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NOZZLE MOMENTUM EFFICIENCY DEFINITION, MEASUREMENT AND IMPORTANCE FOR DIESEL COMBUSTION

Godfrey Greeves*, Gavin Dober*, Simon Tullis*, Nebojsa Milovanovic* and Stefan Zuelch*
*Delphi Diesel Systems, Gillingham, Kent, UK
godfrey.greeves@delphi.com

ABSTRACT
An understanding of fuel injection related effects on conventional diffusion-controlled diesel combustion and soot emissions is reviewed. From this it is concluded that at normal hot engine operating conditions the liquid phase length of the diesel fuel spray /jet is relatively short. The review identifies the importance of the axial momentum input by the nozzle to the fuel spray/jet. This is a key factor determining the rate of air/fuel mixing for control of soot formation in the fuel vapour jet. A new parameter named nozzle momentum efficiency (η_n) is defined. This is essentially a measure of the efficiency of a nozzle in converting the fuel injection pressure into axial momentum in the fuel spray/jet. Measurements of η_n from a newly developed test rig are presented. The η_n values for different spray-hole shapes show an increase for more convergent spray-hole shapes and a correlation with measured HSDI engine exhaust smoke levels. The values of η_n are found to decrease at part needle lift conditions explaining the soot emission benefits seen with the fast opening and closing characteristics of an advanced piezo direct-acting injector. The new test rig is also used to measure an initial jet angle of the emerging fuel spray/jet and the measurements show that this initial jet angle decreases with higher momentum efficiency such as produced with convergent spray-hole shapes. Also the initial jet angle increases at part needle lift conditions. Both η_n and the initial jet angle are factors that affect the spray/jet penetration and rate of air/fuel mixing in the combustion system. The advancement in understanding of these nozzle spray jet characteristics will contribute to future diesel fuel injection nozzle design and manufacture.

INTRODUCTION

The continuing advancement of diesel engine combustion and fuel injection systems is required to meet future more stringent vehicle emissions regulations for particulate and NOx as well as demands for lower fuel consumption and CO2. There have been substantial advances in fuel injection technology providing the capability for much higher fuel injection pressures that together with high levels of EGR have allowed combined reductions in engine-out emissions of soot and NOx [1,2,3]. These advances include new injector concepts such as a two-actuator EUI system for heavy-duty engines [1] and common rail injection systems [3] with fast acting solenoid injectors and the latest directed-acting piezo injector technology for light-duty engines.

The diesel fuel injection nozzle has also been the subject of a continuous design and development improvement process that includes development of more flow-efficient nozzles. This paper is concerned with the diesel fuel injection nozzle and certain characteristics of diesel fuel sprays that are relevant to the control of engine exhaust emissions.

First a review is given of important aspects of diesel fuel sprays in terms of what the engine combustion system and in particular the soot emission responds to. This paper considers conventional diffusion type diesel combustion with fuel injection timings near to top dead centre (TDC) crank angle such as used with high injection pressures and high levels of EGR for low soot and NOx emission. Also the context is light-duty passenger car HSDI engines and heavy-duty truck DI engines that use multi-hole nozzles with a conventional petroleum-based automotive diesel fuel.

From the review it is concluded that the axial momentum in the fuel spray/jet is an import characteristic and this leads to the definition of a new nozzle parameter named the “nozzle momentum efficiency”.

A test rig has been developed to measure the nozzle momentum efficiency to the desired accuracy and initial test results obtained with this test rig are presented.

IMPORTANT ASPECTS OF THE DIESEL FUEL SPRAY/JET FOR CONTROL OF SOOT EMISSION

Soot formation and emission is very sensitive to the fuel injection and combustion system design and operating parameters. It is important to advance the understanding of the key controlling processes. The diesel fuel spray atomisation, fuel evaporation, air/fuel mixing, ignition, combustion, heat release and emission formation are a complex combination of multiphase physical and chemical processes involving hundreds of chemical species.

One approach is to construct detailed 3-D CFD computer models and sub models that aim to simulate this complex sequence of processes. Currently this approach cannot predict engine results to the required level of accuracy for all design and operating variables of interest. In particular the very dense liquid phase part of the diesel fuel spray/jet is not modelled well by the usual CFD atomisation and evaporation models and requires new approaches including the way the mass exchange is done between the Lagrangian liquid phase and the Eulerian gas phase co-ordinate systems [4].

An alternative/parallel approach used here is to start from measured results obtained from different types of experiment and then by a technique somewhat like reverse engineering deduce and model these important processes that control soot formation and emission. In the end the measured engine emission result is what matters to achieve the required future emission regulation. So the approach used here is to identify those key in-cylinder processes that explain the measured changes of the engine–out emissions with various changes of fuel injection and engine design and operating variables. Then construct models of the important processes and devise test techniques to measure certain aspects of these processes. This provides a well-validated combined experimental/theoretical understanding of the means to achieve the next reduction in emission level relative to a given baseline engine technology and system.

The following is based on an understanding of in-house experimental data from optical engine studies and from the responses of single-cylinder and multi-cylinder engines to a large number of fuel injection and engine design and operating variables as well as computer modelling of diesel fuel sprays and combustion.

