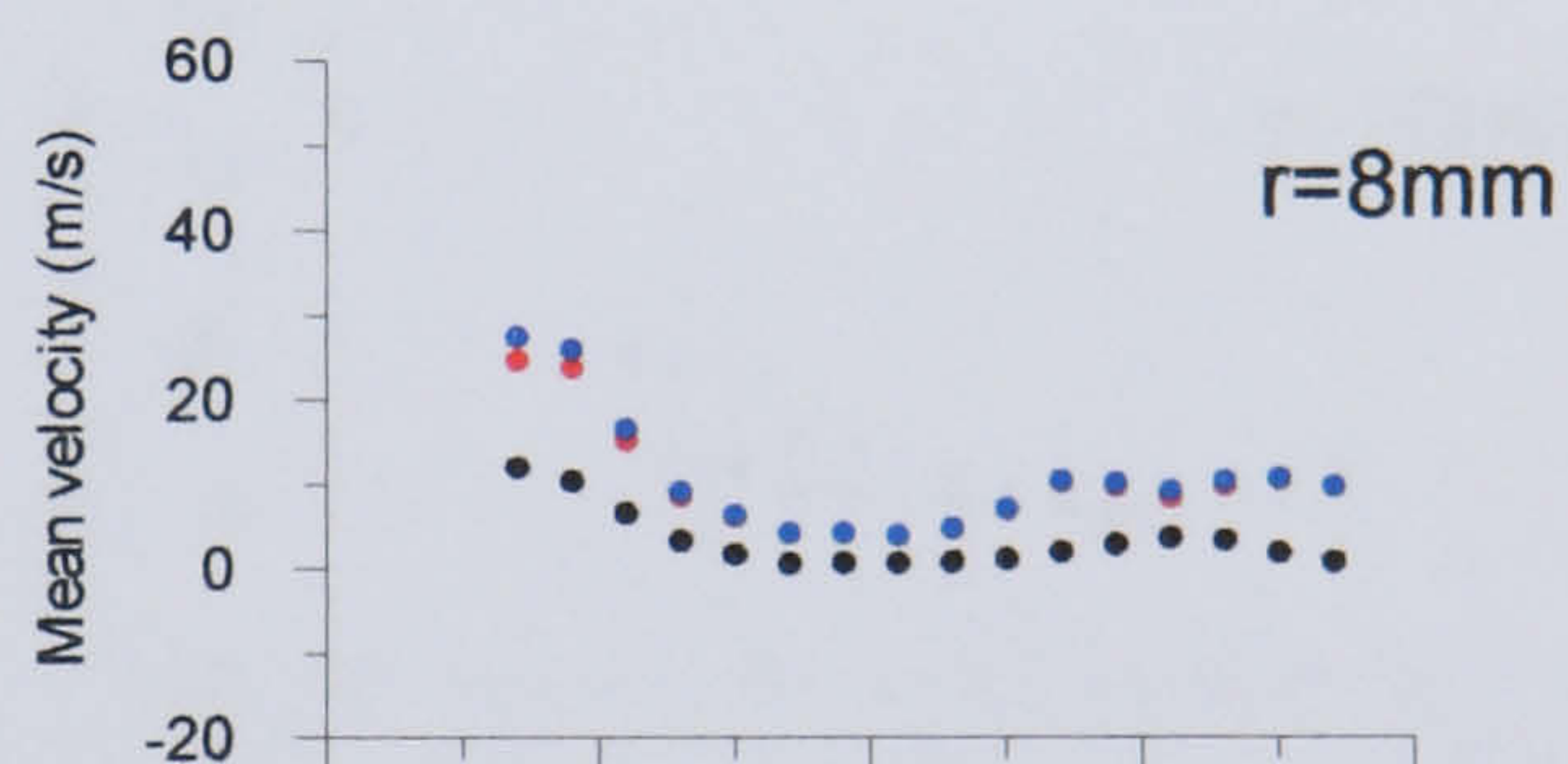
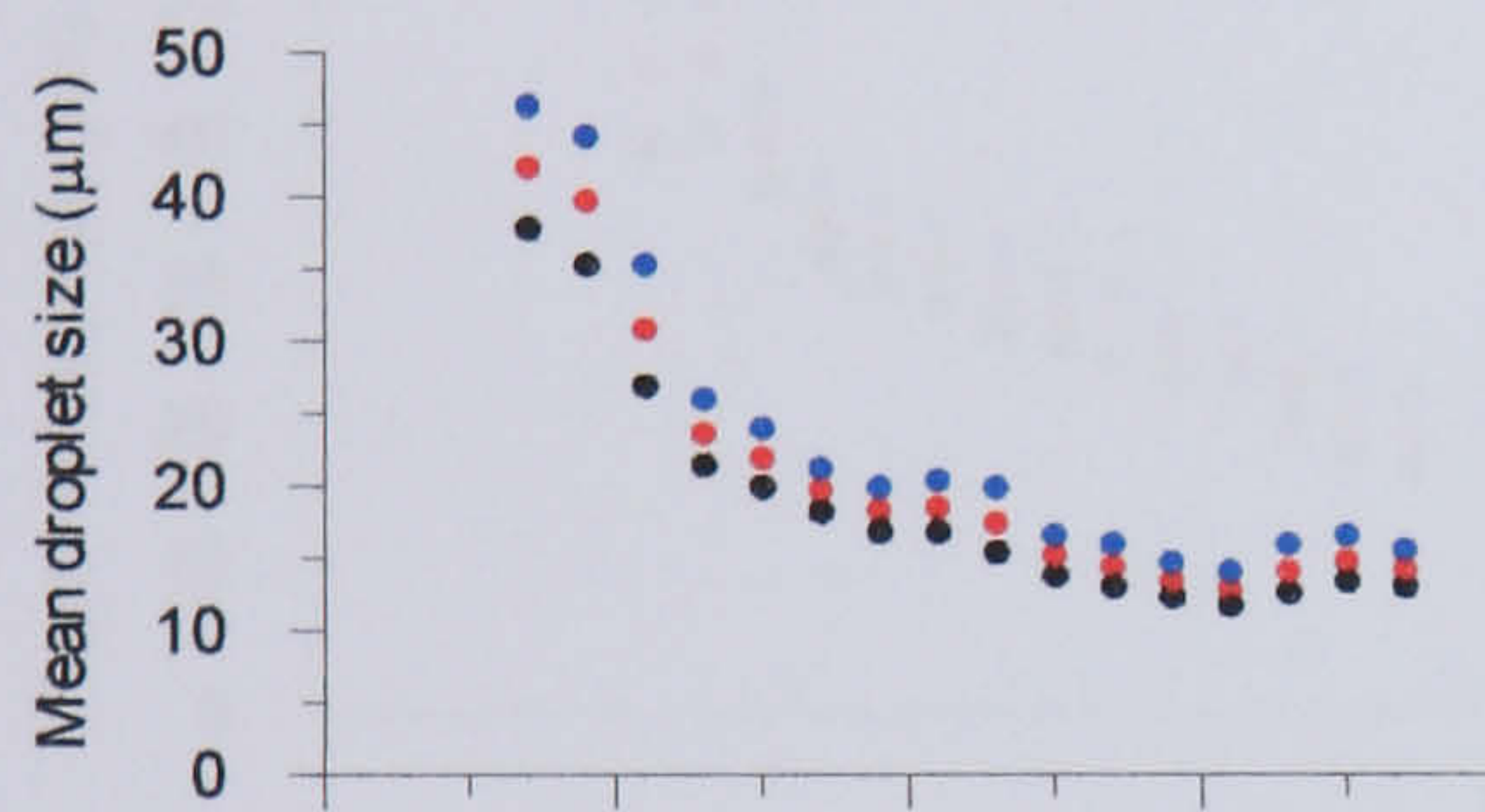
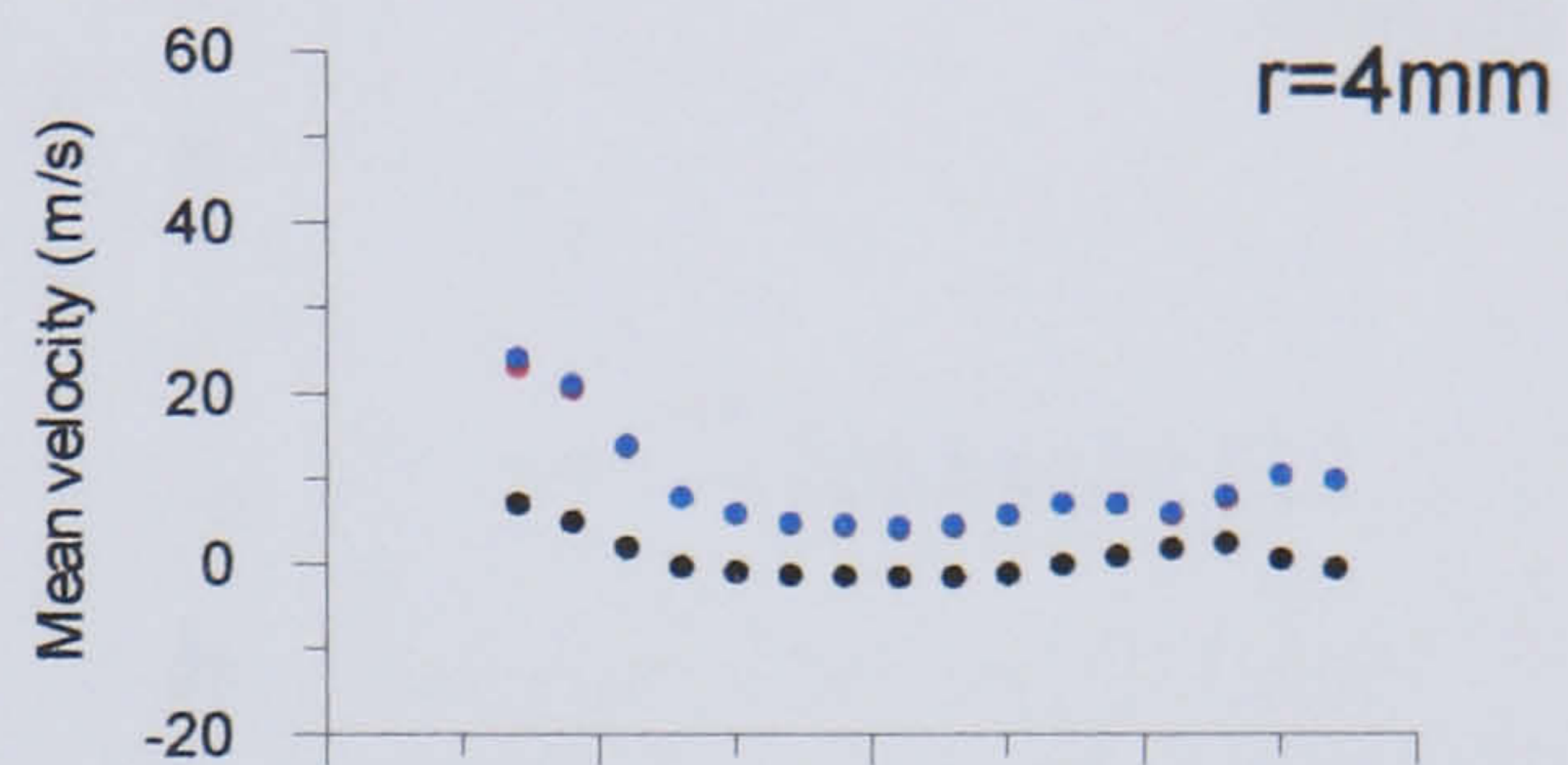
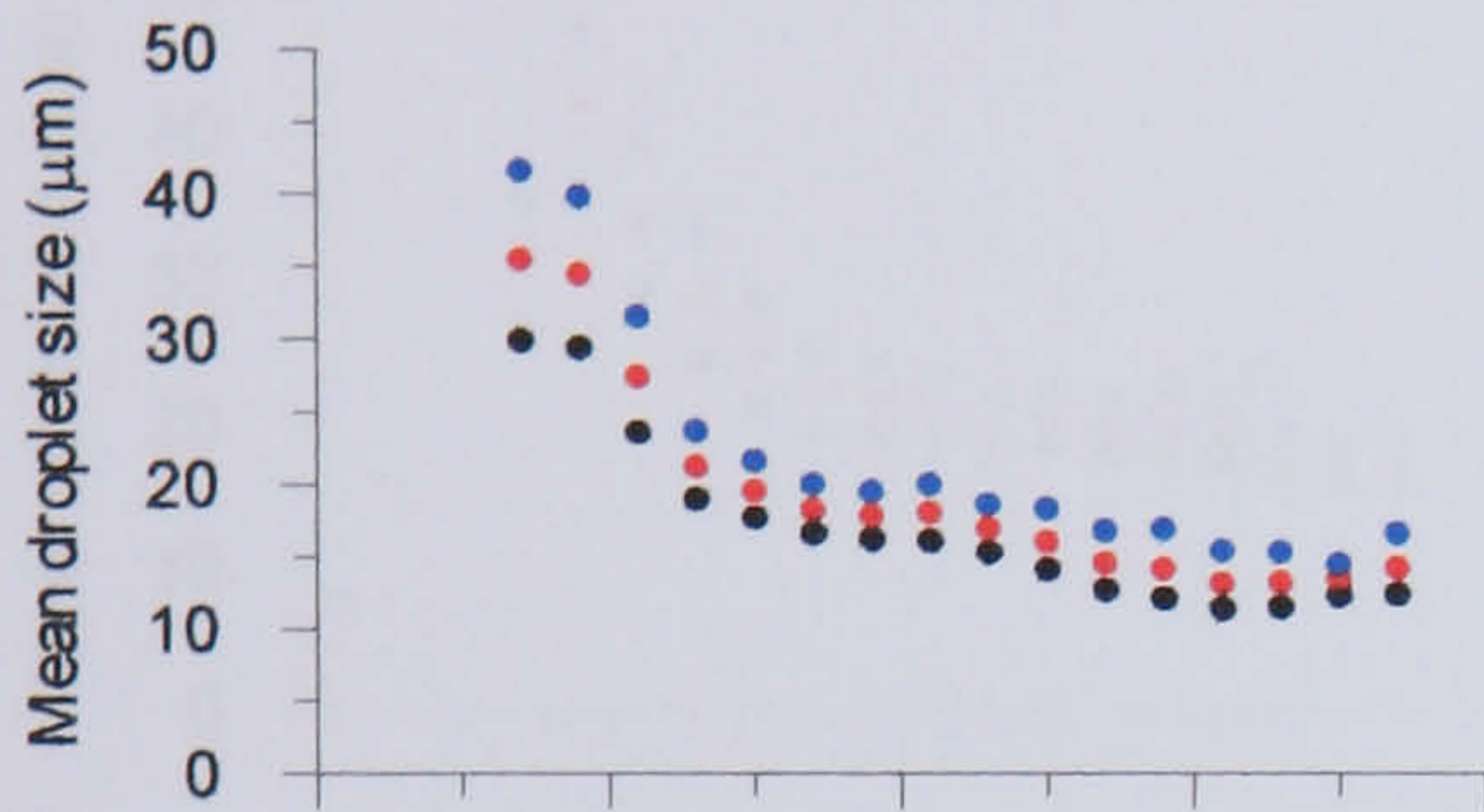
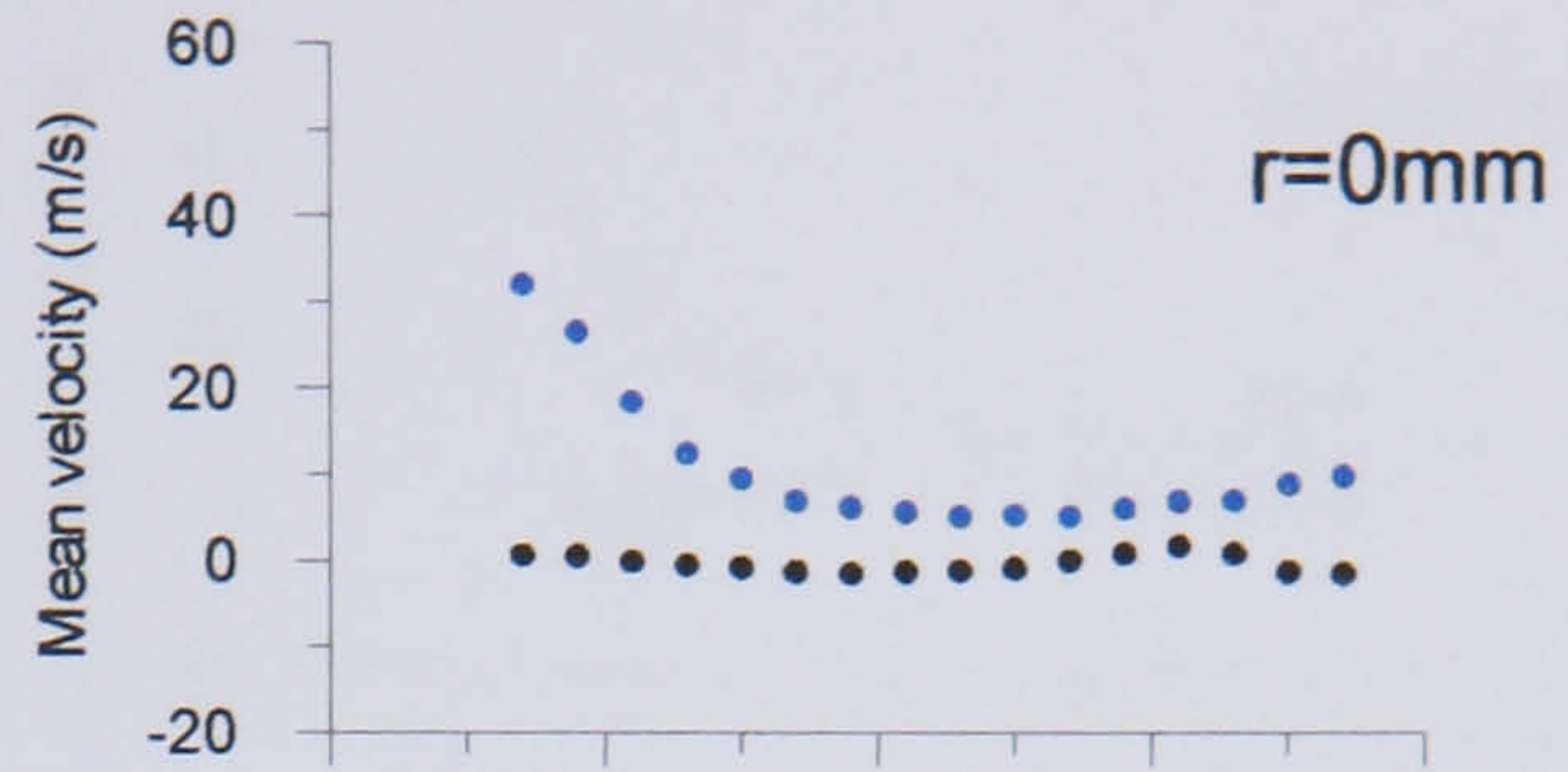
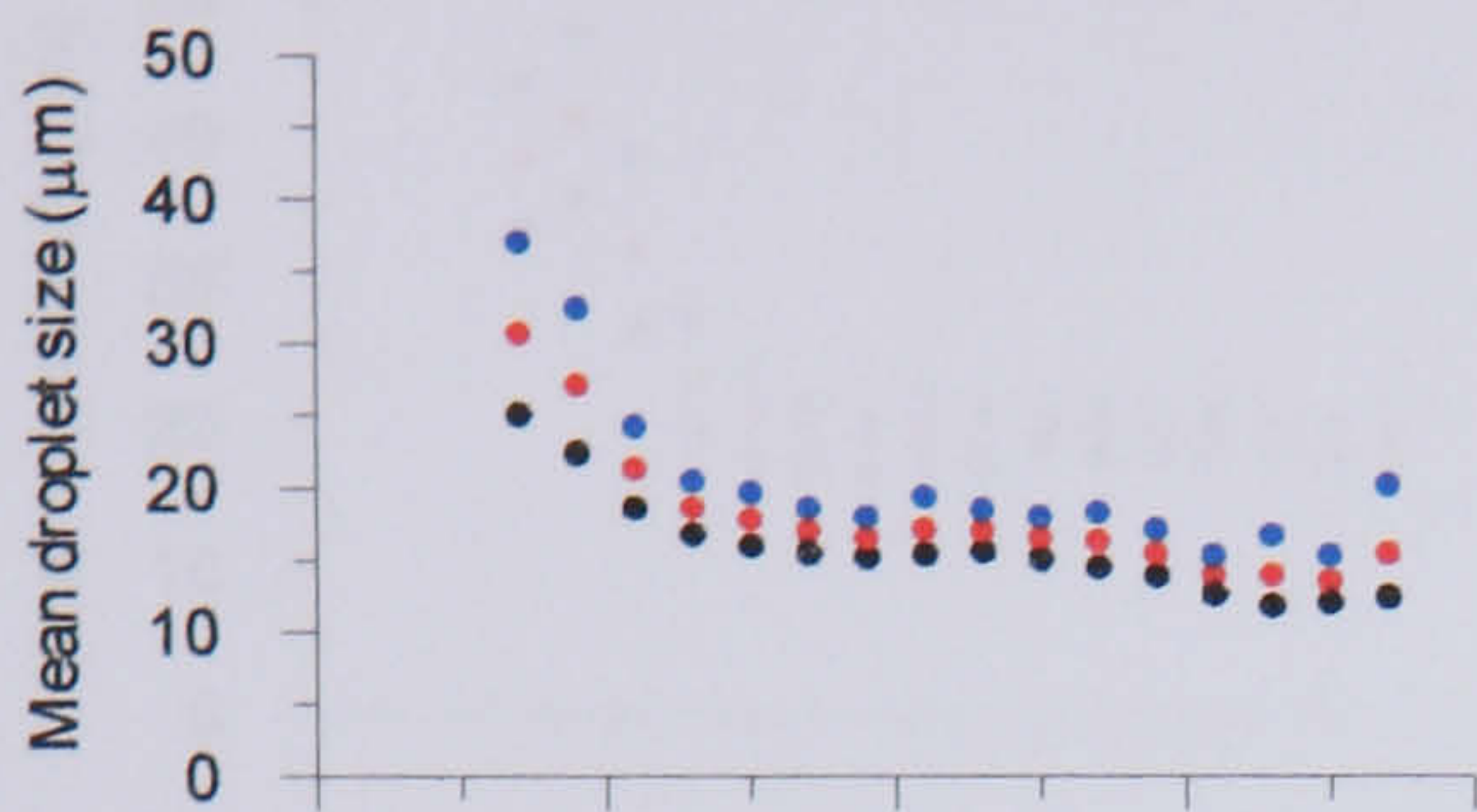
Temporal variation of mean droplet diameters and velocities

at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=15\text{mm}$ (continue)



