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Hybrid RANS-LES Simulation of In-Cylinder Air Flow for Different Engine Speeds at Fixed Intake Flow Pressure

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***Abstract*—**The in-cylinder flow and mixture formations are significant in view of today's increasing concern on environmental issues and stringent emission regulations. In this paper, the numerical simulations of a SI engine at different engine speeds (2000-5000 rpm) at fixed intake flow pressure of 1 bar are studied using the AVL FIRE software. The simulation results show that when the engine speed at fixed intake flow pressure is increased, the volumetric efficiency of the engine decreases. This is due to a richer fuel conditions near the engine cylinder wall when engine speed is increased. Significant effects of impingement are also noted on the upper and side walls of the engine cylinder. These variations in mixture formation before ignition could affect the thermodynamics efficiency and specific fuel consumption that would lead to a reduced engine performance.

***Keywords*—**AVL FIRE, fuel mass, IC engine, LES, RANS, turbulent intensity.

I. Introduction

A. Problem Statement

ESCALATED environmental issues and stringent emission regulations, have driven the development of more efficient Internal Combustion (IC) engines, with better performances and emitting less pollutants. The performances and emission characteristics of IC engines are determined by the complex interactions of the in-cylinder flow, mixture formation, and combustion process [1]. Example of these complex interactions can be seen in the study of [2] , where stoichiometric air-to-fuel ratio around the spark plug, and a leaner conditions close to the cylinder wall are able to increase the engine volumetric efficiency, at the same time reducing the wall heat losses, unburned hydrocarbons and nitrogen oxide formation.

The variations of in-cylinder mixture have been known to affect the cycle-to-cycle combustion variations [3]. In some cases, the study stated that a wide cyclic variation can have a negative impact on the drivability of the vehicle, and lead to increased levels of pollutant emission and fuel consumption. The elimination of cycle-to-cycle variability could lead to a 10% increase in the power output for the same fuel consumption, as suggested in [4].

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Apart from variation of in-cylinder mixture, spark discharged characteristic, inhomogeneity mixture at spark plug vicinity, turbulence intensity, mean flow speed, angle of spark plug vicinity, and overall in-cylinder flow pattern are some of the factors that could cause cycle-to-cycle combustion variations [3]. Therefore an insight understanding of the governing processes of in-cylinder flow, mixture formation and combustion are therefore significant importance in view of today's environmental and regulations concern. The main objectives of this work are to numerically study the distribution of fuel under different engine speeds (2000-5000 rpm) at fixed intake flow pressure of 1 bar using the AVL FIRE software and to investigate the turbulent intensity of the resultant flow.

II. Approach and Methods

A. Species Transport Model

The physically correct description of multi component diffusion becomes important when diffusion is the dominant part of the species transport, and the mixture consists of more than two chemical components [5]. The species transport equations used in AVL to describe the multi-component diffusion for ideal gases are the Maxwell – Stefan equations, as given in (1):

$$\frac{\partial}{\partial t}(\rho y_k) + \frac{\partial}{\partial t}(\rho(U_i - U_{\delta i})y_k) = \frac{\partial}{\partial x_i}\left(\Gamma_{y_k}\frac{\partial y_k}{\partial x_i}\right) + S_{y_k} \tag{1}$$

where y_k represents the mass fraction of an individual chemical species $k$and Γ_{y_k} is as defined in (2).

$$\Gamma_{y_k} = \left(\rho D_{k,m} + \frac{\mu_t}{S_{c_t}}\right) \tag{2}$$

where S_{c_t} is the the turbulent Schmidt and is set to 0.7 as default, and $D_{k,m}$ is the diffusion coefficient of species k in the mixture. The mass source is as given in (3), where $\dot{\mathrm{I}}$ and M_k are reaction rate and molar mass of species k, respectively.

