International Journal of Engineering Innovation & Research
Volume 3, Issue 4, ISSN: 2277 – 5668

Investigation of Different of Spray - Wall Impingement on Performance and Emissions in DI Diesel Engines

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***Abstract* – For Prediction of experimental results in diesel engines, have been suggested the different theoretical models for combustion, fuel injection, spray breakup and spray's wall impingement, that the accuracy accurate results comparing to the actual condition, is one of the issues interest to researchers in the simulation of combustion process. In this paper, by theoretical, different regimes of spray's wall impingement have been studied and then, based on experimental observations and calculations, different models of spray's wall impingement are examined. Finally, the best model to simulate the spray's wall impingement at the operating conditions is proposed. The results show that the O'Rourke model are the better ability to predict the decomposition of the fuel spray's droplets and considered all aspects wall impingement and predicts the droplet size produced by penetration and decomposition, correctly. Therefore, it can be used for simulation of spray's wall impingement at engine's operating conditions.**

***Keywords* – Wall Impingement, CFD, Emissions, DI Diesel Engine.**

I. INTRODUCTION

Spray-wall impingement is an important process during mixture formation in direct injection diesel engines and port injection gasoline engines. Usually, two main physical processes are involved spray development and wall film evolution. Both processes may strongly influence combustion efficiency and the formation of pollutants. In a small direct injection diesel engine, the liquid penetration is sometimes longer than the distance between the nozzle tip and the piston cavity wall, especially in engines with low swirl or during cold start. In this case, the spray-wall impingement may cause a significant increase of unburned hydrocarbon and soot emissions, especially if a wall film is formed. On the other hand, if no liquid wall film is generated, it promotes combustion under hot engine conditions, because spray heating and vaporization are intensified by drop shattering, and the large scale gas vortex, which forms in the near-wall region, enhances gas entrainment.

Hence, in diesel, a detailed modeling of spray-wall impingement processes is necessary in order to predict their effects on engine performance and on the formation of pollutants. Because of the widespread use of DI diesel engines and tightening emission standards, it is necessary to will be studied spray-wall impingement processes for the understanding of its overall effect on engine performance.

It is extremely difficult to get information of spray-wall impingement through the experiment. Therefore, computational modeling can be used for this purpose.

II. IMPINGEMENT REGIMES

Diesel spray impinging on a cylinder wall that consists of the set of droplets on a surface, collide sequentially or simultaneously and each droplet collision, feels the impact of neighboring droplets.

The behavior of a droplet at wall interaction depends on several parameters like droplet velocity, diameter, droplet properties, liquid properties such as viscosity, temperature, surface tension, wall properties like surface roughness and temperature, wall film thickness etc.

Figure 1 shows the various impingement regimes of a droplet-wall interaction. In the stick regime, a droplet with low kinetic energy adheres to the wall in nearly spherical form and continues to evaporate. In the case of spread, the droplet impacts with moderate velocity on a dry or wetted wall, spreads out and mixes with the wall film (wetted wall) or forms a wall film (dry wall). If rebound occurs, the droplet bounces off the wall (reflection) and does not break up. This regime is observed in the case of dry and hot walls, where the contact between drop and wall is prevented by a vapor cushion. Rebound also occurs in the case of a wet wall if the impact energy is low and an air film between drop and liquid film minimizes energy loss. In the boiling-induced break-up regime, the droplet disintegrates due to a rapid liquid boiling on a hot wall. The wall temperature must be near the Nakayama temperature T_N, at which a droplet reaches its maximum evaporation rate. In the case of break-up, the droplet deforms into a radial film on the hot surface, which breaks up due to thermo-induced instability. The splash regime occurs at very high impact energy. A crown is formed, jets develop on the periphery of the crown, become unstable and disintegrate into many droplets [1].