Temporal variation of mean droplet diameters and velocities

at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=30\text{mm}$

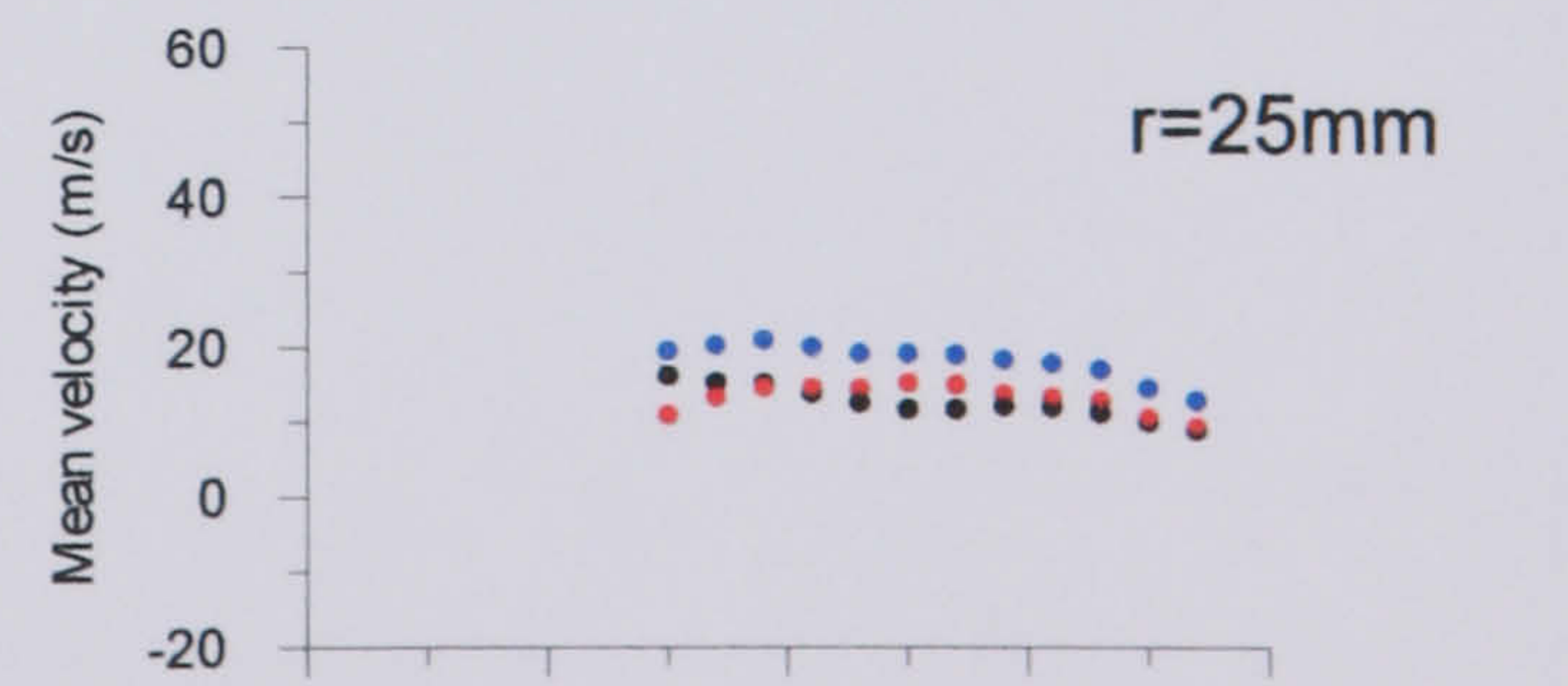
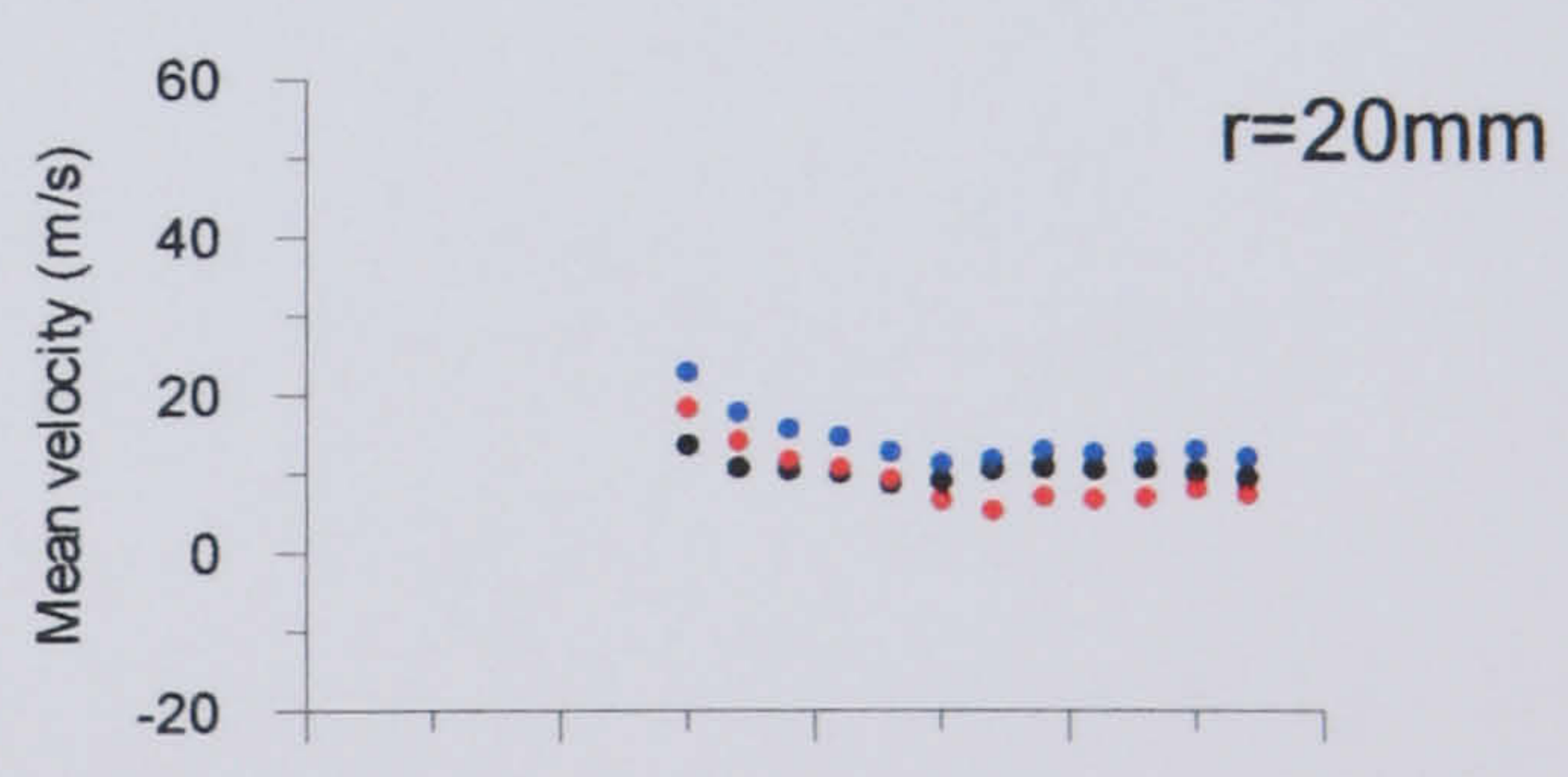
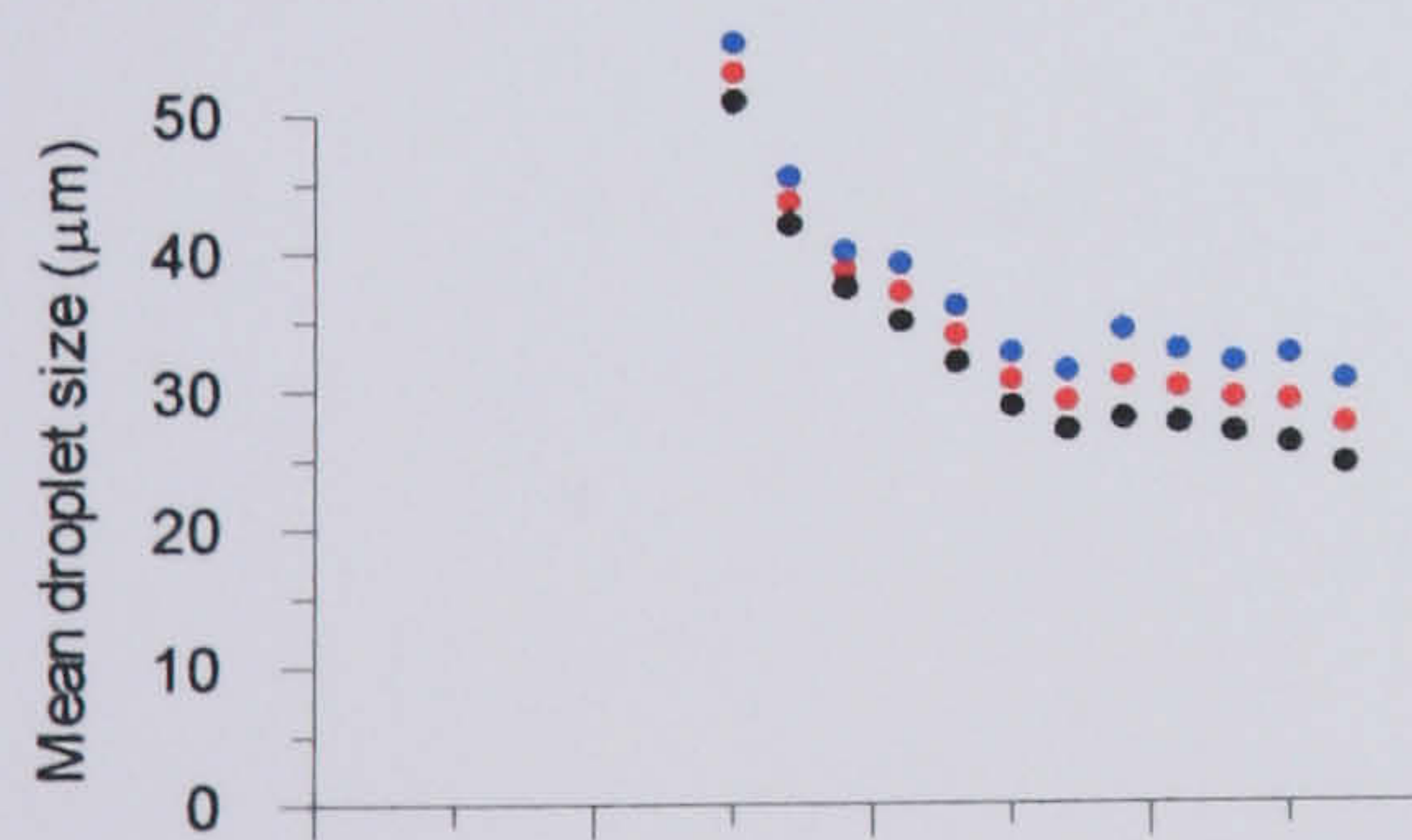
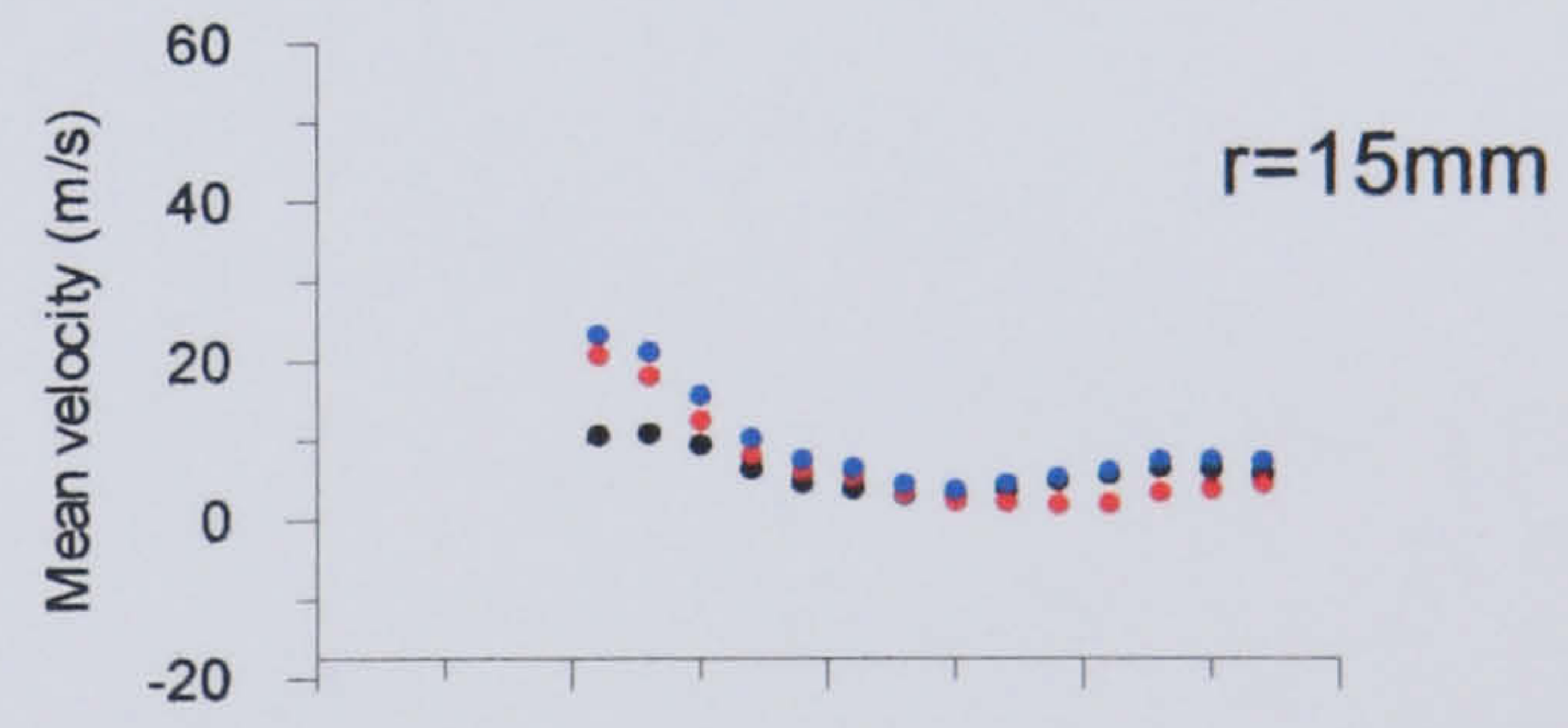
