Numerical modeling of diesel spray wall impaction phenomena

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A model for diesel spray impaction on walls is presented that is based on experimental observations. The model, which is incorporated into a two-phase CFD code employing a discrete droplet model for the liquid spray and the implicit noniterative PISO algorithm for gas equations, is assessed against experiments for a number of test cases including normal or angled injection into a quiescent space or a cross-flowing gas. In most of the cases studied, the wall spray radius is reasonably well predicted, but the wall spray height is always underpredicted.

In order to capture the dispersive nature of the wall spray that contributes to the wall spray height, the original drop–drop collision model of O'Rourke and Bracco is extended to allow for the effects of multicollisions between two drop parcels in a thick spray. The extended collision model works well in two-dimensional cases but is not fully assessed in three-dimensional cases.

The wall impaction model is also implemented in the three-dimensional EPISO engine code. Preliminary results on the simulation of diesel spray impingement in an actual engine suggests that the model is there also capable of producing reasonable wall sprays.

Keywords: diesel spray; wall impaction; modeling

Introduction

Diesel spray impingement onto cylinder walls is an important feature affecting engine combustion efficiency and emissions. The deposition of liquid fuel on the wall was generally regarded as having negative effects on the formation of a combustible mixture and also as being the main cause of the higher level of hydrocarbon (HC) emissions often seen in small high-speed direct-injection engines. In such engines, wall impingement of diesel sprays is quite unavoidable for the reasons of compact combustion chamber and high-speed fuel injection designed for good atomization. However, there have also been attempts to deliberately employ wall impaction to achieve a controlled evaporation and therefore combustion rate, such as in the "M" combustion system (Meurer 1956), or to promote fuel/air mixing (Kroeger 1986).

Due to its complexity, experimental studies on the issue of spray–wall interaction, which began only in recent years, are almost all conducted with high-speed photographic techniques and carried out in laboratory test rigs at room temperature (Kuniyoshi et al. 1980; Fujimoto at al. 1988; Sakane et al. 1988; Mirza 1991). Presentations of experimental results are therefore concentrated on the distribution pattern of a wall spray, and the two fundamental parameters: wall spray radius and wall spray height, which are nevertheless very useful measures for the assessment of relevant analytical models.

As far as the models are concerned, the present paper shall focus only on the numerical models that come under the framework of multidimensional techniques. The first such attempt is due to Naber and Reitz (1988) who considered three alternative ways of tracking droplets after wall impingement, namely,

(1) *stick model*: drops that reach the wall stick to the wall and continue to evaporate;
(2) *reflect model*: drops that reach the wall rebound specularly with the components of normal and tangential velocity unchanged (only the sign of the normal velocity changes); and
(3) *jet model*: the impinging droplet is assumed to move tangentially along the surface in a similar way as fluid flow in an inviscid impinging liquid jet—that is, the tangential velocity magnitude of the droplet after impaction is set equal to the speed of the same droplet just before impaction.

These models were employed in the KIVA code (Amsden et al. 1985) and applied in cases involving diesel spray impaction either normally or at an angle onto a wall placed at a distance of 63 mm from the injector inside a pressurized chamber at room temperature, where comparisons with the experimental data were available, or onto the piston top inside an engine without combustion. On the whole, the JET model produces the best results. However, in a separate case study (Naber et al. 1988), in which a diesel spray inside a simulated combustion chamber impacts normally onto a wall only 6.43 mm from the injector, the JET model had to be modified such that a normal velocity component 0–34 percent (randomly chosen) of the impinging velocity was imposed on the leaving droplet to achieve a reasonable agreement with the corresponding experiments on spray distribution and tip penetration. Alloca et al. (1990) also used this modified JET model in their simulation of spray impaction normally onto a wall. But it seems that the results were not satisfactory, with many of the simulated drops appearing well outside the envelope of the spray, as determined by experiment.

The present authors reported a different drop–wall impaction model (Watkins and Wang 1990), which is based on

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relevant experimental observations. The model was implemented in a two-phase CFD code currently being developed at UMIST (Watkins 1989; Watkins et al. 1991), which features a discrete droplet model for liquid phase, the PISO algorithm for solving the gas-phase governing equations, and the k–ε turbulence model, and was assessed against experiments in cases of normal and angled impactions. In order to capture the dispersive nature of the wall spray, the original drop-collision model of O'Rourke and Bracco (1980) was modified to allow for the effects of the local drop concentration and stochastic characteristics of collisions. This modified collision model produced very good results in some two-dimensional (2-D) cases.

In this paper, the wall impaction model and the modified drop-collision model that were reported in Watkins and Wang (1990), as well as the rationale, will be discussed in detail. Performance of the models is further studied in a variety of wall impaction cases including normal and angled spray impaction in quiescent space, spray injection into a cross-flowing gas, and impingement onto piston walls in a direct injection engine in a cold state.

General approach

The simulation of diesel sprays is based on the discrete droplet model (DDM), in which the liquid phase and the gas are dealt with separately but interphase effects are accounted for in a full scale. For the gas phase, the transport equations of mass, momentum, energy, and turbulent kinetic energy and its dissipation rate are discretized using the finite-volume approach. The PISO algorithm of Issa (1986) (in the engine case it is the EPISO algorithm of Ahmadi-Befrui at al. 1990), which uses noniterative but implicit methods, is applied to solve the discretized gas-phase governing equations.

For the liquid phase, the spray is represented by a number of drop parcels containing many identical droplets. Each droplet's momentum and trajectory equations are marched in time implicitly using the Euler two-point scheme. These equations are solved as a part of the spray EPISO method (Watkins 1989). The secondary breakup of droplets (Reitz 1987) and drop–drop collisions (O'Rourke and Bracco 1980) are also taken into account.

Modeling spray wall impaction

It is desirable, in the context of the discrete-droplet spray model, that the wall impaction should be based on the fundamentals of individual droplets. Wachters and Westerling (1966) first carried out an experimental study of drop–wall interaction, in which water drops (about 2 mm in diameter) impinging on a hot polished metal surface (~400°C) were photographed using stroboscopic lighting. It was observed that, depending on the approaching Weber number, the droplet would either reflect, or stick and shatter on the wall. It was confirmed later by other experimentalists (e.g. Araki and Moriyama 1982) using different liquids that the critical value of Weber number, We, is around 80.

On the other hand, high-speed photographic observations on wall impingement of diesel fuel sprays injected into chambers at elevated pressure—for example, Katsura et al. (1989) and Mirza (1991)—have indicated that a substantial amount of liquid is entrained into the air after impaction on the wall. In contrast, the feature of wall wetting is far less important.

It is understood, as those photographs recording the development of wall sprays have suggested, that droplets that first arrive at the wall form a cushion and those that come later, pushed by others behind them or carried by the gas-phase flow near the wall, move along the cushion, which exerts a shear drag force from underneath. Thus, the wall spray gets thicker as later droplets catch up and collide over those in the front. This process is somewhat similar to that of a free spray where a cone area is formed. The difference is that in the wall impaction case, there are also drops reflected after impaction, which tends to make a wall spray more dispersed than a free spray.

Based on this understanding, the basic principle for modeling the impinging droplet is that it is allowed to either reflect from or attach to the wall, depending on the approaching We number. In the latter case, the droplet is allowed to float over the wall. The droplet velocity and diameter after impaction can then be expressed as

$$\mathrm{We} \le 80 \quad \begin{aligned} v_{an} &= -\alpha v_{bn} \\ v_{a\tau} &= \alpha v_{b\tau} \\ D_{da} &= D_{db} \end{aligned} \qquad (1)$$

$$\mathrm{We} > 80 \quad \begin{aligned} v_{an} &= 0 \\ v_{a\tau} &= v_{b\tau} \\ D_{da} &= C D_{db} \end{aligned}$$

$$\mathrm{We} = \frac{\rho_d v_{bn}^2 D_{db}}{\sigma}$$

Here subscripts a, b stand for "after," "before," and τ, n for "tangential," "normal," respectively. ρ_d and σ are the liquid density and surface tension, respectively.