$$S_{y_k} = \dot{\mathrm{I}}.\mathrm{M_k} \tag{3}$$

B. Turbulence Model

An improved version of the $\overline{v^2} - f$ model was developed as in [6], which offers improvement of numerical stability by solving a transport equation for velocity scale ratio $\zeta = \overline{v^2}/k$

instead of the velocity scale $\overline{v^2}$ [5]. The eddy-viscosity equation is obtained in (4).

$$\nu_t = C_\mu \zeta \frac{k^2}{\varepsilon} \tag{4}$$

And the rest of variables from the following set of model as given by (5-7).

$$\rho\frac{Dk}{Dt} = \rho(P_k - \varepsilon) + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] \tag{5}$$

$$\rho\frac{D\varepsilon}{Dt} = \rho\frac{C^*_{\varepsilon 1}P_k - C_{\varepsilon 2}\varepsilon}{T} + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] \tag{6}$$

$$\rho\frac{D\zeta}{Dt} = \rho f - \rho\frac{\zeta}{k}P_k + \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\zeta}\right)\frac{\partial \zeta}{\partial x_j}\right] \tag{7}$$

The elliptical relaxation function f can be represented as in (8), where L and T are the characteristic turbulence length and time scale respectively. AVL FIRE recommended using hybrid wall treatment when applying k-zeta-f turbulence model.

$$f - L^2\frac{\partial^2 f}{\partial {x_k}^2} = \left(C_1 + C_2\frac{P_k}{\zeta}\right)\frac{\left(\frac{2}{3}-\zeta\right)}{T} \tag{8}$$

Further details on the description of the models and techniques can be found in [7].

C. Model Specifications

Table I refers to the specifications of the engine model whilst Fig. 1 and Table II refer to the CFD meshing characteristics.

TABLE I
ENGINE MODEL SPECIFICATIONS

Stroke (mm)	81.4
Bore (mm)	80
Connecting rod (mm)	137
Number of valve	4
Inlet valve open (deg.)	-359
Inlet valve close (deg.)	-139
Exhaust valve open (deg.)	149
Inlet valve light height (mm)	6
Fuel	Gasoline
Equivalence ratio	1 (Stoichiometric mixture)

TABLE II
INDIVIDUAL SURFACES OF THE VOLUME MESH

Selections	Representations
Boundary conditions	
1	BND_Inlet
2	BND_Chamber
3	BND_Piston
4	BND_Intake_ports
5	BND_Intake_valves
6	BND_Liner
7	BND_Outlet
8	BND_Ex_seats
9	BND_In_seats
10	BND_Squish
Dynamic geometry	
3	MOV_Piston
5	MOV_Intake_valves