Air/Fuel Mixing – Liquid phase

Figure 1 shows an evaporating fuel spray/jet image taken in an optical engine similar to that of [5] using a nitrogen charge to suppress combustion to give a clear view of the details of air/fuel mixing. This illustrates the nature of the air/fuel mixing process for an evaporating diesel fuel spray/jet in hot end of compression conditions for a swirling charge as in the piston bowl of a HSDI engine. The coloured lines and marks in Figure 1 are an over-plot of calculated results from an in house phenomenological computer model of the fuel spray/jet called SPRY2 [6] as shown in Figure 2 and will be referred to later.

The dark region of the image of the fuel jet in Figure 1 is the only place where liquid phase fuel exists. This region is too optically dense for optical techniques to penetrate and is likely to be a mixture of fuel ligaments, droplets, fuel vapour and entrained gas. The end of this dark image represents the end of the liquid phase of the evaporating diesel fuel spray/jet. Measurements of this maximum liquid phase length of the fuel

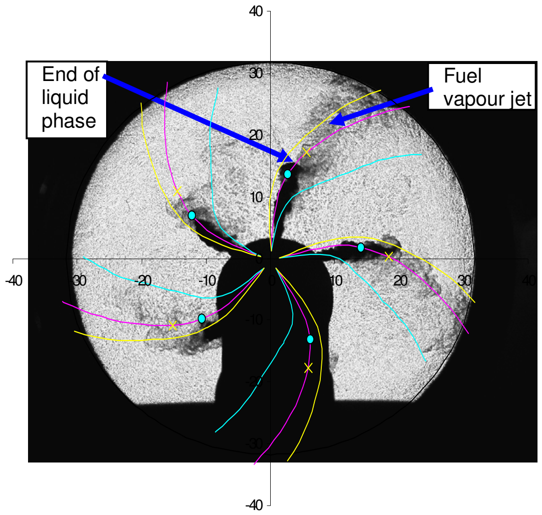

Figure 1 Shadowgraph/Schlieren image of evaporating diesel fuel spray/jet taken in optical engine with superimposed SPRY2 spray model calculations, 1000 bar rail pressure, 78 bar cylinder pressure

spray/jet show that it reaches a maximum quasi-steady length after the maximum lift of the nozzle needle is reached [5]. The SPRY2 model of the fuel spray/jet (Figure 2) applied to the hot end of compression swirling charge conditions in the optical engine gives a good simulation of these liquid length results and trends in the optical engine as in Figure 1.

Also in an independent study liquid lengths have been measured in a heated high pressure spray chamber for a wide range of conditions [7]. In this case a scaling law model [8] has been developed for the hot quiescent charge conditions in the spray chamber and the results obtained are in agreement with the experimental results and trends in the spray chamber for quiescent charge conditions.

Both of these liquid length measurements in an optical engine and in a hot high-pressure spray chamber and the corresponding modelling results show that the maximum liquid length is a function of the spray hole diameter, gas density, gas temperature and the thermal and vapour pressure properties of the fuel [5,8]. The observed maximum liquid length is correctly modelled by an assumption of local thermal equilibrium (liquid, vapour and entrained charge) and the local concentration of fuel (liquid + vapour). There is no evidence of a dependence on droplet size since neither of these simulations requires an input of droplet size to predict the experimental liquid lengths.

The initial angle of the liquid fuel jet as it emerges from the spray hole is however important and this is an input to the SPRY2 model of the fuel spray/jet [6]. This initial jet angle can change with nozzle design parameters and affects the initial rate of air entrainment into the liquid phase of the fuel jet.

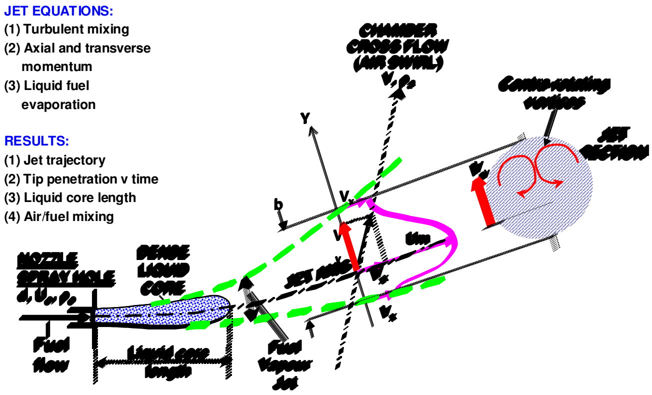

Figure 2 Phenomenological model of fuel spray/jet (SPRY2)

An important conclusion from the optical engine experiments and the validated phenomenological model [5,6] is that the maximum liquid lengths are much shorter than the distance to the piston bowl wall for the spray hole diameters and conditions that exist in automotive diesel DI engines of interest (LD, MD, HD). This conclusion can also be made from the high-pressure chamber measurements and scaling law model [7,8] when a spray hole size is used as for today's modern diesel DI engines. Also in both these optical engine and hot high-pressure spray chamber studies it was found that no fuel droplets escaped beyond the dense liquid phase region referred to above.

This does not mean that the atomisation characteristic of the diesel fuel spray is unimportant. Instead it can be said that at normal hot diesel engine conditions with near TDC injection and standard automotive diesel fuel the atomisation appears to be more than adequate, liquid lengths are short and other factors become controlling.

Also from observations with diesel ignition and combustion in an optical engine with a single well-defined fuel spray/jet

[5] it was concluded that it is too cold for soot to form in the liquid phase of the jet. In particular soot did not form in the fuel jet until well after the end of the maximum liquid length.

well-validated representation of the airborne part of the diesel fuel spray/jet as observed in the optical engine.