The coefficient α is to account for the kinetic energy loss and is calculated from Jayaratne and Mason (1964):

$$\alpha = (1 - 0.95 \cos^2 \beta)^{1/2} \qquad (2)$$

which is based on water drops falling on a water surface, since

Notation

D_d	Droplet diameter
p	Random variable
t	Time
V	Velocity vector
We	Weber number

Greek symbols

θ	Void fraction
v	Velocity (speed)
ρ	Density
σ	Surface tension

Subscript

a	After
b	Before
d	Drop
imp	Impaction
n	Normal
τ	Tangential

no data are available for diesel drops impinging on solid walls. β in the equation is the impinging angle to the normal.

C is designed to account for the droplet shattering in high-incident Weber number cases. While its value is not known, here it is assumed to be 1/4, so that a drop shatters into 64 smaller drops all of the same size. A limited parametric analysis has been undertaken by Park et al. (1993), in which the value of C was varied between 1/3 and 1/4. For the particular case examined, that of spray impacting on the land of a piston bowl pip set 10 mm from the nozzle, the indications are that the value of C exerts little influence over the subsequent velocities and sizes of the drops, due to later interdrop collisions.

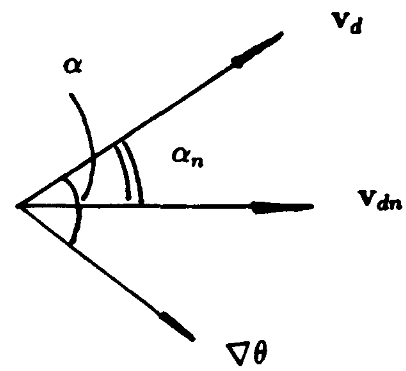


Figure 1 Definition of new grazing velocity

Extended collision model

As has been shown extensively in Wang (1992), the above wall impaction model does not suffice to give a dispersed wall spray that would match experiments. The droplets after wall impaction tend to accumulate in a thin layer along the wall. The reason is that the approach Weber number is often very high and most droplets lose their normal velocity component immediately after impaction. The subsequent motion of the drops along the wall induces a head vortex that carries some drops away from the wall. However, this dispersion mechanism is insufficient to match the experimental data. The only other mechanism available as a source of subsequent droplet dispersion is the interaction among the droplets. Therefore, the collision model of O'Rourke and Bracco (1980) has been examined to discover whether it is capable of simulating this process.

In the drop-collision model of O'Rourke and Bracco, collision is taken to be between two drops, the sizes of which are neglected. Thus, in the case of the so-called "grazing" collision in which coalescence is followed by the immediate breakup of the drops, if the velocities of the two drops are in the same line (or parallel to each other as along a wall) before collision, they will still be in the same line after grazing, and no third direction will result. In reality this may not always be the case if the drop sizes are considered, because a small drop with higher velocity colliding with a big drop may well result in a large deflection of the smaller drop and a small deflection of the larger.

Here, rather than going into the considerable detail of drop size's influence on the collision, which in practice would require much more computational effort, the droplet density distribution around the collision site is used as a factor of "blocking," or changing the droplet in its original route. That is, the droplet after collision is assumed to have a tendency of moving toward the direction of the least concentration of droplets around it. In addition, given the fact that in the present discrete droplet model, collision between two drops actually represents collisions between two parcels of many droplets, the outcome of collision should have some statistical feature.

Let v_d (Figure 1) be the velocity vector of one of the two drops obtained from the original grazing collision and $\nabla\theta$ the void fraction gradient, which is a measure of local droplet distribution. Then the velocity v_{dn} after the new grazing consideration is decided from the following equations:

$$\alpha_n = (1 - \bar{\theta}^b)p\alpha \tag{3}$$

$$v_{dn}(v_d X \nabla\theta) = 0 \tag{4}$$

$$|v_{dn}| = |v_d| \tag{5}$$

where p is a random variable uniform in (0, 1), and α, α_n are, respectively, the angle between v_d and $\nabla\theta$, and that between v_d and v_{dn}. $\bar{\theta}$ is the average void fraction of all the neighboring cells, and b is a prescribed constant (= 1 here).

Equation 3 is in fact a kind of weighting between the direction of v_d and that of $\nabla\theta$. The use of random number p as a part of the weight is to reflect the indeterminate nature of collisions in which many droplets are involved. The average level of local droplet concentration is also taken into account and is in the form of $1 - \bar{\theta}^b$, which implies that for low drop density (thin spray) case in which $\theta \approx 1$, the effect of drop distribution nearby would be very small. The power b is here to either enhance or weaken this effect.

Equation 4 requires that the new velocity vector lies in the same plane composed by vectors v_d and $\nabla\theta$, which is what one would expect to see. With the magnitude of the new velocity set equal to that of the old collision velocity (Equation 5), Equations 3–5 form a closed set and a unique velocity will be obtained.

Table 1 Test-rig cases

Case No.	N1	N2	N3	A1	C1	C2	C3	C4	C5	AC1	AC2	AC3
Wall distance (mm)	24	32	50	32	32	32	32	32	32	32	32	32
Gas pressure (MPa)	1.0	1.38	2.64	1.38	0.69	1.38	2.07	0.69	1.38	1.38	1.38	1.38
Injection pressure (MPa)	14.3	A[a]	B[b]	A	A	A	A	A	A	A	A	A
Nozzle diameter (mm)	0.3	0.26	0.29	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26
Injection duration (ms)	1.2	1.36	7.8	1.36	1.36	1.36	1.36	1.36	1.36	1.36	1.36	1.36
Injection angle (°)	0	0	0	30	0	0	0	0	0	30	30	30
Cross-flow (m/s)	0	0	0	0	13.5	13.5	13.5	13.5	9.4	9.4	13.5	9.4
Turbulent intensity (%)					≤2	≤2	≤2	16	≤2	≤2	≤2	16

[a] Flow diagram of Mirza (1991).
[b] Pattern B in Sakane et al. (1988).

