IMAGING OF CAVITATION, HOLLOW JETS AND JET BRANCHING AT LOW LIFT IN A REAL SIZE VCO NOZZLE

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Abstract
By grinding a standard VCO single hole nozzle for a large Diesel engine and the subsequent gluing of a transparent glass nozzle (4mm x 4mm with a hole of 1 mm in length and 0.2 mm diameter) on the injector it was possible to observe the flow within and outside the nozzle hole for a real size injector geometry in the same way Arcoumanis et al. did for the flow [1] for a submerged injector. The present results are obtained for the injection into the atmosphere and a pre-set lift of 50 μm. However, the injection pressure range was chosen to give a range of cavitation numbers that encompass the operating conditions of the injector in an engine. There are marked effects of the injector internal flow on the development of the jet outside of the nozzle ranging from a laminar jet, jet branching, a hollow jet and jet break-up with increasing discharge. The internal flow shows in the same order cavitation onset, string cavitation with partial hydraulic flip and fully developed cavitation that exits the nozzle.

Introduction

It is now an established fact that cavitation in the holes of Diesel injection nozzles, plays an important role in the process of atomization of the fuel in the engine. However, the precise mechanism by which it acts upon the jet is not yet completely clear. The conjecture is tempting that cavitation pockets or bubbles, that exit the nozzle and collapse, break-up the jet. However, it is also well known that the spray angle and the size of atomized droplets is mainly a function of the surrounding gas density and jet velocity and not of the exit pressure, [2]. The variation of the exit pressure implies a variation of cavitation number. Therefore, the most probable explanation is that cavitation serves as a trigger for large surface disturbances of the jet which are then amplified by and or lead directly to an aerodynamic interaction with the surrounding gas. Cavitation free nozzles also atomize but not as effectively. Therefore, cavitation is only a part of the puzzle. The typical hydrodynamic Reynolds numbers of the nozzle flow for atomizing jets correspond to nominally turbulent flow. Nominally because the nozzles are too short for turbulence to develop fully. For a nozzle length to diameter ratio of 5 one would need a Reynolds number of about 100.000 to obtain a fully developed turbulent profile, see Prandtl [3]. This value is higher than that of injection nozzles. On one hand fully flipped (cavitation free) nozzle flow (nozzle length tending to zero) produces smooth jets, Bergwerk [4]. The flow in an atomizing nozzle of finite length is therefore unstable, but needs a minimal length for transition to turbulence. On the other hand nozzles with a large length to diameter ratio do not atomize as well. In this case cavitation appears at the nozzle but collapses within the nozzle for the same cavitation number for which in a shorter nozzle cavitation would reach the nozzle exit, [5]. It appears that there is a subtle interplay between cavitation and turbulence in the nozzle but also a minimal disturbance amplitude of the jet needed for aerodynamic forces to produce prompt atomization. An increase of the curvature of the inlet corner is evident within this framework. It both reduces the extent of cavitation within the nozzle and can even suppress it as well as it reduces secondary flows which lead to a faster transition to turbulence. The aim of this work is to collect evidence that may help in clearing up the complex phenomenon of atomization.

Experimental Set-up

The geometry of the injector tip is sketched in Figure 1. It is a standard VCO one hole injector with a large hole of 0.55 mm diameter for a large engine. The tip was ground down to leave 150 μm of steel between the ground surface and the needle seat surface. This was necessary to avoid changing the geometry of the seat. As a result there is a small volume of 0.55 mm diameter and 150 μm in length at the transition between the flow at the seat and the inlet of the sharp inlet of the transparent glass nozzle. A similar small volume is nowadays being used to equalize the sprays from multi-hole injectors but it is placed as a depression of rotational symmetry on the needle itself. These measures influence in both cases the inflow conditions to the nozzle hole.