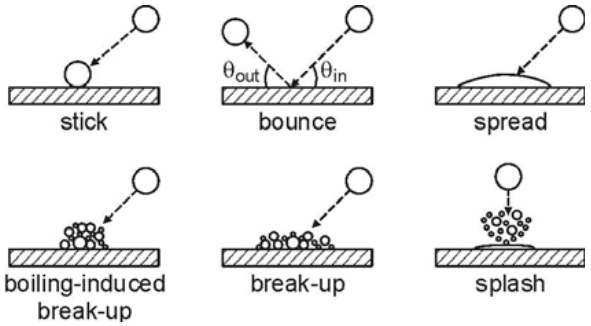


Fig.1. Schematic illustration of different impact regimes [1]

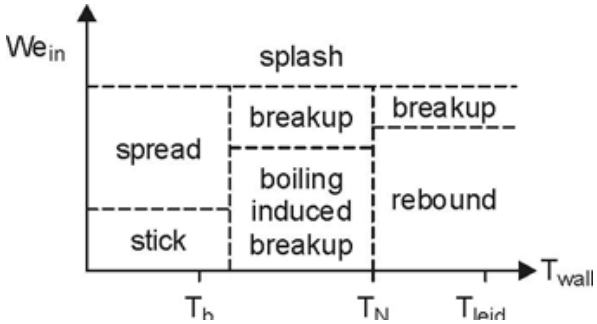


Fig.2. Droplet impingement regimes and transition conditions for a dry wall [1]

There are two most important numbers associated to impingement regimes. Weber number (We),

$$We = \rho_d d_d u_n^2/\sigma \quad (1)$$

which represents the ratio of the droplet's kinetic energy (u_n: velocity component normal to the surface, ρ_d: liquid density, d_d: droplet diameter) and its surface energy, and the Laplace number,

$$La = \rho_d \sigma d_d/\mu_d^2 \quad (2)$$

which measures the relative importance of surface tension and viscous forces acting on the liquid (μ_d dynamic viscosity of liquid). The Laplace number is also represented by the Ohnesorge number $Z = La^{-1/2}$. Another important parameter influencing the impingement process is the wall temperature.

$$T_b < T_N < T_{leid} \quad (3)$$

The characteristic temperatures are the liquid boiling temperature T_b, the Nakayama temperature T_N at which a droplet reaches its maximum evaporation rate, and the Leidenfrost temperature T_{leid} at which a thin layer of vapor forms between the surface and the drop and evaporation is minimized. Figure 2 gives an overview of droplet impingement regimes and transition conditions for a dry wall and fixed Laplace number and surface roughness [2]. In internal combustion engines the wall temperatures during injection are usually below the fuel boiling point [2]. This reduces the number of relevant impingement regimes in case of a dry wall to stick, spread and splash. In the case of a wet wall, Kolpakov et al. [3] revealed that with an increasing impact Weber number the regimes stick, rebound, spread, and splash are important.

III. SPRAY-WALL IMPINGING MODEL

A. *Walljet Model*

This model in principle is based on the spray-wall impingement model of Naber and Reitz. The Walljet model makes the engine working conditions; a vapor layer is formed between the droplet and the wall, and depending on the droplet Weber number drops back or slide on the wall [4].

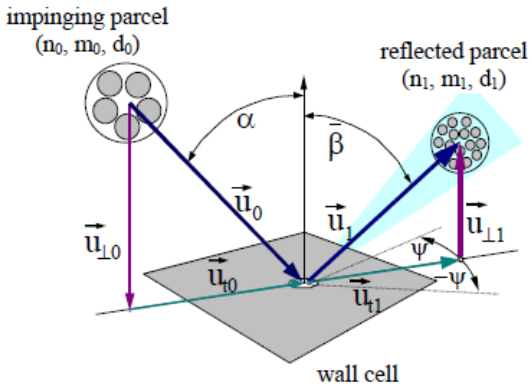


Fig.3. Wall Interaction of Droplets [4]

Weber number is criterion 80. Phenomenon back in less than this number drops rapidly ,but the tangential component of the wall have the vertical velocity component in the previous photo and can change as a function of Weber number drops.

