Combustion Modeling for Modern Direct Injection Diesel Engines

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ABSTRACT: *In order to comply with stringent pollutant emissions regulations, a detailed analysis of the engine combustion and emission is required. In this field, computational tools like CFD and engine cycle simulation play a fundamental role. Therefore, the goal of the present work is to simulate a high speed DI diesel engine and study the combustion and major diesel engine emissions with more details, by using the AVL-FIRE commercial CFD code. The predicted values of the in cylinder pressure, heat release rate, emissions, spray penetration and in-cylinder isothermal contour plots by this code are compared with the corresponding experimental data in the literature and is derived good agreement. This agreement makes the model a reliable tool that can use for exploring new engine concepts.*

KEY WORDS: *Diesel engine, Direct injection, CFD, Emission, Combustion, HCCI.*

INTRODUCTION

In the last decade, the performance of engines has been improved, and the consumption and the emissions have been reduced drastically. To deal with this situation, Computational Analysis Engineering (CAE) in general, and CFD and combustion simulation in particular, are playing a more important role in the vehicle and power train development process. Hence, CFD has been successfully established for three dimensional simulation of mixture formation, combustion and pollutant formation in direct injection diesel engines [1, 2]. Limiting our attention to diesel combustion and pollutant formation computations, some of the main literature findings will be briefly reviewed in the following. *Beatrice et al.* [3] studied the combustion and pollution of a DI diesel engine with multi-dimensional CFD code KIVAII. The predicted values by the modeling, showed a good agreement with the experimental data. *Jung & Assanis* [4] investigated the combustion and pollution of a DI diesel engine by the CFD code KIVAII, with multi- zone models and experimental results. *Taklanti & Delhaye* [5] presented an overview of CFD and combustion applications to automotive power train development with several numerical tools, a 1D code and two different 3D CFD codes (FIRE and KIVAII). *McCracken & Abraham* [6] investigated the effects of swirl and spray in DI diesel engines by KIVAIII. *Cantore et al.* [7] studied the combustion, pollution and flow field of a 6 cylinder Diesel engine with KIVAIII and STAR-CD CFD codes.

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The goal of this work is to simulate and investigate the combustion and pollution of a modern six injector holes DI diesel engine with the CFD code.

PROBLEM STATEMENT

The test engine was a single cylinder direct injection diesel engine. The specification and operating conditions of the diesel engine are described in Table 1.

As described, the central injector nozzle, having six holes, 0.17mm in diameter was selected for this work.

The computational mesh was created using AVL ESE Diesel Tool [9]. Details of the computational mesh and spray injection are shown in Fig. 1. The computation used a 360 degree whole mesh with 35 nodes in the radial direction, 20 nodes in the azimuthal direction and 5 nodes in the squish region at top dead center. The ground of the bowl has been meshed with two continuous layers for a proper calculation of the heat transferred through the piston wall. The final mesh consists of a hexahedral dominated mesh. Number of cells in the mesh was about 80,000 and 50,000 at BDC and TDC, respectively. The present resolution was found to give adequately grid independent results.

According to the experimental reference, the governing equations for unsteady, compressible, turbulent flow and thermal fields were solved from 340ºCA to EVO by the commercial CFD code AVL-FIRE [10]. The k-ε model was used for taking the turbulence field into account. More details about the in-cylinder flow in a DI engine with the similar specifications to the related diesel engine have been studied in the reference [11].

Spray and Combustion Models

The standard WAVE model, described in [12] was used for the primary and secondary atomization modeling of the produced droplets. Drop parcels are injected with characteristic size equal to the Nozzle exit diameter (blob injection). The Dukowicz model was applied for treating the heat-up and evaporation of the droplets, which is described in [10, 13]. A stochastic dispersion model was employed to take the effect of interaction between the particles [10]. The spray-wall interaction model used in the simulations was based on the spray-wall impingement model described in [14]. The Shell auto-ignition model was used for modeling of the auto-ignition [10, 15]. The Eddy Break-Up (EBU) model based on the turbulent mixing was used for modeling of the combustion in the bowl [10]. The Zeldovitch mechanism [10] was used for prediction of NOx formation. The net soot formation rate was predicted according to reference [10].