A transparent glass nozzle of 4mm x 4mm and 1 mm thickness with a hole of 0.2 mm diameter was glued onto the ground surface. The index of refraction of the glass for the nozzle matches that of Diesel fuel. Together with the geometry of the nozzle this allows an unhindered observation of the flow in the hole from two different directions that are perpendicular to each other.

The observation of the flow was performed with a CCD camera and back lighting from a high efficiency LED driven at 10 amperes for a duration of 200 ns. The light pulse is short enough to freeze the motion and sufficiently bright to obtain a good contrast of the images. Due to the small depth of field of the imaging long distance microscope lens and due to the difference in optical path between for the light going through the transparent nozzle and the light traversing the spray region (air) it is normally not possible to obtain a sharp image of both the flow in the hole and of the spray simultaneously. However, a compensation was achieved by placing a glass wafer between the lens and the spray. This wafer has a thickness of 2 mm. It was adjusted by introducing a hair into the nozzle and carefully shifting the wafer until the optical disturbances at the edge of the nozzle on the images were minimal and a sharp picture of the hair within and outside of the nozzle was obtained. The injector was then rotated about its axis until the image of the hair was continuous at the edge of the nozzle.

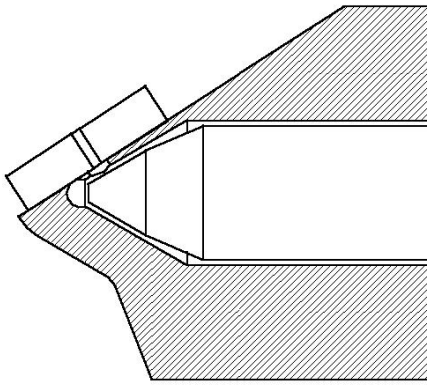


Figure 1. Cross-section through the injector tip with glued transparent nozzle

Most of the experiments were performed with a fixed needle at a pre-set lift of 50μm. The injection was initiated by a solenoid valve upstream of the injector. A fuel reservoir was pressurized by nitrogen from a bottle. The opening time of the solenoid valve determined the duration of the injection. This was monitored by a pressure transducer between the injector and the solenoid. Picture acquisition was started after the opening transients of the system had decayed and it was clear that any air which might have entered the injection system had been ejected. The total duration of the injections was around 5 seconds. Picture acquisition started 2 seconds after energizing of the solenoid valve. 50 pictures were taken at a framing rate of 25 fps.. During this period the pressure showed no fluctuations and decayed slowly by at most 3%.

Results

Figure 2 shows images of the flow in the hole and of the jet exiting the nozzle for increasing pressure difference. On these images the effect of increasing injection pressure and, hence, of increasing velocity on the cavitation phenomena can be seen as well as the effect of the low lift (50 μm). The latter causes the generation of a vortex within the nozzle hole due to the asymmetric inflow conditions at the hole inlet. At low velocities the jet is smooth but slightly disturbed. The increase of disturbances of the jet with velocity is obvious. Initially when the cavitation number is high enough this vortex (low pressure region at its core) is visible as a cavitation string, see also [1]. But as it emerges from the nozzle it disrupts the jet causing a hollow cone jet as well as jet branching in most cases. The flow flips in the sense that air can now enter the core of the vortex when it comes out of the nozzle, best seen on Figure 2h. In this case the flow is no longer cavitating, because the external pressure can now propagate upstream in the vortex core. With further increasing pressure difference the vortex core becomes more and more wavy until it actually breaks up, Figure 2i. This flow is no longer flipped but fully cavitating because there is no longer a connection between the outside and the inlet of the nozzle. On Figure 2j the surface of the cavitation pocket in the hole is strongly disturbed and rough. This is an indication that the flow has become turbulent. The break-up of the pocket can also be seen on this picture close to the exit of the hole.