The following empirical relationship between the droplet Weber number and the Weber number drops before dealing relate to:

$$We_{norm,out} = c_1 . We_{norm,in} . e^{-c_2 We_{norm,in}} \quad (4)$$

where C_1 and C_2 as empirical constants respectively 0.687 and 0.04415 were obtained. Given the above, the reflection angle in the range $0 < \beta < 5$ will change [5,6]. The tangential angle ψ on the surface that gets reflected in the changing range of $-180 < \psi < +180$ will be determined. This angle is determined by a probability distribution function:

$$\psi = -\frac{\pi}{k} ln\left[1 - p\left(1 - e^{-k}\right)\right] \quad (5)$$

In this regard, a random number between 0 and 1which can change the parameter k is calculated from the following relationship:

$$\sin\alpha = \left(\frac{e^k + 1}{e^k - 1}\right) \frac{1}{1 + \left(\frac{\pi}{k}\right)^2} \quad (6)$$

After the droplet diameter at different, Weber numbers will change as follows:

$$\begin{cases} We < 50 & d_1 = d_0 \\ 50 \le We \le 300 & d_1 = d_0 \ f(We_{norm,in}) \\ We > 50 & d_1 = 0.2\ d_0 \end{cases} \quad (7)$$

B. Mundo Tropea Sommerfeld Model

This model is based on the experiments of Mundo, Tropea, Sommerfeld [7]. These authors distinguish between two regimes, deposition and splash. In the deposition regime all of the liquid remains on the wall while in the splash regime a part of the droplet is deposited and another part reflected away from the wall. According to the authors the transition from one regime to the other can be described by the splashing parameter K denoted as W_1 in the model parameter list.

$$K = W_1 = \sqrt{We\sqrt{Re}} = OhRe^{1.25} \quad (8)$$

where the Reynolds and the Ohnesorge numbers are calculated as

$$Re = \frac{\rho_1 D_{pi} v_{ni}}{\mu_1} \quad (9)$$

$$Oh = \frac{v_{ni}}{\rho_1 \sigma D_{pi}} \quad (10)$$

Whereas K =W_1 =57.5 is supposed to be independent of the surface roughness there are different equations for a smooth and a rough wall when it comes to determining the amount of mass that is splashed as well as the diameter of the splashed droplets.

$$\text{smooth wall} \quad \frac{m_o}{m_i} = 3.9896_{10}{}^{-21}.W_1{}^{9.2133}$$
$$\frac{d_o}{d_i} = 0.88 - 0.013.W_1{}^{0.8} \quad (11)$$
$$\text{rough wall} \quad \frac{m_o}{m_i} = 8.035_{10}{}^{-11}.W_1{}^{4.1713}$$
$$\frac{d_o}{d_i} = 0.43 - 0.0003.W_1{}^{0.9}$$

with diameter ratio limited by a minimum splashing diameter $W_3 < \frac{d_o}{d_i}$

During the experiments the ratios of the incoming and outgoing normal and tangential velocities have also been measured:

$$\text{smooth wall} \quad \frac{v_{to}}{v_{ti}} = 1.068$$
$$\frac{v_{no}}{v_{ni}} = 0.208 \quad (12)$$
$$\text{rough wall} \quad \frac{v_{to}}{v_{ti}} = 0.965$$
$$\frac{v_{no}}{v_{ni}} = 0.407$$

