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CFD MODELING OF THERMAL CYCLE OF SUPERCHARGED COMPRESSION IGNITION ENGINE

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Abstract

Results of modelling of thermal cycle of turbocharged compression ignition IC engine are presented. The object of investigation was a 6CT107 turbocharged auto-ignition internal combustion engine powered by diesel oil, installed on an ANDORIA-MOT 100 kVA/ 80 kW power generating set in a portable version. The performed simulations of the combustion process have provided information on the spatial and time distributions of selected quantities within the combustion chamber of the test engine. The numerical analysis results have been juxtaposed with the results of indicating the engine on the test stand.

Modelling of the thermal cycle of an auto-ignition piston engine in the AVL FIRE was carried out within the study. Advanced numerical submodels were used to analysis of combustion process, such as: Extended Coherent Flame Model (ECFM-3Z), turbulence model k-zeta-f, injection submodels with evaporation, collisions, coalescence and other. Intake and exhaust processes were included during modelling. This resulted in a lot of information about the intake, fuel mixing, ignition process and the exhaust process. Results of modelling were compared with results from real engine.

Keywords: *combustion, modelling, CFD, diesel, injection*

1. Introduction

To model the thermal cycle of IC engine are used own or commercial programs. The complexity of the processes occurring in the IC engine significantly reduce the possibilities of creating own universal programs. The authors used Fire program form AVL company. In literature is possible to find works in which the authors have used other programs. Modelling is one of the most effective and readily used research methods. Advanced numerical models allow researches to analyze the flow processes coupled with combustion and spray. These models require a number of initial and boundary parameters. Therefore, before using the model to optimize the engine cycle should be verified experimentally [3, 4, 11, 14, 15]. Binesh and Hossainpour [10] presents the results of modelling fuel mixture formation and combustion in the turbocharged direct-injection compression-ignition engine. The numerical analysis was performed using the FIRE program. As a result of computations, cylinder pressure variations and the curves of NO_x and soot formation in the engine exhaust gas were obtained; these results were then compared with the result of research work carried out on the real engine. As a result, fairly good agreement between the modelling results and experimental test results were achieved; and what the engine model reflected best was the variation of pressure in the engine. Kusaka and Daisho [9] reported the results of modelling the combustion and exhaust gas emission processes in the CI turbocharged engine with the common-rail system. The numerical analysis was performed using the KIVA-3V program, as modified by being supplemented with chemical reaction sub-models taken from the CHEMKIN-II program. The numerical analysis results were compared with the results of experimental tests. Good consistence, both qualitative and quantitative, was obtained for pressure variation in the cylinder, the heat release degree and the NO contents of exhaust gas. Considerable discrepancy was obtained between the results of soot formation in the exhaust gas. The modelling yielded an exhaust gas soot concentration being ten times that obtained in reality. Hélie and Trouvé [8] presented a modifications of the coherent flame model (CFM) to account for the effects of variablemixture strength on the primary premixed flame, as well as for the formation of a secondary non-premixed reaction zone downstream of the premixed flame. The modeling strategy was based on a theoretical analysis of a simplified problem by Kolmogorov, Petrovskii, and Piskunov (KPP). The KPP problem corresponds to a one-dimensional, turbulent flame propagating steadily into frozen turbulence and frozen fuel-air distribution, and it provides a convenient framework to test the modified CFM model. In this simplified but somewhat generic configuration, two radically different situations were predicted: for variations in mixture strength around mean stoichiometric conditions, unmixedness tends to have a net negative impact on the turbulent flame speed: in contrast, for variations in mixture strength close to the flammability limits, unmixedness tends to have a net positive impact on the turbulent flame speed [8].

