Cristian Andreescu
Adrian Clenci *Editors*

Proceedings of the European Automotive Congress EAEC-ESFA 2015

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Editors
Cristian Andreescu
University Politehnica of Bucharest
Bucharest
Romania

Adrian Clenci
University of Piteşti
Piteşti
Romania

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Foreword

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The European Automotive Congress was thought to be a common scientific event of the traditional biannual European Automotive Engineers Cooperation (EAEC) Congress (this year at its 14th edition) and the Annual International Conference of the Society of Automotive Engineers of Romania (SIAR)—ESFA.

The Congress was organized by the SIAR and the Automotive Engineering Department of the University “Politehnica” of Bucharest in cooperation with EAEC under the patronage of Fédération Internationale des Sociétés d’Ingénieurs de Techniques d’Automobile (FISITA).

The motto of the Congress (*Academia, Industry and Government: together for automotive engineering development*) was an indication of the organizers belief that there is a need for meetings and discussions about the current and future challenges of the automotive world in order to find the best solutions. In other words, we are convinced that the challenges of the future can only be overcome if these tripartite discussions occur permanently.

The papers included in this volume are selected by the Scientific Committee among the almost 130 technical papers proposed to be presented at the Congress.

The authors of these articles are experts from research, industry, and universities coming from 14 countries among which nine from Europe. Their papers are covering the latest issues such as fuel economy and environment, automotive safety and comfort, automotive reliability and maintenance, new materials and technologies, traffic and road transport systems, advanced engineering methods and tools, advanced powertrains, and hybrid and electric drives.

Therefore, we hope these papers will generate fruitful discussions about an exciting topic: the automotive engineering in the light of the future challenges.

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EAEC-ESFA 2015 Congress Chairman

Assoc. Prof. Adrian Clenci
SIAR President

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Prof. Nicolae Vasiliu, University "Politehnica" of Bucharest, Romania
Dr. Ludwig G.E. Vollrath, FISITA Vice President, Europe

Contents

Abbreviations

AEM	Advanced engineering methods and tools (CAD-CAM-CAE)
APT	Advanced power trains (engine and transmission)
ARM	Automotive reliability and maintenance
FEP	Fuel economy and pollution control
HEV	Hybrid and electric vehicles
MAT	New materials and technologies (lightweight solutions)
NVH	Noise vibration and harshness (NVH)
TTS	Traffic and road transport systems (including automated and cooperative driving and future sustainable mobility concepts)
VDS	Vehicle dynamics safety and comfort

Investigation of a Mechanism Through a Transient Thermal Analysis and an Equivalent Steady-State Thermal Analysis

Alexandra-Raluca Moisescu, Stefan Sorohan and Gabriel Anghelache

Abstract The thermal analysis of a rotary engine mechanism requires taking into consideration the transfer of heat from the combustion gas to the engine parts, which include rotating parts and fixed parts, as well as the transfer of heat to the environment. During an engine mechanism rotation, the conditions of convective heat transfer are variable, and the surfaces of fixed parts exposed to combustion gas are continuously changing. In this case, the transient thermal analysis using the finite elements method is very complex because of the permanent modification of surfaces covered with combustion gas, as a consequence of mechanism rotation. Therefore, in the current paper, an equivalent model for steady-state thermal analysis is developed, so that the same results are obtained as in the long transient thermal analysis, but with significantly smaller requirements of time and computational resources. The transient thermal analysis performed for a large number of rotations, which provides the stationary thermal conditions of mechanism parts, is compared with the equivalent steady-state thermal analysis performed using the equivalent film coefficients and the equivalent convection temperatures. The distributions of fixed part temperature and heat flux obtained from the steady-state thermal analysis are compared to those obtained from the transient thermal analysis, and very good similarities are ascertained. In conclusion, the equivalent steady-state thermal analysis provides similar results, compared with the transient thermal analysis, but with significantly lower computational effort.