Figure 3 shows the SPRY2 model calculations for a HSDI

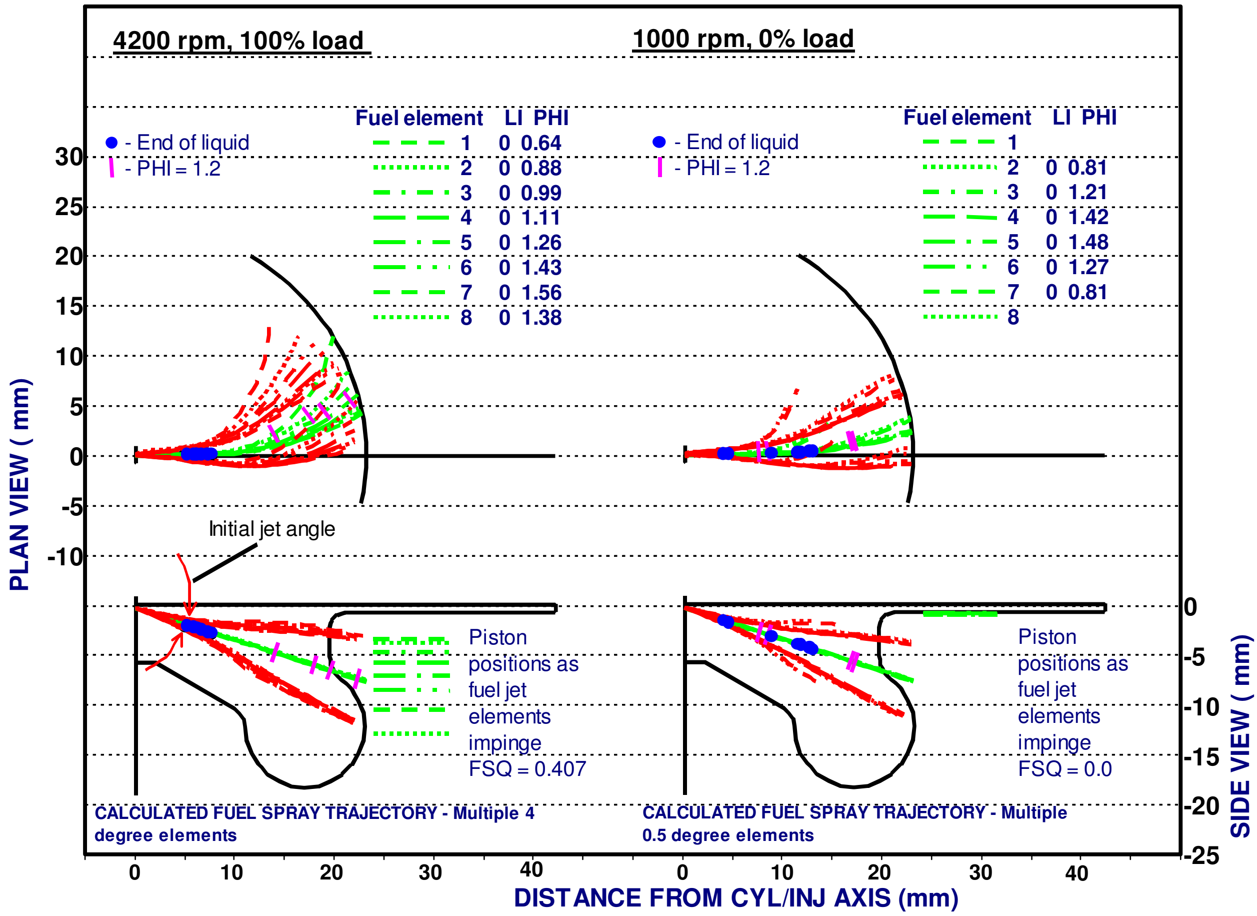

Figure 3 Spray/jet model calculations for an HSDI engine, full load rated speed condition on LHS, engine idle condition on RHS, blue dots are the calculated maximum liquid phase lengths for each of the injected fuel elements

Air/fuel mixing – Vapour phase fuel jet and SPRY2 model

In Figure 1 the mottled image is the fuel vapour jet. This mottling is caused by the variations in fuel vapour concentration differences between adjacent turbulent eddies which cause refractive index changes and light steering in the Schlieren/shadowgraph image.

The SPRY2 model in Figure 2 has been developed in–house over a period of years. It calculates the liquid and vapour phases of the fuel jet, the jet trajectory and air/fuel mixing in a hot swirling air charge. In Figure 1 the lines/marks calculated by the SPRY2 model are the upstream edge (yellow line), the jet centreline (red), downstream edge (blue) and end of the liquid phase (blue circle). These lines/marks show a good general agreement with the observed image taken from the optical engine. This comparison image is for one particular set of conditions. A similar level of agreement between the calculation model and the optical images is achieved over a wide range of conditions including nozzles with different spray hole diameters and different injection pressures, injection timings, charge densities, charge temperatures, levels of air swirl and fuel types with widely different fuel vapour pressure characteristics. This is achieved with no change of the model constants and means that the SPRY2 model gives a

engine. The two diagrams on the left-hand side (LHS) are for a full-load rated speed engine condition (4200 rpm). The upper diagram is a plan view of one sector of the piston showing the trajectory of one of the fuel sprays. The lower diagram is a corresponding side elevation view. The blue solid dots represent the maximum liquid phase length for each of the injected fuel elements and these liquid lengths are much shorter than the distance to the piston bowl wall. The two diagrams on the right-hand side (RHS) show a similar calculation for a hot engine idling condition that is an opposite extreme of the engine speed/load range. At this condition the maximum liquid lengths are longer because of the lower end of compression temperature and lower charge density (low boost pressure), but the liquid lengths are still much shorter than the distance to the piston bowl wall.

