Inter-cylinder Distribution of Di-Ethyl-Ether Injected into the Intake Manifold of a Diesel Engine Using CFD Simulation

Victor Iorga-Siman, Adrian Clenci, Rodica Niculescu and Alina Trică (Tuţă)

Abstract This paper is a consequence of a study on assessing the cold-starting performance of a compression ignition engine fuelled with different blends of fossil diesel fuel and biodiesel. Through experimental investigations, it was found that the engine starting at −20 °C was no longer possible in the case of using B50 (50 % diesel + 50 % biofuel made from sunflower oil). In order to determine the engine starting in this particular situation, Di-Ethyl-Ether (DEE) was injected into the intake manifold. DEE being a highly flammable substance, the result was a sudden and explosive engine starting, the peak pressure in the monitored cylinder in the first successful engine cycle being almost twice the one which is usually considered as normal. As a consequence of this observation, we wondered what happened in the other 3 engine's cylinders which were not monitored with pressure sensors. Since the cause of the sudden and explosive engine starting was the DEE, our question is in which way the DEE injected into the intake manifold was distributed to each of the 4 cylinders of the engine. Does the extremely high peak of pressure occur in the other 3 cylinders, as well? Since only one cylinder was monitored with a pressure sensor, the method which was used to find the answer to the question mentioned before was to use a CFD approach. Thus, this paper's objective is to present the method used in order to find the inter-cylinder distribution of the injected DEE.

Keywords Biodiesel · Cold start · DEE · DEE inter-cylinder distribution · CFD

Notations

Bx	Biodiesel blend ratio (*i.e. for x = 0, B0, meaning no biodiesel; for x = 100, B100, meaning no diesel fuel*)
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics

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CO	Carbon monoxide
CO_2	Carbon dioxide
DEE	Di-Ethyl-Ether
HC	Unburned hydrocarbons
NOx	Nitric Oxides
Pmax	In-cylinder absolute pressure peak, [bar]
PM	Particulate Matter
PN	Particulate Number

Turbulence model notations

Z	Normalized wall-normal velocity scale
κ	Turbulent Kinetic Energy, TKE [m^2 s^{-2}]
ν	Fluid kinematic viscosity
ω	Turbulent dissipation
f	Elliptic relaxation function
RNG	Re-normalisation Group
SST	Shear Stress Transport
y	Dimensionless wall distance

Introduction

The main challenges of today's automotive transport are first the reduction of pollution (CO, HC, NO_x, PM and PN) and then the reduction of greenhouse emissions (e.g. CO_2 emissions). The reduction of the energetic dependency on petroleum products is another challenge. And finally, the transportation industry needs to find a solution in acceptable economic conditions; therefore biofuels appear to be a good answer.

In 2010, biofuels represented about 3 % of the worldwide energy consumption in the automotive transport. In the future, several scenarios are proposed going from 5 to 15 % in 2025. In Europe 10 % of energy for the transport sector should be renewable in 2020.

The work presented in this paper is part of a larger research program that is running at the University of Pitesti (Niculescu and Clenci 2009–2011). Its purpose is to highlight one of the problems encountered when blending biodiesel with commercial petroleum diesel: the deterioration of the cold starting performance of the compression ignition engine. Worldwide there are many areas where really low sub-zero ambient temperatures are encountered during winter (countries at high latitudes, regions at high altitudes and far from the moderating effect of the open sea). In this case, the engine start time and repeatability become the key performance attributes.

In this context, one goal of the authors was to assess the starting performance at −20 °C of a common automotive compression ignition engine, fuelled with different blends of fossil diesel fuel and biodiesel. Another goal was to determine the biodiesel blend ratio limit at which the engine would not start at −20 °C, and subsequently, to investigate the impact of di-ethyl-ether (DEE) injection into the intake manifold on the engine's start (Clenci et al. 2014a).

Figure 1 presents the results obtained during a starting test at −20 °C. For each of the tests performed (commercial petroleum diesel fuel, B30, B50), the glow plug states were the same i.e. all off, preheat, wait for cranking, cranking, post heat on, post heat finished.

As shown in Fig. 1, the engine did not manage to start with B50. Therefore, as already mentioned, the solution used to help the engine to start was the injection of 150 mg of DEE into the intake duct just before the air filter and before pushing the engine's start button. Concerning the effect of DEE on engine behavior, Fig. 1 shows that two cycles with very high peak pressures (>200 bar!) were recorded. As seen, it was enough to help the engine to start but at the cost of generating extreme mechanical stress. It is for this reason that the actuation of glow plugs is not recommended when using DEE as ignition improver.