Keywords Finite elements · Thermal analysis · Heat transfer · Film coefficient · Temperature

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A.-R. Moisescu (✉) · S. Sorohan · G. Anghelache
University Politehnica of Bucharest, Splaiul Independentei 313, 060042 Bucharest, Romania
e-mail: raluca.moisescu@upb.ro

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Introduction

The development of a rotary engine involves simulation, performed using the finite element method, of mechanism behaviour under thermal and mechanical loads during engine cycles. The simulation provides the temperature distribution of engine parts, so that thermal expansion can be investigated in conjunction with mechanical loads applied on engine mechanism parts.

The thermal analysis requires taking into consideration the transfer of heat from the combustion gas to the engine parts, which include rotating parts and fixed parts, as well as the transfer of heat to the environment (Heywood 1988). Both the rotating and the fixed parts are exposed to combustion gas over an interval of the mechanism rotation. Throughout this interval, the conditions of convective heat transfer are variable, and the surfaces of fixed parts exposed to combustion gas are continuously changing. In this case, the transient thermal analysis using the finite elements method (Kurowski 2004) is very complex because of the permanent modification of surfaces covered with combustion gas, as a consequence of mechanism rotation. Furthermore, the transient thermal analysis needs a large number of engine cycles to be performed, in view of obtaining quasi-constant temperatures for each point of the mechanism (Reddy and Gartling 2010).

Therefore, an equivalent model for steady-state thermal analysis has to be developed, so that the same results are obtained as in the transient thermal analysis, but with significantly smaller requirements of time and computational resources. The temperature distribution from this steady-state thermal analysis must correspond to the results from the transient thermal analysis, which provides the stationary thermal conditions of mechanism parts.

Mechanism Model for Transient Thermal Analysis

The hypothetical mechanism, shown in Fig. 1, consists of a cylindrical rotor inside a fixed part. The 300 mm diameter rotor has a pocket on 90° of its circumference. The fixed part has been modelled as a cylindrical ring with 300 mm inner diameter and 330 mm outer diameter. Both rotor and fixed part are made of aluminium, with the following properties: density 2700 kg/m^3, thermal conductivity 200 W/(m K), specific heat capacity 800 (W s)/(kg K). The rotor angular speed corresponds to rated engine speed 60 min^{-1}, and each mechanism rotation corresponds to an entire engine cycle.

Heat is generated in the cavity corresponding to the rotor pocket (considered combustion chamber), on an angular interval between 0° and 90° of mechanism rotation, as shown in Fig. 2. Throughout the interval of combustion process, convective heat transfer of combustion gas (with film coefficient 300 W/(m^2 K) and temperature 800 °C) towards the fixed part is considered on the inner surface corresponding to the rotor pocket, and neglected on the rest of the fixed part inner surface. Throughout the remaining interval of mechanism rotation, heat transfer is