C. Bai and Gosman Model

This model takes also into account splashing for values greater than the user defined splashing parameter W_1. Further smooth and rough as well as dry and wet conditions can be adjusted by model parameters W_3 and W_4. In this model for a dry wall there are two modes, adhesion and splash [8]. If the Weber number is smaller than Wet, the droplet is deposited on the wall, if it is bigger, the droplet undergoes splashing. Wet is calculated by

$$We_c = A.L\alpha^{-0.18} \quad (13)$$

where, A is a constant that depends on the surface roughness. (smooth A = 5264, rough A = 1322). For a wetted wall three modes are taken into consideration. For rebound Weber number W_2 less than 5 the droplets rebound. The droplet is supposed to behave like a solid particle, and the velocities after interaction are:

$$v_{to} = \frac{5}{7} v_{ni} \quad (14)$$

$$v_{no} = -e v_{ni} \quad (15)$$

$$e = 0.993 - 1.76\theta_i + 1.56\theta_i^2 - 0.49\theta_i^3 \quad (16)$$

In the splashing regime which occurs above splashing parameter K>W_1 =57.5 the mass ratio of the incoming and outgoing droplets is calculated as

$$\text{dry wall} \quad \frac{m_o}{m_i} = 0.2 + 0.6R$$
$$\text{wet wall} \quad \frac{m_o}{m_i} = 0.2 + 0.9R \quad (17)$$

Instead of finding the correct product droplet size this model calculates the total number of product droplets N that are produced by a splashing event

$$N = a_0\left(\frac{We}{We_t} - 1\right) \quad (18)$$

For the constant a_0 the authors suggest a value of 5. The next step is to initialize two product parcels. The number of droplets in the first parcel is sampled randomly from the total number of droplets and the second parcel gets the remainder of droplets. The absolute velocities U_1 and U_2 of the product droplets result from energy conservation considerations and momentum conservation.

D. O'Rourke and Amsden Model

Instead of using the dimensionless parameter K these authors calculate a splash Mach number E [9]. This is assigned to model parameter W_1 with the default value 57.5. This model can be used together with the wallfilm model, if model parameter W_2 is set to 1. For W_2 equal to zero wet conditions can be approximately treated by

prescribing a fixed average wallfilm thickness with model parameter W_3. The transition film thickness between wet and dry conditions is set by model parameter W_4. For the splash Mach number holds

$$E^2 = \frac{\rho_1 D_{pi} v_{ni}}{\sigma} \frac{1}{min\left(\frac{h_0}{D_{pi}},1\right) + \frac{\delta_{b1}}{D_{pi}}} \quad (19)$$

where h_0 is the film thickness and δ_{bl} the boundary layer thickness.

$$\delta_{b1} = \frac{D_{pi}}{\sqrt{Re}} \quad (20)$$

For a dry wall with h_0 equal to 0, E is similar to the splash criterion. The thicker the wall film the more the splashing of the droplets is suppressed as the energy dissipates by viscous damping. The ratio of the incoming to the outgoing mass is defined by

$$\frac{m_o}{m_i} = min\left(1.8.10^{-4}\left(E^2 - E_c^2\right), 0.75\right) \quad (21)$$

The size of the outgoing droplets is sampled from a Nukiyama-Tanasawa distribution. The outgoing normal and tangential velocities of the droplets are also sampled from Nukiyama- Tanasawa distributions. The final tangential droplet direction is calculated by the same method as in the Naber Reitz model.

IV. 3D Modeling of Diesel Engine

For three dimensional modeling of combustion chamber in a diesel engine, at first, one engine cylinder modeled in Solid Works. According to the strategy that is meant to create a mesh in AVL FIRE software, needed to create a surface mesh of the model, which produced by ANSYS ICEM CFD software, when the piston is at TDC. Then surface mesh at this stage, is applied by AVL FIRE software. In the later stages, 3D modeling of engine geometry and creating the moving mesh is done. At this step the boundary surfaces with the appropriate names will be selected for applying boundary conditions. In the case of the engine, the zones on the final mesh are shown in Figure 4.