As a consequence of this observation, we wondered what happened in the other 3 engine's cylinders which were not monitored with pressure sensors. Since the cause of the sudden and explosive engine starting was the DEE, our question is in which way the DEE injected into the intake manifold was distributed to each of the 4 cylinders of the engine. Does the extremely high peak of pressure occur in the other 3 cylinders, as well? Since only one cylinder was monitored with a pressure sensor, the method which was used to find the answer to the question mentioned

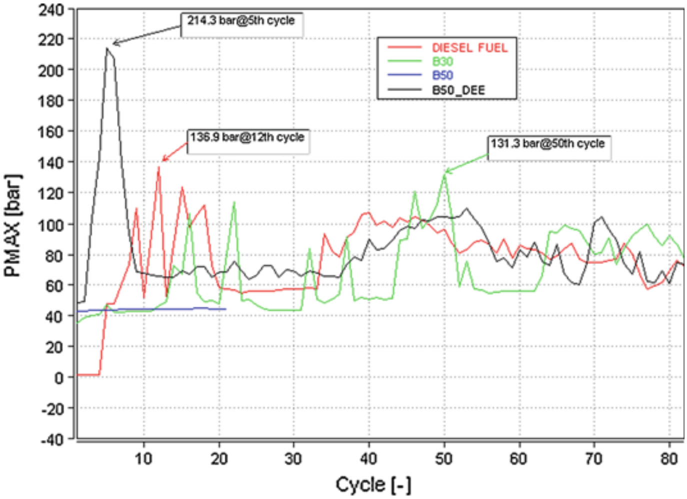


Fig. 1 Pressure peaks cyclic evolution, Clenci et al. (2014a)

before consisted in a CFD approach. Thus, this paper's main objective is to present the method used in order to find the inter-cylinder distribution of the injected DEE.

Our paper is organized in 2 sections, followed by the conclusions drawn from the study and future works. After this first section framing the work in the current context, Sect. 2 presents the Computational Fluid Dynamics (CFD) approach, which has been used in order to find the inter-cylinder distribution of the DEE injected into the intake manifold. This area is divided in two subsections: in the first one, the simulation is described and in the second one, the results are presented and discussed in detail.

Computational Fluid Dynamics Simulation

Computational fluid dynamics, usually abbreviated as CFD, is a branch of fluid mechanics that uses numerical methods and algorithms to solve and analyze problems that involve fluid flows. The fundamental basis of almost all CFD problems are the Navier–Stokes equations, which define any single-phase (gas or liquid, but not both) fluid flow. Despite considerable advances in computer technology and mathematical modeling during the past twenty years, this numerical method only aims to provide approximate results because the exact resolution of the Navier-Stokes equations under specified boundary conditions is still an impossible task. However the numerical approach is a good alternative in fluid flow study.

According to Clenci et al. (2014b), the experimental techniques of fluid flows investigation are able to provide high quality results (even the spatial structure and the temporal revolution of the velocity field) but require good optical access for large fields of view, high speed photography, innovative data analysis methods, and state-of-the-art equipment, which makes them quite expensive. Performing flow measurements in an engine can therefore be difficult because of the complexity of the equipment involved. The advantage of numerical investigations is that an expensive and time-consuming measurement set-up is not necessary. Thanks to the increasing power of computers, nowadays the processes occurring in an internal combustion engine can be modeled more and more accurately and simulated faster. One may note, however, that even the numerical simulations need significant computational cost.

In the current study, the CFD simulation was performed AVL-Fire® 2013 software. FIRE® is a powerful multi-purpose thermo-fluid dynamics software with a particular focus on handling fluid flow applications related to internal combustion engines and powertrains.

The details of the computer used to simulate the flow phenomena of the air-DEE mixture flow into the intake manifold of our engine are presented in Table 1.

As mentioned before, the aim of the simulations is to have an idea of the inter-cylinder distribution of the injected DEE; thus, to extrapolate on the effect of DEE in each of the engine's 4 cylinders.