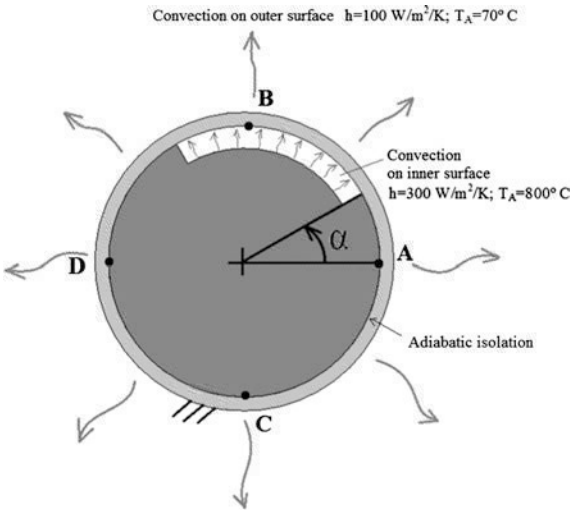

Fig. 1 Model for thermal analysis of a simplified rotary engine mechanism

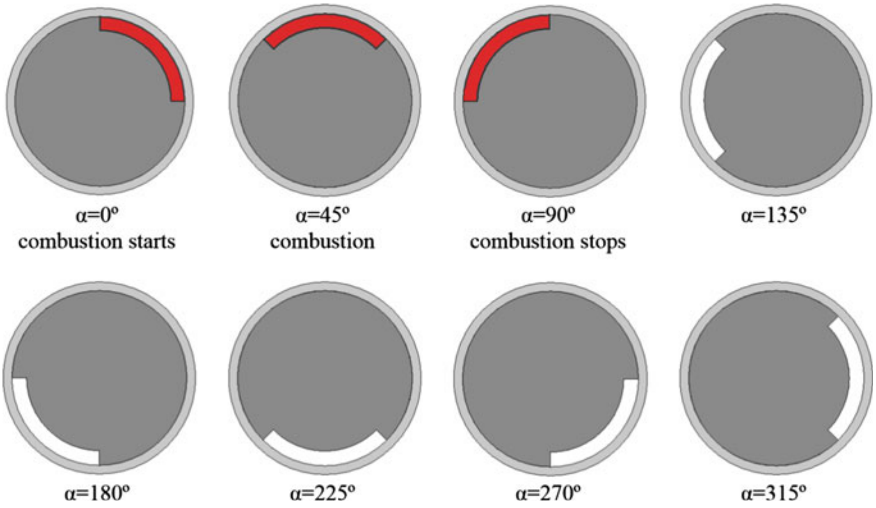

Fig. 2 Combustion phase with respect to rotor position throughout mechanism rotation

neglected on the entire inner surface of the fixed part. Convective heat transfer (with film coefficient 100 W/(m^2 K) and temperature 70 °C) is considered constant on the outer surface of fixed part.

The transient thermal analysis is performed for 1000 rotations, in view of obtaining quasi-constant temperatures for each point of the mechanism after many

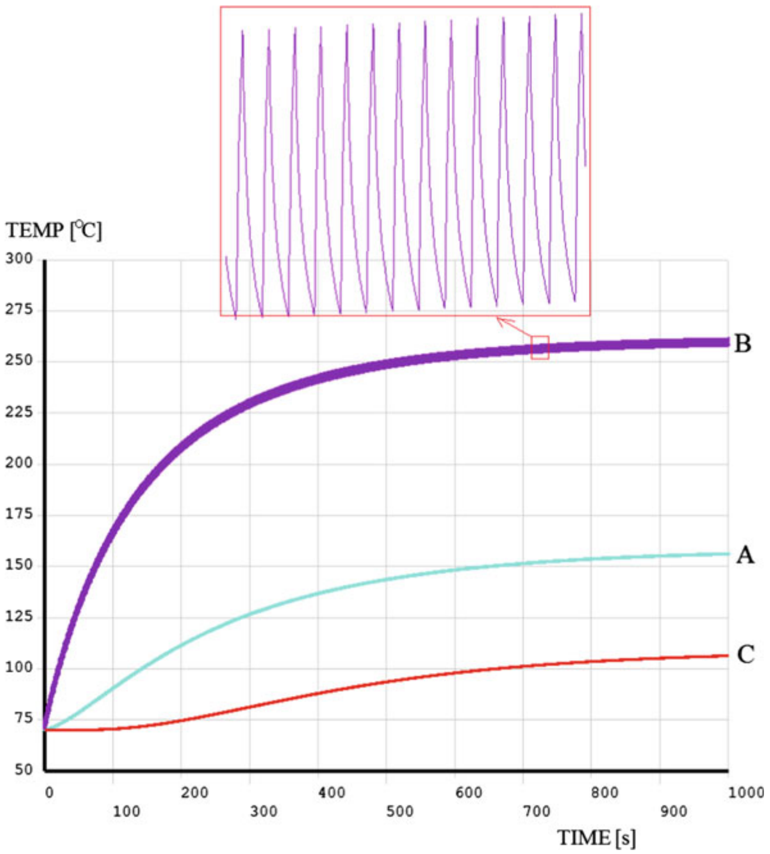

Fig. 3 Evolutions of temperatures during transient thermal analysis

engine cycles. The evolutions of temperatures for points A, B, and C of the fixed part during this analysis are shown in Fig. 3. It can be noticed that point B has pulsating temperature variation, but all three points tend to stabilize to quasi-constant temperature values.