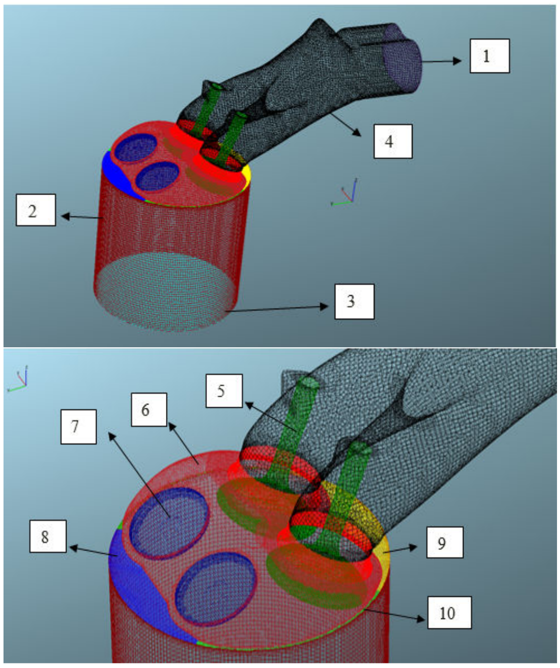

Fig. 1 Volume mesh

III. RESULT AND DISCUSSION

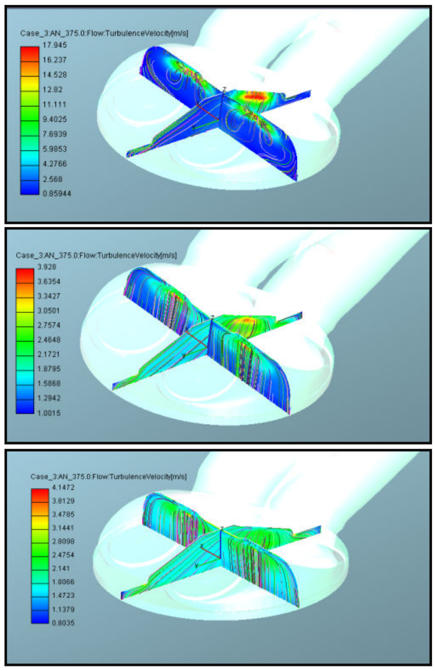

Fig. 2 Turbulent velocity for engine speed of 2000rpm, 4000rpm, and 5000rpm during -345 deg. crank angle intake stroke

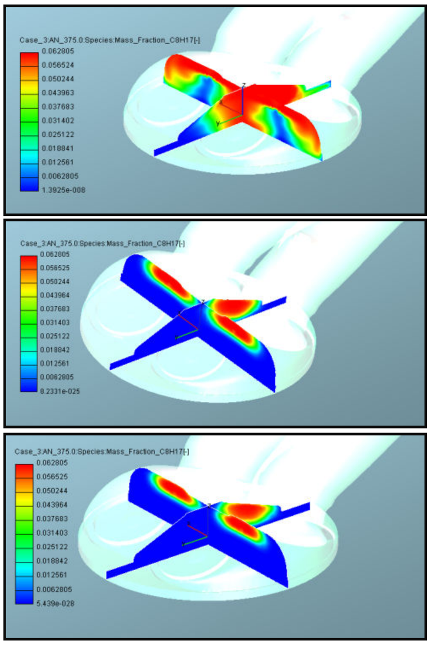

Fig. 3 Mean fuel mass species for engine speed of 2000rpm, 4000rpm, and 5000rpm during -345 deg crank angle intake stroke

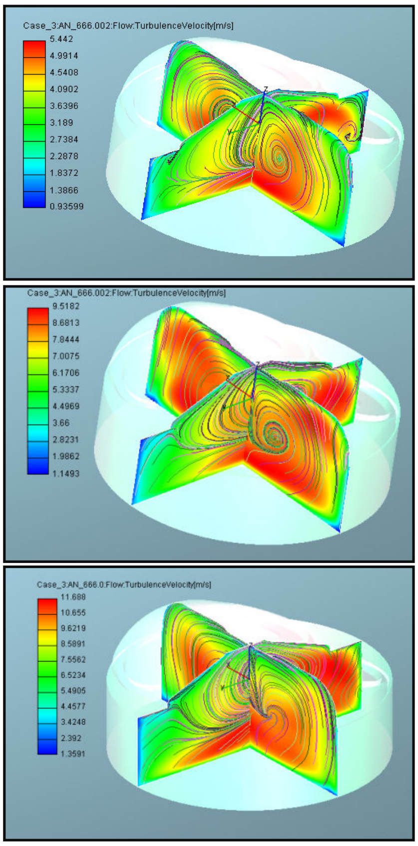

Fig. 4 Turbulent velocity for engine speed of 2000rpm, 4000rpm, and 5000rpm during -54 deg. crank angle compression stroke just before ignition