Air/fuel mixing – Rate of vapour phase air/fuel mixing and soot formation

An earlier phenomenological fuel jet-based computer model of diesel combustion and a semi-empirical sub model of soot formation showed that it correctly calculated the main trends of the measured exhaust soot emission from engines over a range of engine and fuel injection variables [9]. This calculation model provided a useful concept of soot formation in diesel combustion and in explaining the observed soot emissions from engines. In particular high molecular weight gaseous soot precursors form rapidly in the gas phase fuel-rich fuel vapour regions of the fuel jet when these are at a sufficiently high temperature [10]. As an injected fuel parcel travels along the fuel jet centreline more of the hot compressed air charge is entrained into the jet. This added entrained charge causes an increase of the mixture temperatures promoting soot formation but eventually further along the jet there is sufficient added oxygen to prevent soot formation through oxidation of the gaseous soot precursors. This means that there is a certain range of fuel-rich air/fuel ratios (e.g. ϕ 1.4 – 1.2) along the fuel jet where soot forms most rapidly.

Consequently the amount of soot formed from an injected fuel parcel is very dependent on the residence time of when it is in a fuel-rich high-temperature condition and hence on its rate of air/fuel mixing as it passes through the combustion system. This explains the very strong effect that an increase of fuel injection pressure has on reducing soot emission since it increases the axial fuel jet velocity and reduces the residence time of the gas phase fuel-rich mixtures [1].

The SPRY2 model has been used to calculate the rate of air/fuel mixing corresponding to the measured engine exhaust soot emissions as shown in Figure 4 [1]. The x-axis values are derived from the rate of change of local air/fuel ratio as each injected fuel parcel passes through a fuel-rich air/fuel ratio of 9 (RAF9) in the fuel jet. This calculated rate of air/fuel mixing shows a good correlation with the measured exhaust soot for a range of different levels of injection pressure and nozzles with different spray hole diameters.

Consequently an understanding of the basis of the SPRY2 model and its input/result helps to define the important features for fast fuel/air mixing and low soot emission. In particular it is the fuel jet entrainment process that drives the rate of air/fuel mixing. The axial fuel jet momentum that is input by the nozzle spray hole is important because this drives the turbulent fuel jet entrainment process.

For the SPRY2 model calculations, the fuel mass flow and the axial momentum input from the spray hole are derived

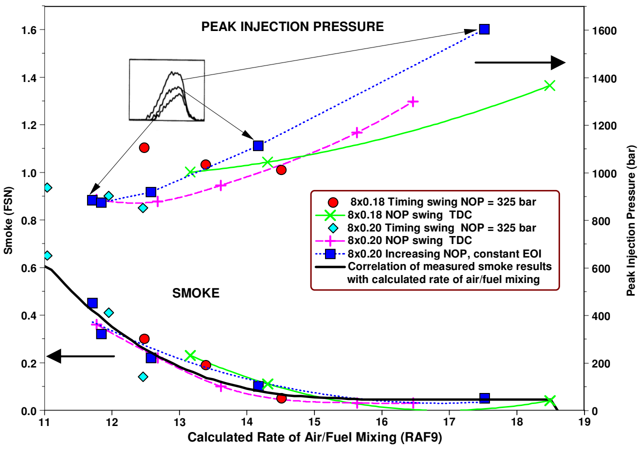

Figure 4 Correlation of HD engine (with EUI) exhaust smoke with rate of air/fuel mixing calculated by SPRY2 model [1]

from various inputs. These include the nozzle flow measured in a test rig and/or the nozzle discharge coefficient (C_d) and a Bernoulli velocity calculated from the injection pressure difference across the nozzle. With the use of these data certain assumptions are needed to derive the axial momentum input by the nozzle. There is a need to obtain a more direct measure of the axial momentum input by the nozzle to the combustion system to advance understanding and take account of different detailed nozzle design effects.

NOZZLE MOMENTUM EFFICIENCY DEFINITION AND MEASEMENT TECHNIQUE

A comparison of interest here is the effect of different nozzle internal flow details and in particular the effect of different spray hole shapes for the same injection pressure and the same fuel mass flow. It is found that the engine-out soot emission can be significantly reduced with a nozzle having convergent spray holes compared with a nozzle having parallel spray holes. From the review and the SPRY2 model above it is concluded that these smoke differences should be related to the axial momentum input from the spray hole.

Nozzle momentum efficiency definition

An efficient nozzle that minimises soot formation should be one which maximises the axial momentum from the spray hole for a given fuel injection pressure and fuel mass flow. In order to capture this property of a nozzle a new parameter named nozzle momentum efficiency is defined here below.

The nozzle discharge coefficient (C_d) for an individual spray hole can be defined as follows.