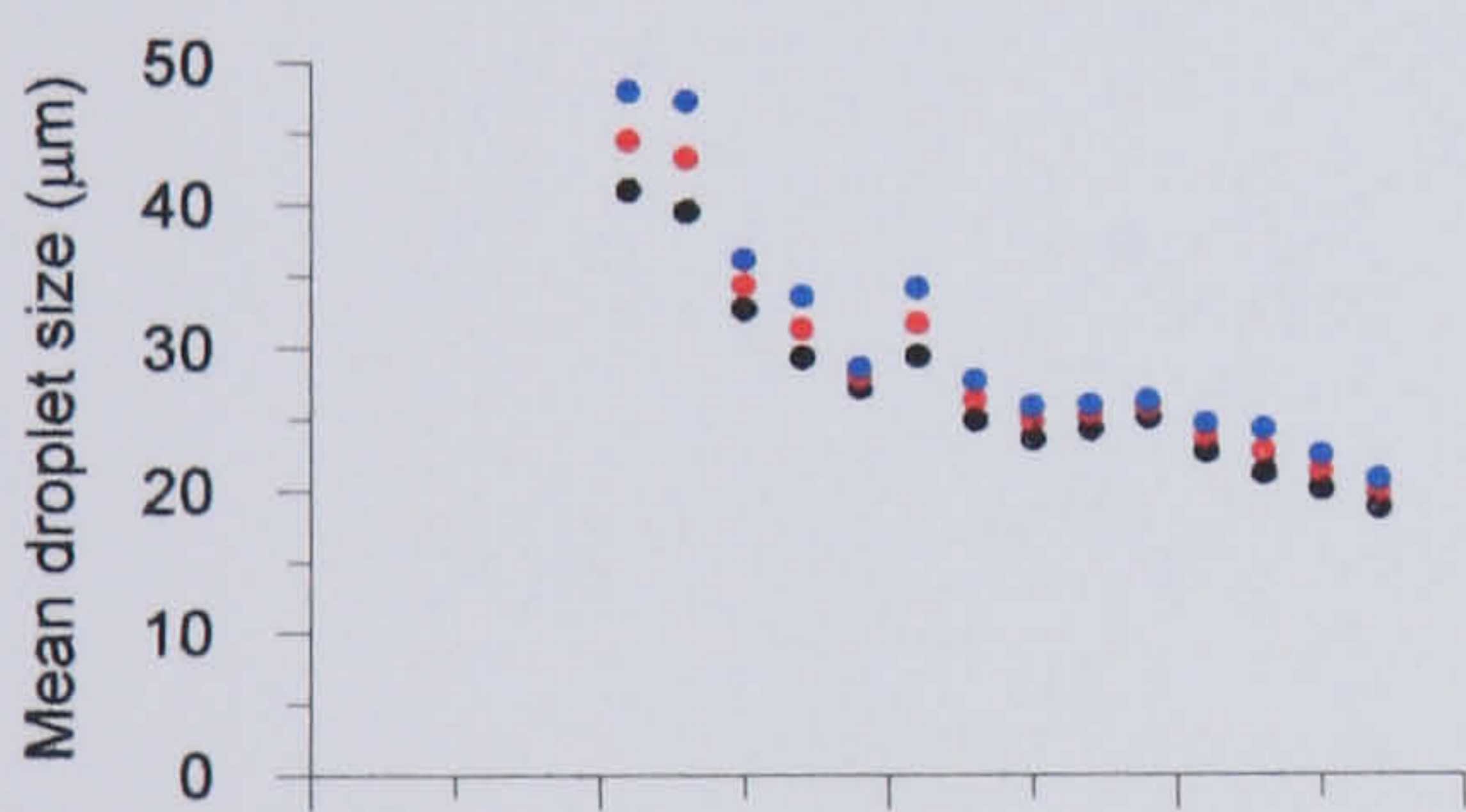
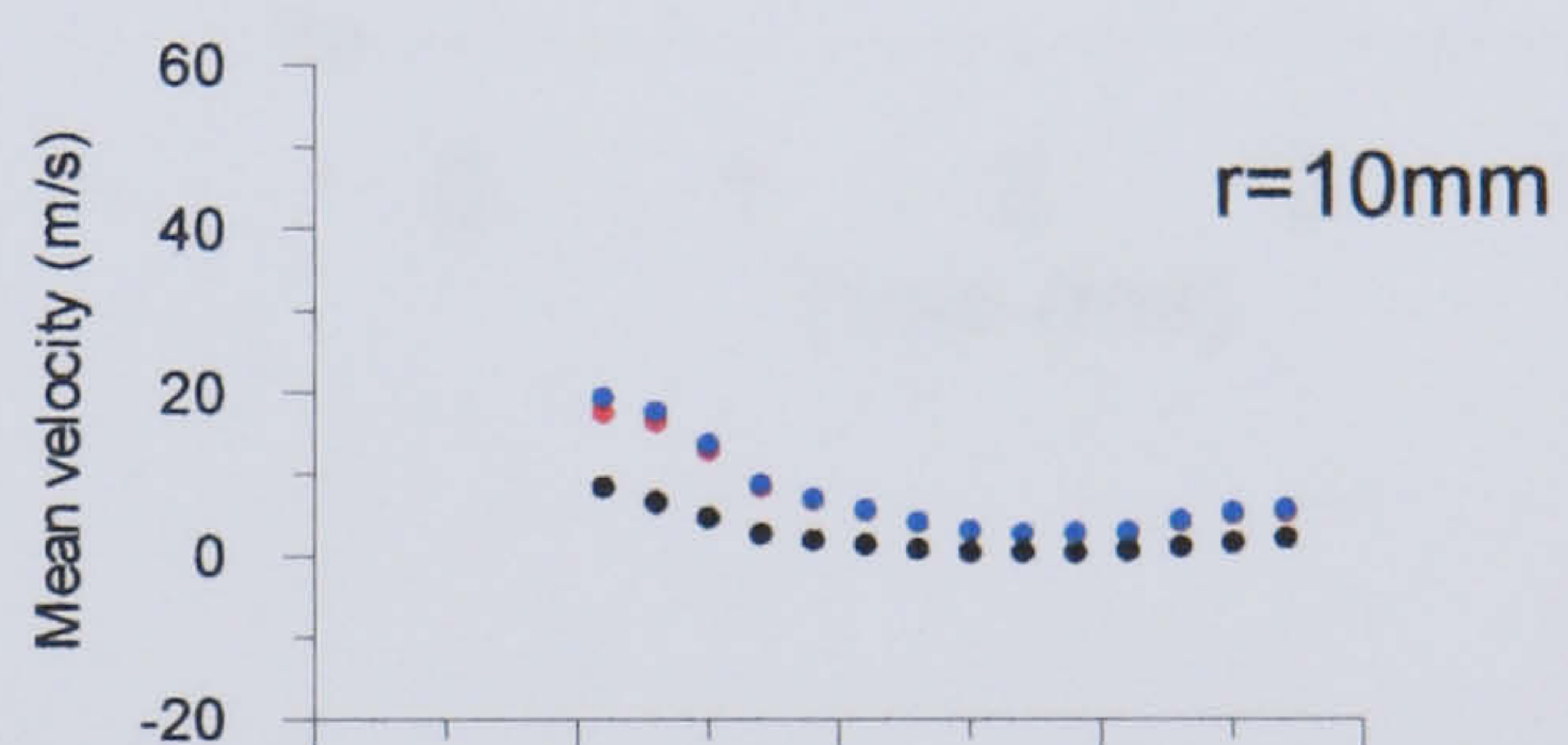
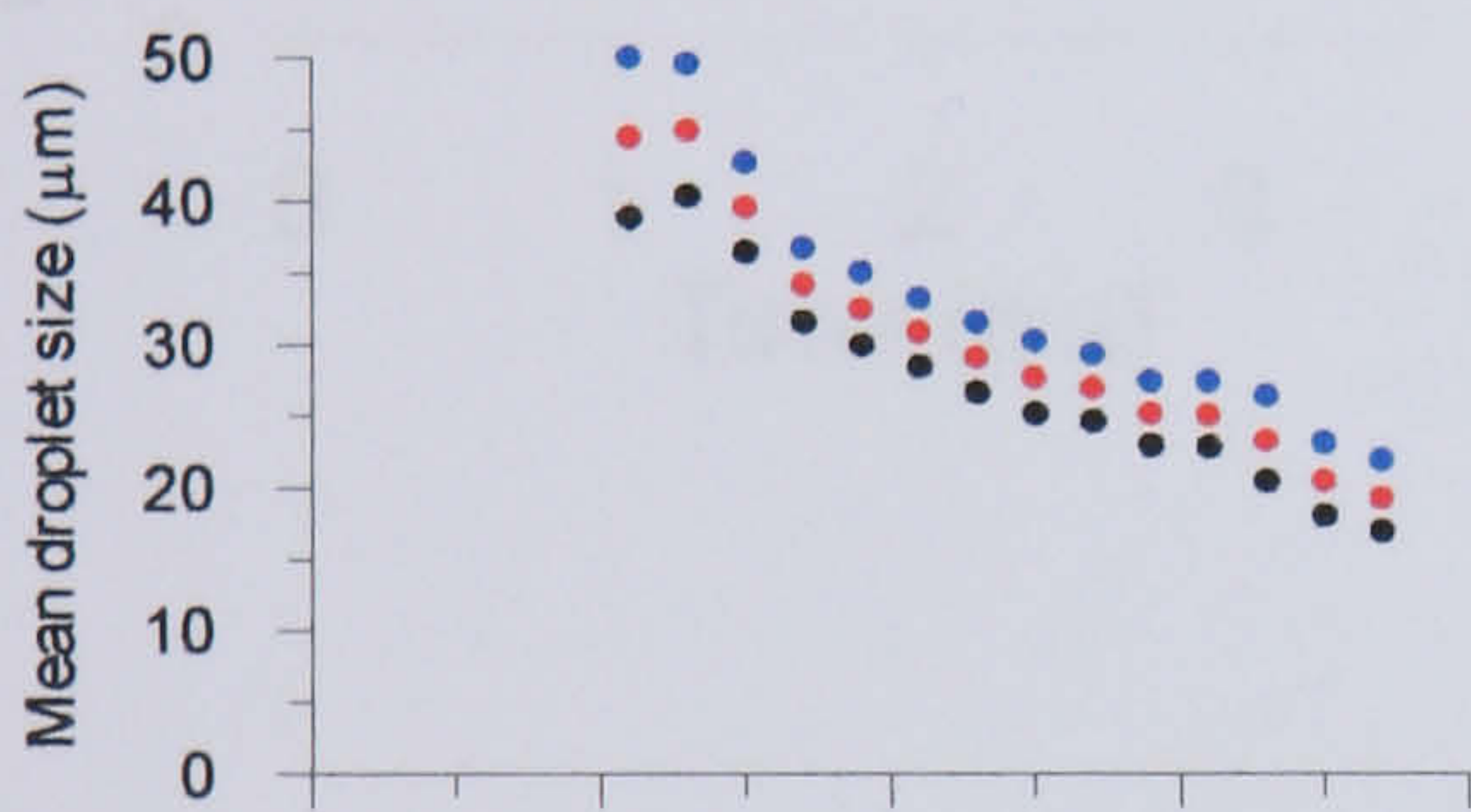
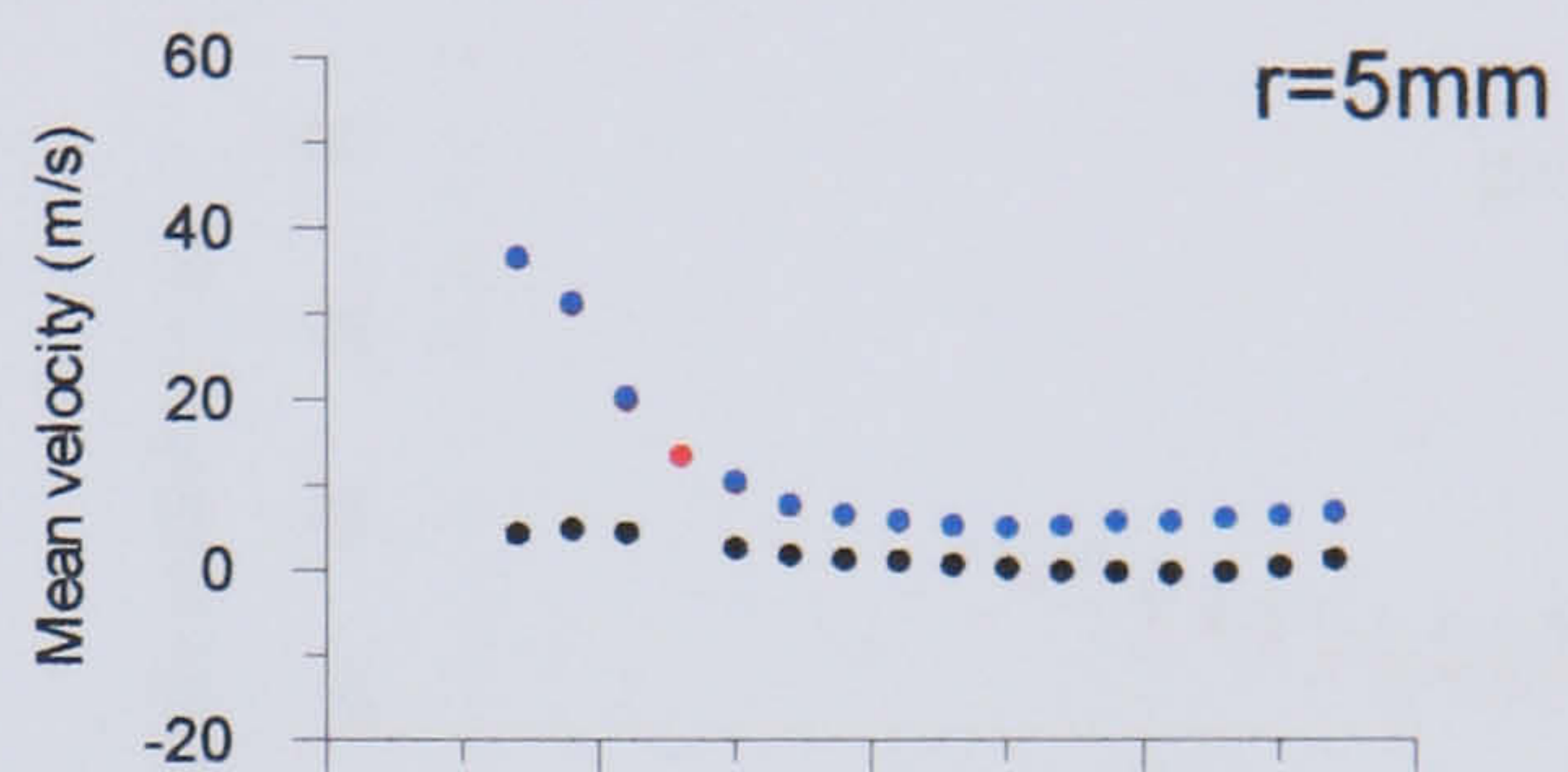
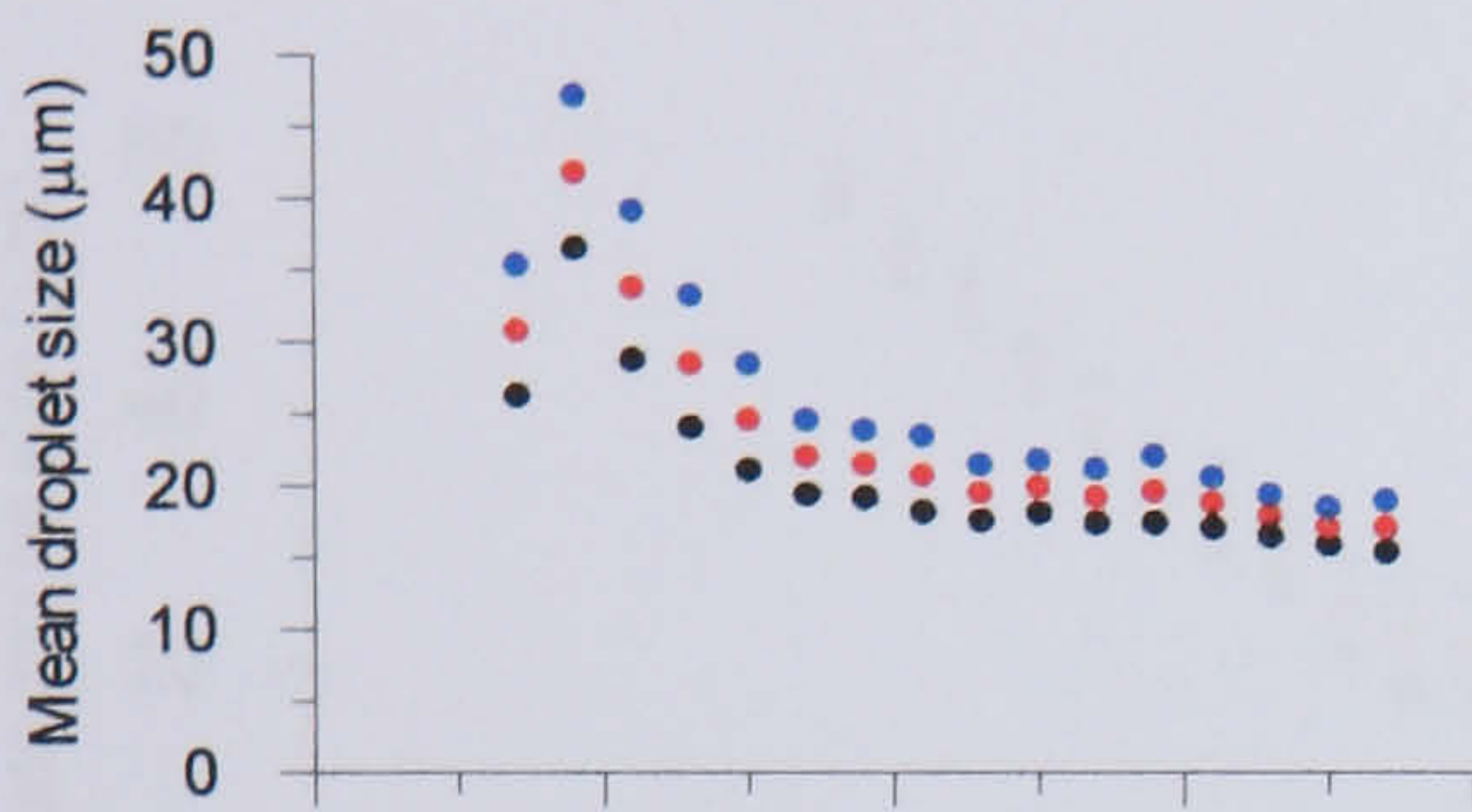
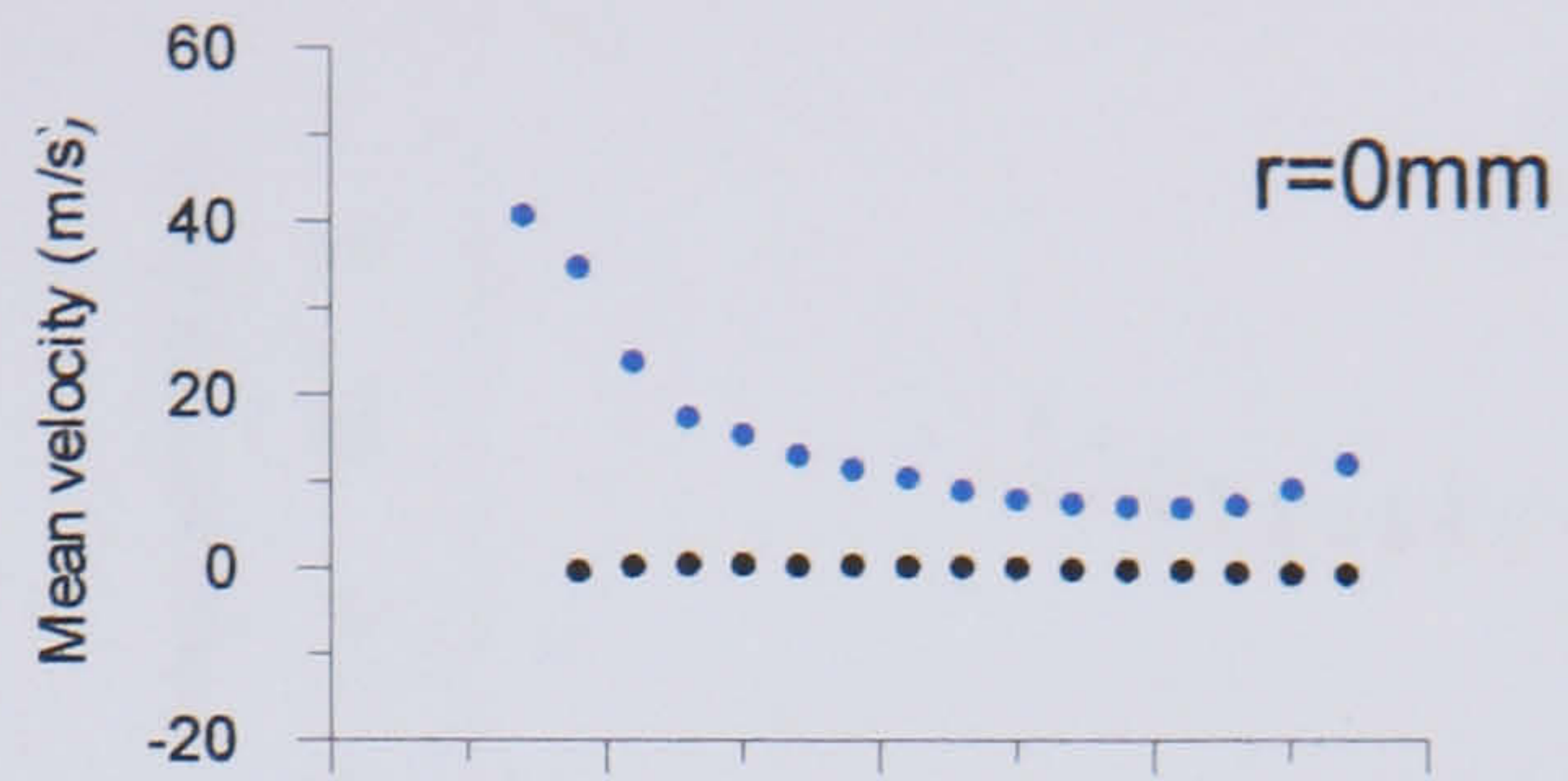
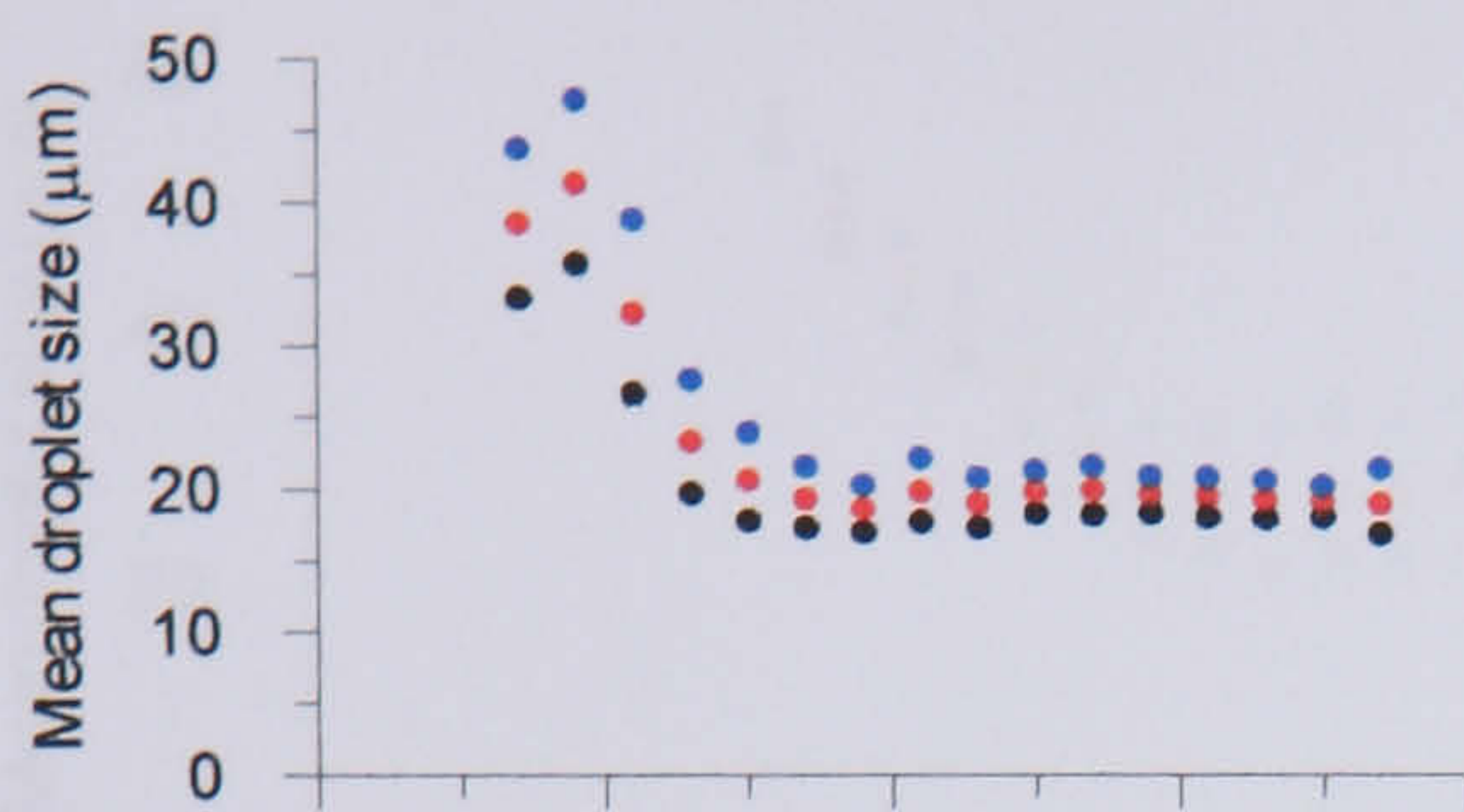