At moderate pressure differences the disturbances that can be seen are mainly on the thin sheets between the branches of the jet. The waves grow in amplitude with increasing distance from the nozzle exit. However, when cavitation reaches the exit the break-up starts at the nozzle exit and all of the fluid is involved. Something like Medusas head composed of twisting ligaments is present already close to the nozzle. This would of course not be stable in a high gas density environment. Aerodynamic forces would break the ligaments into drops before reaching this state. In the present case aerodynamic forces amplify the waves in the moderate manner we see for lower pressure differences. Their effect is not abrupt, so that what we observe in Figure 2j is mostly the effect of increasing internal flow fluctuations as well as the reaction of the jet to these strong disturbances.

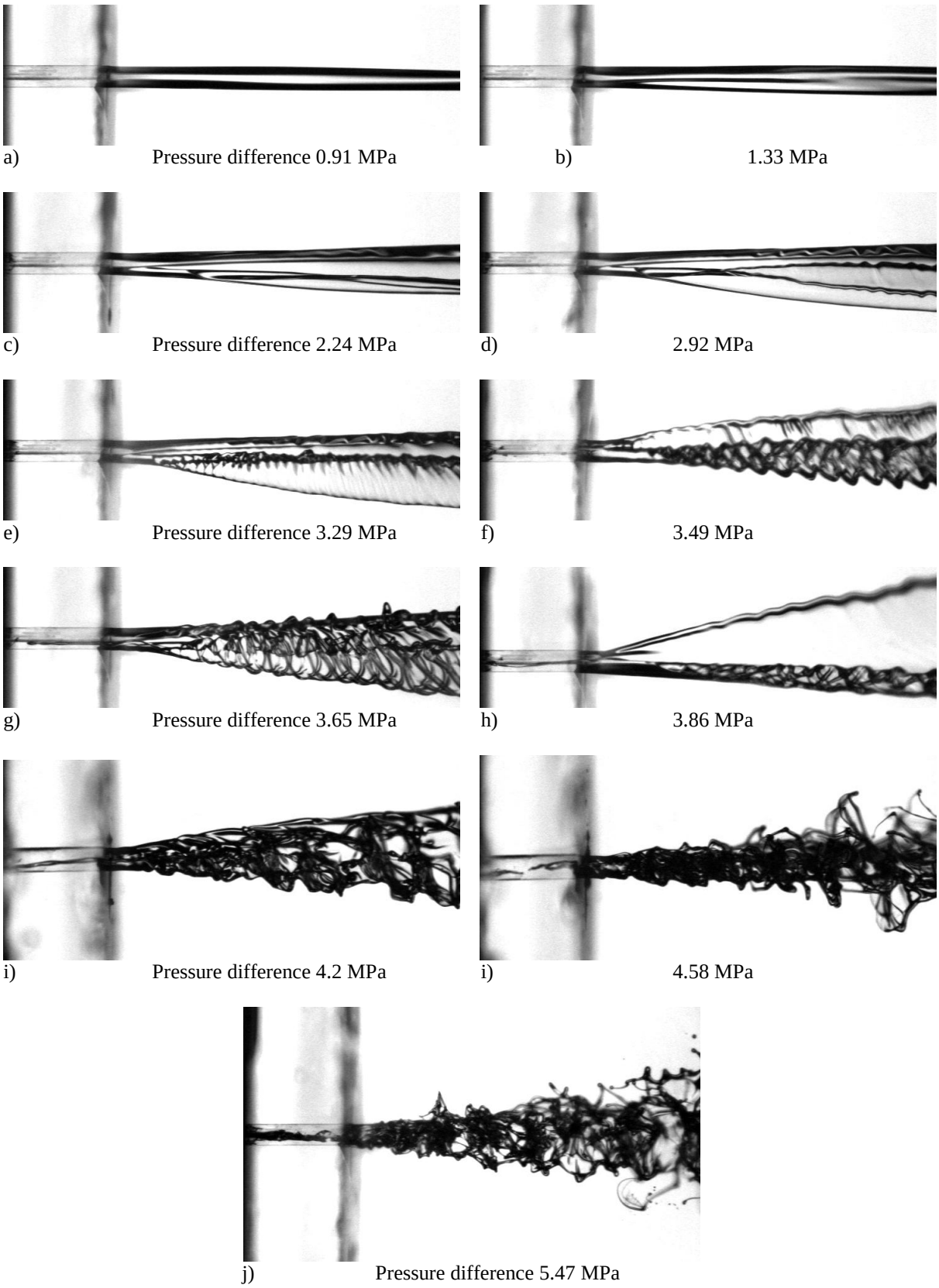
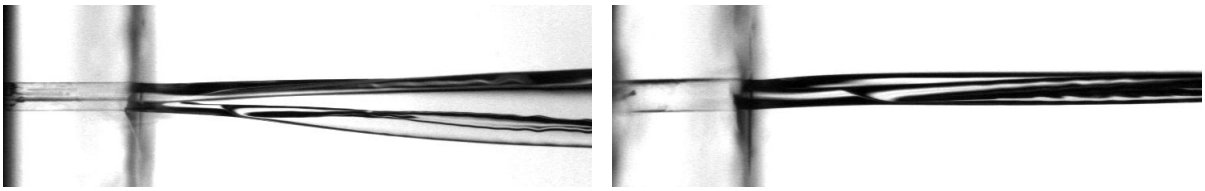