C_d = (actual spray-hole mass flow) / (theoretical spray-hole mass flow)

$$C_d = \dot{m} \;/\; \dot{m}_{th} = \dot{m} \;/\; (\rho_f \,.\, A_h \,.\, U_{th}) \qquad (1)$$

$$U_{th} = \sqrt{[\, 2\,(P_1 - P_2) \,/\, \rho_f \,]} \qquad (2)$$

For a momentum efficiency value to be useful in comparing engine results there is a need to hold the fuel mass flow constant at a given fuel injection pressure. Otherwise for a given total fuel quantity delivered to the combustion system the injection period will change which means the smoke comparison is no longer valid because of the other in-cylinder effects that changes of injection period will have on soot formation. Thus both the theoretical momentum flux and the actual momentum flux need to be based on the actual or measured fuel mass flow.

The new definition for the nozzle spray-hole momentum efficiency (η_n) is given here as follows.

η_n = (actual spray-hole momentum flux) / (theoretical spray-hole momentum flux based on the actual fuel mass flow)

$$\eta_n = \dot{M} \;/\; \dot{M}_{tham} \qquad (3)$$

$$\eta_n = \dot{m} \,.\, U_{effM} \;/\; (\dot{m} \,.\, U_{th}) = U_{effM} \,/\, U_{th} \qquad (4)$$

$$\eta_n = \dot{M} \;/\; (\dot{m} \,.\, \sqrt{[\, 2\,(P_1 - P_2) \,/\, \rho_f \,]}\,) \qquad (5)$$

The definition for η_n given by equation (5) compares the actual momentum flux to a theoretical momentum flux based on the actual mass flow $\dot{m}$ for a spray hole.

The effects that lead to a nozzle spray-hole discharge coefficient C_d, as defined by equations (1) and (2), of less than

1.0 include a vena-contracta (this effect restricts the use of the full geometric area of the spray hole) and flow losses that reduce the effective exit velocity such as from secondary flows. Further effects are a non-square velocity profile at the minimum spray-hole area and cavitation in the spray hole if it exists at the minimum area since this reduces the effective fuel density. Also a choked condition means that the mass flow is no longer a direct function of (P_1 - P_2).

Some of these effects will also play a part in reducing the momentum efficiency, as defined in equations (3-5), below 100%. This will mean that nozzle design changes that increase C_d may also tend to improve η_n. Nevertheless by considering theoretical nozzles it can be argued that the C_d value for a nozzle cannot be assumed to directly represent the momentum efficiency of a nozzle. For example, if a square axial velocity profile equal to the Bernoulli velocity existed in a central region of the spray hole with zero velocity outside this region then the momentum efficiency would be 100%, while the discharge coefficient would be less than unity. In general if both C_d and η_n are expressed as ratios then the values of these parameters should be related as follows.

$$C_d < \eta_n <= 1.0 \qquad (6)$$

In essence it can be said that C_d is particularly important for the fuel metering function of the nozzle while η_n is important for the nozzle function that gives efficient conversion of fuel

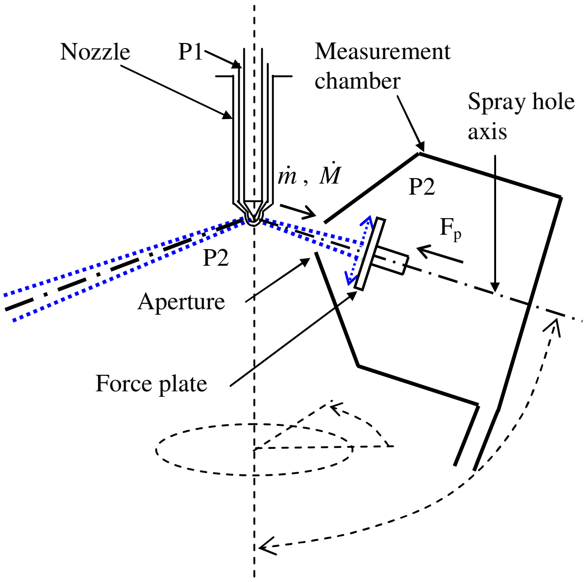

Figure 5 Schematic of new steady-state flow rig to measure spray force, mass flow and momentum efficiency

injection pressure into axial momentum in the fuel jet.

A nozzle design with non-tapered cylindrical spray holes with a sharp spray-hole entry and exit would have a nozzle flow discharge C_d value in the region of ~ 0.7 and a nozzle spray hole momentum efficiency η_n of ~ 0.8. An advanced nozzle design with convergent spray holes would have a C_d value in the region of ~ 0.8 and an η_n value of ~ 0.9 giving about 50% reduction of the engine exhaust soot emission. So from this a relatively small change in the nozzle momentum efficiency can have a large effect on the soot emission.

Consequently for momentum efficiency measurements to be useful there is a need to measure η_n to a target accuracy of 1%.

Measurement of nozzle momentum efficiency

Axial spray momentum can be measured by directing the fuel spray on to a fixed plate that is arranged to measure the impact force on the plate necessary to destroy all of the axial momentum in the fuel spray/jet [11]. In this type of technique the sprays from a dynamically operating injector are aimed perpendicularly towards a plate with a force transducer. The dynamically varying electrical output signal from the force transducer with each injection event can be used to give a spray force versus time diagram for each of the fuel sprays. This type of technique is useful to indicate differences in the dynamic characteristics of different injector designs. For example results in [12] show the benefit that the advanced piezo directing-acting common rail injector gives over the solenoid injector in terms of a much more square spray momentum diagram.