0 1 2 3 4
Time (ms)
D32
D30
D32

0 1 2 3 4
Time (ms)
Axial
Radial
total

Temporal variation of mean droplet diameters and velocities

at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=30\text{mm}$

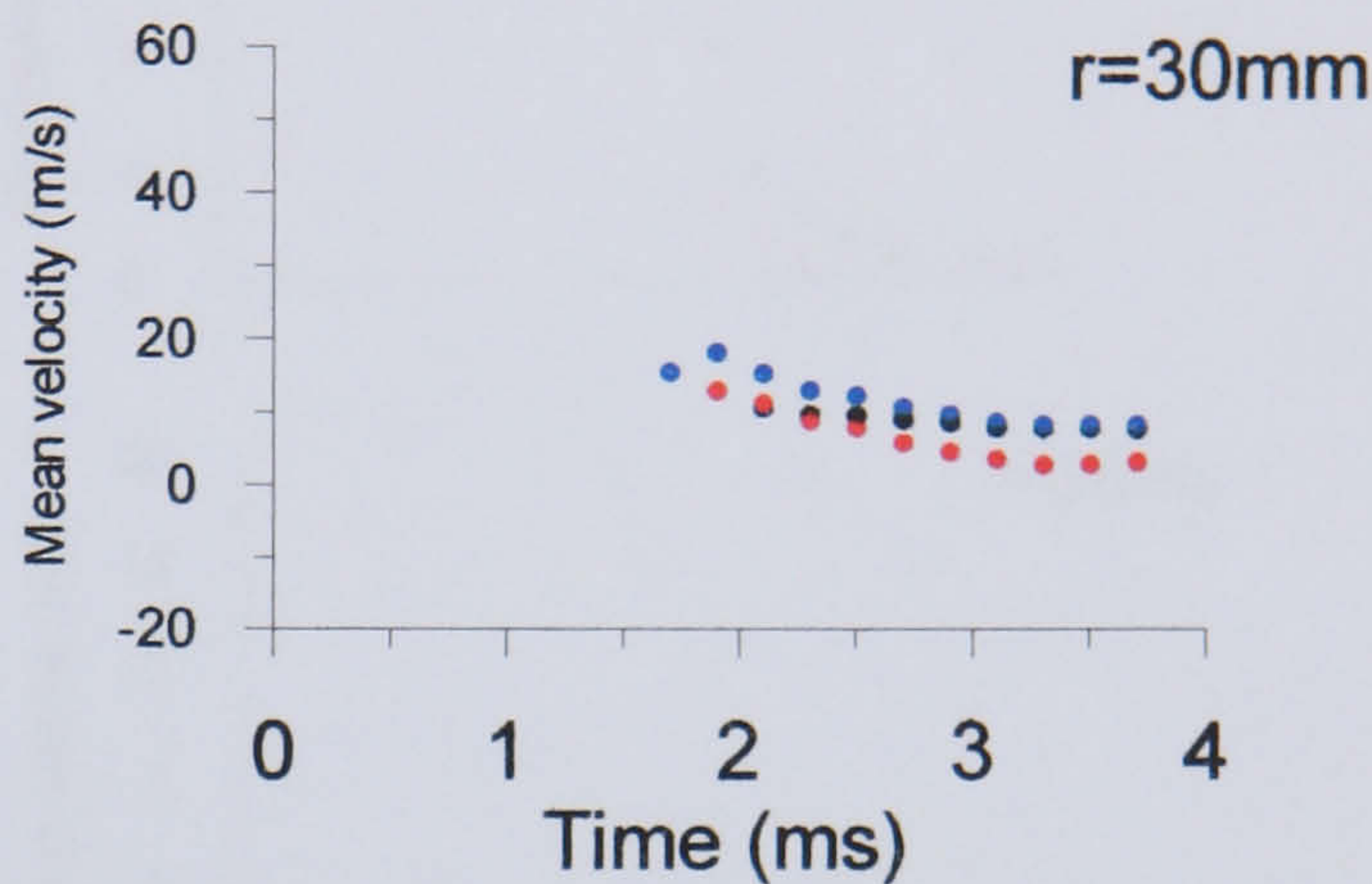
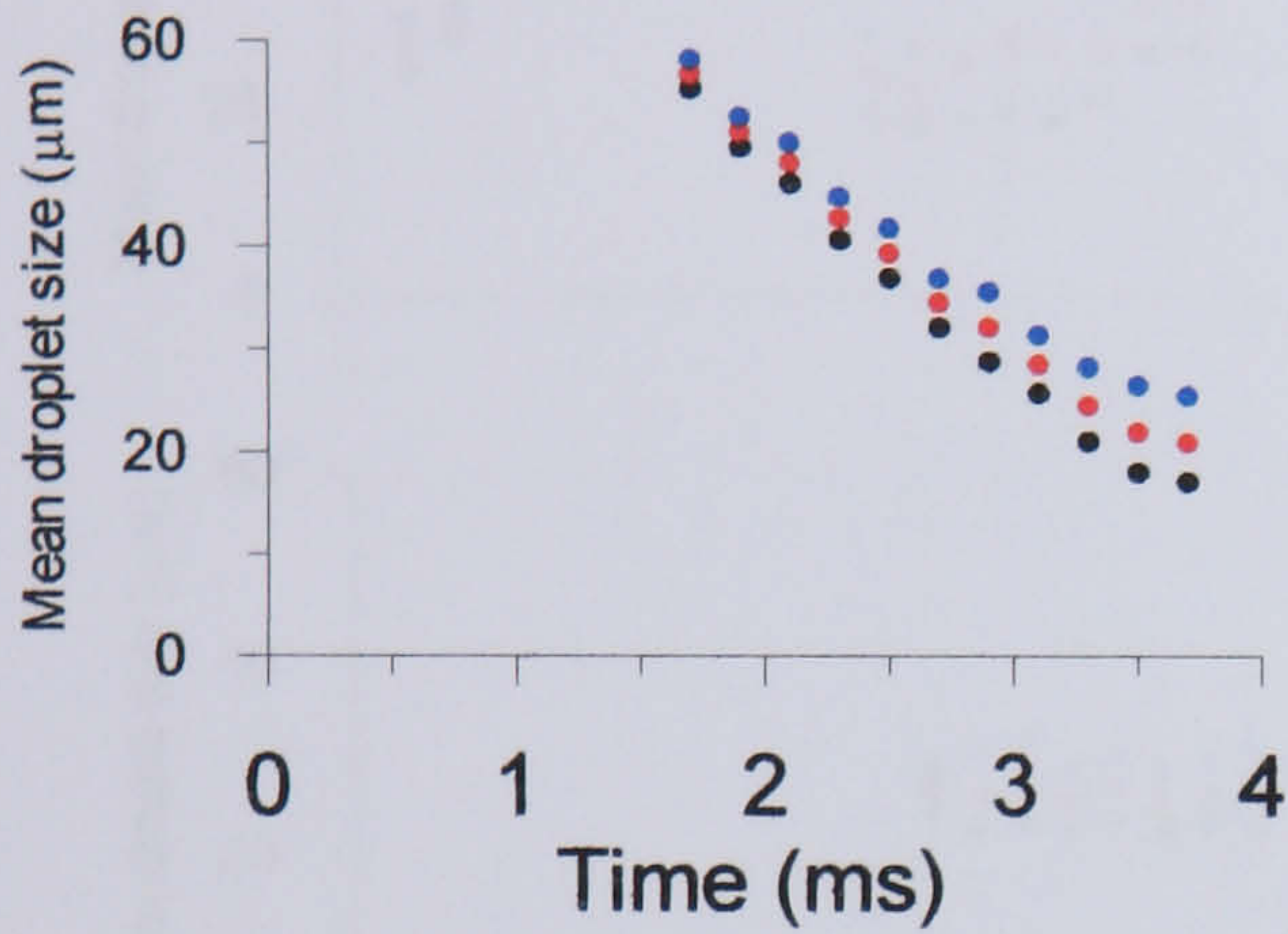
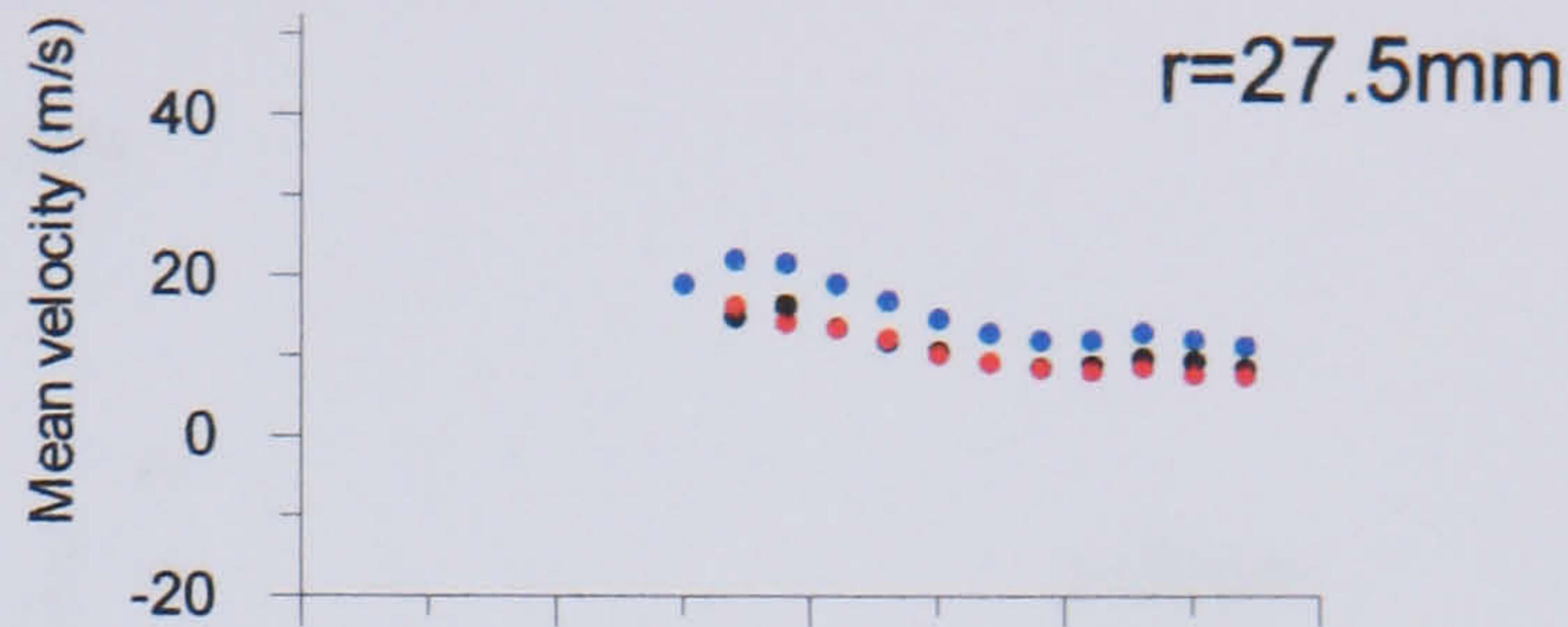
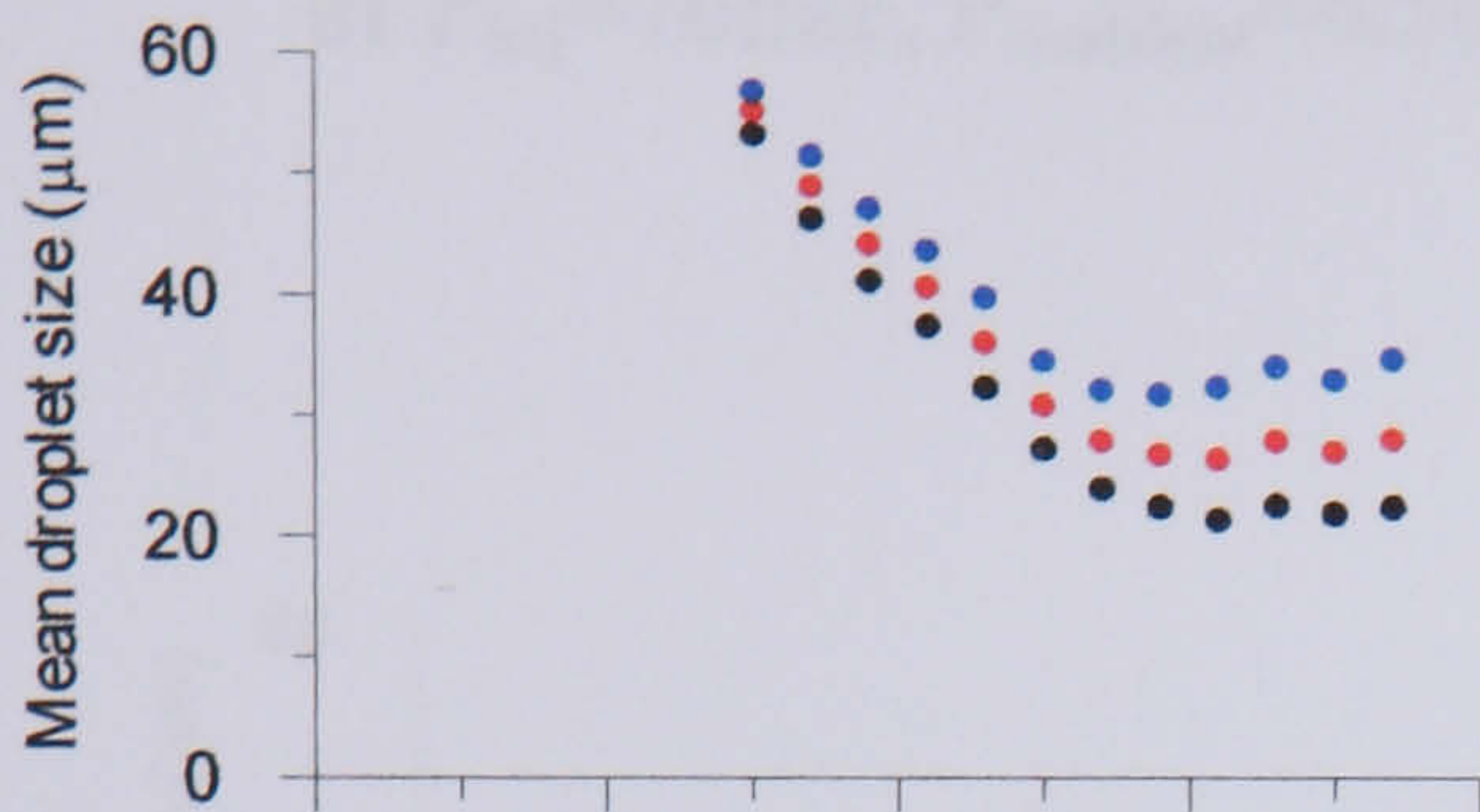


0 1 2 3 4
Time (ms)
• D32
• D30
• D32

0 1 2 3 4
Time (ms)
• Axial
• Radial
• total

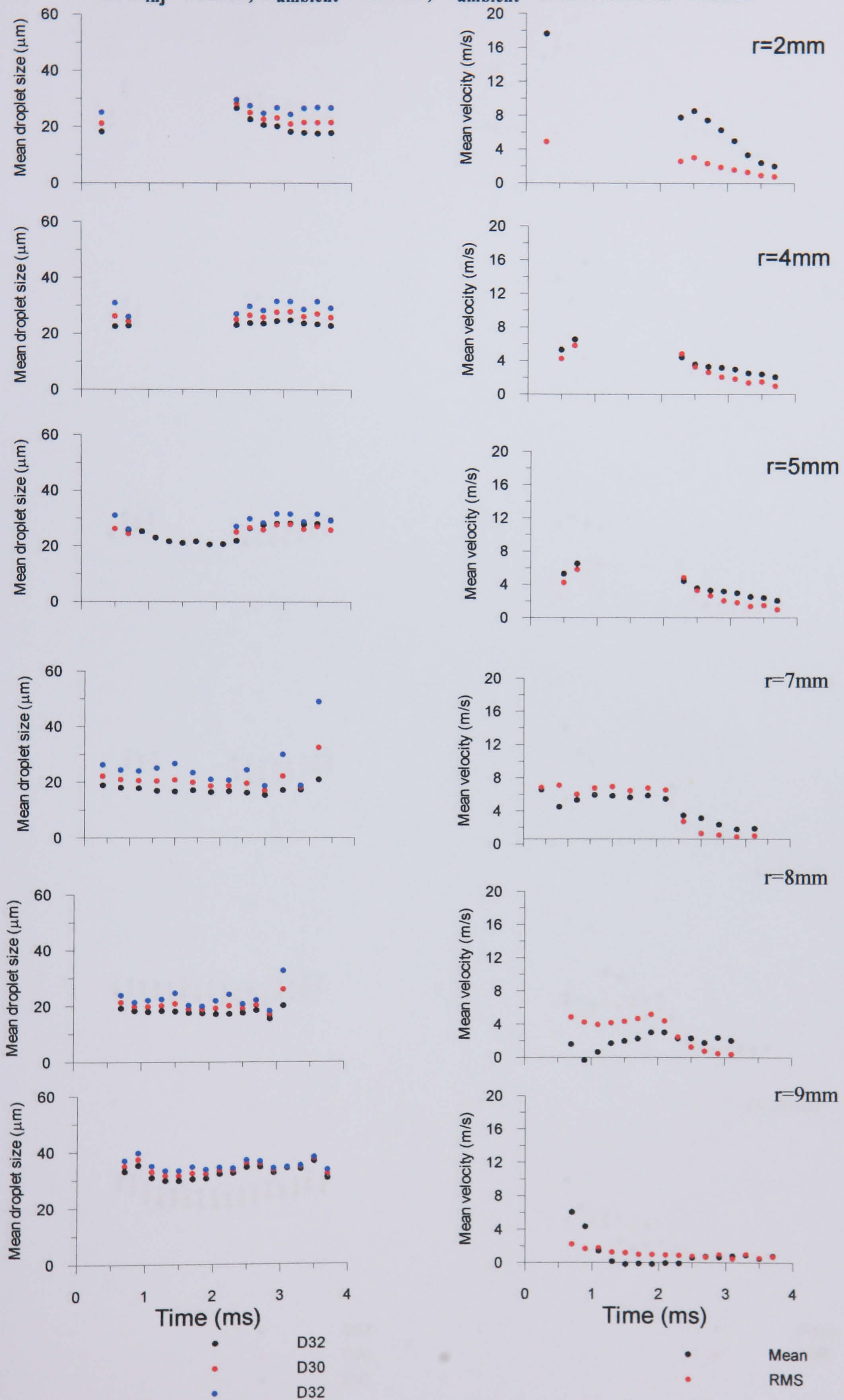
Temporal variation of mean droplet diameters and velocities

at $P_{inj}=20\text{bar}$, $P_{ambient}=1\text{bar}$, $T_{ambient}=293\text{K}$ and $X=30\text{mm}$ (continue)