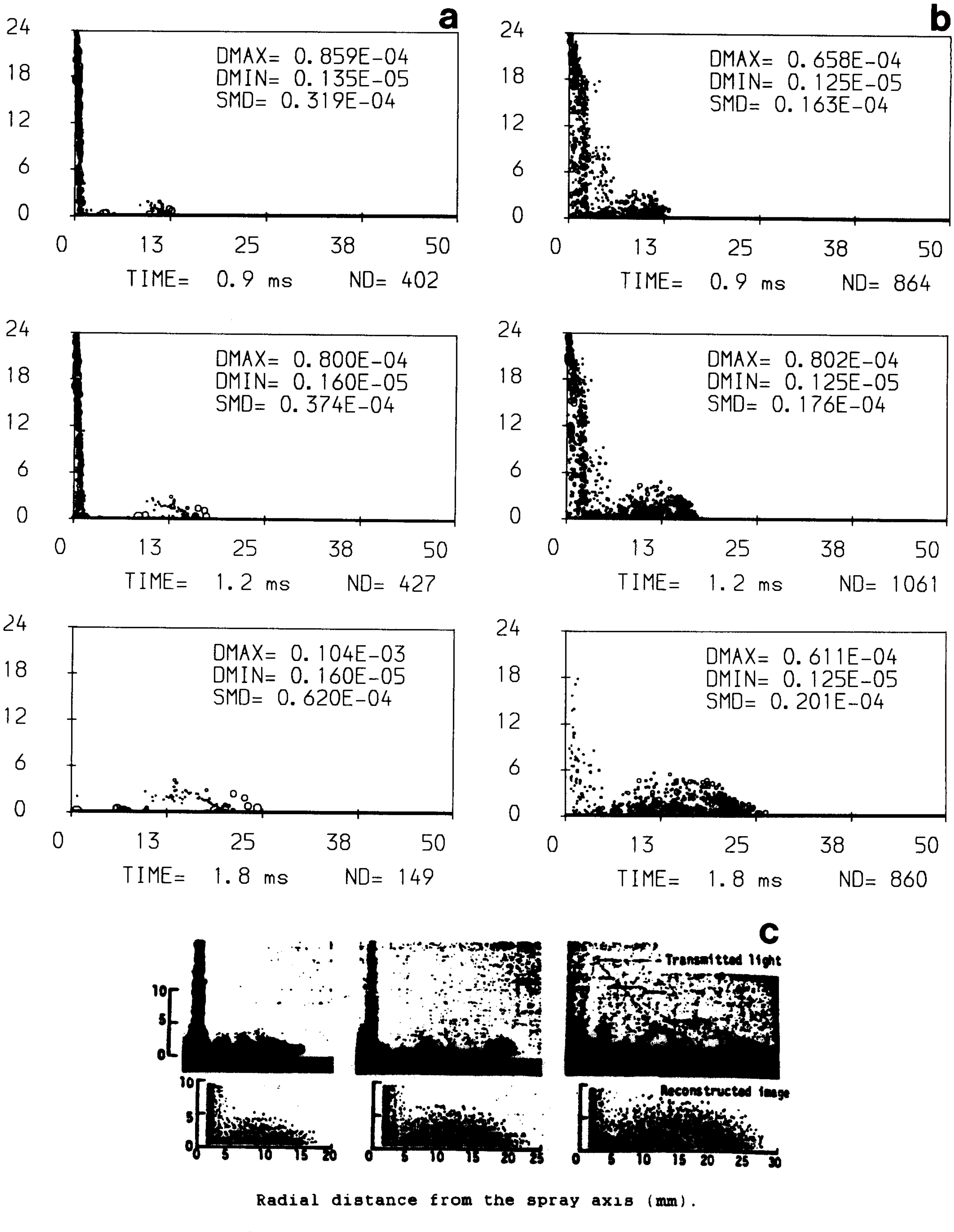


Figure 2 Spray development for Case N1. (a) Old grazing; (b) new grazing; (c) reconstructed image (Fujimoto et al. 1988)

Case studies

Test rig

In order to assess the performance of the above numerical models, a variety of test cases are studied for which experimental results are available. These include normal injection into a quiescent space or injection into a cross-flowing gas (Table 1).

Since the relevant experiments were carried out in fixed chambers at room temperature, the constant-volume version of the EPISO code with either Cartesian (in 3-D cases) or cylindrical (in 2-D cases) grids were used (Wang 1992). This saves greatly on the running cost.

Normal impaction without cross-flow. Figure 2 shows, for case N1, the spray development in terms of droplet distribution at different times from the start of injection as obtained from (a) applying only the wall impaction model with no change made to the grazing-collision model (old grazing); (b) incorporating the extended grazing-collision model (new grazing); and (c) reconstructed images based on photographs of Fujimoto et al. (1988). While both calculations agree well with the experiment in the extent of spray spreading out along the wall (wall spray radius), the significance of applying the new grazing model is that the wall spray shape is much closer to that of the experiment with the wall spray height, calculated to be some 20 percent less than in the experiment. In the old grazing case, only droplets near the wall spray tip are entrained into the gas, which coincides with the leading-edge gas-phase vortex (see Wang 1992). In other words, the spray behaves like an inviscid wall jet, and smaller droplets are carried away by the wall vortex with bigger ones, having more inertia, moving further downstream. This feature is less obvious using the new grazing model, where the diffusive nature of the wall spray in the direction normal to the wall is better captured by allowing droplets to move away from the wall, which justifies the rationale in the proposition of the extended collision model.

DMAX, DMIN, and SMD in the figure stand for, respectively, the maximum, minimum, and Sauter mean diameters of the droplets, and ND stands for the number of drop parcels in the calculation. In both calculations the SMD increases with time, which is consistent with experiments on wall spray development (Hiroyasu et al. 1990). But the actual SMD values in the new grazing model case are substantially smaller than in the case of the old grazing model. This may be explained by the fact that the new grazing-collision model leads to more dispersed droplet distribution (larger distance between droplets and higher relative velocity), which as a result is in favor of less collisions and, hence, coalesced drops.

The effects of model alterations are shown in Figure 3, in which the droplet distribution at $t = 3.9$ ms is plotted for Case N3. Figures 3a and 3b are, respectively, the results of the old and new grazing model. Comparing with Figure 3e, which is a profile of equivalent air/fuel ratio indicating concentration of fuel mass, it is seen that the wall sprays obtained from the two grazing models are not very different, although the improvement in terms of dispersion is still noticeable. Further investigation shows that, for this case, the number of grazing collisions is relatively low, which could be assumed to be the reason, since the new grazing model cannot be effective unless there are a sufficient number of grazing collisions. In an effort to increase grazing collisions, the balance of the outcome of collision is shifted from coalescence toward grazing. The probability E_{coal} (O'Rourke and Bracco 1980) that a collision results in coalescence is multiplied by a factor of 0.1. This results in much smaller droplets (Figure 3c), and the wall spray height becomes even lower. Since droplet breakup and coalescence are two competitive mechanisms of decreasing/increasing drop size, the breakup model is now switched off to maintain a higher mean drop size. Figure 3d shows the resulting drop distribution, and the wall spray does seem better. Comparing with the results of Case N1, the improvement of Figure 3d over Figure 3a is similar to that of Figure 2b over Figure 2a. But in the present case, the number of grazing collisions has to be increased, or the collision model has to be tuned to achieve this result. It would appear that the new model is less effective for sprays impinging on walls at a distance from the injector as large as 50 mm. The results from calculations with the wall at 63 mm have been compared with those of Naber and Reitz (1988) by Watkins and Wang (1990). Very little difference could be detected. The wall spray radius is a little better predicted by the present model, whereas the wall spray height is marginally better predicted by the Naber and Reitz model.

It is also noted that the extended collision model causes some droplets near the injector to be scattered from the main stream, as can be seen in Figures 2b and 3d. While the extended collision model was designed to capture the dispersive nature of a wall spray, such influence on the free spray is not expected and indeed should be avoided. This reveals the uncertainty toward the effectiveness of the extended collision model as to how to maximize its good influence on the wall spray and minimize the side effects to the free spray. It is probably best to apply the extended model only in the wall spray and to leave the free spray unaffected.

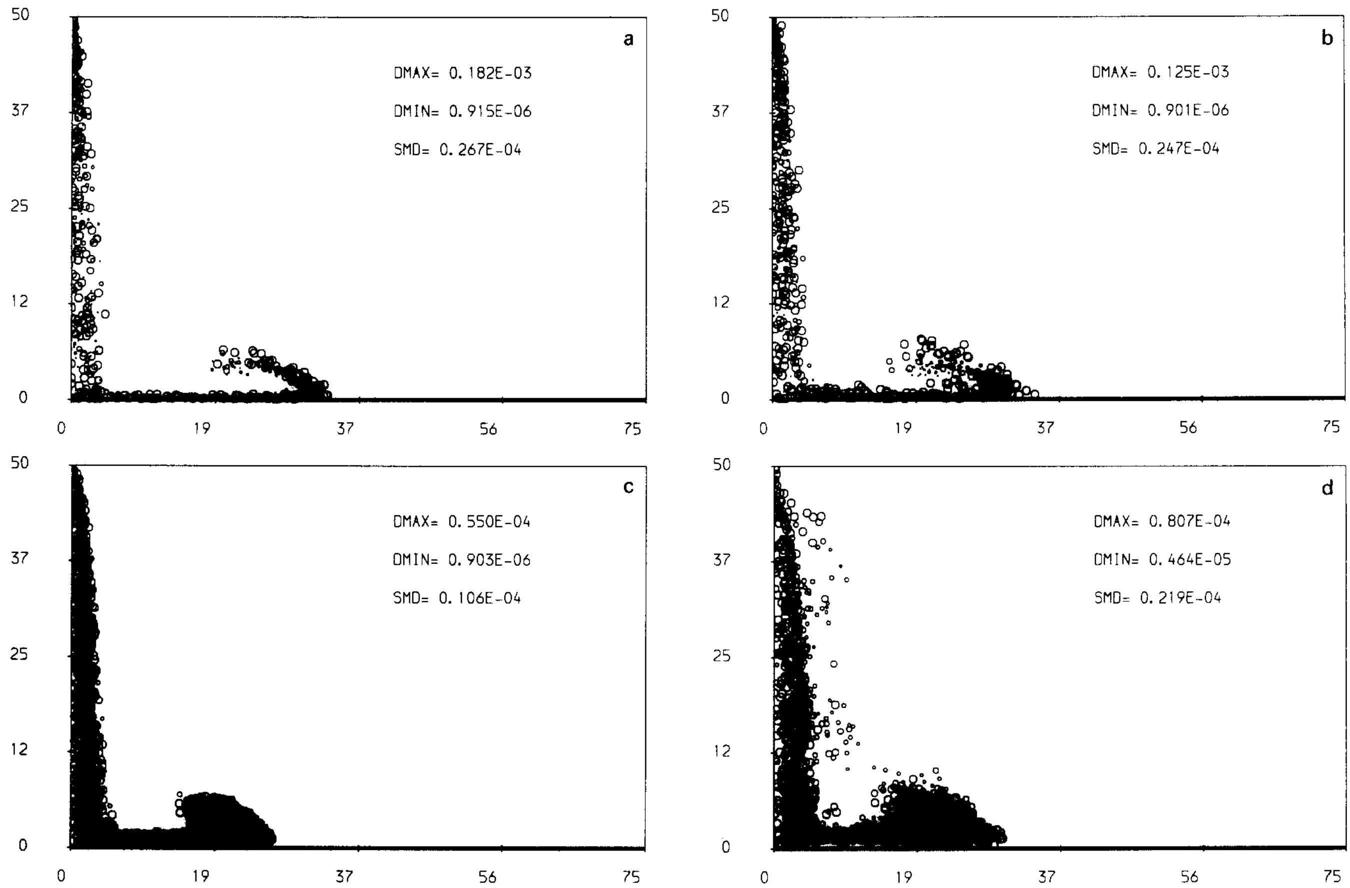
Normal impaction with cross-flow. For cross-flow cases, there is very little influence of the extended collision model. This is because the number of collisions is greatly reduced in the 3-D computation where the grid has to be coarser and the number of drop parcels cannot be increased in proportion to that used in 2-D computations. Changing constant b in Equation 3 could increase the effect of the extended collision model, but that would also have significant side effects on the free spray. For this reason the extended collision model is switched off for cross-flow cases for the time being since tuning of the model would incur substantial computational work. The objective here is then to investigate the effects of gas pressure, cross-flow velocity, and turbulence through a parametric study.