Dynamic spray momentum rigs measure the varying spray force during a single injection event with the needle opening and closing. Because of the relatively small spray forces involved that must be measured in the presence of rig vibrations and unevenness of the turbulent structure in the spray itself, it is difficult to get a high level of absolute accuracy. Also to derive momentum efficiency it is necessary to measure spray force and mass flow rate simultaneously at different stages of the nozzle needle lift. This is very difficult to do with a dynamic injection event with the uncertainty of the needle lift value at any instant in time. Consequently the absolute value of η_n for each spray hole cannot be measured to the required accuracy of the order of 1% as specified above.

A first step towards this objective is to measure at steady-state conditions for preset nozzle needle lifts since it is the efficiency of the nozzle flow paths that are of interest here rather than the dynamic operation of the injector as a whole.

Figure 5 shows a schematic of the new in house developed steady-flow momentum rig that is the subject of a patent application [13]. This spray momentum rig is designed to measure spray force, fuel mass flow, momentum efficiency and initial jet angles.

In this test rig a fuel spray/jet emerges from a nozzle spray hole with a steady upstream fuel pressure P1, a steady downstream ambient pressure P2 and for a fixed position of the lift of the nozzle needle within the nozzle body. The force measurement plate is arranged to be perpendicular to the nozzle spray hole and fuel jet axis. The fuel spray to be measured enters through the centre of the aperture in a measurement chamber cover and impacts normally on the spray target. This target is fitted with a high accuracy steady-state force sensor to measure the force F_p.

By considering a momentum control box it can be shown that the actual axial momentum flux $\dot{M}$, leaving the nozzle spray hole, is equal to the difference in measured force (F_p) on the target plate between with and without the fuel spray/jet. This gives the following:

$$\dot{M} = \dot{m} \, . \, U_{effM} = F_p \qquad (7)$$

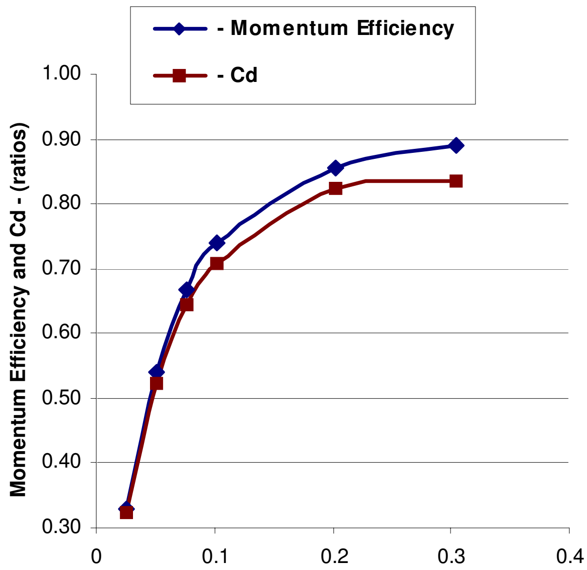

Figure 6 Momentum efficiency and Cd versus needle lift for hole 1 of nozzle 8921-3

F_p is the required opposing axial force on the force measurement plate to bring the axial momentum of the fuel jet flowing along the fuel spray/jet path to zero and U_{effM} is an actual effective velocity relevant to the momentum flux. The momentum efficiency as defined by equation (5) above combined with equation (7) gives:

$$\eta_n = (F_p / \dot{m}) / \sqrt{2(P1-P2)/\rho_f} \qquad (8)$$

The equation (8) can be used to calculate the nozzle spray-hole momentum efficiency from measurements of the following.

F_p – Force measured on a plate to destroy all of the axial spray/jet momentum
$\dot{m}$ – Mass flow rate of fuel measured to emerge from the spray hole
P1 – Upstream fuel pressure
P2 – Downstream pressure for spray hole
ρ_f – The fuel density at entry to the spray hole

It should be noted that this definition of nozzle spray hole momentum efficiency is particularly convenient because, unlike the nozzle discharge coefficient C_d as defined in equations (1-2), it does not require the minimum geometric spray hole area A_h or diameter to be known or measured. The latter can be difficult to do with certain shapes of spray hole, if the spray hole surface is lacquered from running on an engine or at part needle lift conditions.

This new steady-state flow rig (Figure 5) measures the spray force and in addition has a measurement chamber with an aperture. For a multi-hole nozzle this separates and collects the spray that is being measured to give simultaneous spray force and spray mass flow measurements to derive the momentum efficiency as equation (8). Some special features of the test rig give inherent damping characteristics that remove rapid time-varying unevenness in the spray itself to give a consistent and repeatable measurement of the total spray force. The measurement chamber can be inclined in both horizontal and vertical planes as indicated in Figure 5 relative to the spray hole axis. For momentum efficiency measurements the measurement chamber axis is aligned with the spray hole axis.

Measurement of initial jet angle

This test rig can also be used to measure an initial angle of the fuel spray/jet as it emerges from the spray hole. This is done by measurement of the decline in spray force as the measurement chamber is deviated away from the axis of the fuel/spray jet. At first the force will drop very slowly with the cosine of the angle between the measurement chamber and the spray/jet, but then much more quickly as the aperture of the measurement chamber cuts off the entry of the spray/jet cross section. The angle over which the force declines from 90% to 10% of its value is a measure of the spray/jet angle.