Table 1: Engine Specifications [8].

Engine type:	DI, Single cylinder, 4-Stroke
Bore*Stroke	135mm*140mm
Displacement	2004 cm^3
Cylinders head	2 inlet, 2 exhaust valves
Combustion chamber	98 mm Shallow dish (with nozzle recess)
Compression ratio	16.5:1
Injection pressure	150 Mpa

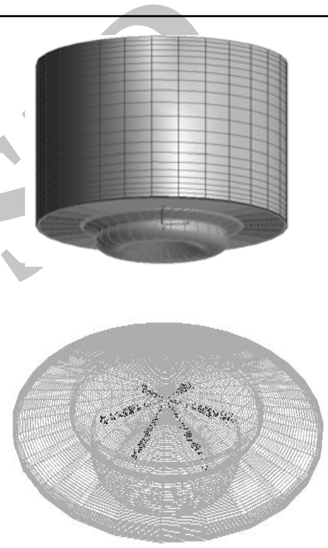

Fig. 1: Computational mesh and fuel spray at two different views.

RESULTS AND DISCUSSION

Fig. 2 shows the mean pressure in cylinder. After start of injection at 363ºCA, ignition delay is 8.333E-4s and premixed combustion causes the increase of cylinder pressure again and maximum pressure reaches to 5.12 Mpa. The good agreement of predicted in-cylinder pressure with the experimental data [8] can be observed.

The fuel injection timing is from 363ºCA to 384ºCA and injection profile is rectangular type (common rail system).

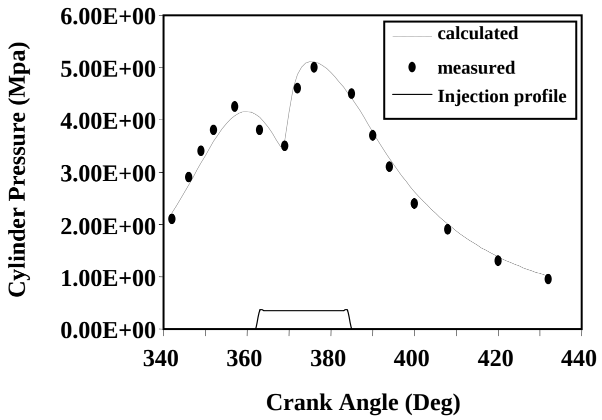

Fig. 2: Comparison of calculated and measured [8] in-cylinder pressure.

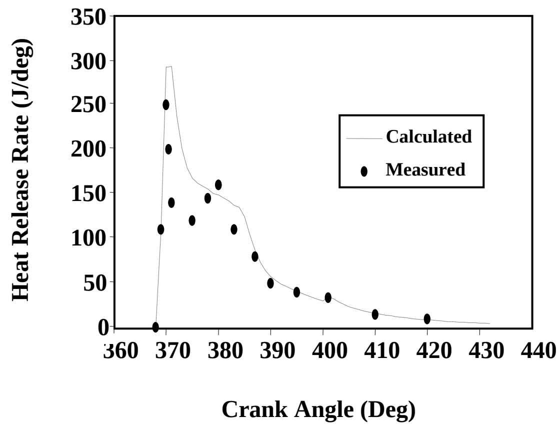

Fig. 3: Comparison of calculated and measured [8] heat release rate.