Equivalent Mechanism Model for Steady-State Thermal Analysis

As the transient thermal analysis requires extended time and computing resources, an equivalent model is necessary for steady-state thermal analysis.

The heat flux of a point or an area element located on the fixed part inner surface is defined as an average considering its variation over an entire mechanism rotation:

$$q_m = h_m(T_0 - T_m) = \frac{\int_0^{360} q(\alpha)d\alpha}{360^\circ}, \tag{1}$$

where

$q_m\left[\frac{\mathrm{W}}{\mathrm{m}^2}\right]$	is the average heat flux of the point over an entire rotation,
$h_m\left[\frac{\mathrm{W}}{\mathrm{m}^2\,\mathrm{K}}\right]$	is the equivalent film coefficient for the point over an entire rotation,
$T_m[\mathrm{K}]$	is the equivalent convection temperature for the point over an entire rotation,
$T_0[\mathrm{K}]$	is the resulting quasi-constant temperature of the area element,
$q(\alpha)\left[\frac{\mathrm{W}}{\mathrm{m}^2}\right]$	is the variation of heat flux of the point as a function of rotor angle over an entire rotation,
$\alpha[^\circ]$	is the rotor angle position.

The variation of heat flux of the abovementioned point as a function of rotor angle over an entire rotation is defined as:

$$q(\alpha) = h(\alpha)[T_0 - T(\alpha)], \tag{2}$$

where

$h(\alpha)\left[\frac{\mathrm{W}}{\mathrm{m}^2\,\mathrm{K}}\right]$	is the variation of film coefficient for the point as function of rotor angle over a rotation,
$T(\alpha)[\mathrm{K}]$	is the variation of convection temperature for the point over an entire rotation.

From (1) and (2), the following equation can be obtained:

$$T_0 360 h_m - 360 h_m T_m = T_0 \int\limits_0^{360} h(\alpha)d\alpha - \int\limits_0^{360} h(\alpha)T(\alpha)d\alpha. \tag{3}$$

Therefore the equivalent film coefficient for the mentioned point over an entire rotation is:

$$h_m = \frac{\int_0^{360} h(\alpha)d\alpha}{360}, \tag{4}$$

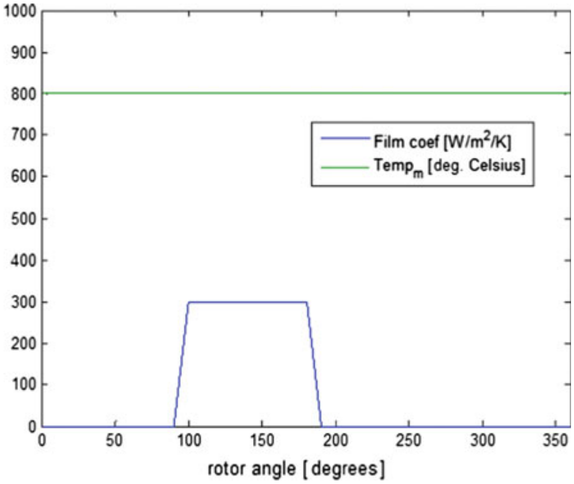

Fig. 4 Variations of convection temperature and film coefficient for point B during a mechanism rotation

and the equivalent convection temperature for the same point over an entire rotation is:

$$T_m = \frac{\int_0^{360} h(\alpha)T(\alpha)d\alpha}{h_m 360}. \tag{5}$$

The Eqs. (4) and (5) allow that the transient thermal analysis, which tends to stationary thermal conditions, is replaced with an equivalent steady-state thermal analysis, which no longer requires $\alpha(t)[^\circ]$ the rotor angle as a variable.

For example, during an entire mechanism rotation the point B undergoes variations of convection temperature and film coefficient as shown in Fig. 4.