Table 1 Specification of used computer

System	Manufacturer	Dell
	Type	X64
OS		Windows 7.6.1
Processor	Type	Intel Xeon X5650
	Speed	2.67 GHz
	# CPU(s)	6
RAM		24 GB

Table 2 Technical characteristics of the engine

Characteristics	Values
# Cylinders	4
Total capacity (cm^3)	1461
Volumetric compression ratio	17.9
Max. power (kW) @ speed (rpm)	48 @ 4000
Max. torque (Nm) @ speed (rpm)	160 @ 1700
Number of injector holes/hole diameter (mm)	5/0.15
Injection pressure (bar)	100–1400

The technical characteristics of the engine are presented in Table 2. Some of these technical characteristics are used in the AVL Fire tool calibration.

Mesh Generation and Simulation Description

The intake manifold has been previously designed using commercial CAD software (CATIA)—Fig. 2. As seen from this figure, there is one inlet that corresponds to the injection surface. In reality, the DEE is injected before this surface, actually, at the entrance in the air filter. Thus, we considered the whole inlet surface as the injection

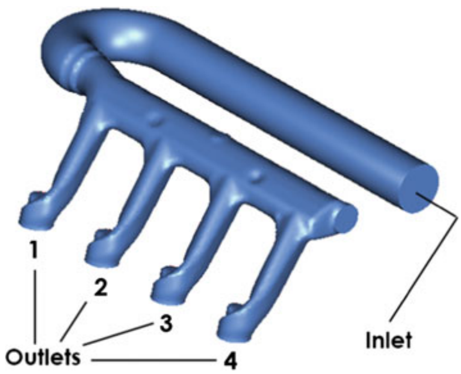


Fig. 2 Description of the volume—Inlet/outlets

surface in order to obtain the largest cloud of DEE. The outlets correspond to the intake of the four cylinders and are numbered from 1 to 4 (the first being the nearest to the bend of the intake manifold, and the fourth, the farthest).

The first step is to mesh the computational domain. In order to study the effect of the meshing on the results, two different meshes have been made and analyzed: a coarse mesh, containing 96,863 cells and a fine mesh of 351,000 cells. For both cases, the structured grid with hexahedral shape cells was used. The numerical results are fairly close. However, the computational time and the average time per step are very different (Fig. 3). Generally, the mesh is a trade-off resulted from the need to obtain good results in reasonable simulation time.

Several selections in addition to inlet/outlets are required to study the inter-cylinder distribution of DEE. Indeed, the software is not able to calculate the mass fraction of DEE in a surface selection because the additional formula accounts for the volume of the cells. So, four cells selections have been added (Fig. 4): the finer the meshing is, the smaller these four volumes are.

The chosen turbulence model is the $\kappa-\zeta-f$ recently developed by Hanjalic et al. (2004). It is a robust modification of the elliptic relaxation model. The aim is to improve numerical stability of the original $\overline{v}^2-f$ model by proposing an eddy viscosity model, which solves a transport equation for the normalized wall-normal velocity scale $\zeta=\overline{v}^2/k$ instead of $\overline{v}^2$. This turbulence variable (ζ) can be regarded as the ratio of the two time scales: scalar k/ε (isotropic), and lateral $\overline{v}^2/k$ (anisotropic). It also introduces a more robust wall boundary condition for f equation, this time f_{wall} is proportional to $1/y^2$ (y is a dimensionless wall distance) instead of $1/y^4$ in the original $\overline{v}^2-f$ model.

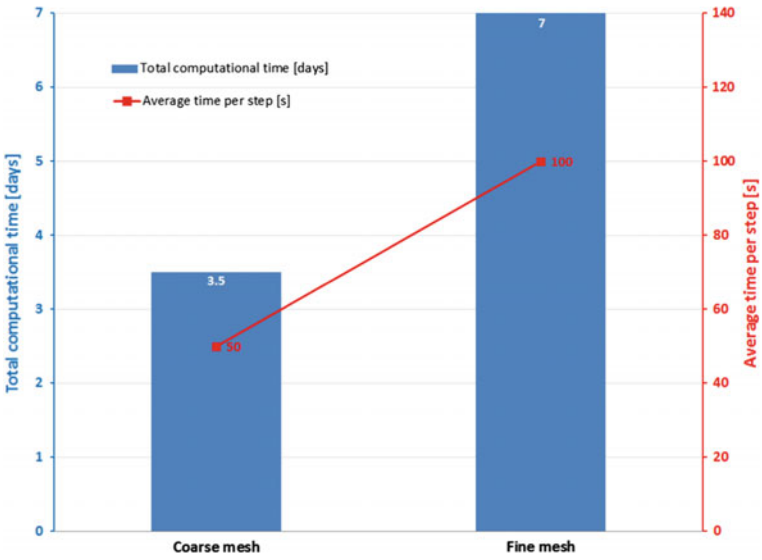
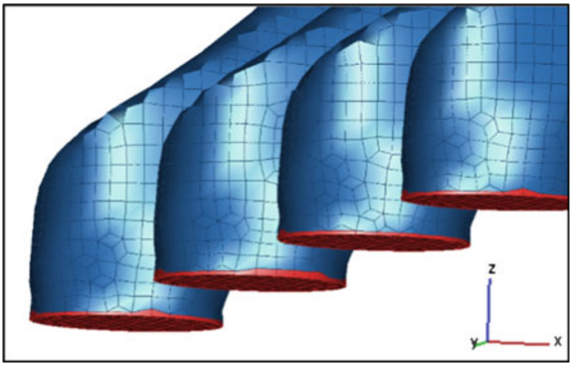