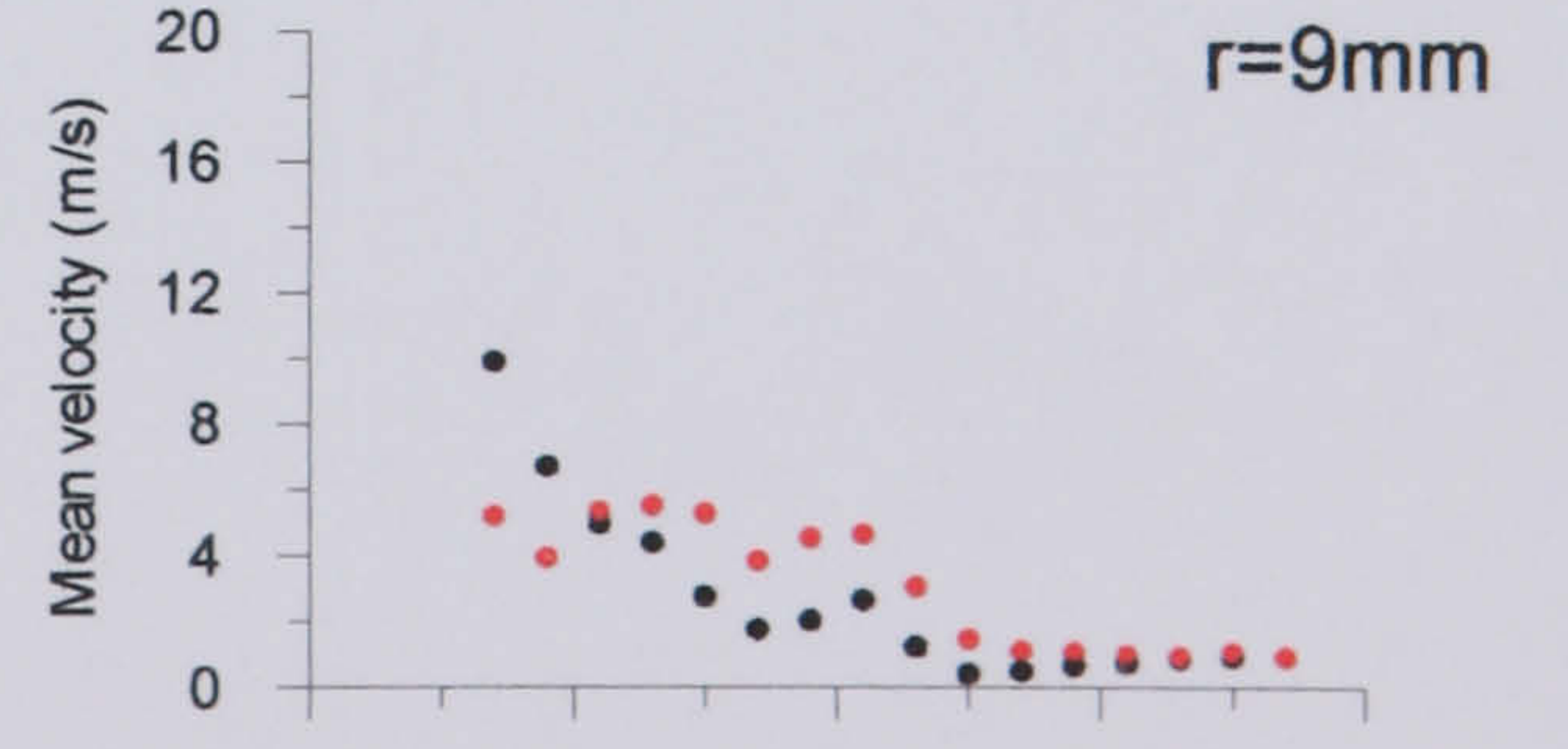
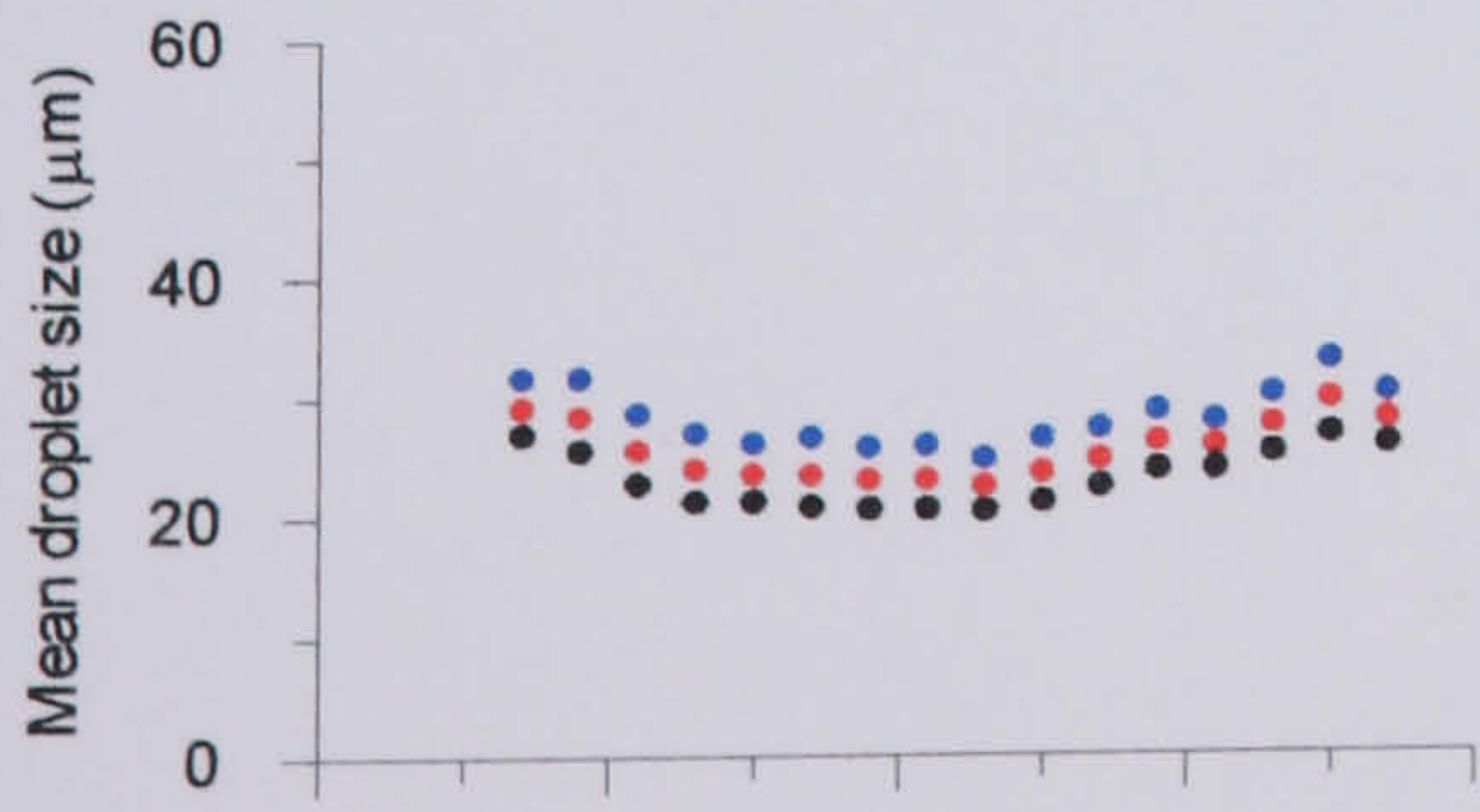
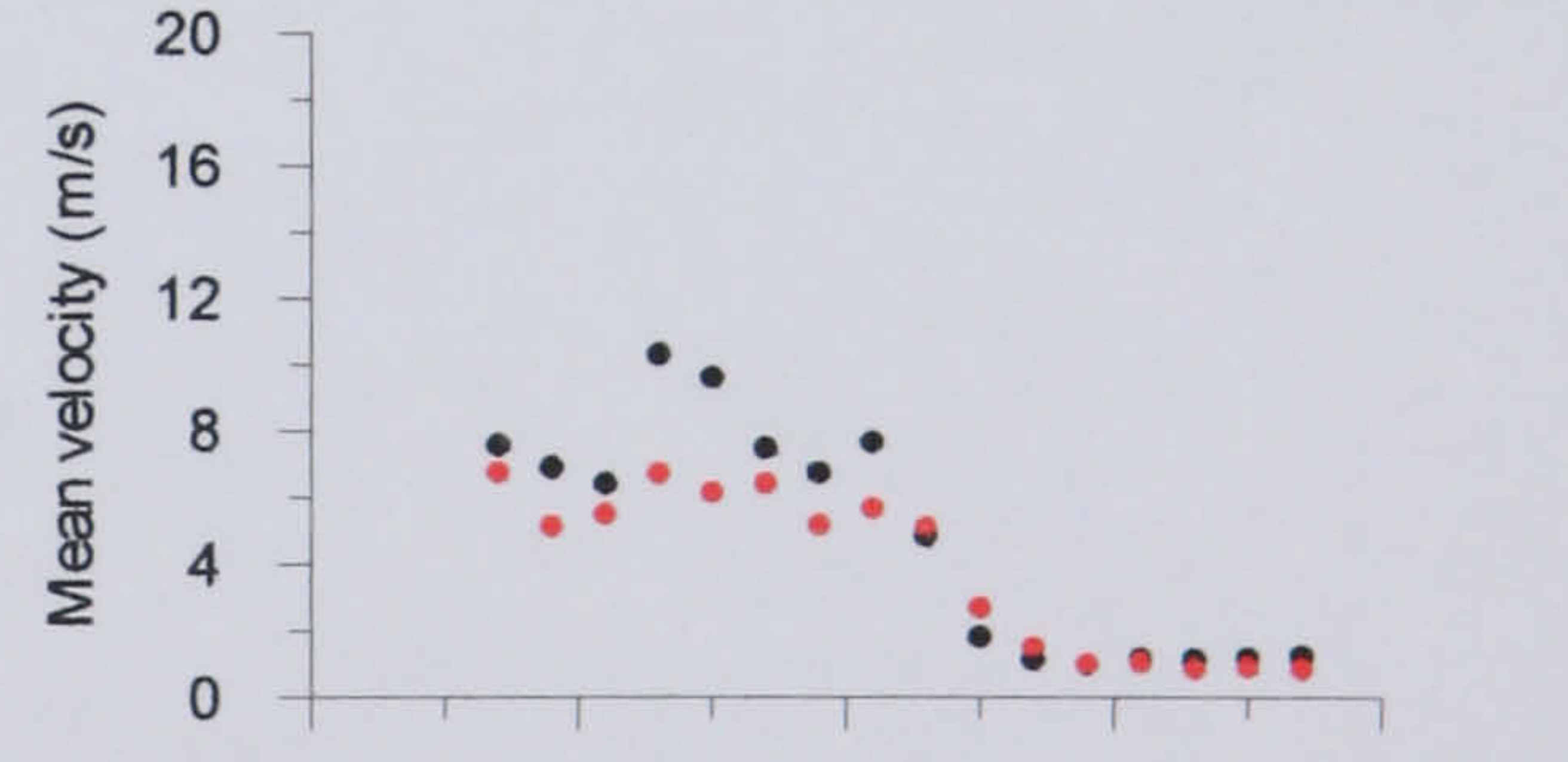
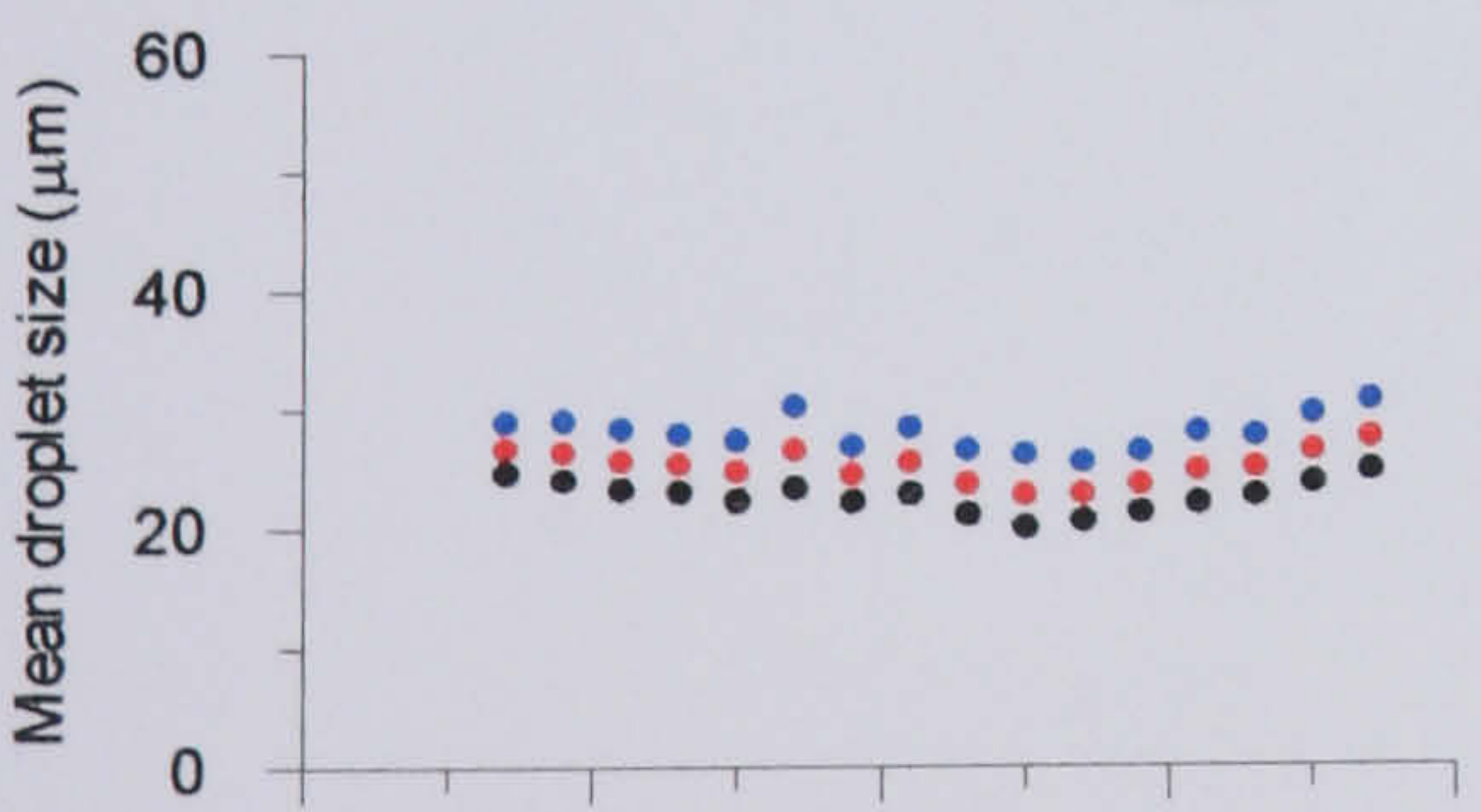
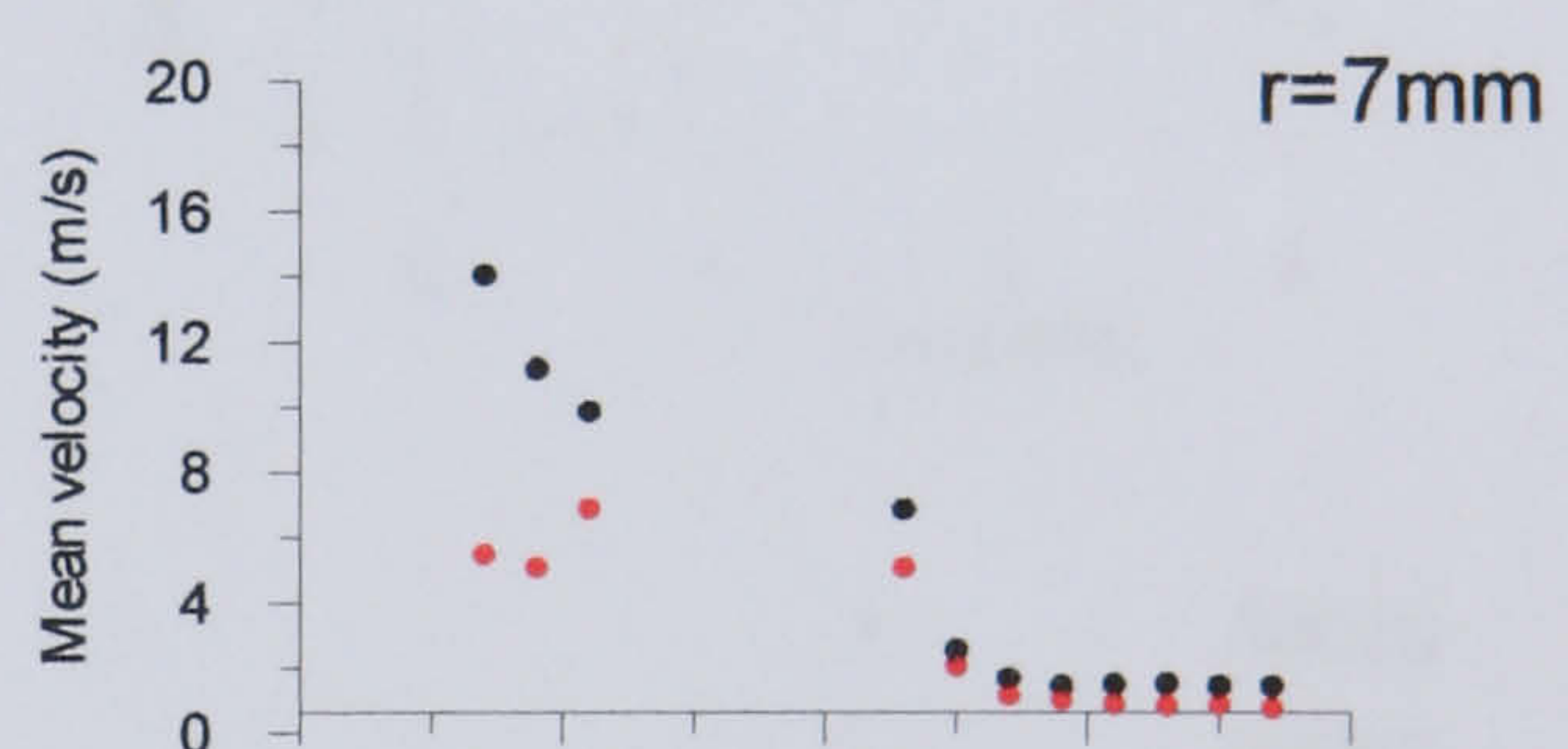
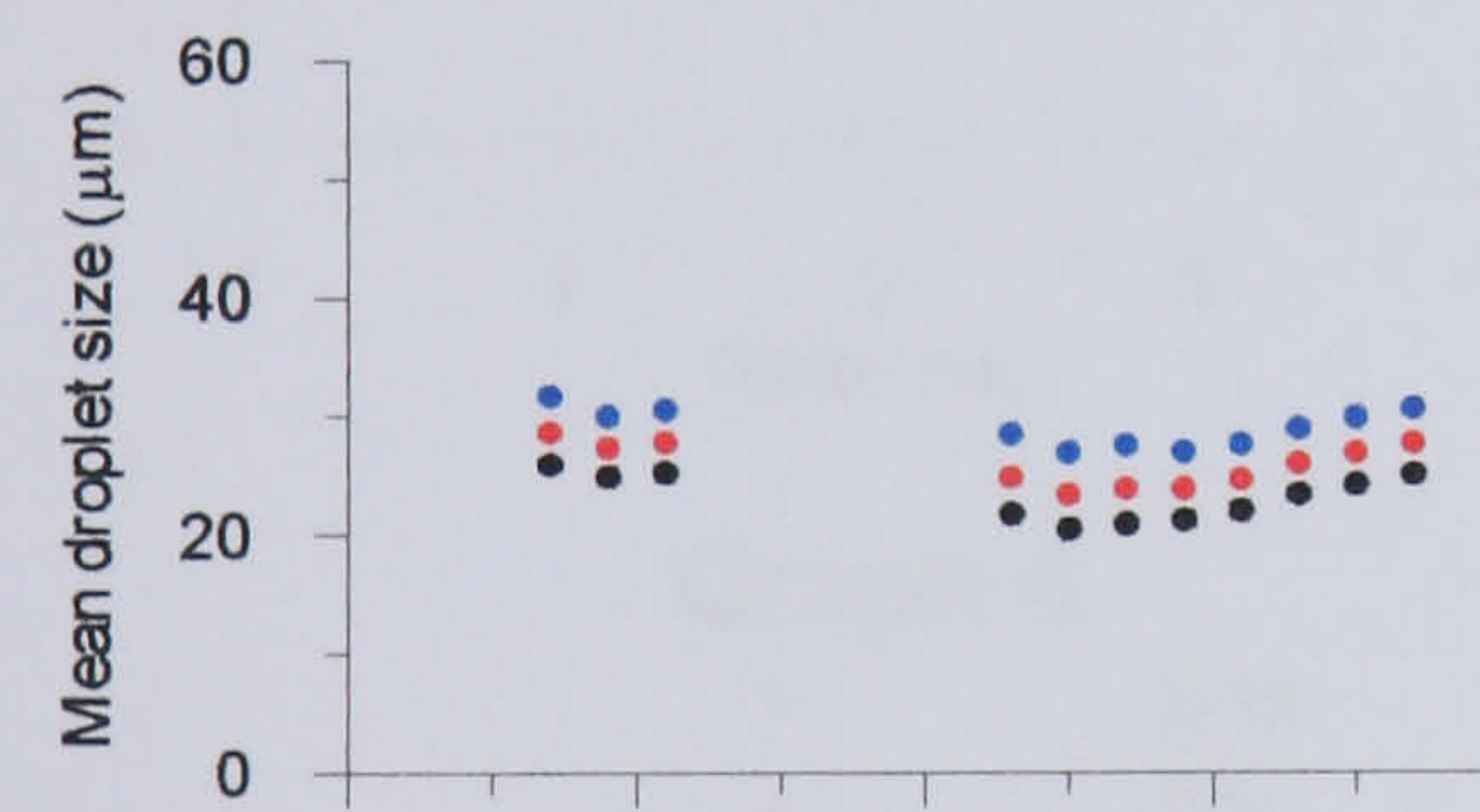
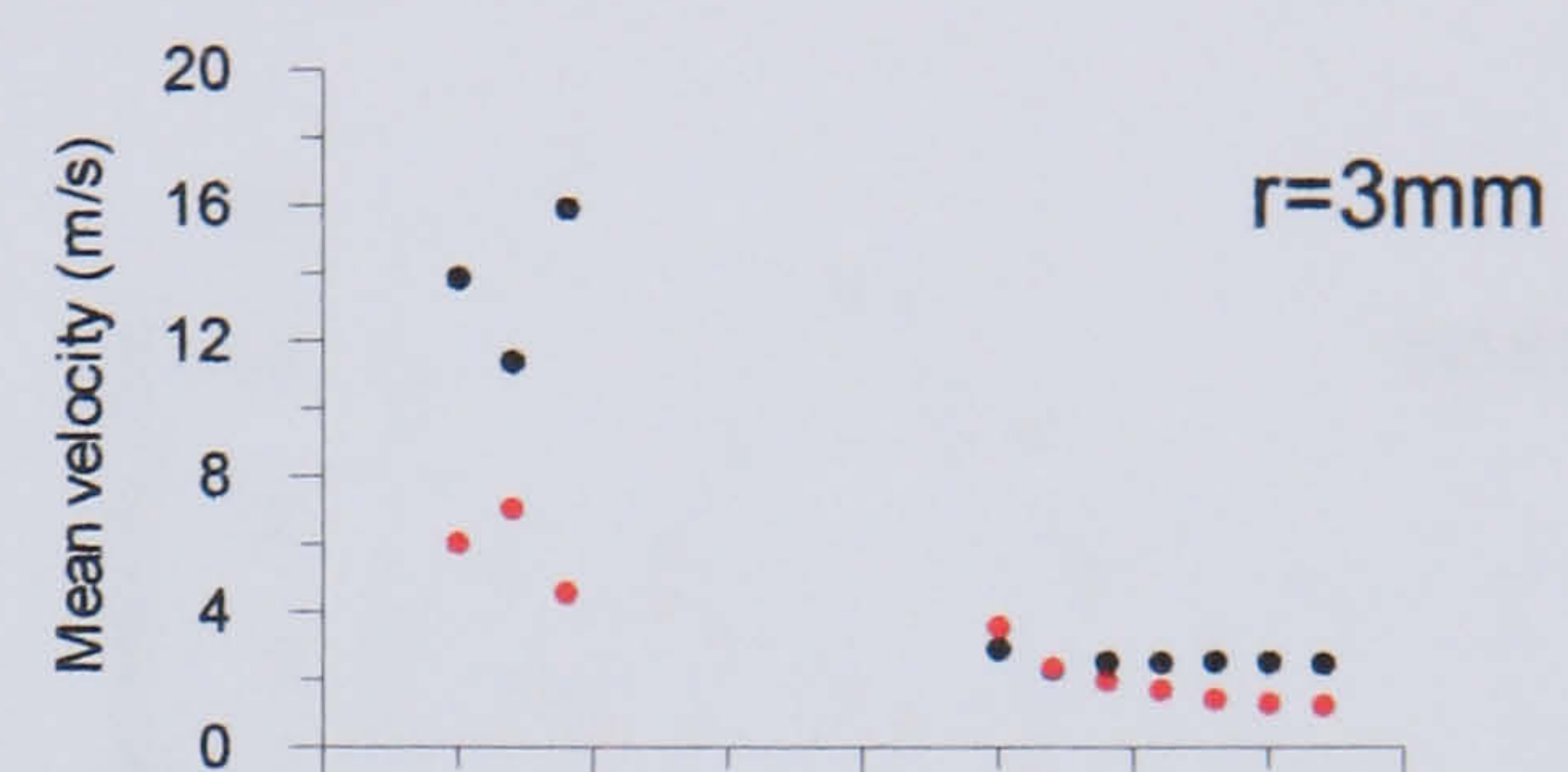
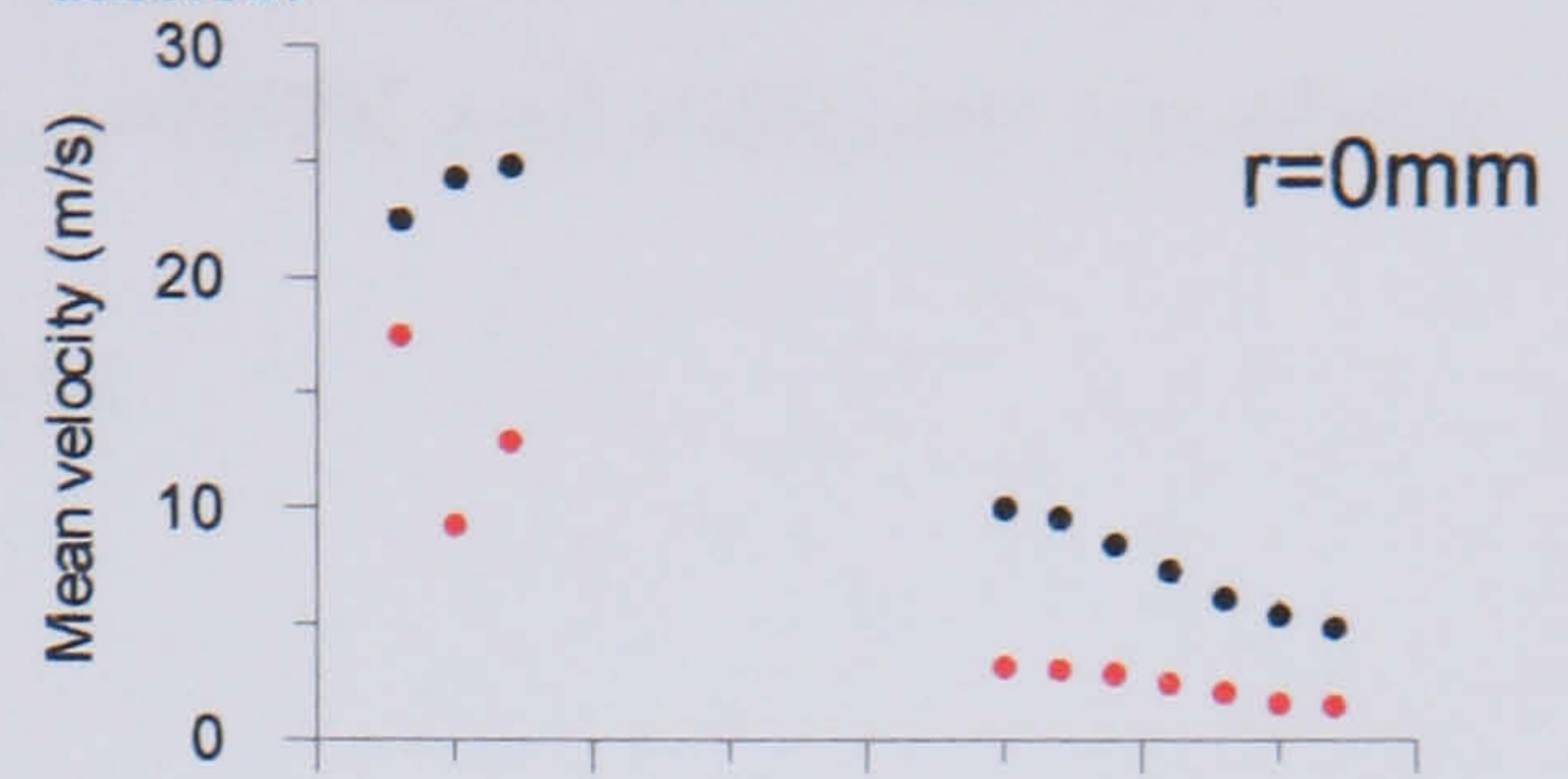
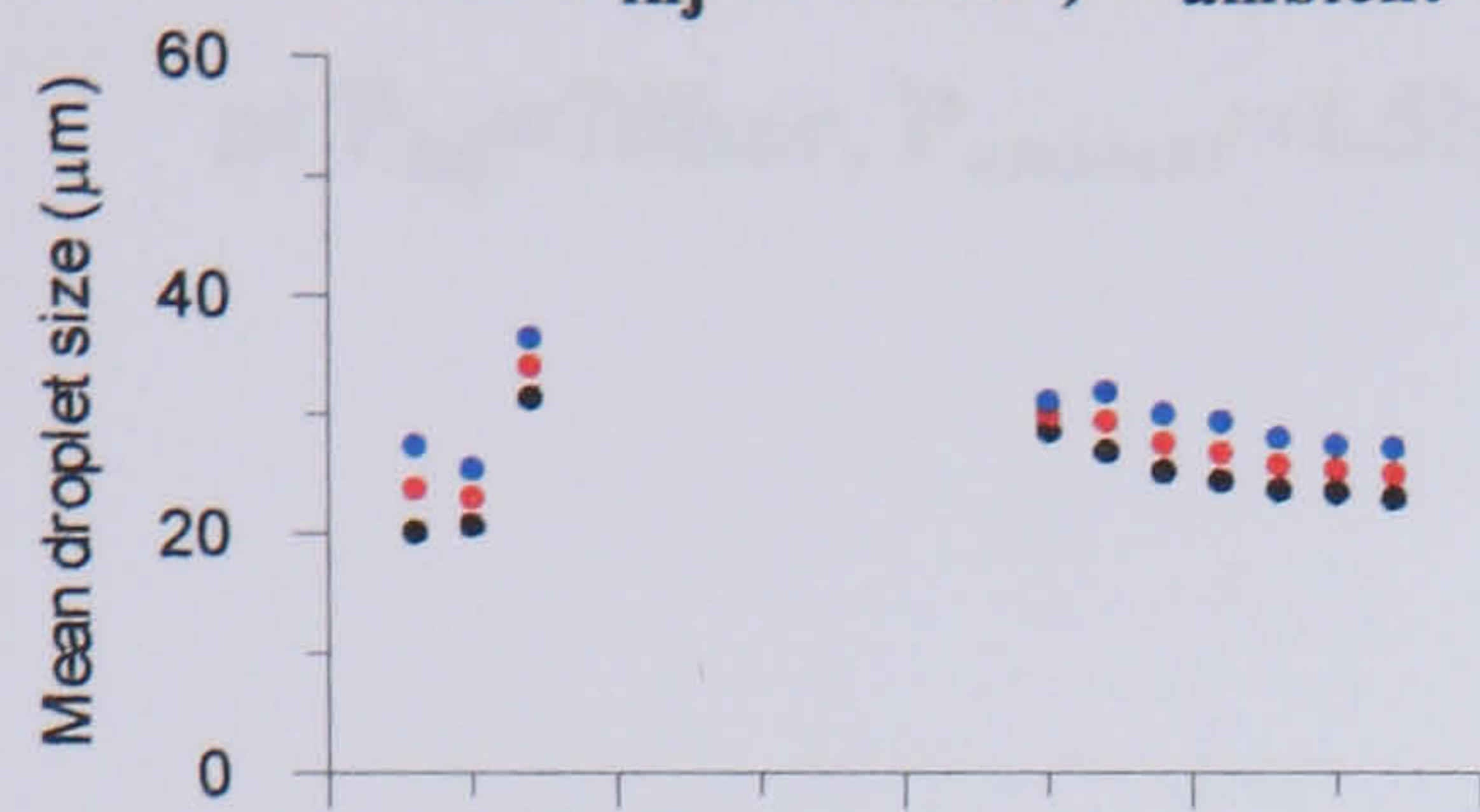
Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=4.5\text{bar}$, $T_{ambient}=293\text{K}$ and $X=10\text{mm}$



Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=4.5\text{bar}$, $T_{ambient}=293\text{K}$ and $X=15\text{mm}$