The average values of the film coefficient and convection temperature for the point B situated on the fixed part at $\alpha = 90°$ (which has the longest exposure to combustion gas) are $h_m = 75$ W/(m^2 K) and $T_m = 800$ °C. The area elements A and D situated on the fixed part at $\alpha = 0°$ and at $\alpha = 180°$ have practically no convective heat transfer, and therefore on the angular intervals [0°; 90°] and [90°; 180°] linear variations of the film coefficient and convection temperature are obtained.

Comparison of Results from Transient and Steady-State Thermal Analysis

The transient thermal analysis performed for 1000 rotations is compared with the steady-state thermal analysis. The distribution of temperatures on the fixed part obtained from the steady-state thermal analysis is shown in Fig. 5 next to the distribution of temperatures obtained from the transient thermal analysis, and very

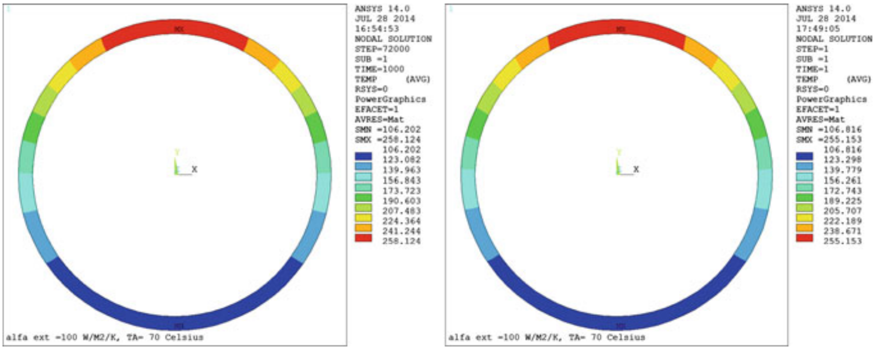

Fig. 5 Temperatures of fixed part obtained from the transient (*left*) and steady-state (*right*) thermal analysis

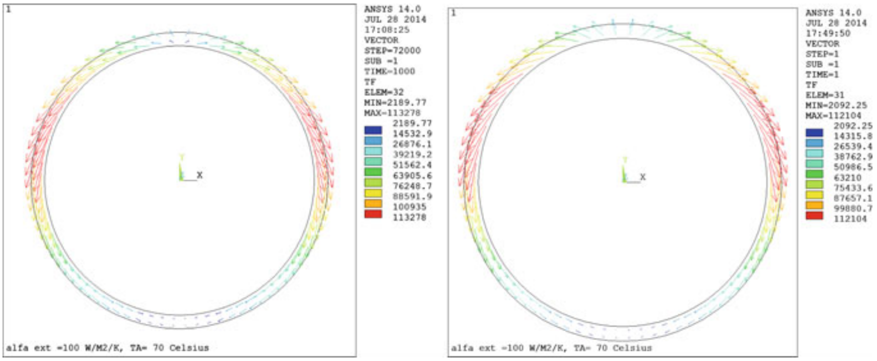

Fig. 6 Heat flux of the fixed part obtained from the transient (*left*) and steady-state (*right*) thermal analysis

good similarities are ascertained. Also, the results obtained from the two analyses for the heat flux of the fixed part are quite similar, as shown in Fig. 6.

Conclusions

The transient thermal analysis takes into account the permanent modification of surfaces covered with combustion gas, as a consequence of mechanism rotation, but requires a large number of engine cycles to obtain quasi-constant temperatures for each point of the mechanism. The complexity of the transient thermal analysis makes it inappropriate for simulations of a realistic rotary engine mechanism, with many parts and complicated geometry.

The equivalent model for steady-state thermal analysis is developed based on the definition of heat flux as an average considering its variation over an entire mechanism rotation. This assumption allows expressing the equivalent film coefficient over an entire rotation and the equivalent convection temperature over an entire rotation. Thus, the transient thermal analysis, which tends to quasi-constant temperature distributions, can be replaced with an equivalent steady-state thermal analysis, which no longer requires the rotor angle as a variable.

The comparison of fixed part temperature and heat flux proves that the steady-state thermal analysis performed using the equivalent film coefficients and the equivalent convection temperatures provides very similar results, compared with the transient thermal analysis, but with significantly lower computational effort.