Fig. 3 Effect of the mesh on computational time

Fig. 4 Description of the additional selections



Since there are numerous turbulence modeling approaches (e.g. *k–ω* model with its two declinations: *k–ω standard* and *k–ω SST* and *k–ε* model with its three declinations: *k–ε standard*, *realizable k–ε* and *k–ε RNG*), in order to sustain the one chosen, several CFD simulations should have been carried out to see which the best

Table 3 Details of simulation

	Parameters	Values	Comments
Run mode	Run mode	Time step	–
	Calculation time step, Δt [ms]	0.5	–
	End time [s]	3.0	The equivalent of 15 cycles @ 600 rpm (based on experimental results, it is expected that DEE to be consumed in this time)
Boundary conditions	Inlet/outlets/wall temperature [°C]	−20	Based on the experimental conditions
	Inlet static pressure [Pa]	100,000	Based on the experimental conditions
	Outlets mass flow [g/s]	2.3	Based on an engine speed of 600 rpm
Spray	Type of fluid	DEE	DEE was introduced in fire's database by user defined functions
	Fluid temperature [°C]	−20	Based on the experimental conditions
	Injected mass [mg]	150	Based on the experimental conditions
	Injection duration [s]	1	Based on the experimental conditions
	Velocity [m/s]	0.5	Based on the experimental conditions
	Number of cloud particles per second	200,000	Arbitrary value

is. However, taking into account the long computational time (Fig. 3) and the novelty of the $\kappa-\zeta-f$ model, we have decided for this stage to not explore the results of other turbulence models.

As for the boundary conditions, spray details and setup of the simulation, Table 3 presents the data.

Concerning the DEE spray, it is underlined that it is injected in a static environment for 1 s. After this injection, the flow is activated by imposing the mass flow boundary condition through the outlets based on the timing resulted from operating the engine at 600 rpm.

Simulation Results. Discussion

This section will present only the inter-cylinder distribution of DEE obtained when using the fine mesh of 351,000 cells and the 1–3–4–2 cylinder filling order.

The CFD software is able to provide numerous 2D and 3D results. For instance, Fig. 5 present the life time of spray droplets and the mass fraction of the gaseous DEE obtained at certain moments in time.

In the above pictures, the four circles corresponding to the four cylinders, represent the opening and closing of the intake valves, i.e. when a circle is black the valves are closed (the air-DEE mix is not sucked into the cylinder), while when a circle is white, the valves are opened (the corresponding cylinder sucks the air-DEE mix). These results are presented just to show an example of post-processing.

Detailed results concerning the DEE inter-cylinder distribution were obtained by using the four additional cells selections created to calculate the mass fraction of DEE passing through each of the intake manifold's outlets (Fig. 4). Thus, Fig. 6 presents the percentage of the evaporated DEE found in each of the 4 cylinders, starting from the 1st cycle till the 10th cycle.

In the figure above, only 10 cycles were taken from the 15 simulated due to the fact that in the 11th engine cycle there is almost no more DEE in the intake manifold.

Now, by summing up the DEE quantities found in each cylinder after these first 10 cycles, one may obtain the DEE inter-cylinder distribution during the duration of the 10 engine cycles (Fig. 7).

By analyzing Figs. 6 and 7, it is obvious that cylinder number 1 (this is the one that was monitored with pressure sensor during the cold starting tests) is absorbing more DEE compared to the others 3 engine's cylinders.