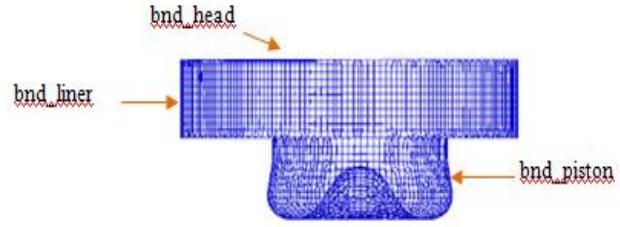


Fig. 4. Computational grid with boundary condition zones.

FIRE software such other action codes to finite volume approach, attempting to discrete the continuity equations of mass, momentum and energy with the turbulence model and then by an iterative algorithm to solve the resulting algebraic equations. However, in this study had been used the Shell model for ignition [10], Eddy breakup model for combustion [11], the Standard k-□ model for turbulence [1], Dukowicz model for heat transfer and evaporation of fuel droplets [12], the extended Zeldovich mechanism for the formation of NO_x emission [13], the Hiroyasu mechanism for the formation of Soot [14]. The main engine specification and operating conditions are represented in Table I.

Table I: Engine Specifications and operating conditions

Bore × Stroke (mm)	100 × 127
Compression ratio	17.5:1
Engine Speed (rpm)	2000
Intake Pressure (kPa)	128
Intake Temperature (k)	320
Start of Injection (deg btdc)	11
Duration of Injection (deg CA)	17
Number of Nozzle orifice diameter (mm)	5 × 0.276
IVC to EVO (deg CA)	240 to 478
Displacement (lit)	3.99
Combustion chamber	Reentrant

V. Results and Discussion

Figure 5 shows predicted in-cylinder pressure for the different spray-wall impingement models for the diesel engine in comparison with experimental data. All the models are in good agreement with the measured data. In particular, the simulation correctly models the time of auto-ignition and the peak pressures in this study. This very good agreement in ignition delay and peak is achieved by time step and computational grid independency of obtained results. The initial pressure rise is also in good agreement with the measured data for all models. As shown in these figures, the difference between predicted in-cylinder pressures in the spray-wall impingement models is significant from 4° crank angle ATDC. According to experimental data, the O'Rourke model predicts in-cylinder peak pressure more accurately than the other models. Only slight difference in predicted peak pressure the other models is due to differences in predicted vaporization rates.

Figures 6 and 7, show the effect of the different spray-wall impingement models on NO_x and Soot pollutant per crankshaft angle. As shown in these figures, the O'Rourke model because of the better estimate for post-impingement characteristics such as rebound velocity, fraction of the mass deposited on the wall, size and velocity distributions of the secondary droplets for the splash regime than the other models has lower soot emission.

Figure 8 shows the interaction of streamline flow and spray droplet at 5 ATDC for the different spray-wall impingement models. As it can be seen, the evolution in piston bowl and cylinder cause to distribute the spray droplet from the center line of it. The further distribution of spray droplet, the better fuel-air mixing and combustion occurred in O'Rourke Model.