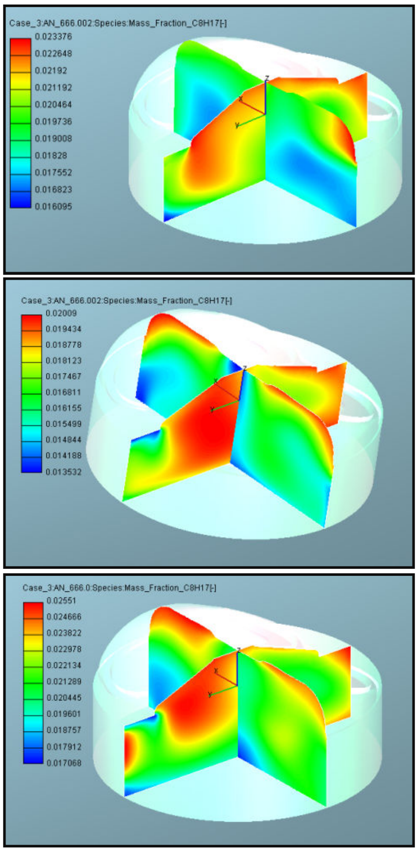

Fig. 5 Mean fuel mass species for engine speed of 2000rpm, 4000rpm, and 5000rpm during -54 deg crank angle compression stroke just before igition

In the numerical investigation, the effects of increasing operating load from 2000 rpm to 5000 rpm at an intake flow pressure of 1 bar on the mixture formation in a SI engine are studied. It is observed that at -345 deg. crank angle intake stroke as shown in Fig. 2, at increasing load, the turbulent velocities are more widely spread in the engine cylinder. The effects of this on the fuel distribution is shown in Fig. 3 that the fuel distribution region becomes smaller. This is due to the more dispersed turbulent velocity, and the smaller difference in flow concentration gradient that may damper the distribution of fuel.

At -54 deg. crank angle (upon nearing the end of the compression stroke), it is observed that increasing the engine speed does not affect much on the distribution of fuel. As the operating load increases, the tumble flow near the vicinity of the spark plug becomes more intense, and therefore creating highly turbulence region (Fig. 4). Fig. 5 shows the fuel is mainly concentrated near the vicinity of the spark plug for engine speed of 2000rpm. However, at higher operating load, more fuel are seen impinged on the surface of the upper and side walls of the engine cylinder, causing an increased wall heat loss, unburned hydrocarbons and nitrogen oxide formations. The finding is consistent with [2].

IV. CONCLUSION

Thus, it can be concluded that increasing the engine speed at fixed intake flow pressure, the volumetric efficiency of the engine decreases. The fuel is not widely spread when the load is increased at a fixed intake flow pressure. This is due to a richer fuel conditions near the engine cylinder wall. Based on the numerical investigation conducted, it be concluded that the hybrid modelling in AVL is able to be used to predict the phenomenon inside an internal combustion engine successfully.

REFERENCES

[1] R. Tatschl, and E. V. Berg. 2002. CFD in IC - Engine Spray and Combustion Simulation - Current status and future development. WCCM V - 5th World Congress on Computational Mechanics.
[2] M. Costa, U. Sorge, and L. Allocca. 2011. Numerical study of the mixture formation process in a four-stroke GDI engine for two-wheel applications. Simulation Modelling Practice and Theory, 19, 1212-1226.
[3] O. Vermorel, S. Richard, O Colin, C. Angelberger, A. Benkenida, and D. Veynante. 2009. Towards the understanding of cyclic variability in a spark ignited engine using multi-cycle LES. Combustion and Flame, 156, 1525-1541.
[4] N. Ozdor, M. Dulger, and E. Sher. 1994. SAE Paper 950683.
[5] Avlfiremanual 2013. edited by AVL LIST GmbH. v2013.1 ed.
[6] K. Hanjalić, M. Popovac, and M. Hadžiabdić. 2004. A robust near-wall elliptic-relaxation eddy-viscosity turbulence model for CFD. International Journal of Heat and Fluid Flow, 25, 1047-1051.
[7] Daniel, LVF, Ulugbek A. and Sharul SD. 2013. Numerical simulation of cycle-to-cycle variations of in-cylinder air flow in naturally aspirated and highly boosted SI engine. Proceedings of the 8th CUTSE International Conference, Curtin University Sarawak, Malaysia.