The effects of varying gas pressure on wall spray for cases C1, C2, and C3 are shown in Figure 4. The wall spray radius for the high-pressure case C3 agrees well with the experimental correlation of Mirza (1991), although the influence on the calculation of lower back pressures is clearly not as great as the experiment suggests.

Mirza's correlation is used here because it is the only one produced to date that contains the effects of cross-flowing gas on the wall spray.

For the wall spray height, the calculation shows an opposite trend to the correlation. That is, for higher gas pressure, the wall spray height is found to be lower in the calculation but higher in the correlation. In this respect the predictions are more in line with the correlation of Kuniyoshi et al. (1980) for wall sprays without cross-flow. In their correlation, the gas density ρ_g appears in the term $\rho_g^{-0.076}$ rather than the term $\rho_g^{0.52}$ as in Mirza's correlation. Katsura et al. (1989) give $\rho_g^{0.048}$. Figure 4b also shows a substantial underprediction in the calculation of the wall spray height, although the shape of the curves is similar to that of the correlation. This suggests that the term $(t - t_{imp})^{0.365}$ in Mirza's correlation and $(t - t_{imp})^{0.36}$ in Katsura's is corroborated by the calculations. On the other hand, the spray height as evidenced in the photographs of Mirza (1991) looks much less than that represented in the correlation. The argument here is that in measuring the wall spray height from those pictures, there is usually a very faint area (claimed to be composed of fuel vapor and tiny droplets) that has been counted as part of the wall spray, while in modeling cold flow cases the fuel vapor is not calculated and tiny droplets below 1 μm are simply neglected. It should also be noted that other correlations for wall spray height do not give such large values as does Mirza's correlation. For example, Katsura et al. (1989) obtained a correlation from their experiments that gives values of wall spray height approximately 65 percent of those of Mirza. This is for wall sprays without cross-flow. As is shown in Figure 5, the effect of cross-flow is to reduce the wall spray height. Thus Katsura's correlation, if extended to cross-flow cases, could be expected to give even lower values.

Figure 5 shows the effect of different cross-flow velocity on normal injection (N2, C2, C5). The calculated wall spray radius agrees reasonably well with that of the correlation, with more extent of underprediction for higher cross-flow velocity. The wall spray height is underpredicted as discussed above in comparison with both Mirza's correlation and Katsura et al.'s (1989) for the case N2, without cross-flow. In the experiment



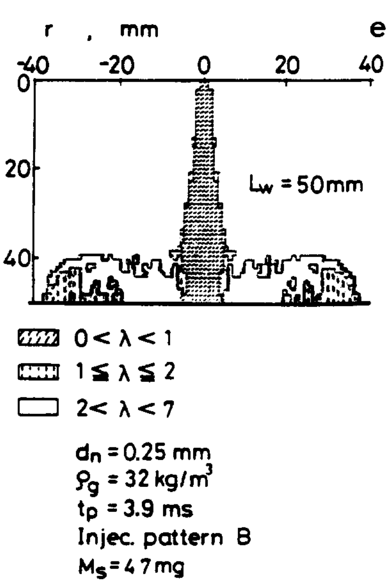
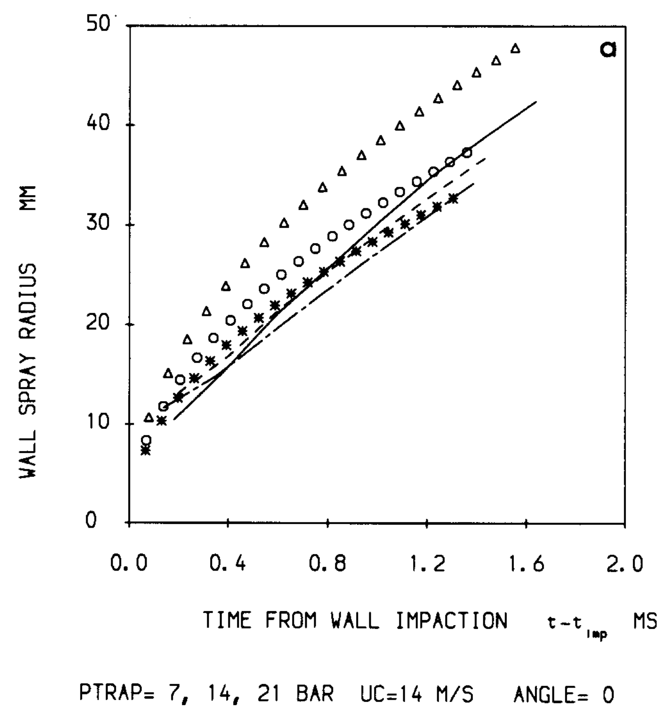


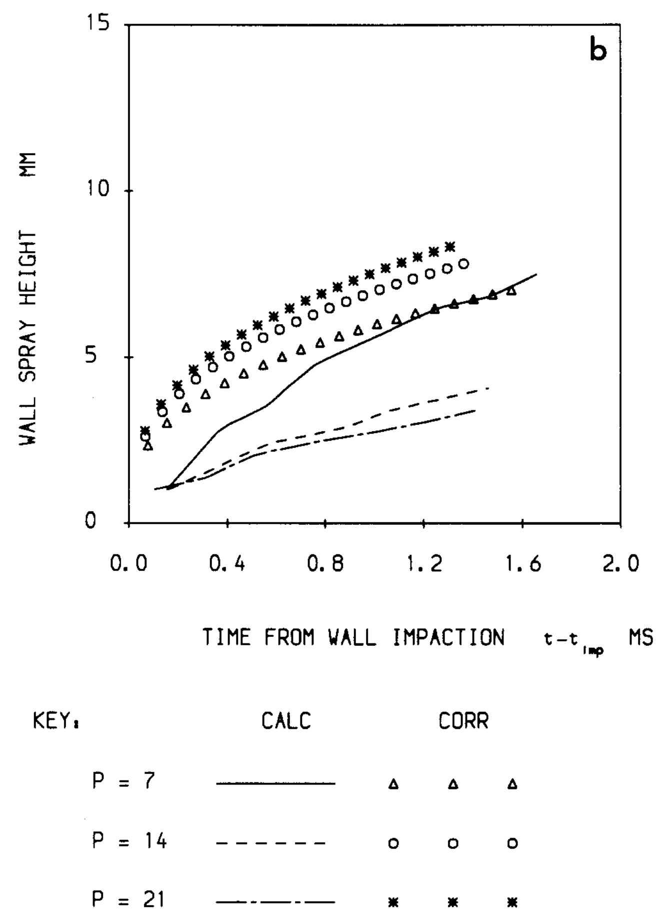
Figure 3 Spray structure for Case N3. (a) Old grazing model; (b) new grazing model; (c) new grazing model, $E_{coal} \times 0.1$; (d) new grazing model, $E_{coal} \times 0.1$, no breakup; (e) equivalent air/fuel ratio of experiment (Sakane et al. 1988)

the effect of increasing the cross-flow velocity is to decrease the wall spray height. This seems intuitively to be correct. The calculations, however, give a mixed response, with a reduction in height for a cross-flow of 9.4 m/s compared to no cross-flow, but with an increase again for a cross-flow of 13.5 m/s. The three droplet distributions at 2.1 ms after the start of injection are shown in Figure 5c, and these confirm the above result. However, it is clear that extracting a definite value of wall spray height from these distributions is quite difficult. Better, clearer, results would be obtained if many more drop parcels could be used. However, this drives up the computing times required.