● D32
● D30
● D32

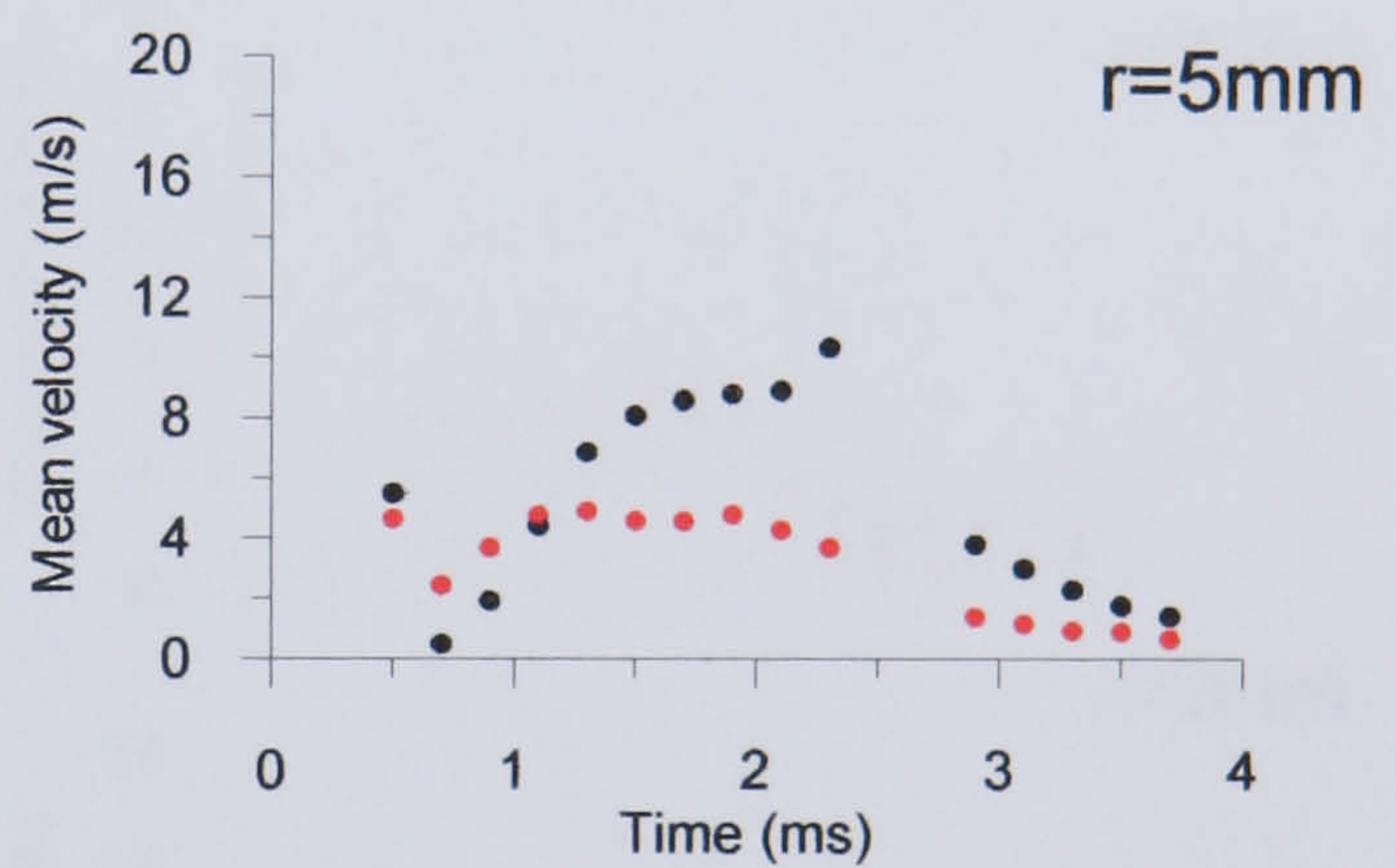
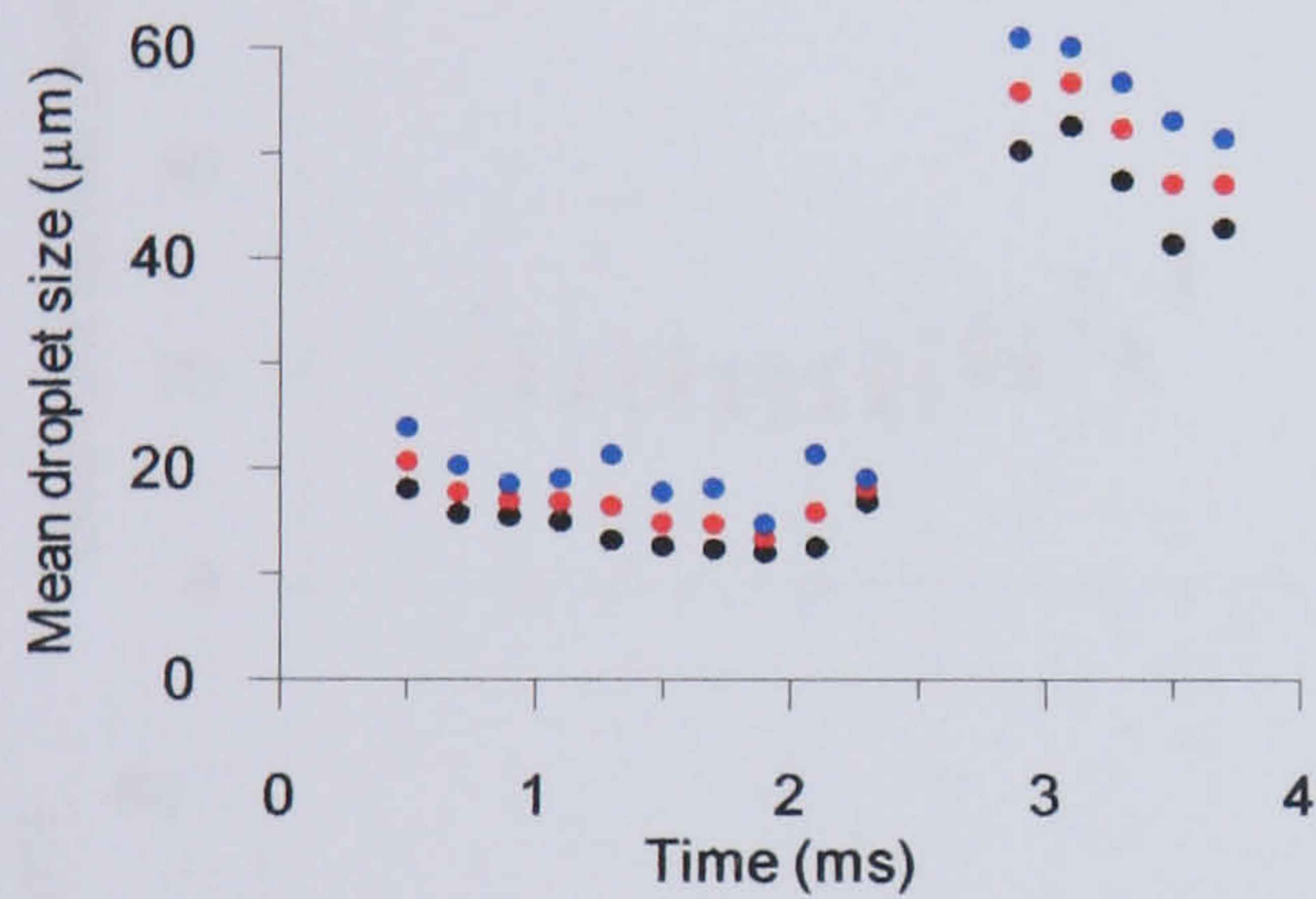
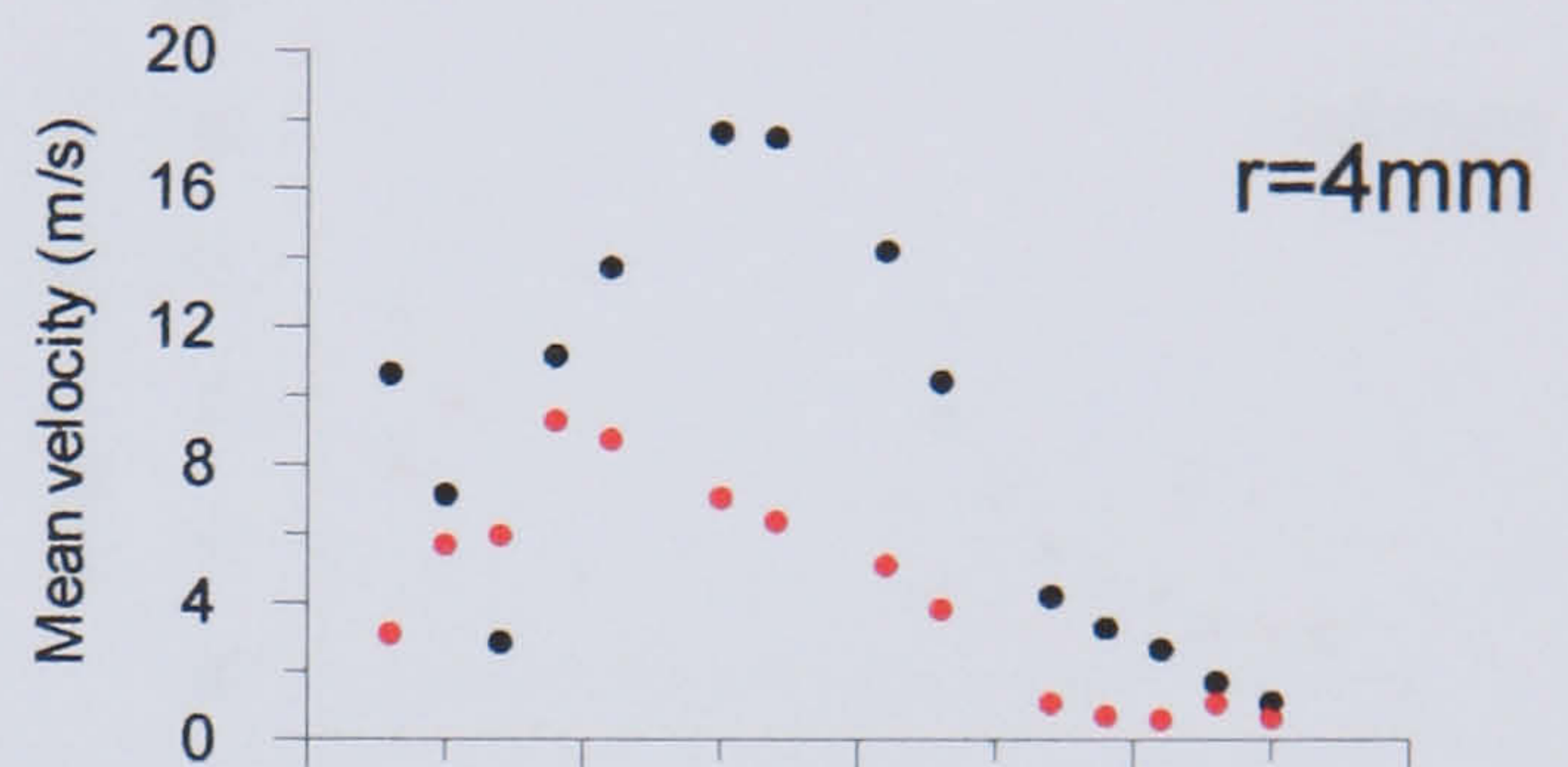
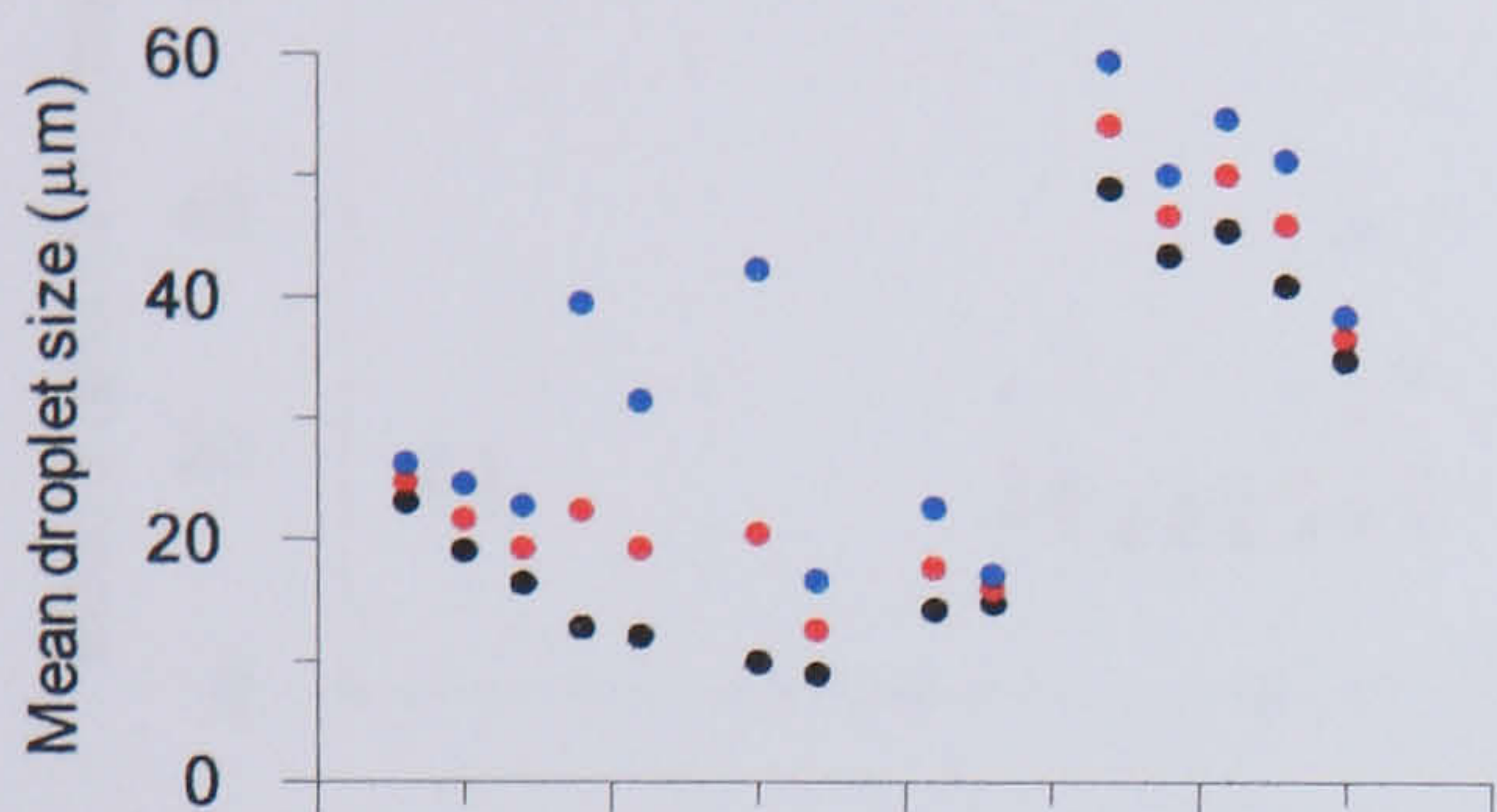
● Mean
● RMS

Appendix 4-D

Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=4.5\text{bar}$, $T_{ambient}=500\text{K}$ and different locations