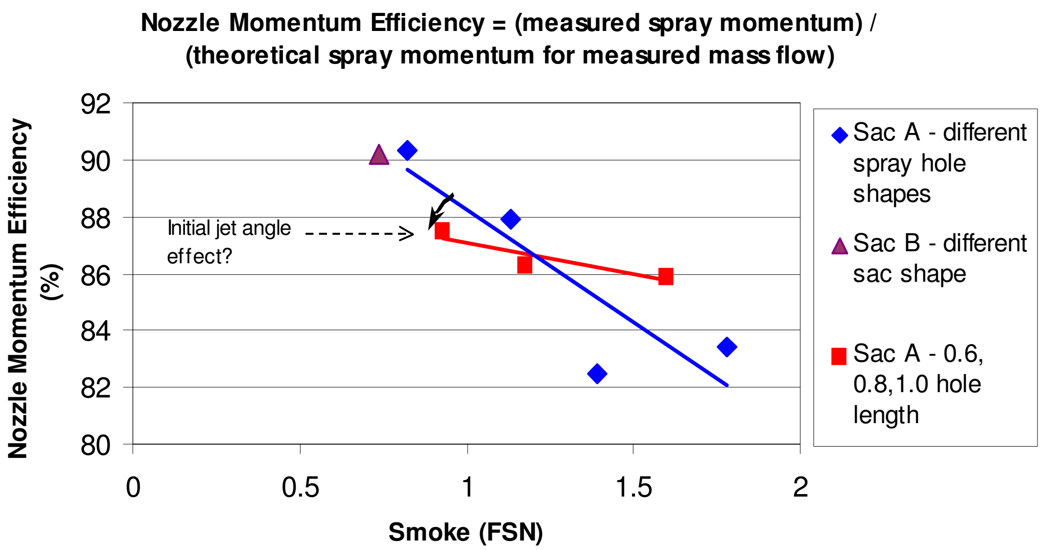

Figure 7 Correlation of measured engine smoke with measurements of nozzle momentum efficiency for different nozzle designs – single-cylinder HSDI engine, 2800 rpm, 10.8 bar BMEP

RESULTS FOR NOZZLE MOMENTUM EFFICIENCY

Figure 6 shows some sample momentum efficiency and Cd values measured with this new test rig. These measurements were taken for an upstream injection pressure of P1 = 100 bar and atmospheric back pressure for a range of needle lift conditions. The momentum efficiency values decrease with a reduction of the nozzle needle lift and the momentum efficiency values are greater than the Cd values as stated in equation (6). At higher engine load conditions the nozzle needle lifts to a full lift condition and the greater part of the fuel is injected at full needle lift. Of particular interest is the extent to which momentum efficiency values at full lift explain changes of exhaust smoke levels for different nozzle design details.

Figure 7 shows the exhaust smoke values measured on a single-cylinder HSDI engine plotted versus the measured momentum efficiency values at a full needle lift condition. The engine test results were taken for the same injection pressure and for nozzles with the same flow, so that the differences in smoke should be the effect of the different spray hole shapes and sac shapes. These results show there is a significant reduction of the engine exhaust smoke for nozzle designs with higher momentum efficiency (as expected from Figure 4).

The nozzle designs with different spray-hole shapes include the effect of spray holes with parallel and convergent tapers and entrance rounding. Also shown is the effect of a different sac design and different spray hole lengths. A separate trend line is indicated for the results with different spray hole lengths. A possible explanation for this is that the different spray hole lengths can change the initial fuel spray/jet angle as it emerges from the spray hole. This angle is an additional spray hole input characteristic to that of momentum efficiency since it has a separate effect on the fuel jet and air/fuel mixing

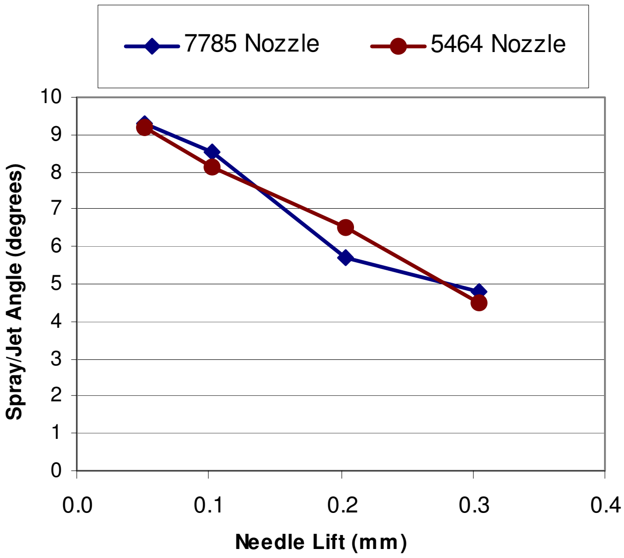

Figure 8 Initial spray/jet angle versus needle lift

process. For example, in the SPRY2 model mentioned earlier, the initial jet angle is an independent input to that of the input spray momentum by the spray hole.

RESULTS FOR INITIAL JET ANGLE

It is difficult to get an accurate measure of the initial jet angle by optical techniques since the edge of the spray/jet can be indistinct. As an alternative the new test rig as in Figure 5 has been used to measure the initial jet angle results given below. This was done by variation of the angle of the measurement chamber such that the aperture gives a progressive cut off of the jet section and spray force as described above. It should be made clear that this initial jet angle is measured close to and within 5 mm of the spray hole exit (as illustrated in Figure 3). The intention is to capture the output from the spray hole rather than the wider jet angles seen further downstream (Figure 3) as these latter are more to do with the gas phase jet angles and are a function of the entrained gas charge density etc.