One of the more advanced programs and readily used for modelling thermal cycle of internal combustion engine is Fire. In this program to modelling the combustion process, an advanced combustion submodel is used. The ECFM (Extended Coherent Flame Model) model [1, 2] was developed specially for modelling the combustion process in a compression ignition engine. The ECFM-3Z model belongs to a group of advanced models of the combustion process in a compression ignition engine. For several years, ECFM-3Z combustion model has been successfully used, constantly modified and improved by many researchers [2, 6, 14]. Together with turbulence process sub-models (e.g. the k-zeta-f), exhaust gas component formation, knock combustion and other sub-models, they constitute a useful tool for modelling and analysis of the thermal cycle of the compression ignition internal combustion engine. To adapt the model for the modelling of the combustion process in the auto-ignition engine, a sub-model has been added, which describes the process of mixing fuel to be injected to the combustion chamber. The turbulent combustion process is defined by the time scale of chemical reactions, the time scale of turbulent processes, and turbulence intensity. The flame front is formed by the turbulent effect of load vortices and interaction between the burned zone and the unburned part of the load. The time scale of chemical processes is much smaller than that defining the load turbulence. This model is based on the concept of laminar flame propagation with flame velocity and flame front thickness as the average flame front values. It is also assumed that the reactions occur in a relatively thin layer separating unburned gases from the completely burned gases [2]. The model relies on the flame front transfer equation, as well as on the mixing model describing the combustion of an inhomogeneous mix and the diffusion combustion model. The model assumes the division of the combustion region into three zones: a fuel zone, a zone of air with a possible presence of exhaust gases remained from the previous engine operation cycle, and an air-fuel mixture zone, where combustion reactions occur following the ECFM concept. The air-fuel mixture formation model provides for gradual mixing of fuel with air. The created combustion model is called ECFM-3Z (3-Zones Extended Coherent Flame Model). In this model, the mixture zone is additionally divided into a burned and an unburned zone. To initiate the combustion process, the auto-ignition model for the forming mixture zone and for the diffusion flame zone is used [1, 2]. The ECFM makes use of the 2-stage fuel oxidation mechanism ($C_{13}H_{23}$) [1]:

$$C_{13}H_{23} + 18{,}75O_2 \rightarrow 13CO_2 + 11{,}5H_2 0\,, \tag{1}$$

$$C_{13}H_{23} + 11{,}5O_2 \rightarrow 13CO + 11{,}5H_2\,. \tag{2}$$

The reaction of formation of CO and H_2 is taken into account for stoichiometric and fuel-rich mixtures, while for lean mixtures this reaction is omitted. In the ECFM-3Z model, transport equations for the chemical components O_2, N_2, CO_2, CO, H_2, H_2O, O, H, N, OH and NO are also solved. The concept of the injected fuel and air-mixing model relies on the characteristic time-scale of the turbulence model. Because of the occurring process of fuel evaporation, it is necessary to determine the amount of fuel entering the mixture zone and to the pure fuel zone. In the injected fuel stream, fuel droplets are situated so close to one another as to form altogether a fuel zone. After the fuel has evaporated, a specific time is still needed for mixing of the pure fuel zone fuel with air and formation of the combustible mixture [5]. It is additionally assumed that the composition of gas, fuel + EGR is identical both in the mixture zone and in the zone being still unmixed. The mixture auto-ignition delay is calculated from the empirical correlation [12]. The combustion model for the auto-ignition engine has been complemented with the unburned product zone. The exhaust gas contains unburned fuel and O_2, N_2, CO_2, H_2O, H_2, NO, CO. The fuel oxidation occurs in two stages: the first oxidation stage leads to the formation of large amounts of CO and CO_2 in the exhaust gas of the mixture zone, at the second stage in the mixture zone exhaust gas, the previously formed CO is oxidized to CO_2.

This paper presents results of thermal cycle modelling of turbocharged internal combustion diesel engine. A number of remarks about the geometry and mesh creation of a piston engine are mentioned.