$X=5\text{mm}$



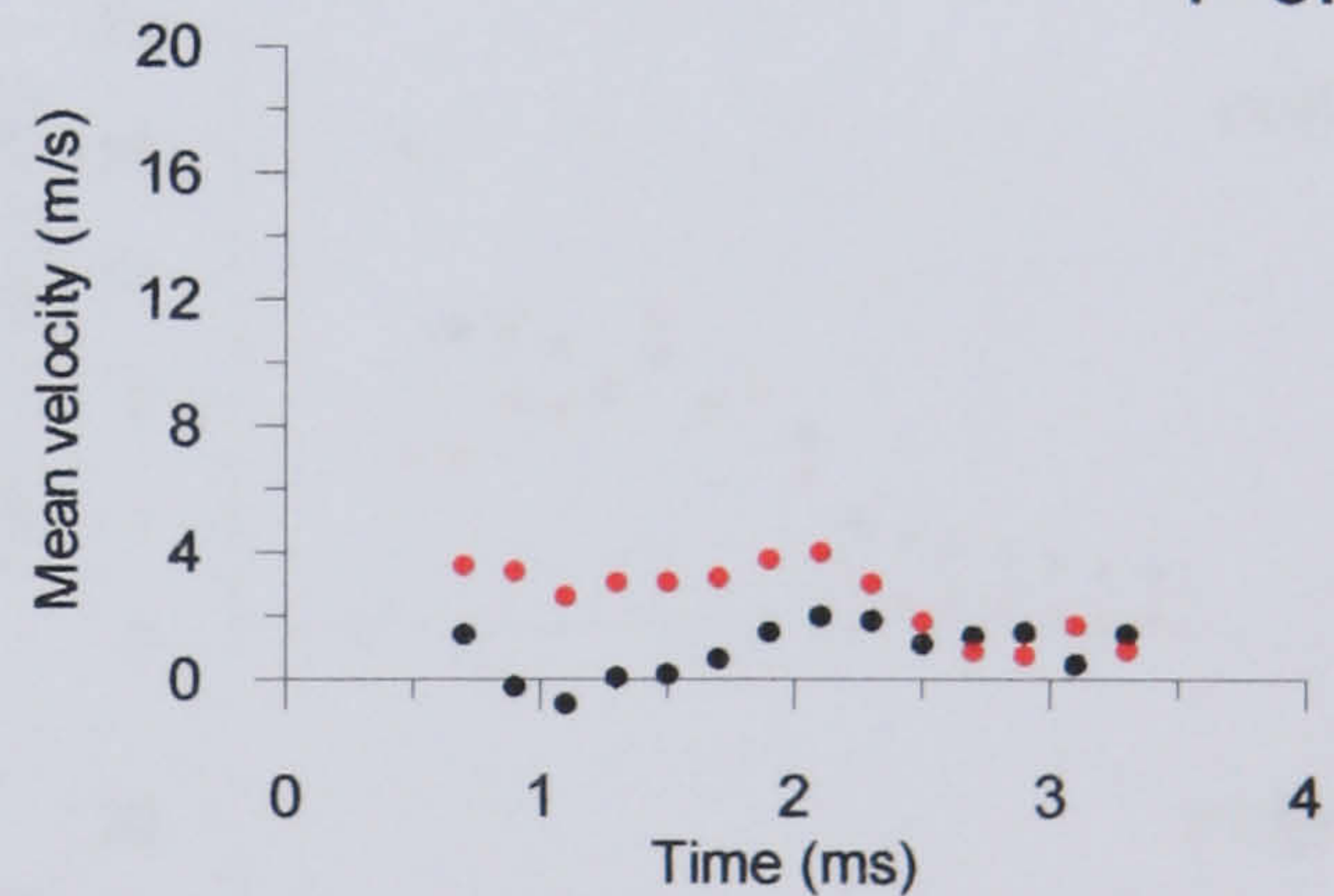
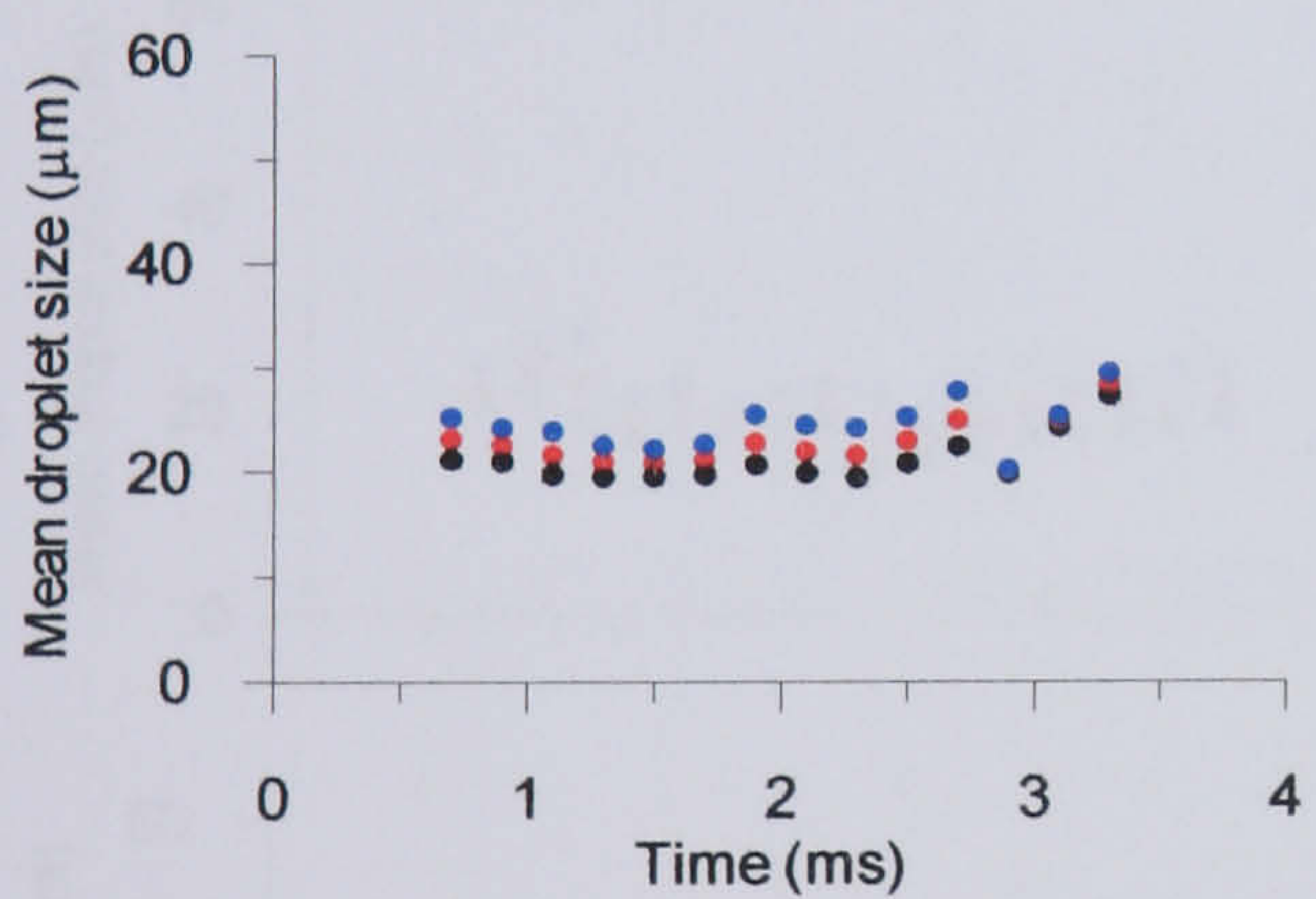
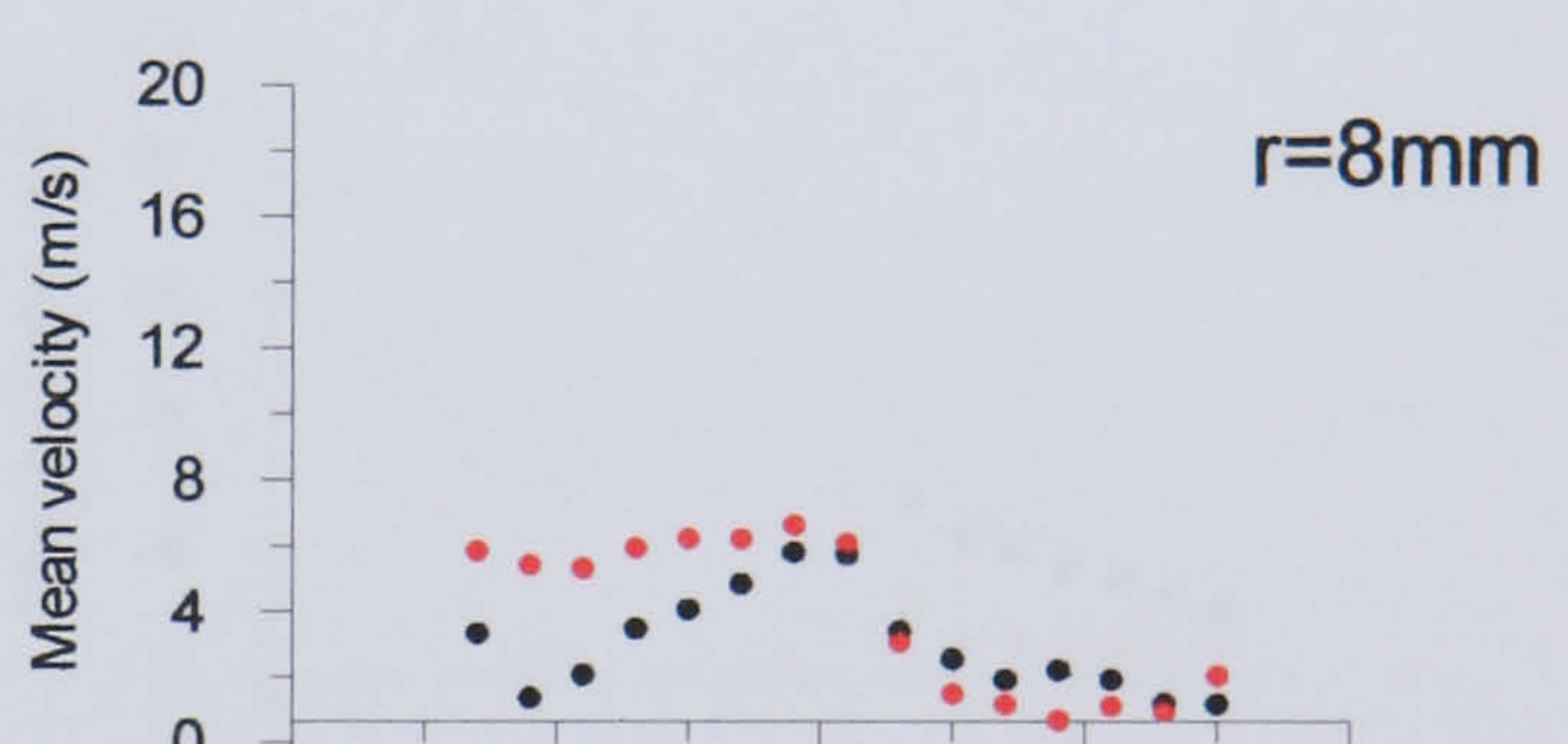
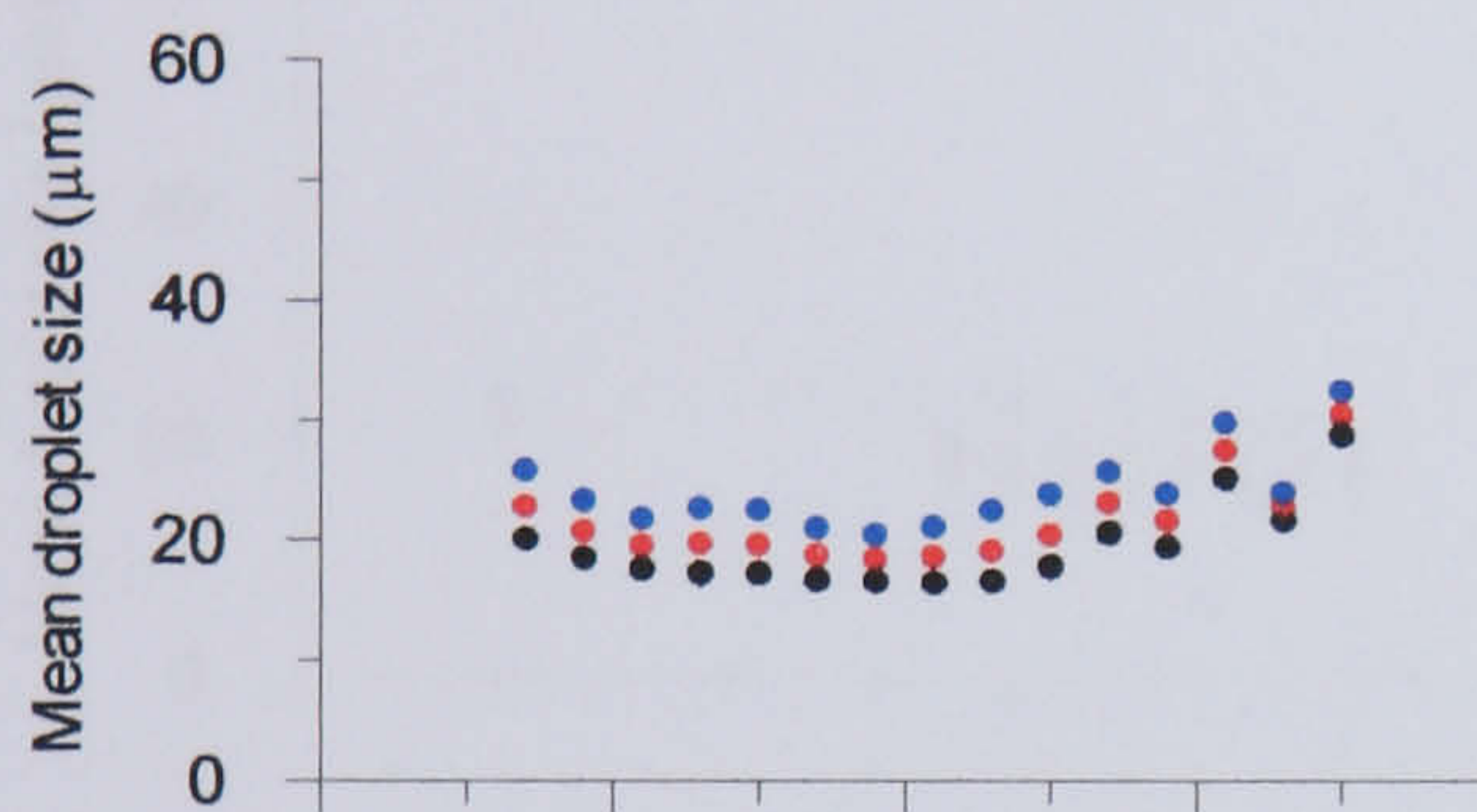
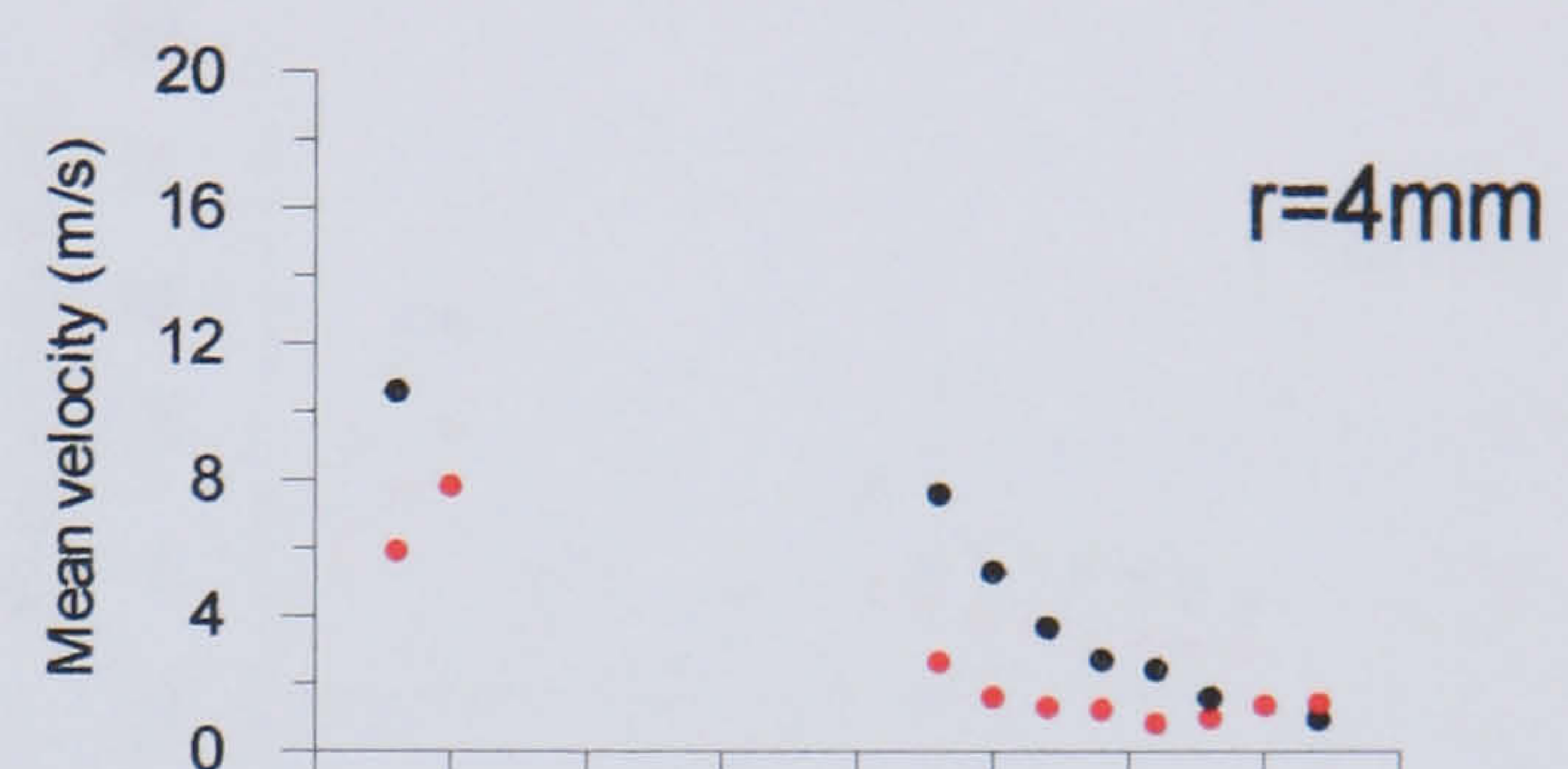
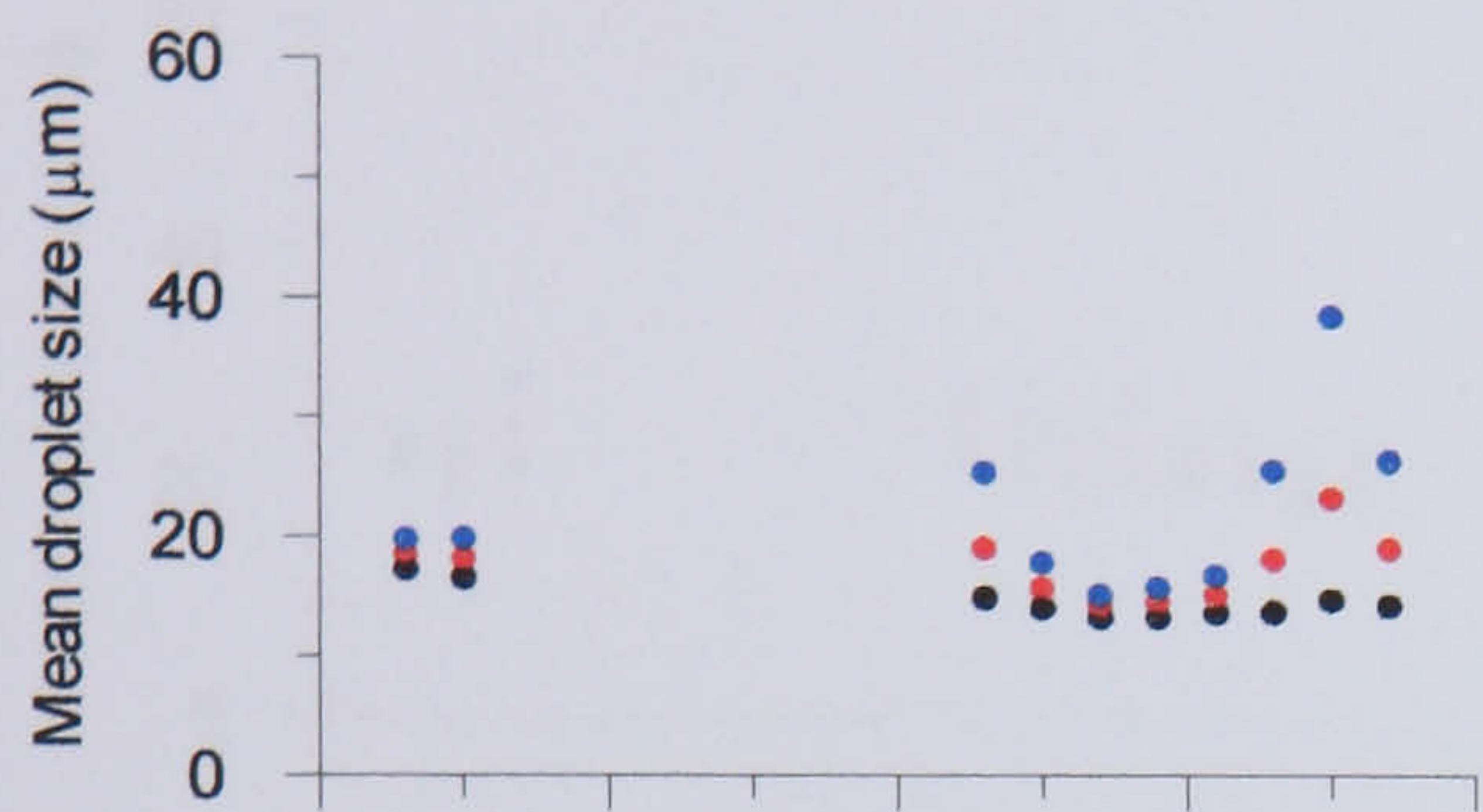
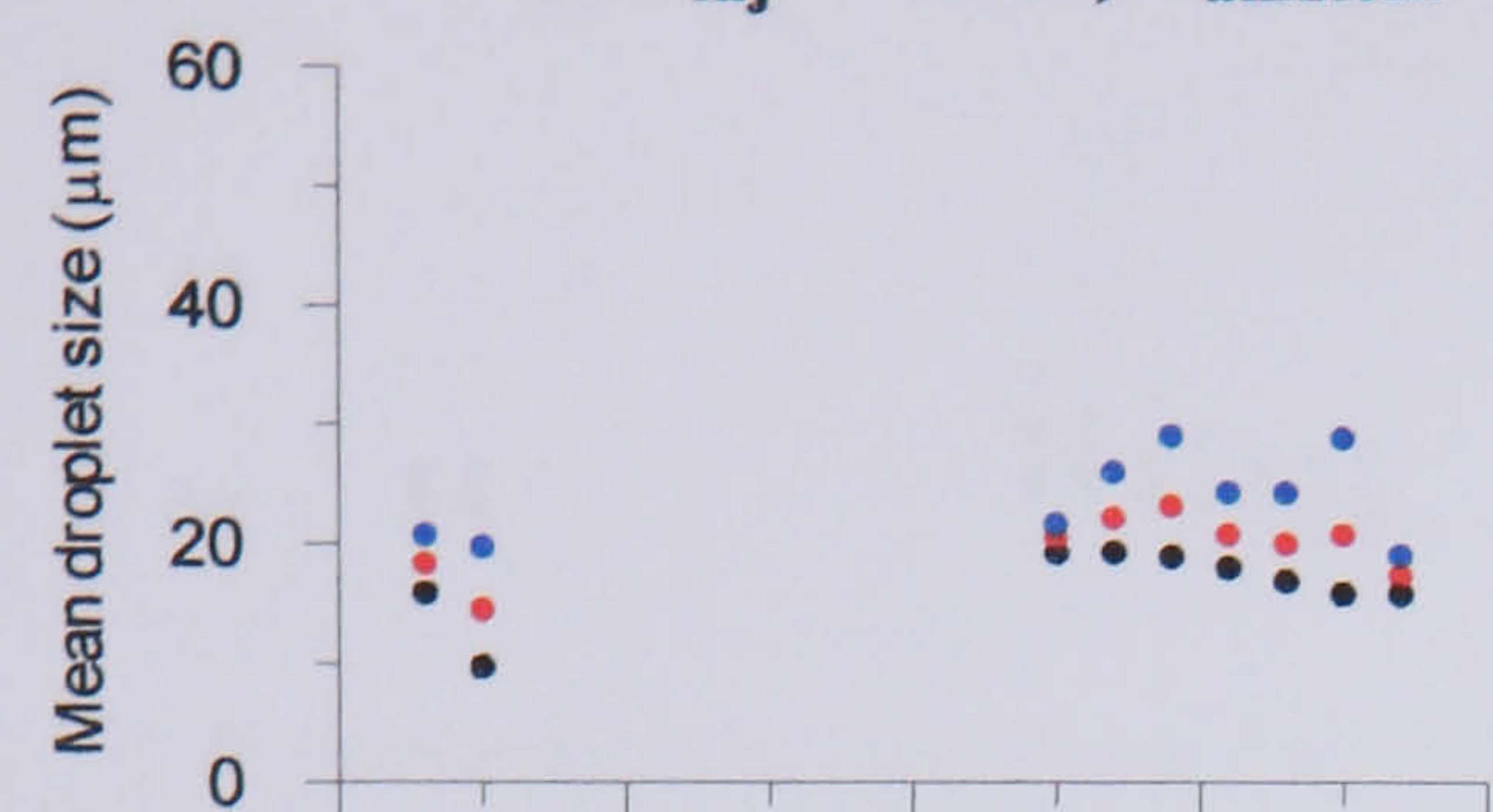
Graph 6

• D10
• D30
• D32

• Axial
• RMS

Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=4.5\text{bar}$, $T_{ambient}=500\text{K}$ and $X=10\text{mm}$