The relative strengths of the present wall impaction model in handling the effects of cross-flow are difficult to state. No other impaction model has, to our knowledge, been subjected to the same scrutiny and comparison with data as is presented here.

Angled impaction. In Figure 6 are shown the results for three cases of angled (30°) impaction on the wall. One of the cases (A1) is without cross-flow, whereas cases AC1 and AC2 have cross-flow velocities of 9.4 m/s and 13.5 m/s, respectively. The gas pressure is the same in all three cases, at 1.38 MPa, or an air equivalent of 14 bar. Only comparison with Mirza's correlation can be shown here, since it is the only one that includes the effect of angled impaction. The agreement with





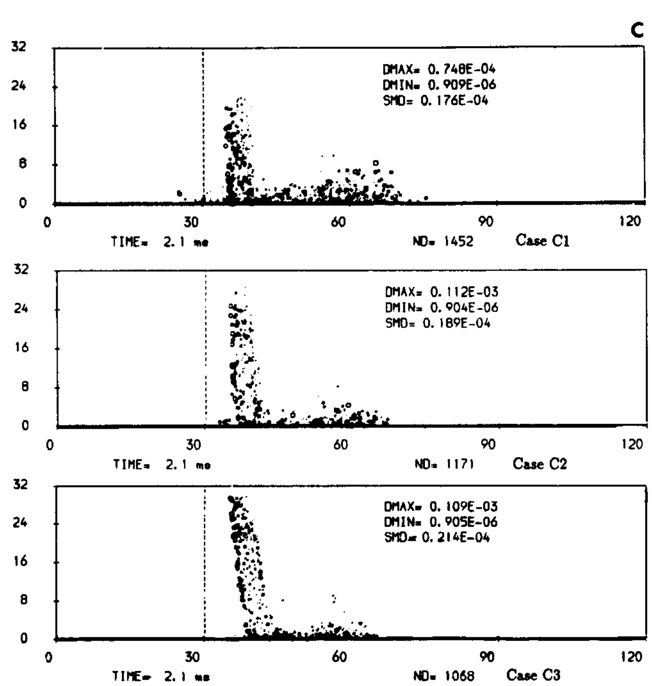
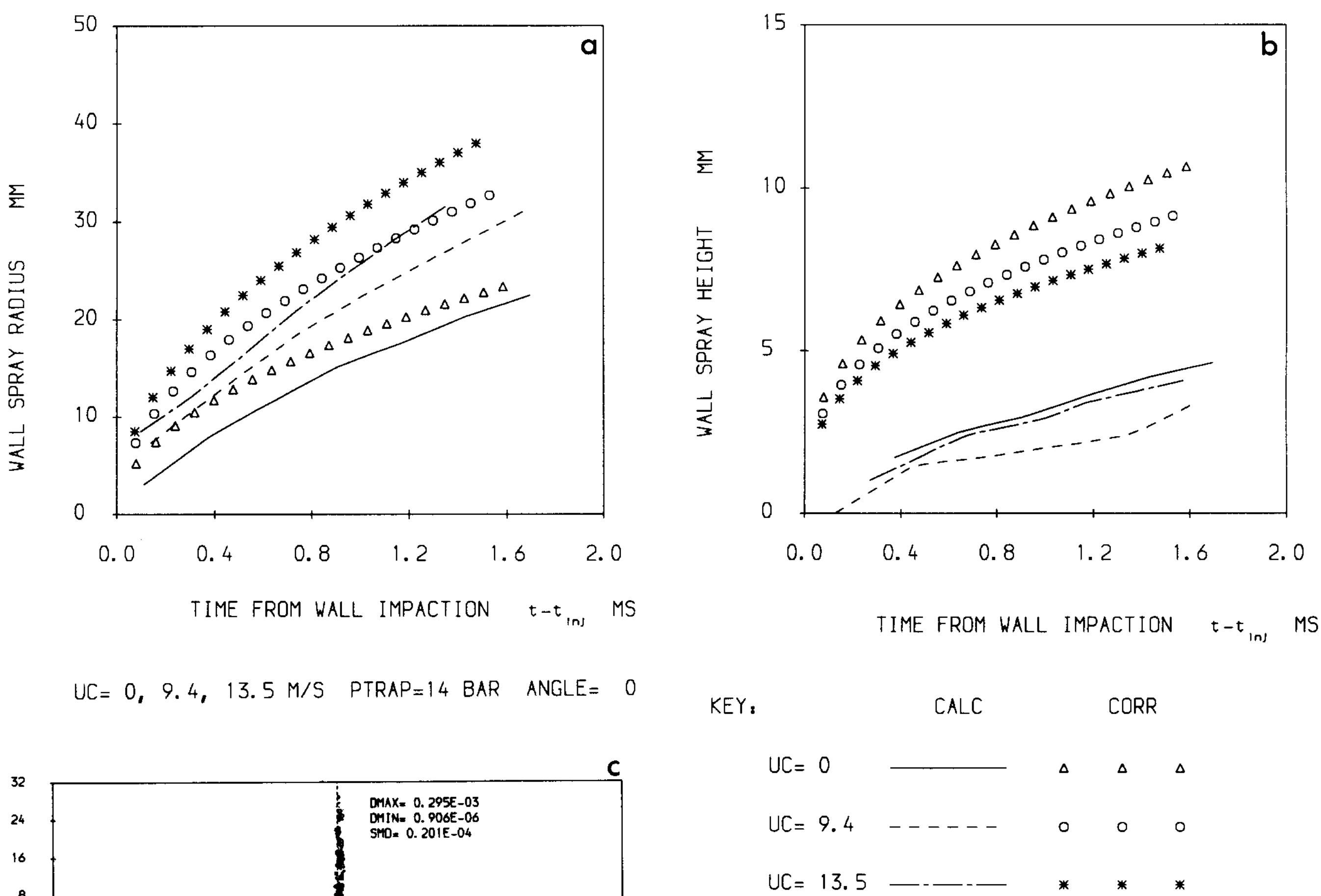


Figure 4 Effect of gas pressure on wall spray, Cases C1, C2, C3. (a) Wall spray radius; (b) Wall spray height; (c) Drop distribution at $t = 2.1$ ms

experimental data is relatively poorer than in Figure 5, particularly for the spray radius. It is believed that one of the reasons for this is related to the numerical errors induced by the presence of an angle between the main stream spray and the grid lines, as discussed in Watkins and Khaleghi (1990a). The spray height behaves perhaps more consistently than in Figure 5b, but it is clear from the droplet distribution plots of Figure 6c that there is a large margin of error in deciding the actual value of spray height.

Effects of turbulence. The effects of turbulence in the cross-flow gas are studied through comparison of cases C1 with C4, and AC1 with AC3. Figure 7 shows the calculated spray at $t = 2.1$ ms for each case. Basically, there is no significant difference in the spray structures for an increase of up to 16 percent in the gas turbulence levels in the cross-flow. This suggests that the spray tends to form its own turbulence structure in the nearby gas, which is not affected significantly by the turbulence in the cross-flowing gas. The corresponding wall spray radius and height are compared in Figure 8. Wall spray radius is slightly increased in the turbulent cross-flow case, which is consistent with the experiments of Mirza (1991). But wall spray height is seen to decrease, which is contrary to the findings of Mirza. However, this reduction is not significant, as illustrated by Figure 7a, and probably results from uncertainties in determining the spray height.