Thus, coming back to the goal of this study (*What happened in the other 3 engine's cylinders which were not monitored with pressure sensors? Does the extremely high peak of pressure occur in the other 3 cylinders, as well?*), by extrapolating the results, one may say that the extremely high pressure peak of 214.3 bar (see Fig. 1) did not occur in the other 3 cylinders. Certainly, higher pressure peaks than the ones obtained with diesel fuel occurred when DEE was used but apparently not at the extent of the one recorded in the 1st cylinder. For

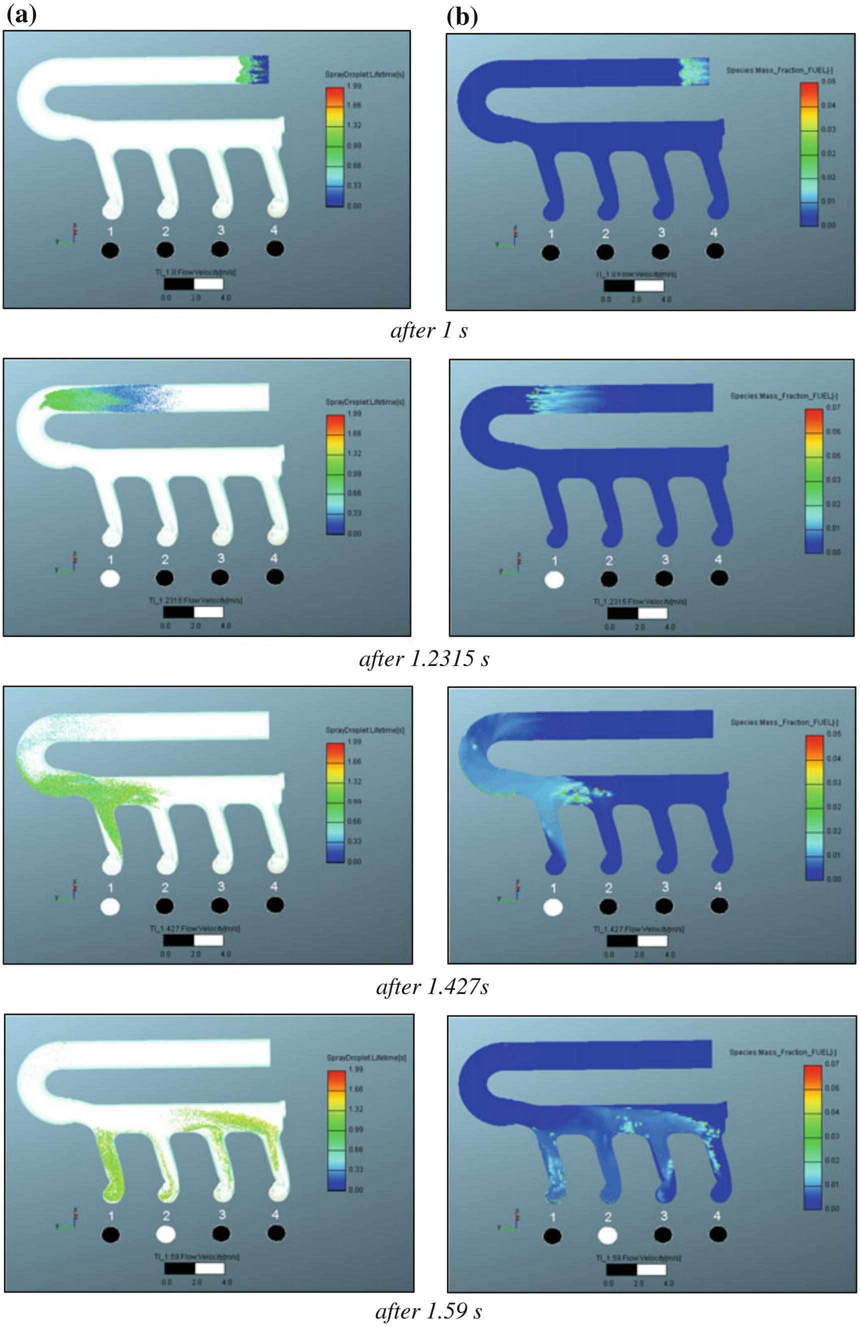


Fig. 5 **a** Spray droplets lifetime. **b** Mass fraction of gaseous DEE

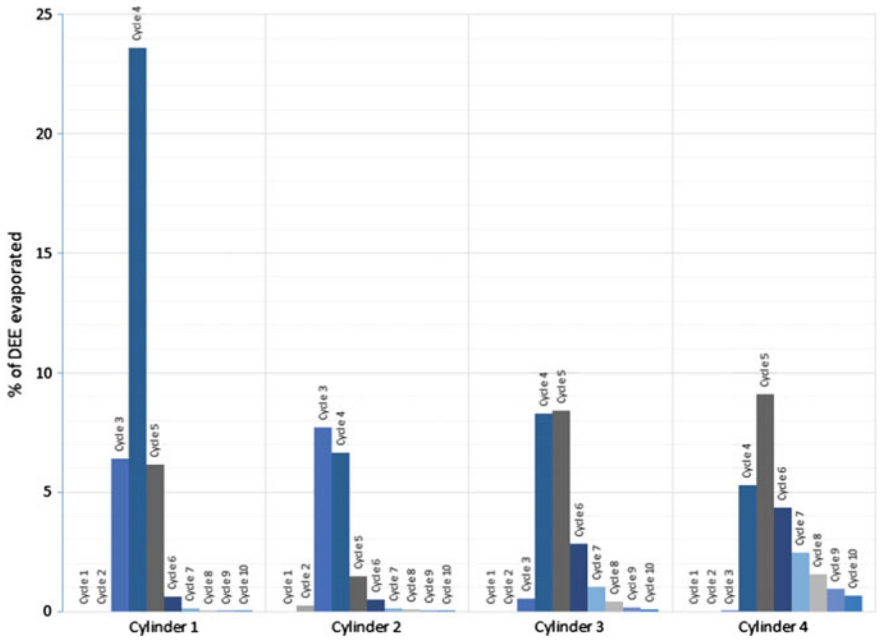


Fig. 6 DEE absorption per cylinder and engine cycle

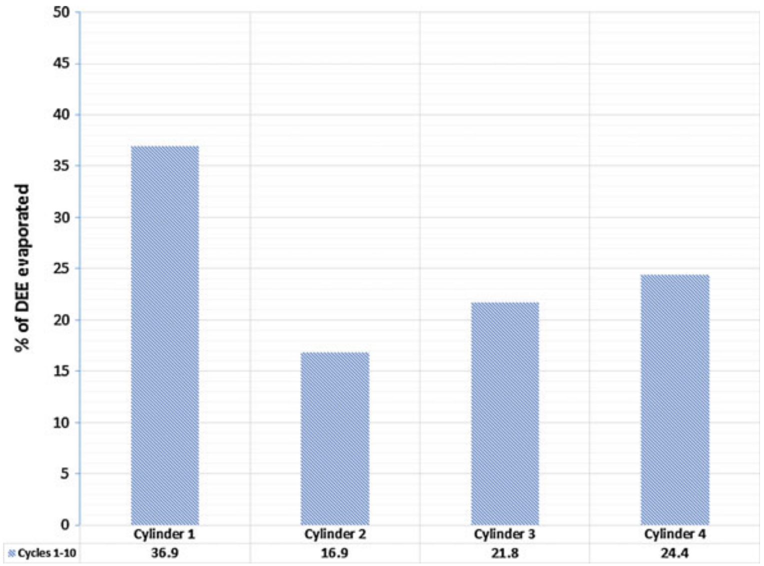


Fig. 7 Total absorption of DEE per cylinder and during 10 cycles

instance, if one associates the 4th engine cycle from the 1st cylinder (see Fig. 6) with the extremely high pressure peak mentioned before, then, the conclusion is that this kind of peak values does not occur in any of the other cylinders or cycles.

Conclusions and Future Works

The CFD simulation helped to understand what would be the impact in each cylinder of the DEE which was used as an ignition improver for engine cold starting.

The simulation showed that the 1st cylinder is taking the highest quantity of DEE, therefore, this represents the worst case scenario regarding in-cylinder's peak pressures.

This kind of simulation should be enough to determine the needed quantity of DEE able to still start the engine when using B50 (for instance) but on the condition of respecting the maximum allowable cylinder pressure capable of ensuring the wanted engine's reliability.

For the validation of the simulation's results, the authors will soon perform once again the cold starting tests, this time by using in-cylinder pressure sensors for each of the 4 cylinders. Obviously, the validation will be a qualitative one since the pressure peaks from each cylinder should be in the same relation as the DEE inter-cylinder distribution shown in Figs. 6 and 7.

The mesh type and the turbulence model are amongst the key points of the CFD simulation. Usually, in order to be sure that the converged solution is trustworthy, the numerical solution must be independent of the mesh size and turbulence model. Therefore, in order to accomplish this goal the authors are currently running other simulations on this particular topic.

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