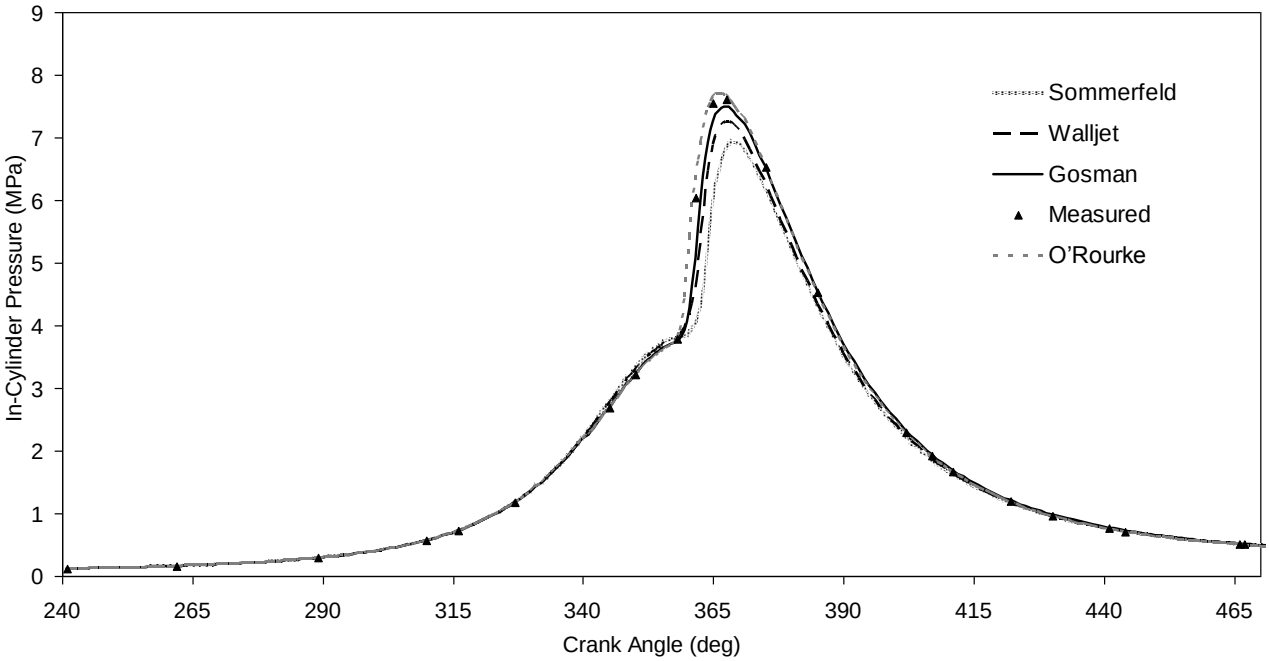


Fig. 5. Predicted in-cylinder pressure for the diesel engine using the different spray-wall impingement models