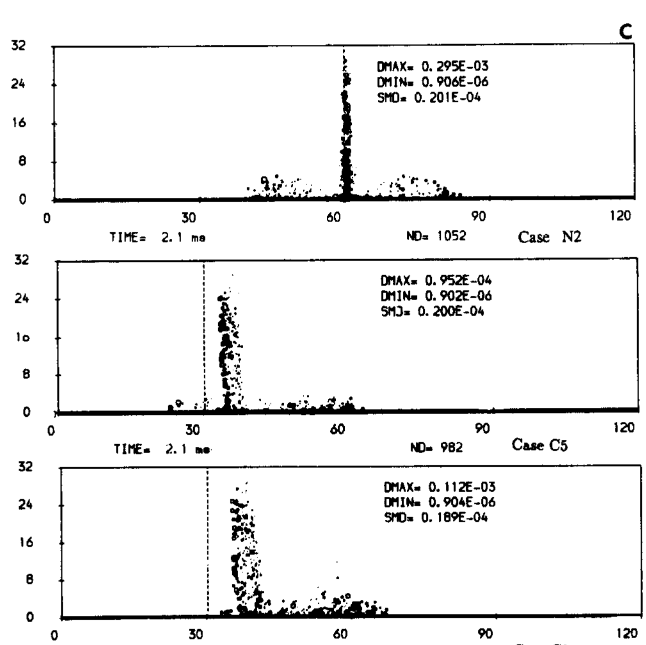
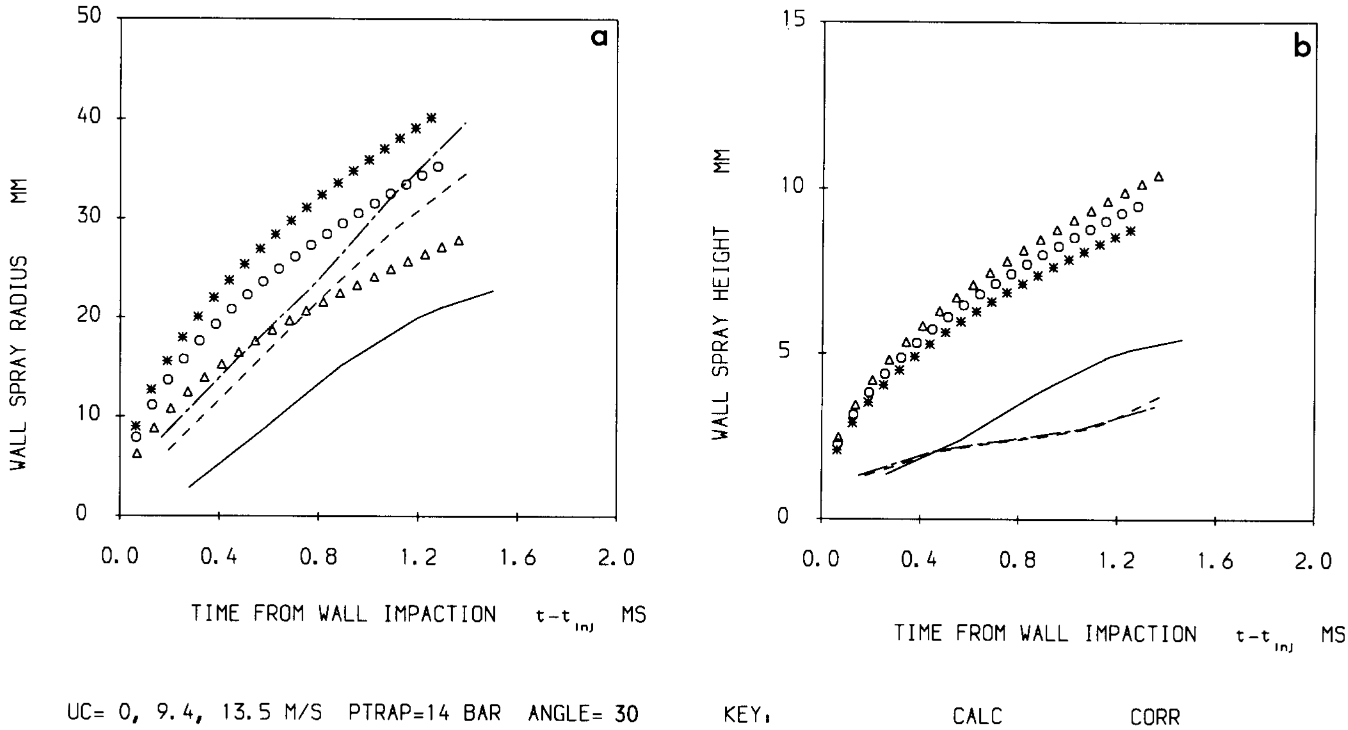
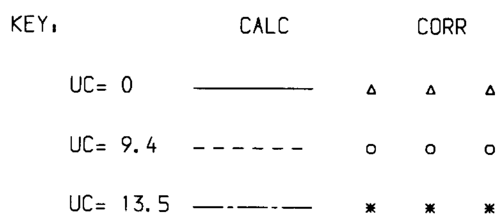


Figure 5 Effect of cross-flow velocity on wall-spray: normal injection, Cases N2, C2, C5. (a) Wall spray radius; (b) wall spray height; (c) drop distribution at $t = 2.1$ ms





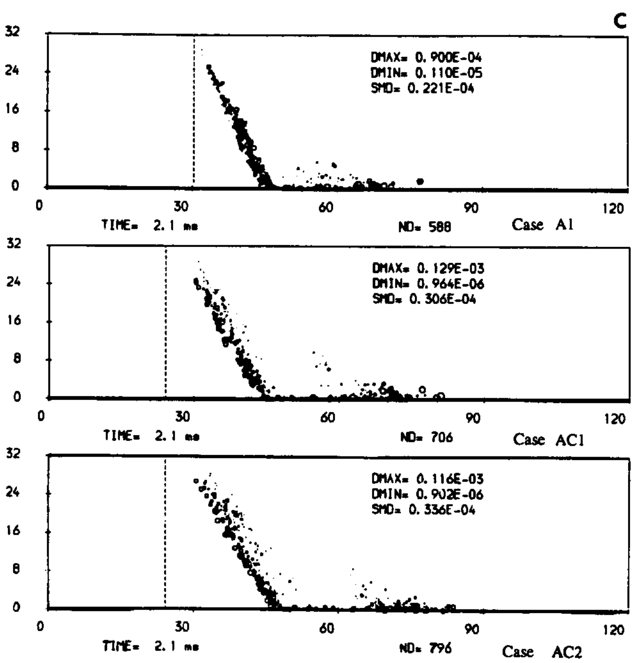


Figure 6 Effect of cross-flow velocity on wall spray; angled injection Cases A1, AC1, AC2. (a) Wall spray radius; (b) wall spray height; (c) drop distribution at $t = 2.1$ ms

Engine

The wall spray impaction model has been incorporated into the 3-D EPISO code described by Watkins (1989). The application case shown here, to demonstrate wall spray development inside an engine cylinder, is of a direct-injection single-cylinder test engine with a shallow-dish combustion chamber. The specifications of the engine are listed in Table 2 (Aoyagi et al. 1980).

The drop heat and the mass transfer and combustion calculations are not activated and the injection velocity is kept at 150 m/s. Thus, the spray would have more momentum and would penetrate further than in the heated flow case, and the events of the spray after wall impaction would be more distinctive.