Figure 2. Pictures of the nozzle and jet flow for increasing pressure difference.

The effect of branching is documented in Figure 3, which shows two pictures taken from two different view directions perpendicular to each other. This branching is observed here for steady state conditions. Lin et al. [4] observed jet branching only for an oscillating nozzle.



view at right angle

Figure 3. Jet branching: view of the flow from two different directions, pressure difference 2.8 MPa

In order to quantify the observed phenomena the discharge of the nozzle was measured for a constant pressure difference of 5 MPa and varying pre-set lift, Figure 4 (left) and for increasing pressure difference at a constant lift of 50 μm, Figure 4 (right).

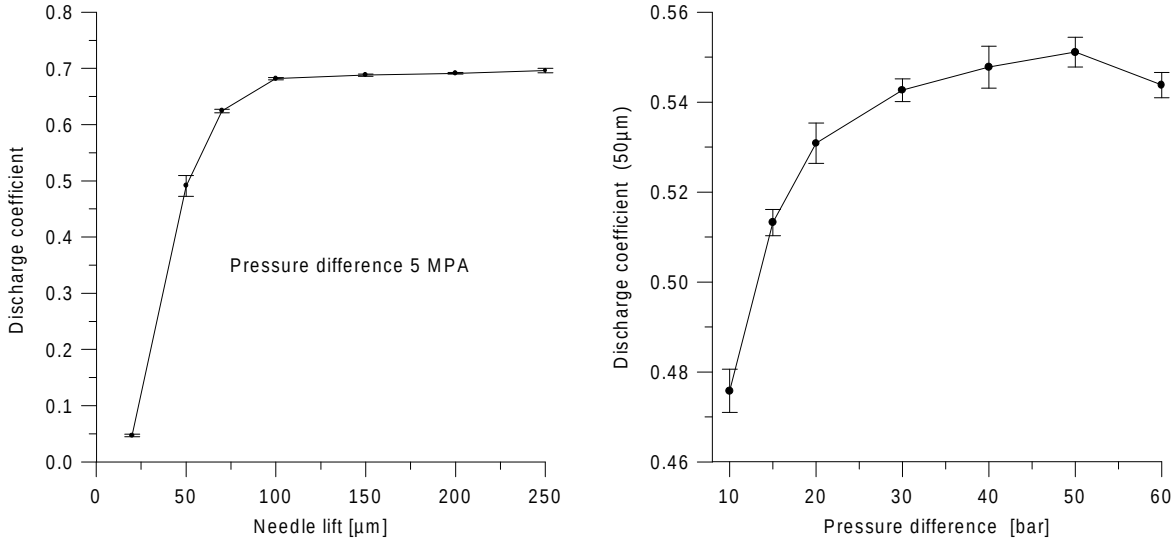


Figure 4. Discharge coefficient vs. needle lift, left and vs pressure difference for 50 μm needle lift, right