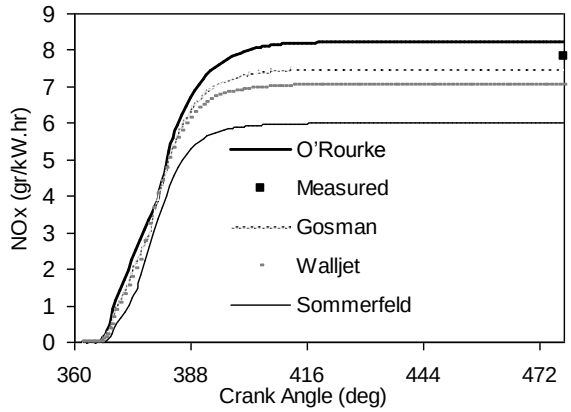


Fig. 6. Predicted NO_x emission using difference models

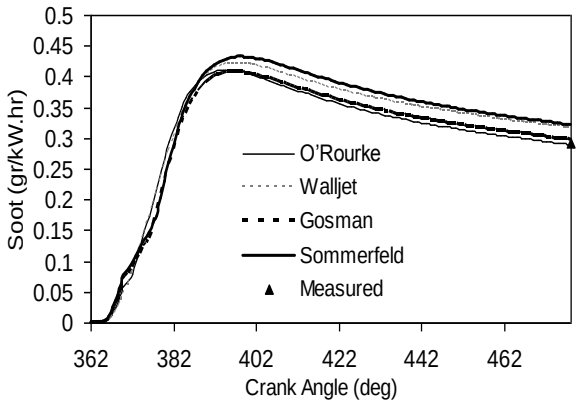
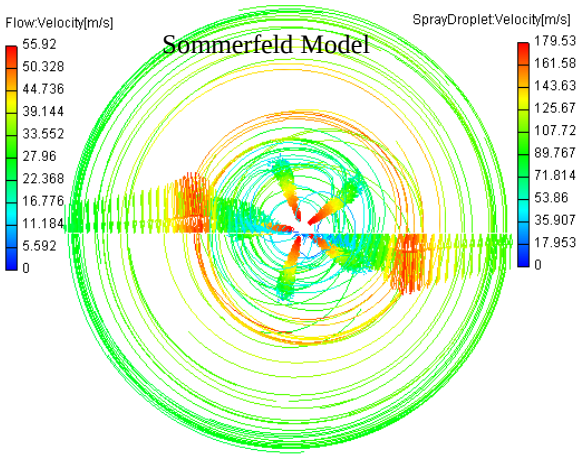
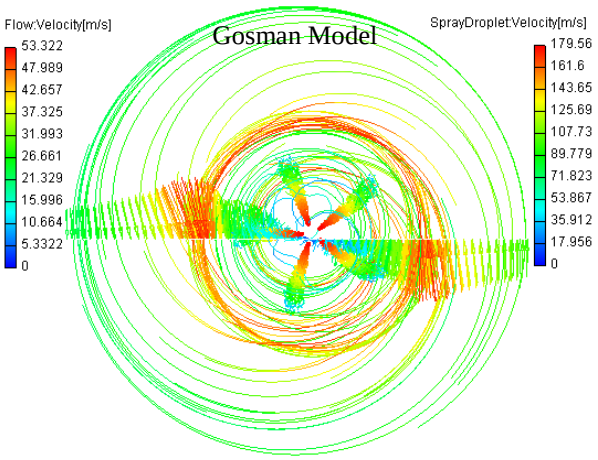


Fig. 7. Predicted Soot emission using difference models





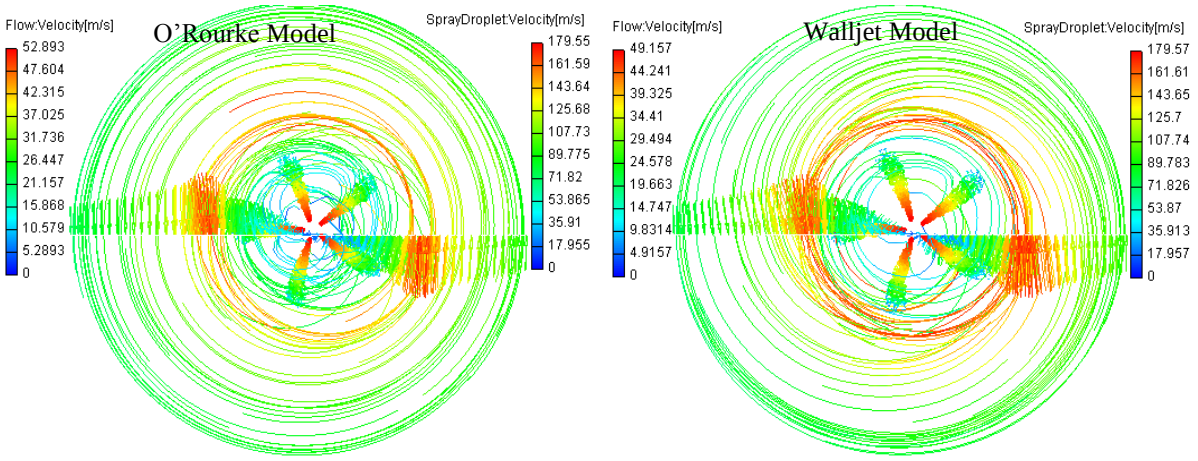


Fig. 8. Interaction between streamline flow and spray droplet at 5 ATDC for difference models