Shown in Figure 9 is a side and top view of the spray development at different crank angles (CAs). Fuel injection starts at CA = 345, and the first drops impact on the side wall of the piston bowl at around CA = 351, which is 0.6 ms after the start of injection. Because of an impaction angle (injection angle is 75° to the cylinder axis) and upward movement of the piston during the compression stroke, in the axial direction the wall spray is seen developing mainly toward the bowl bottom. After top dead center (TDC), when the piston moves downward, the pumping effect assists the wall spray movement upward and, in fact, some drops are seen "entrained" into the

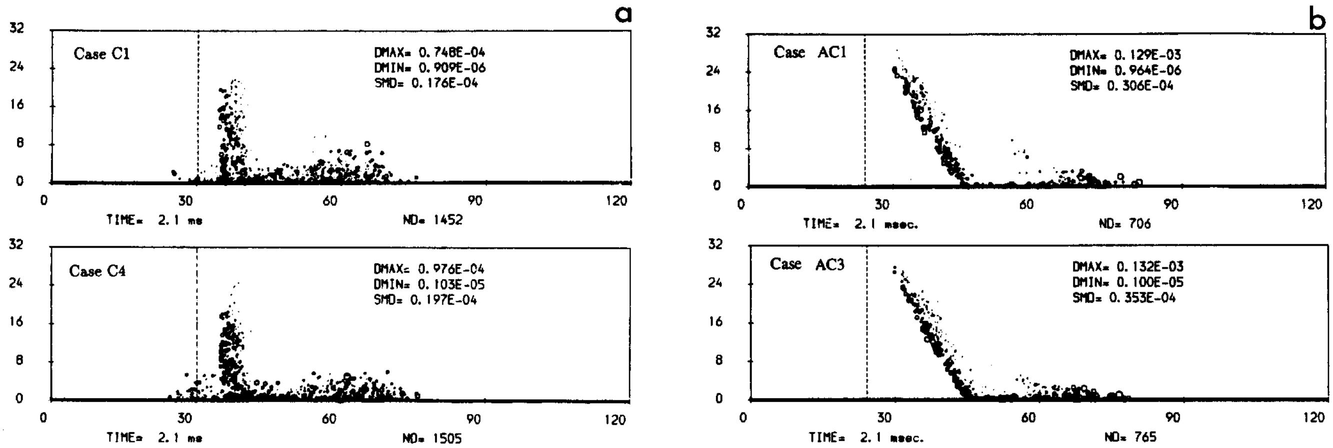


Figure 7 Effect of turbulent cross-flow on wall spray structure. (a) Cases C1 and C4; (b) Cases AC1 and AC3

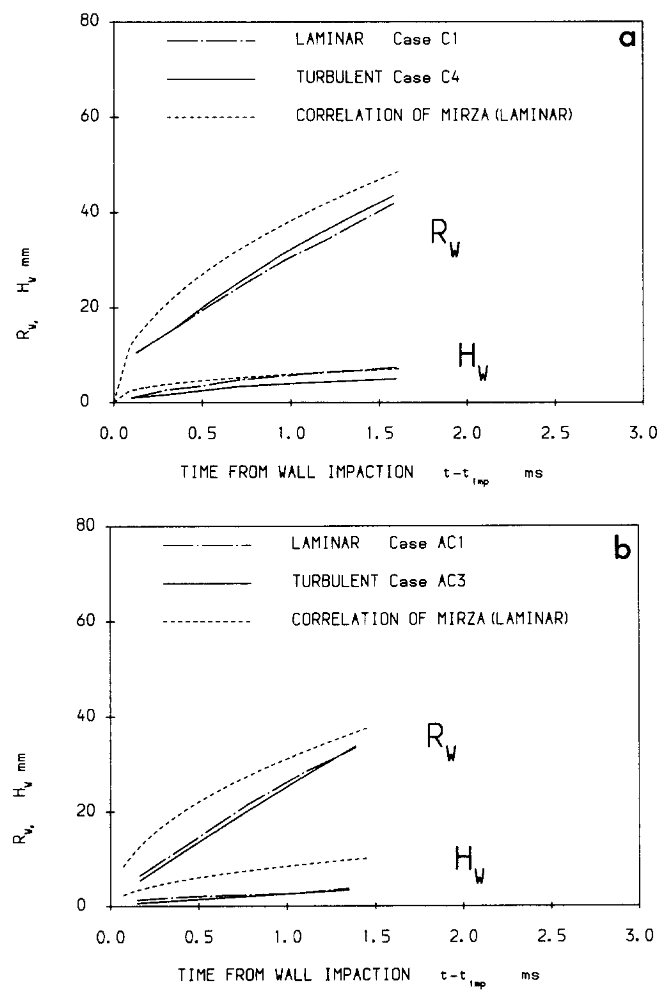


Figure 8 Effect of turbulent cross flow on wall spray radius and height. (a) Cases C1 and C4; (b) Cases AC1 and AC3

clearance space. Viewing from the top, the wall spray exhibits a similar pattern as in the case of normal impaction without a cross-flow, since there is no swirl in the gas flow.

Figure 10 plots only the droplets lying in a slab of 5 mm width in the middle of the piston bowl. This gives a clearer picture of the development of the wall spray. The structure of the wall spray looks similar to that obtained for flat wall impaction in the case of quiescent gas. It would therefore be expected that the model underpredicts the wall spray height.

A further engine test case using the present wall impaction model has been described by Park et al. (1993). One of the test cases of Naber et al. (1988) was recalculated. In this, spray was injected onto a land on a piston bowl pip at a distance of 6.43 mm from the injector. A major conclusion drawn from that analysis was that the present model appears to predict the loss of energy and momentum in the spray at impaction much better

Table 2 Engine specifications and operating conditions

Cylinder bore (mm)	95
Stroke (mm)	110
Piston displacement (cc)	779
Compression ratio	14.6
Engine speed (rpm)	1250
Swirl ratio	0
Number of injector nozzles	4
Nozzle diameter (mm)	0.2
Injection timing	15° BTDC—2.5° ATDC

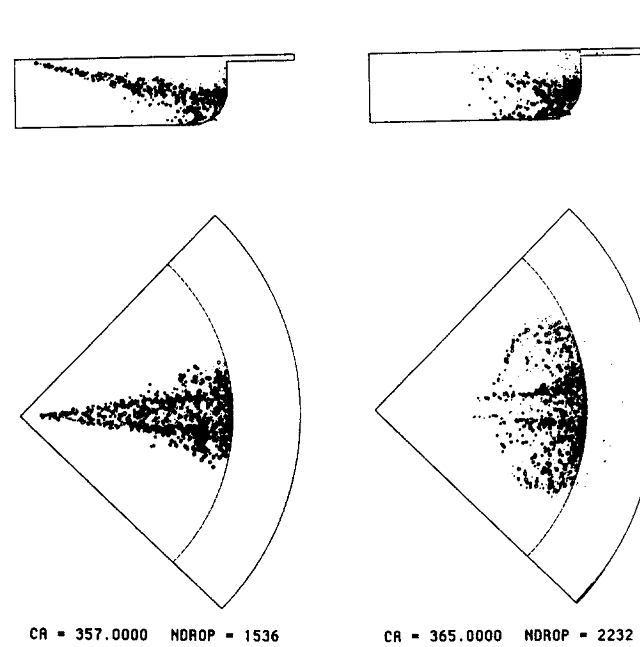


Figure 9 Spray development inside the cylinder

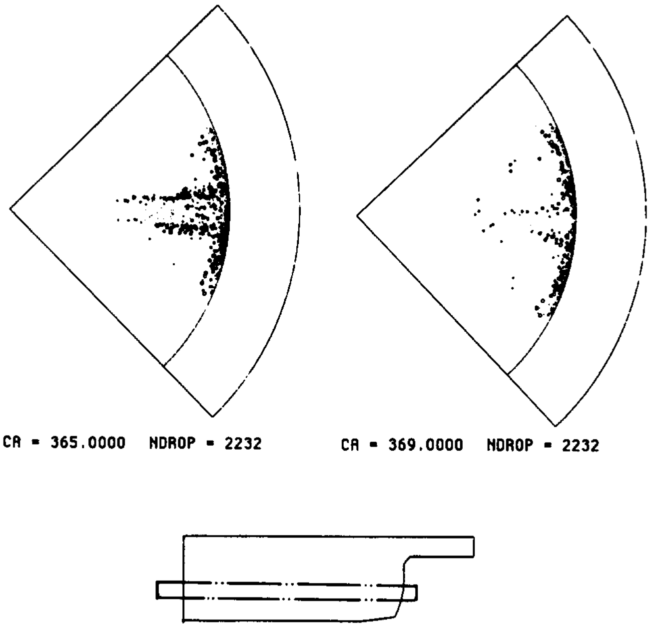


Figure 10 Sectional plots of spray

than the model used by Naber et al. The latter model overpredicted the spray penetration very soon after wall impaction by nearly 200 percent, whereas the current model was within 15 percent.