Acknowledgments The research presented in this paper has been performed within the scientific research project No 266059 funded under the European Union Programme FP7-TRANSPORT.

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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

V. Iorga-Siman · A. Clenci (✉) · R. Niculescu
University of Pitesti, Pitesti, Romania
e-mail: adrian.clenci@upit.ro

A. Trică (Tuţă)
Renault Technologie Roumanie, Ilfov, Romania

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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

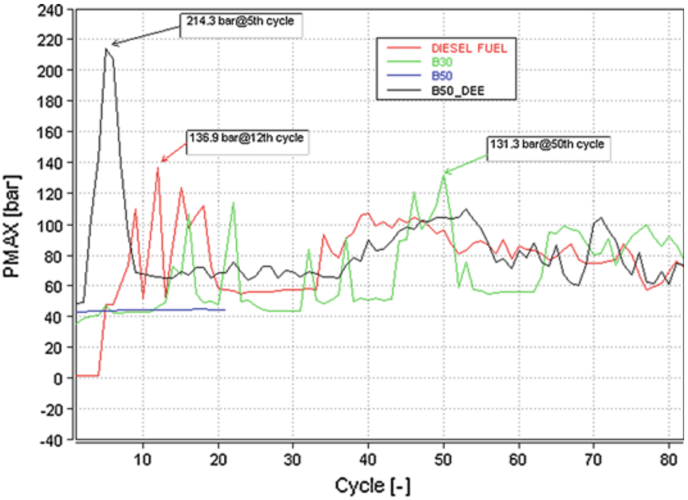

Fig. 1 Pressure peaks cyclic evolution, Clenci et al. (2014a)

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

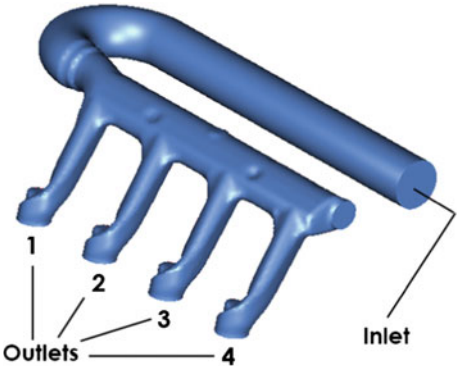

Fig. 2 Description of the volume—Inlet/outlets

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 $1/\mathrm{y}^2$ (y is a dimensionless wall distance) instead of $1/\mathrm{y}^4$ in the original $\overline{v}^2-f$ model.

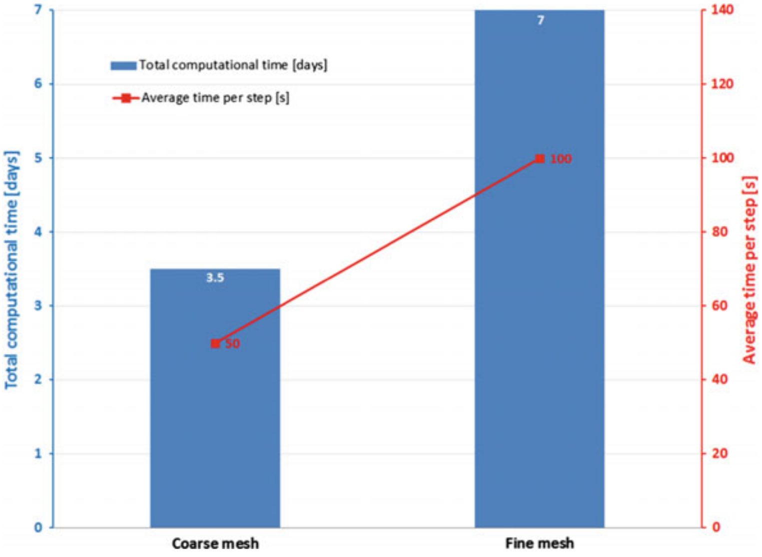

Fig. 3 Effect of the mesh on computational time