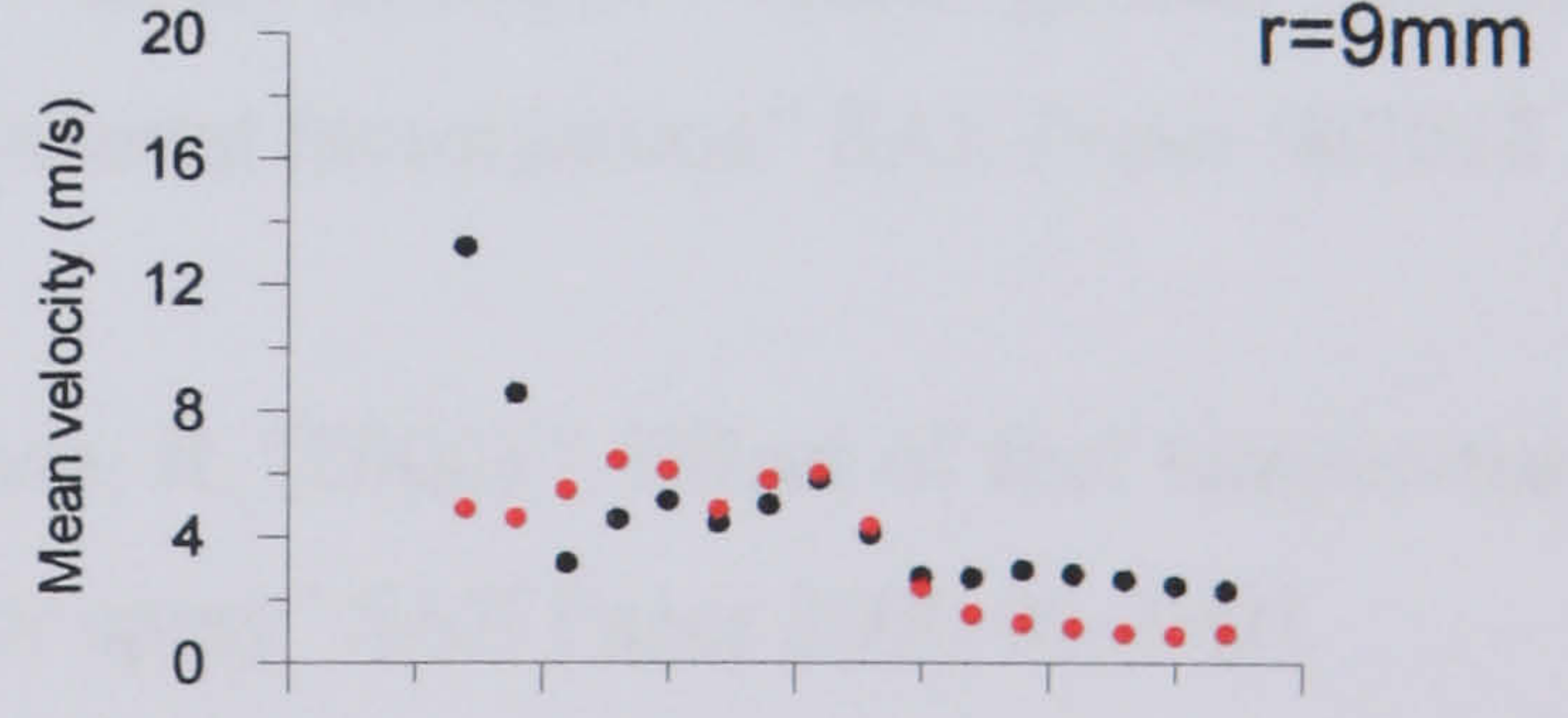
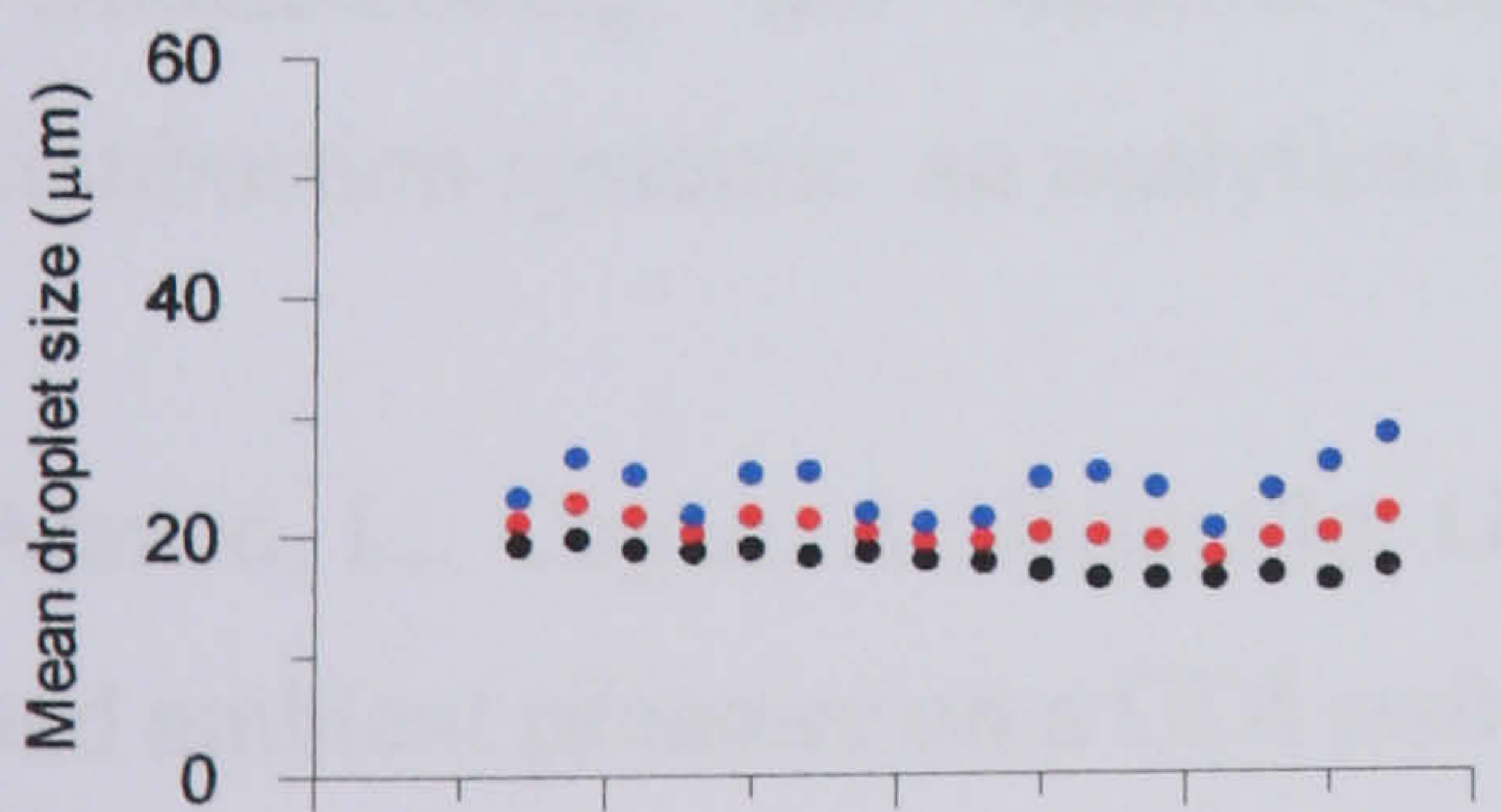
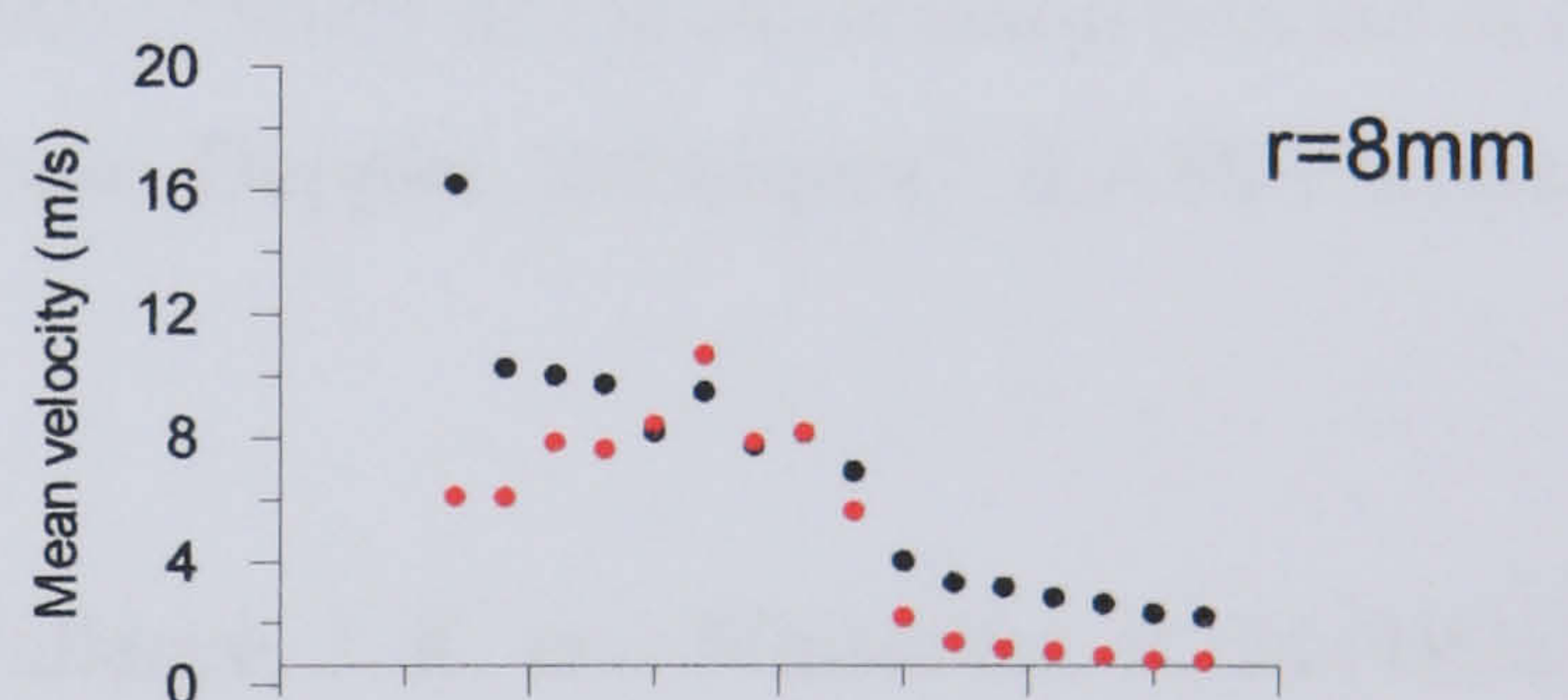
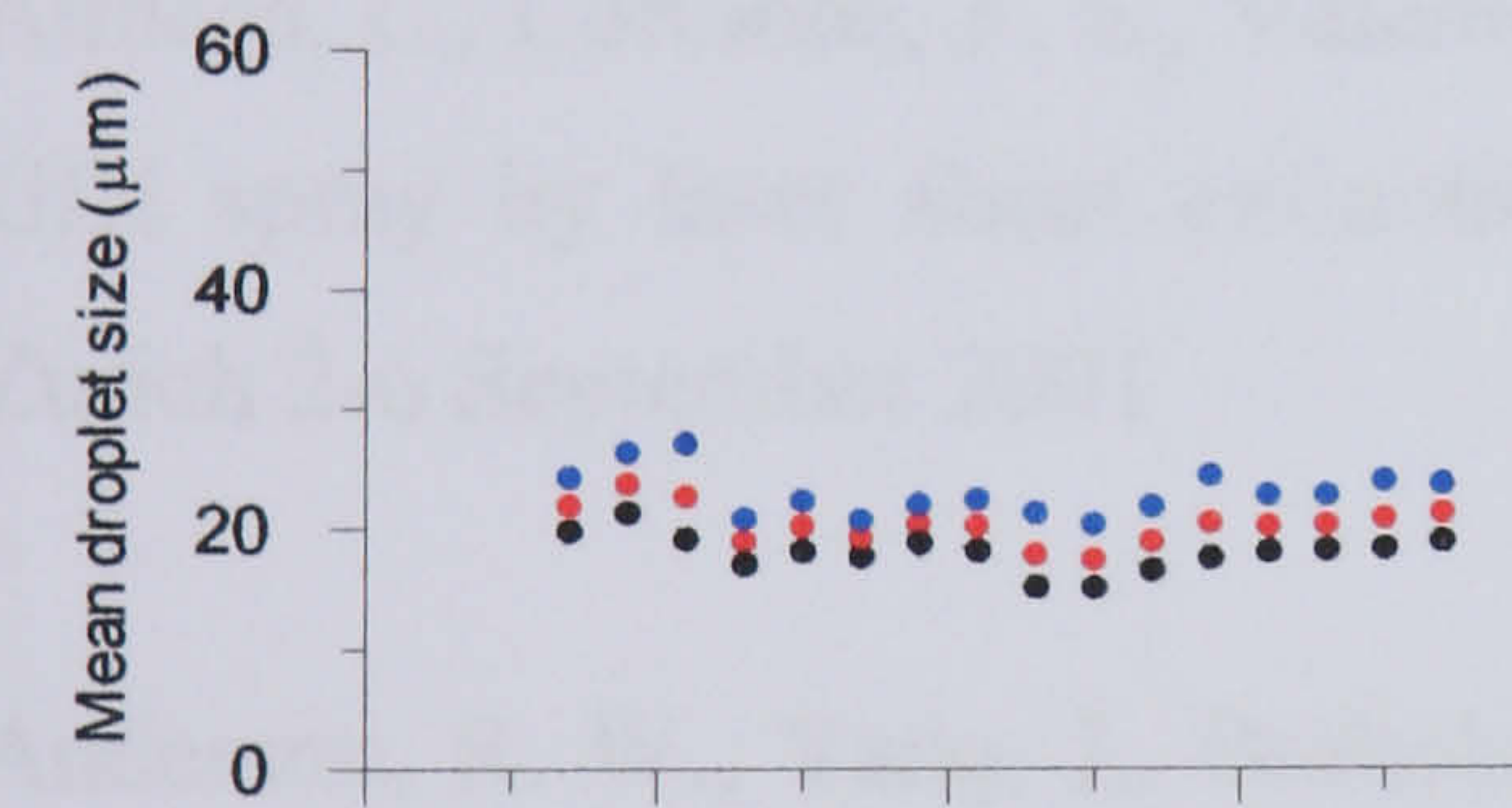
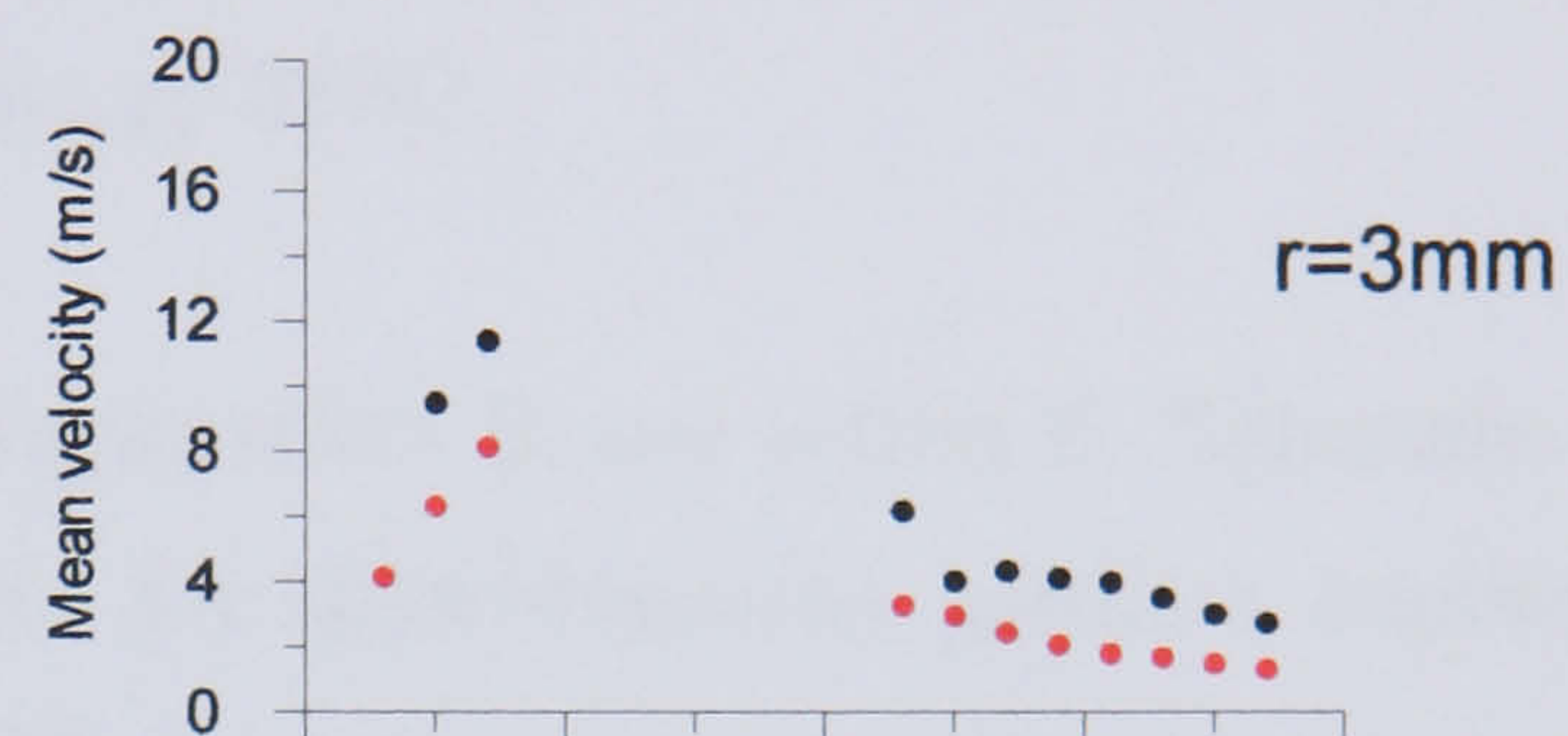
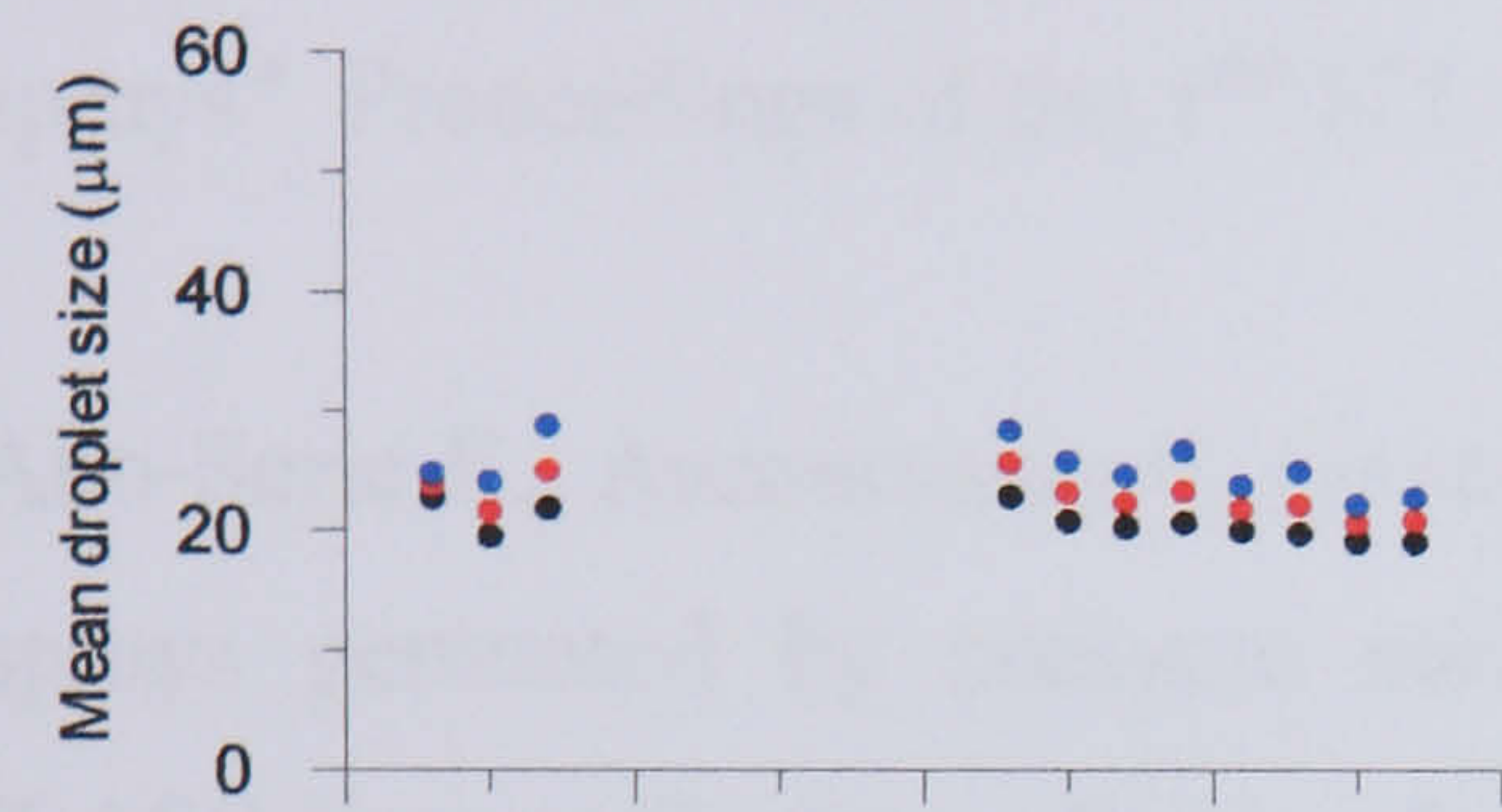
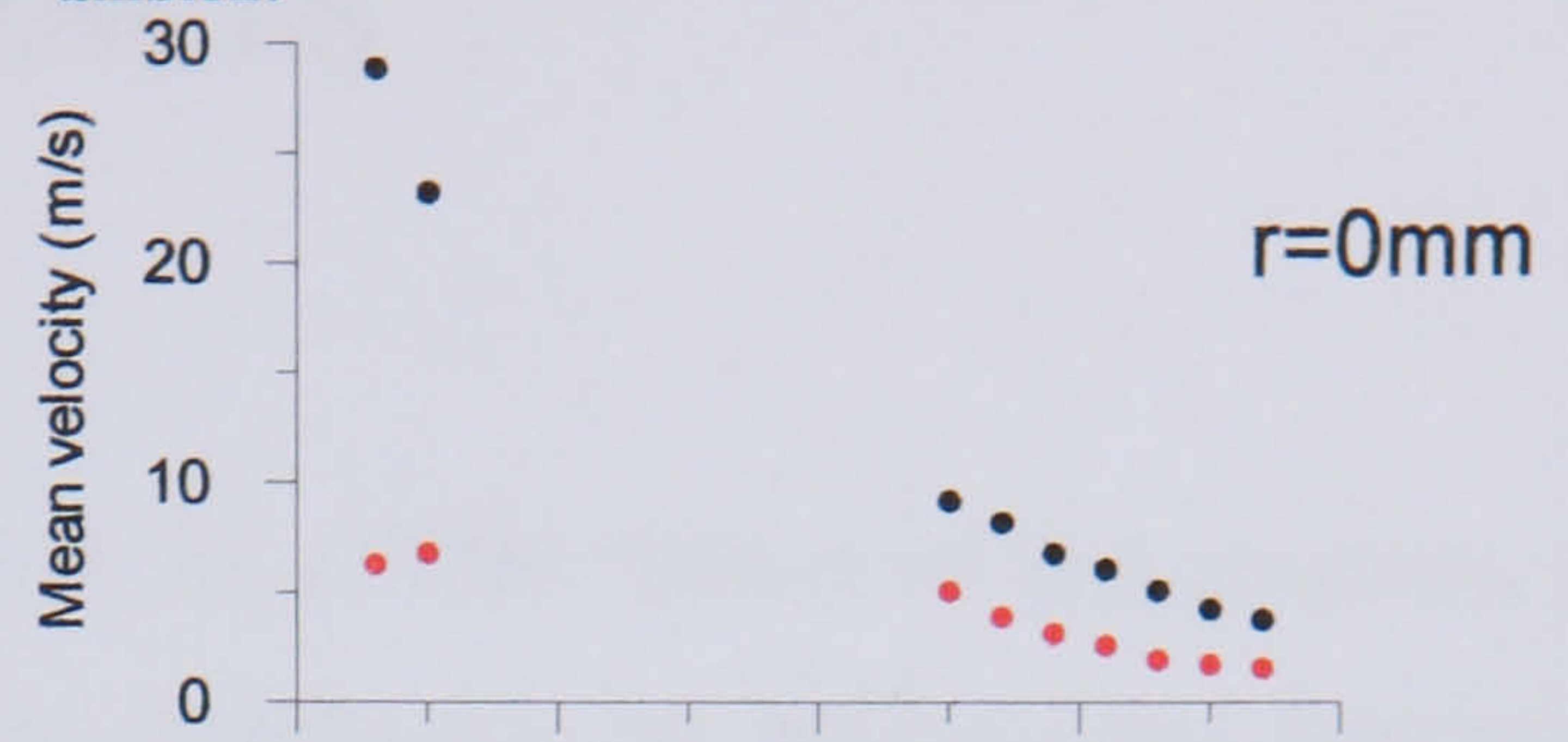
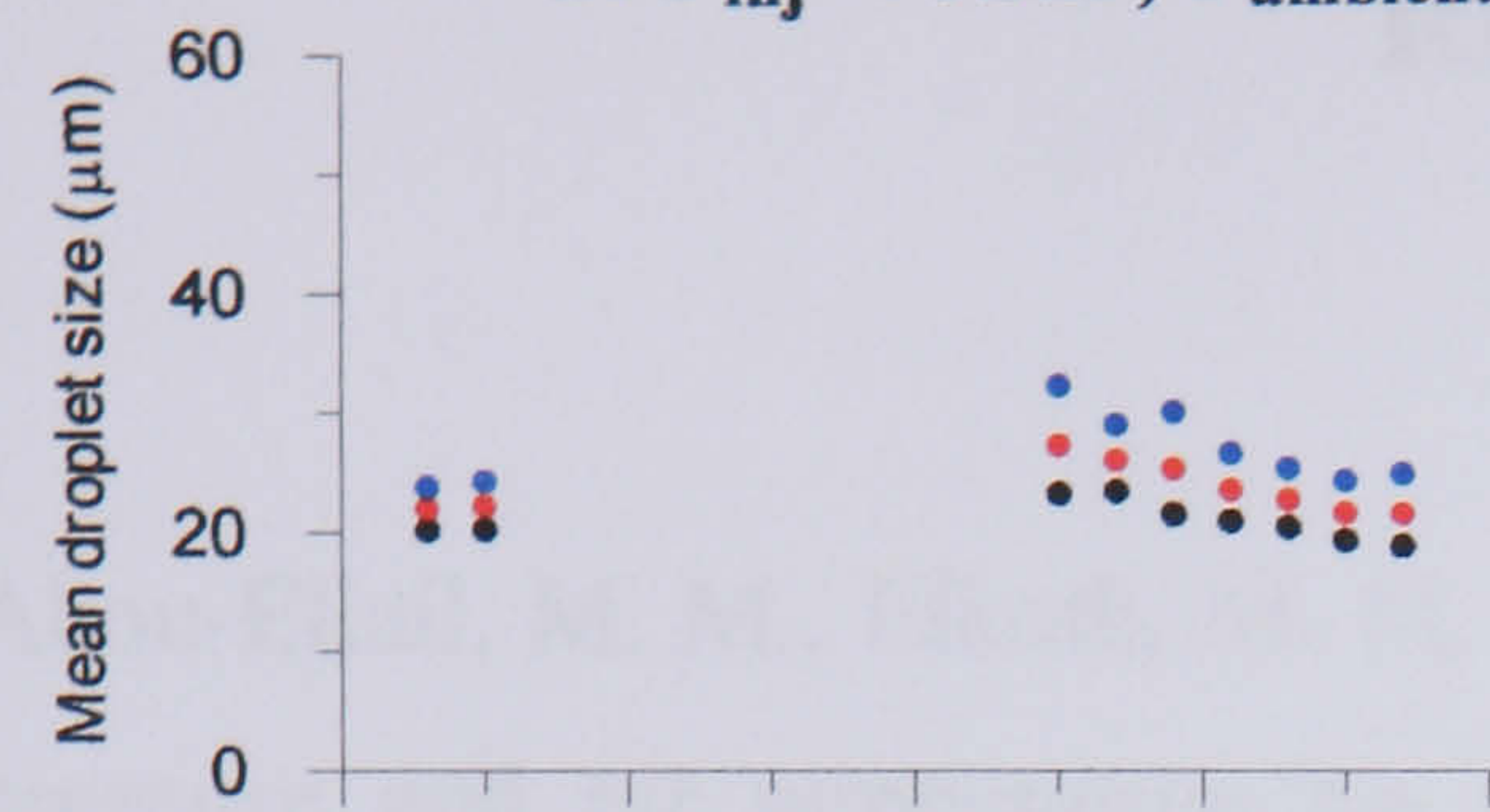


• D10
• D30
• D32

• Mean
• RMS

Temporal variation of mean droplet diameters and velocities

at $P_{inj}=70\text{bar}$, $P_{ambient}=4.5\text{bar}$, $T_{ambient}=500\text{K}$ and $X=15\text{mm}$



Time (ms)

- D32
- D30
- D32

Time (ms)

Mean
RMS

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