In an operating engine the gas is at an elevated temperature compared to that of the spray. For free sprays, consequent evaporation of the liquid drops has been extensively investigated and models developed. These are incorporated in the computer codes employed here but have been deactivated for the present cases. Results employing these models are given by Watkins and Khaleghi (1990b) and Wang, Watkins, and Cant (1993).

In the current model, drops impacting on a wall are assumed to float near the wall. Any effect of heat transfer to the drops from a heated wall is handled by the same evaporation models as used for free sprays. The validity of these assumptions has not been put to the test because of lack of detailed experimental data on the effects of a hot wall on drop evaporation and subsequent drop size. However, recent detailed data obtained by Laser Doppler Anemometry by Ozdemir and Whitelaw (1991) will allow some verification of the model's abilities in this regard to be carried out in the near future.

Conclusions

A droplet wall impaction model is proposed that is based on experimental observations. The original drop–drop collision model of O'Rourke and Bracco is modified to allow for the effects of multicollisions between two drop parcels in a thick spray. For all the cases studied, the wall spray radius is generally well predicted, but the wall spray height is underpredicted. The introduction of the extended collision model can potentially produce much more realistic wall spray structure, as shown in 2-D cases, since it reflects the nature of spray dispersion. However, further studies are needed for its application in 3-D cases.

The parametric study for cross-flow cases suggests that the wall impaction model does not consistently predict the same trend of the wall spray height to that of the experiments, although the wall spray radius and its variations are generally reasonably predicted. In addition, the wall impaction model is demonstrated to simulate reasonably the spray wall impaction process inside an engine cylinder.

It has been demonstrated that the present impaction model has features that make it superior to earlier wall spray impaction models for impactions close to the nozzle. For larger distances, approximately greater than 40 mm, the model behaves very similarly to earlier models. The use of the extended collision model in the free spray region should be avoided.

References

Ahmadi-Befrui, B., Gosman, A. D., Issa, R. I., and Watkins, A. P. 1990. EPISO—an implicit non-iterative solution procedure for the calculation of flows in reciprocating engine chambers. *Comp. Meth. Appl. Mech. Eng.*, **79**, 249–279

Alloca, L., Amato, U., Bertoli, C., and Corcione, F. E. 1990. Comparison of models and experiments for diesel fuel sprays. In *Int. Symp. Diagnostics and Modelling of Combustion in I C Engines*, Japan Society of Mechanical Engineers, Kyoto, 255–261

Amsden, A. A., Ramshaw, J. D., O'Rourke, P. J., and Dukowicz, J. K. 1985. KIVA: A computer program for two and three dimensional fluid flows with chemical reactions and fuel sprays. Technical Report LA-10245-MS, Los Alamos National Laboratory, Los Alamos, NM

Aoyagi, Y., Kamimoto, T., Matsui, Y., and Matsuoka, S. 1980. A gas sampling study on the formation processes of soot and NO in a DI diesel engine. SAE Paper 800254

Araki, K. and Moriyama, A. 1982. Deformation behaviour of a liquid droplet impinging on a hot metal surface. In *Int. Conf. on Liquid Atomization and Spraying Systems (ICLASS-82)*, Univ. of Wisconsin

Fujimoto, H., Saito, M., Minoura, A., Katsura, N., Cho, I., and Senda, J. 1988. Characteristics of a diesel spray impinging on a flat wall. In *Int. Conf. on Liquid Atomization and Spraying Systems (ICLASS-88)*, Fuel Society of Japan

Hiroyasu, H., Nishida, K., Min, J. C., Arai, M., and Yoshida, H. 1990. Computed tomographic study on internal structure of a diesel spray impinging on a flat wall. In *Int. Symp. on Diagnostics and Modelling of Combustion in I C Engines*, Japan Society of Mechanical Engineers, Kyoto, 205–210

Issa, R. I. 1986. Solution of the implicitly discretised fluid flow equations by operator splitting. *J. Comp. Phys.*, **62**,(1) 40–65

Jayaratne, D. W. and Mason, B. J. 1964. The coalescence and bouncing of water drops at an air/water interface. *Proc. R. Soc. Lond. A.*, **280**, 545–565

Katsura, N., Saito, M., Senda, J., and Fujimoto, H. 1989. Characteristics of a diesel spray impinging on a flat wall. SAE Paper 890264

Kroeger, C.A. 1986. A neat methanol direct injection combustion system for heavy duty applications. SAE Paper 861169

Kuniyoshi, H., Yamamoto, H., Fujimoto, H., and Sato, G. T. 1980. Investigation of the characteristics of diesel fuel spray (Third Paper: Impinging on a flat plate). *J. Mech. Eng. Soc. Jpn.*, **15**, 57–64

Meurer, J. S. 1956. Evaluation of reaction kinetics eliminates diesel knock. The M combustion system of MAN. *SAE Trans.*, **64**, 250–268

Mirza, R. 1991. Studies of diesel spray interacting with cross-flow and solid boundaries. Ph.D. thesis, Faculty of Technology, University of Manchester, Manchester, UK

Naber, J. D. and Reitz, R. D. 1988. Modelling engine spray/wall impingement. SAE Paper 880107

Naber, J. D., Enright, B., and Farrell, P. 1988. Fuel impingement in a direct injection diesel engine. SAE Paper 881316

O'Rourke, P. J. and Bracco, F. V. 1980. Modelling of drop interactions in thick sprays and a comparison with experiment. In *Stratified Charge Automotive Engines Conf.*, IMechE, London

Ozdemir, I. B. and Whitelaw, J. W. 1991. Impingement of an unsteady two-phase jet on unheated and heated flat plates. Report No TF/91/20, Mechanical Engineering Department, Imperial College, London

Park, K., Wang, D. M., and Watkins, A. P. 1993. A contribution to the design of a novel direct-injection diesel engine combustion system—analysis of pip size. *Appl. Math. Modelling*, **17**, 114–124

Reitz, R. D. 1987. Modelling atomization processes in high-pressure vaporizing sprays. *Atomization Spray Technol.*, **3**, 309–337

Sakane, A., Hamamoto, Y., and Sumimoto, T. 1988. Behaviour of diesel spray impinging on a wall. *J. Mech. Eng. Soc. Jpn.*, **54**, 1861–1865

Wachters, L. H. J. and Westerling, N. A. J. 1966. The heat transfer from a hot wall to impinging water drops in the spheroidal state. *Chem. Eng. Sci.*, **21**, 1047–1056

Wang, D. M. 1992. Modelling spray wall impaction phenomena and combustion processes of diesel engines. Ph.D. thesis, Faculty of Technology, University of Manchester, Manchester, UK

Wang, D. M., Watkins, A. P., and Cant, R. S. In press. 3D diesel engine combustion simulation with a modified EPISO procedure. *Numer. Heat Transfer*

Watkins, A. P. 1989. Three-dimensional modelling of gas flow and sprays in diesel engines. In N. C. Markatos, (ed.), *Computer Simulation of Fluid Flow, Heat and Mass Transfer and Combustion in Reciprocating Engines*, Hemisphere, Washington, DC, 193–237

Watkins, A. P. and Khaleghi, H. 1990a. An ad-hoc procedure to alleviate false diffusion effects in computer codes using discrete droplet models. In *Int. Symp. on Diagnostics and Modelling of Combustion in I C Engines*, Japan Society of Mechanical Engineers, Kyoto, 237–241

Watkins, A. P. and Khaleghi, H. 1990b. Modelling diesel spray evaporation using a non-iterative implicit solution scheme. *Appl. Math. Modelling*, **12**, 468–474

Watkins, A. P. and Wang, D. M. 1990. A new model for diesel spray impaction on walls and comparison with experiment. In *Int. Symp. on Diagnostics and Modelling of Combustion in I C Engines*, Japan Society of Mechanical Engineers, Kyoto, 243–248

Watkins, A. P., Khaleghi, H., and Wang, D. M. 1991. Modelling spray phenomena in direct-injection diesel engines. In *Internal Combustion Engine Research in Universities, Polytechnics and Colleges.* IMechE, C433/008, 131–142