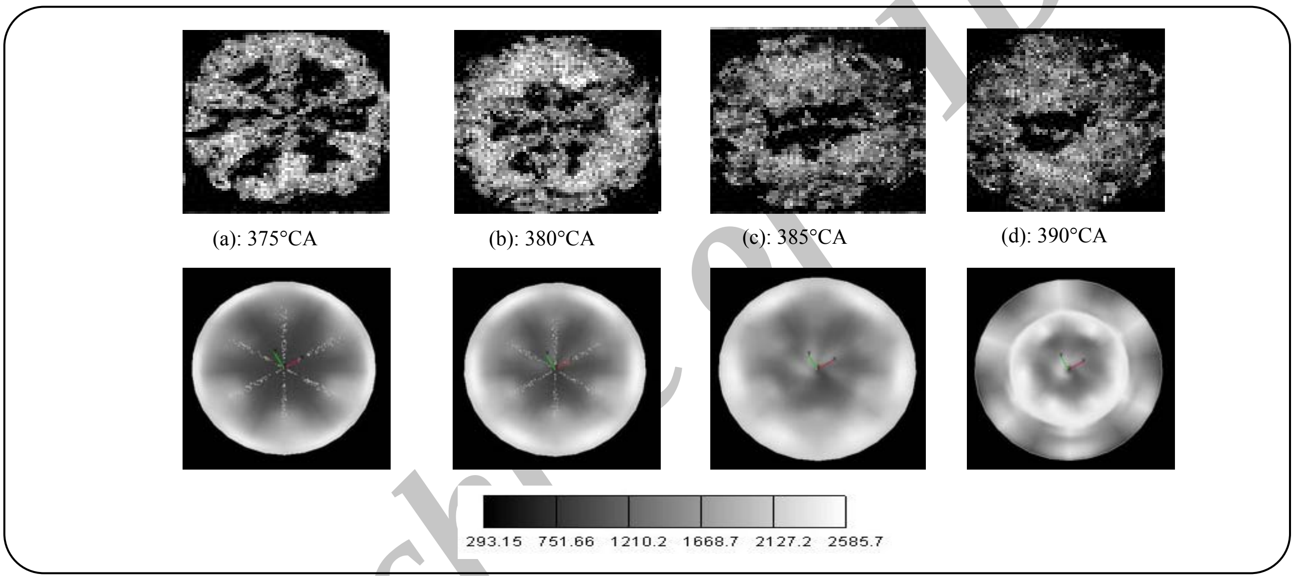

Fig. 4: Comparison of calculated and measured [8] spray penetration and cylinder temperature.

The engine speed fixed at is 1000RPM and the fuel injection quantity is 122mm^3/cycle.

Fig. 3 shows the curve of heat release rate. The first peak which is due to the premixed combustion occurs at 372ºCA. This sudden combustion of well prepared fuel-air mixture result in a strong and sudden increase heat release rate and cylinder pressure. At the present state because of less ignition delay period, the pressure gradient, peak temperature which is response for first production nitric oxides (NOx) and the rate of heat release are decreased. The second peak is the result of the diffusion combustion and occurs at 380ºCA. The good agreement trend of predicted heat release rate with the experimental data [8] also can be observed.

Fig. 4 compares the calculated spray penetration and in-cylinder isothermal contour plots with experimental photographs [8] at different crank angle degrees at a cross-section just above the piston bowl. At 375°CA, It can be seen that fuel sprays from the injector holes are directed towards the bowl of the combustion chamber and premixed combustion starts due to high fuel atomization and evaporation. The rapid increase in temperature due to the stoichiometric combustion can be observed at 380ºCA. At 385ºCA, injection termination can be observed. At 390ºCA, Because of the more availability of oxygen due to the injection termination and mixing, diffusion combustion causes the increase of the temperature towards the center of the cylinder and

Table 2: Comparison of calculated and measured exhaust NOx and soot.

Emission	Calculated	Measured
NOx (ppm)	657	500
Soot (mg/lit)	0.12	0.1

maximum local temperature reaches to 2570K. Spray penetration and combustion photographs at different crank angle degrees show a good agreement with calculated spray penetration and isothermal contour plots.

CONCLUSIONS

1- Calculated spray penetration and in-cylinder isothermal contour plots are compared with experimental photographs at different crank angle degrees and is derived good agreement.

2- Predicted value for average pressure, heat release rate, soot and NOx emissions in cylinder are good agreement with the corresponding experimental data.

3- Model can be predicts exactly start of combustion.

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