Figure 8 shows some results obtained for the initial jet angles of two different nozzles. These results show that the initial jet angle increases as the needle lift decreases. This means that at low needle lift the wider initial jet angle will cause the liquid jet velocity to initially decay more rapidly along the axis of the fuel spray/jet. This combined with the lower momentum efficiency at low needle lift (Figure 6) means that the part of the fuel spray injected at low needle lift will be much less penetrating than at full needle lift. A low penetrating part of the fuel spray is deflected rapidly by the air swirl in the combustion chamber. This leads to interference with the adjacent fuel sprays leading to fuel-rich areas and an additional source of soot formation.

These effects explain one of the benefits of the direct acting piezo injector [12]. This injector has very fast needle opening and closing characteristics. This minimises the quantity of fuel that is injected at low needle lifts (with a wider initial jet angle) and with low momentum efficiency, to give significant reductions in emissions.

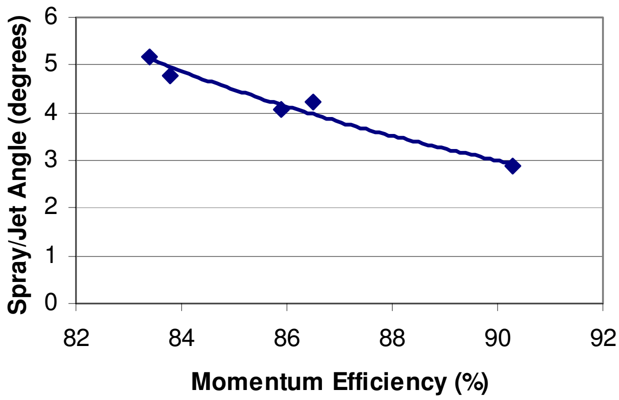

Figure 9 Measurements of initial spray/jet angle versus measured momentum efficiency at full needle lift for five different nozzle designs

Measurements of the initial jet angle at full lift are shown in Figure 9 for five different designs of nozzle. In particular spray-hole shapes that give higher momentum efficiency, as described in Figure 7, also tend to have smaller initial jet angles. This means that the lower engine exhaust smoke values observed with higher momentum efficiency spray-hole shapes (Figure 7) may be influenced also by the narrower jet angles.

CONCLUSIONS

From a review of optical engine experiments and a phenomenological computer model of the diesel fuel spray jet, in a swirling air charge, it is concluded that the liquid phase part of the fuel spray jet is very dense and relatively short at normal hot engine operating conditions. Soot forms further downstream in hot fuel-rich regions of the fuel vapour jet and is reduced by faster rates of air/fuel mixing. The axial fuel jet momentum input by the nozzle spray hole drives the fuel jet entrainment process and the speed of air/fuel mixing.

A new parameter named nozzle momentum efficiency is defined. This is essentially a measure of the efficiency of a nozzle in converting the fuel injection pressure into axial momentum in the fuel spray/jet. In order for this parameter to be useful to evaluate the effect of different nozzle design details a new technique was needed to measure momentum efficiency to an accuracy of ~1%. Results with a new in-house technique show the following.

(1) The momentum efficiency increases for nozzles with more convergent spray-hole shapes and entrance rounding.
(2) Observed engine exhaust smoke levels decrease with nozzles having higher momentum efficiency.
(3) The momentum efficiency decreases and the initial spray jet angle increases as the nozzle needle lift is reduced. This means that fuel injected at low needle lift will have lower penetration and tend to be deflected much more rapidly by the air swirl. The latest direct-acting piezo injector has much faster needle opening and closing that minimises the quantity of fuel injected at low needle lifts giving emissions benefits.
(4) The nozzle design affects the initial jet angle at full lift with higher momentum efficiency nozzles giving a narrower initial jet angle.

The advancement in understanding of these nozzle spray jet characteristics will contribute to future diesel fuel injection nozzle design and manufacture.

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NOMENCLATURE

Symbol	Quantity	SI Unit
A_h	Minimum geometric area of the spray hole	m^2
C_d	Nozzle spray-hole discharge coefficient	ratio
F_p	Force measured on a plate to destroy all of the axial spray/jet momentum	N
FSQ	Fraction of injected fuel from part of jet section going directly over top of piston bowl	ratio
$\dot{m}$	Actual mass flow rate exiting the spray hole	kg/s
$\dot{m}_{th}$	Theoretical mass flow	kg/s
$\dot{M}_{tham}$	Theoretical momentum flux exiting the spray hole based on the actual mass flow	kg m/s^2
$\dot{M}$	Actual momentum flux exiting the spray hole	kg m/s^2
P_1	Upstream or source fuel pressure	N/ m^2
P_2	Downstream or back pressure	N/ m^2
RAF9	Rate of air/fuel mixing through an air/fuel ratio of 9	(A/F ratios)/ms
U_{effM}	Effective velocity for spray momentum	m/s
U_{th}	Theoretical or Bernoulli velocity	m/s
η_n	Momentum efficiency ($\dot{M}$ / $\dot{M}_{tham}$)	ratio or %
ρ_f	Liquid fuel density	kg/m^3

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