The discharge measurements are difficult to perform because the needle is permanently open. This explains the discrepancy in values at 5 MPa. Also the steep increase of the discharge coefficient with needle lift makes this measurement sensitive to tolerances of the lift at low values. The decrease of the discharge coefficient for a pressure difference of 6 MPa correlates well with the pictures of the flow, where one observes that a larger part of the nozzle exit area is occupied by cavitation. These measurements allow to calculate a Reynolds number for the flow. However, only a mean exit velocity can be derived from the discharge measurements, when the volumetric discharge is divided by the geometric nozzle area. In reality the velocity is higher and the values of the Reynolds number have to be used cautiously. This is also valid for the cavitation number because the low lift value causes pressure losses at the seat of the needle and also the small pre-chamber upstream of the hole inlet modifies the conditions at which cavitation appears, shifting it to higher values in comparison to a case without a pre-chamber, e.g. Arcoumanis et al. [1].

Figure 5 shows the branching or jet angle as a function of the cavitation number, defined here as the ratio of pressure difference to external pressure and the hydrodynamic Reynolds number. The tendencies are very similar. The basic problem of the present experimental set-up is that the cavitation number and the Reynolds number can not be varied independently. The value at which the flow becomes unstable in the pictures appears somewhat low. However, in reality the velocity is higher which brings it closer to the classical value for transition to turbulence in pipe flow. Furthermore the classical value for transition does not necessarily have to be valid for this type of two-phase flow.

A plot of the branching or jet angle vs. the discharge coefficient is shown in Figure 6. In this case the plot folds back on itself. The decrease in discharge coefficient for the highest pressure differences corresponds also with a decrease of the jet angle.

Figure 7 is a plot of the discharge coefficient vs. cavitation number that corresponds to the pictures of Figure 2.

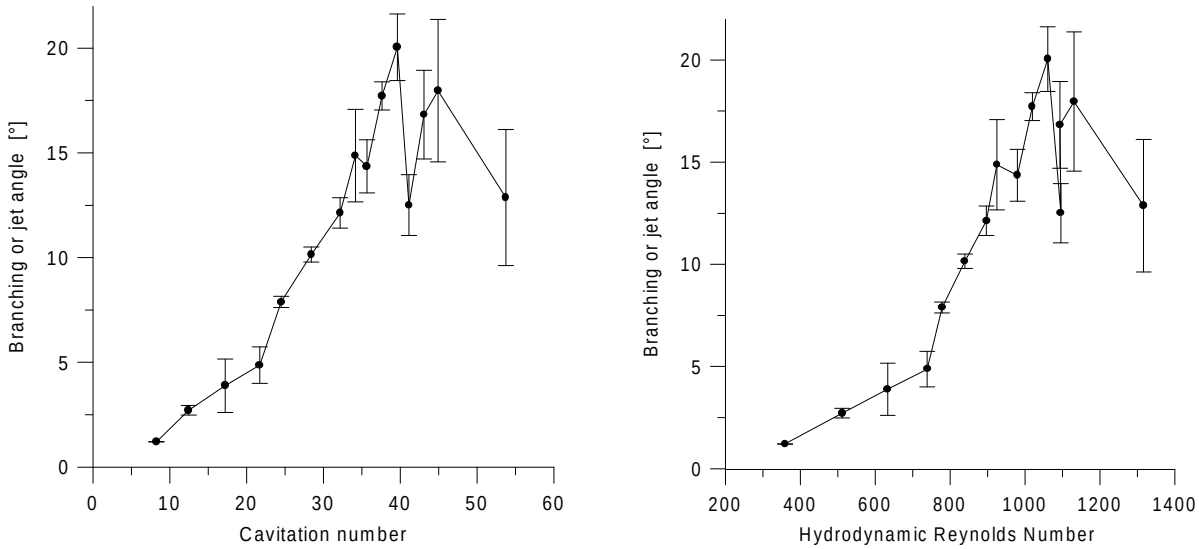


Figure 5. Branching or jet angle vs. cavitation number and hydrodynamic Reynolds number