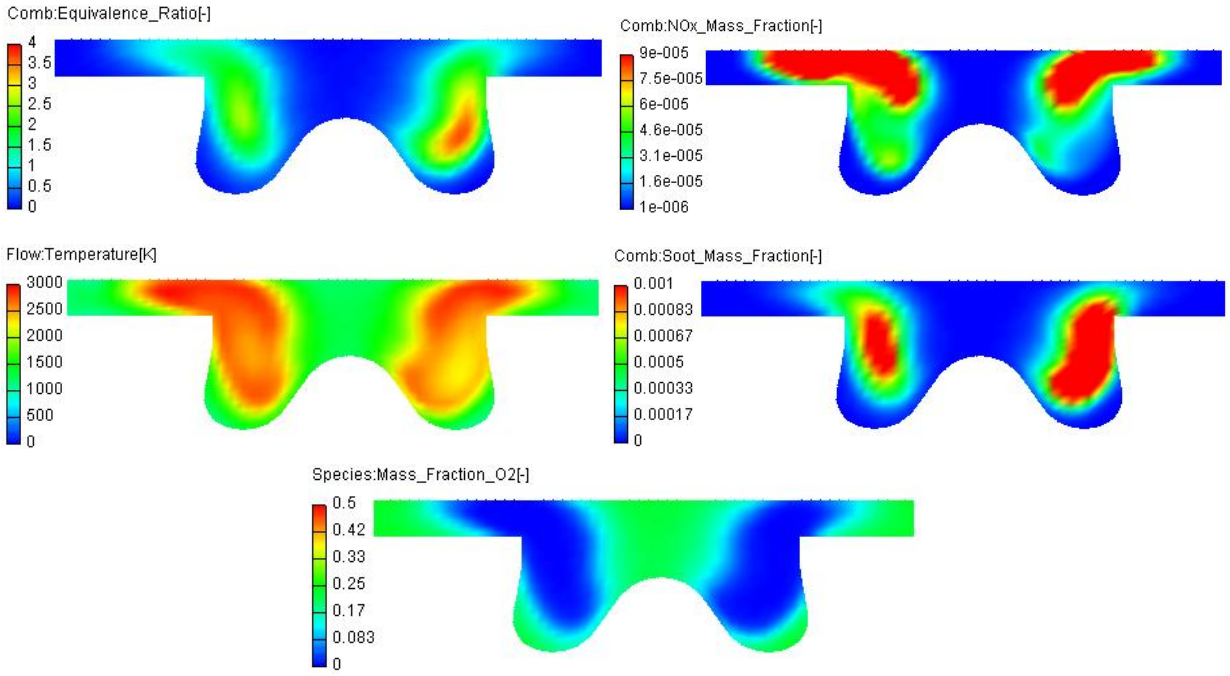


Fig. 9. Quality relevance of numerical results at 20 ATDC for O'Rourke model.

Figure 9 shows quality relevance between O2, NOx, Soot, temperature and equivalence ratio at 20 ATDC for the O'Rourke Model. From this figure, it can be seen that when the local temperature was high and the local equivalence ratio was near to 1 (stoichiometry mixture), the NOx formation was higher than the other zones. It can be observed that soot was formed in rich combustion zones where the oxygen diffusion rate in to combustion zones, was not adequate in order to reach stoichiometry state.

IV. Conclusion

In the present study the spray-wall interaction has been simulated with different spray-wall impingemet models and the effect of these models on DI diesel engine combustion and spray characteristics was investigated. Results were validated and compared with available experimental data for the DI diesel engine for mean cylinder pressure and NOx and soot pollutant. A good agreement between the predicted and experimental values ensures the accuracy of the numerical predictions collected with the present work. From the study on the diesel engine spray-wall impingement using four different models, the following conclusions could be drawn:

1-The Knowledge of the fuel spray-wall impingement mechanism can be a key issue for a successful simulation of all the subsequent processes of mixture formation, and eventually combustion and pollutant formation.

2-All the fuel spray-wall impingement models capture the ignition timing and peak pressures accurately.

3-The Sommerfeld model over predicts the drop sizes and drop mass comparing with predictions obtained from other models.

4- A faster splash and decomposition of big drops is achieved in O'Rourke model, and an increased

evaporation as well as a reduced amount of remaining fuel is calculated allowing a better matching of experimental data.

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