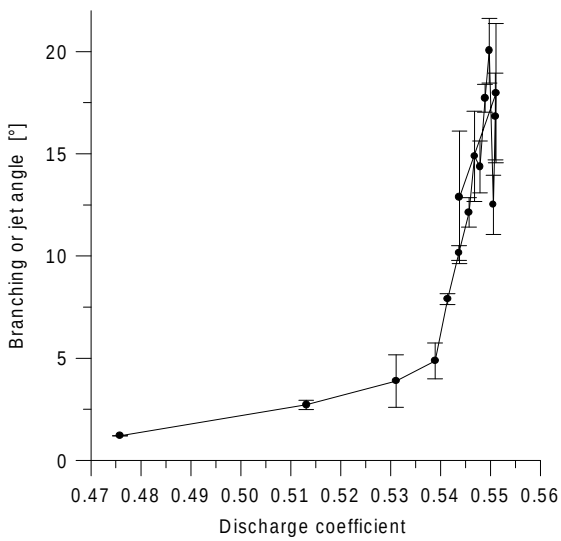


Figure 6. Jet angle vs. discharge coefficient

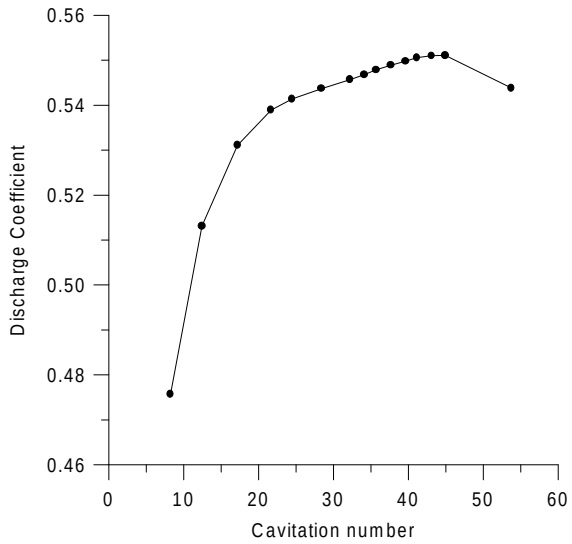


Figure 7. Discharge coefficient vs cavitation number

Conclusions

Large amplitude secondary flows and transverse fluctuations of the nozzle internal flow are responsible for the different phenomena observed in the jet, like jet branching and the onset of atomization. The transition to turbulence appears to be promoted by cavitation in the nozzle.

References

[1] Arcoumanis, C.; Badami, M.; Flora, H. and Gavaises, M., Cavitation in Real-Size Multi-Hole Diesel Injector Nozzles, SAE Paper 2000-01-1249, 2000.
[2] Reitz, R. D.; Bracco, F. V.; Mechanism of Atomization of a Liquid Jet, *Phys. Fluids*, 25, 1730-1742, 1982.
[3] Prandtl, L., Strömungslehre, Vieweg, 1965, p. 225.
[4] Bergwerk, W.; Flow Patterns in Diesel Nozzle Spray Holes, *Proc. Instn. Mech. Engrs.*, 173, 655-660, 1959.
[5] Chaves, H., Knapp, M., Kubitzek, A., Obernmeier, F. and Schneider, T., Experimental Study of Cavitation in the Nozzle hole of Diesel Injectors Using Transparent Nozzles, SAE Paper 950290, 1995
[6] Lin, S. P.; Woods, D. R., A Branching Liquid Jet, *Phys. Fluids*, A 3 (2), 241-244, 1991