2. The object of investigation

Modelling of the thermal cycle of an auto-ignition internal combustion supercharged engine in the AVL FIRE program was carried out within the study. The object of investigation was a 6CT107 turbocharged auto-ignition internal combustion engine powered by diesel oil (Fig. 1), installed on an ANDORIA-MOT 100 kVA/ 80 kW power generating set in a portable version. The engine was equipped with pressure sensors in each cylinder (Fig. 1). The engine was indicated for a few loads. In this paper, results of three loads are presented. The measurements results were used to the model validation. Based on the recorded results of indication, thermodynamic analysis of the engine was performed. It was determined inter alia, the mean cylinder pressure and efficiency of the test engine. Because that the test engine was 6-cylinder engine that to model validation was taken results from one cylinder. It should be noted that in this engine, the peak pressure in all six cylinders are not significantly different from each other.

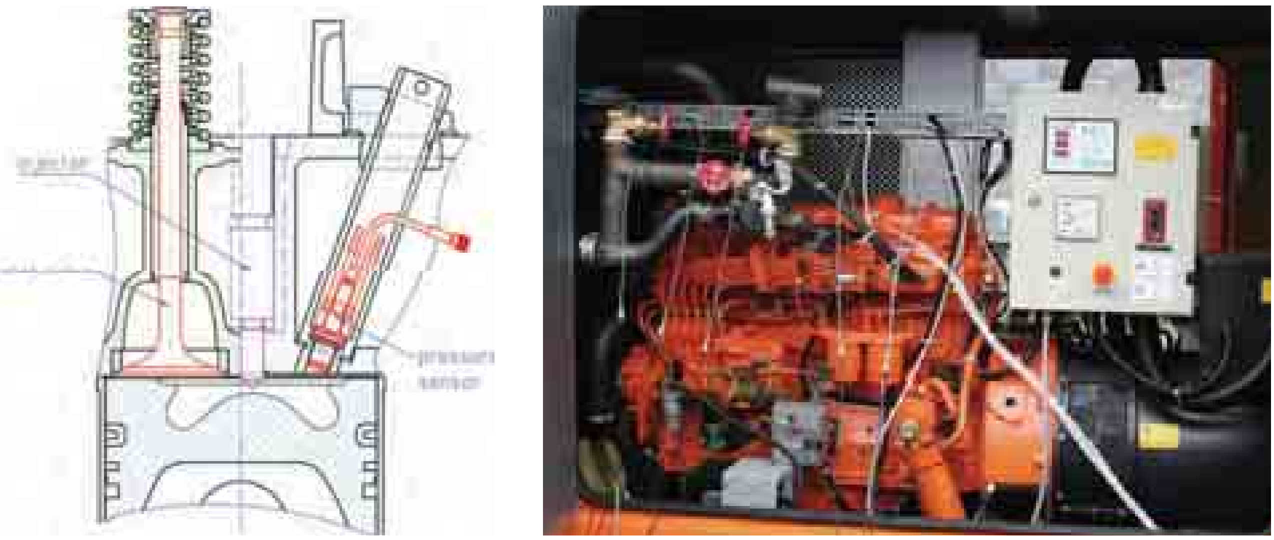

Fig. 1. The 6CT107 test internal combustion engine

Figure 1 presents the test engine equipped with measuring system and section of the engine cylinder with visible valve, injector and pressure sensor. The shape of combustion chamber is visible as well.

In Tab. 1 the main engine parameters with valves timing are presented. It is a stationary engine operates at constant speed of 1500 rpm.

Tab. 1. Engine specification

Parameters	value	
displacement	6.54	dm^3
rotational speed	1500	rpm
crank throw	60.325	mm
cylinder bore	107.19	mm
connecting-rod length	245	mm
compression ratio	16.5	-
intake valve opening	10 ± 4^o BTDC	deg
intake valve closure (IVC)	50 ± 4^o ABDC	deg
exhaust valve opening	46 ± 4^o BBDC	deg
exhaust valve closure (EVC)	14 ± 4^o ATDC	deg
injection angle	$9^o\pm1.5^o$	deg

Figure 2 shows cross sectional of computational CAD domain. There are visible a ports, combustion chamber and valves. Both valves are partly opened. It is necessary because of mesh generation process requirements. The size of this gap also affects the required size of the cells. In this case the gap was equal 0.3 mm, the same for both valves. This gap is better visible in Fig. 5a.

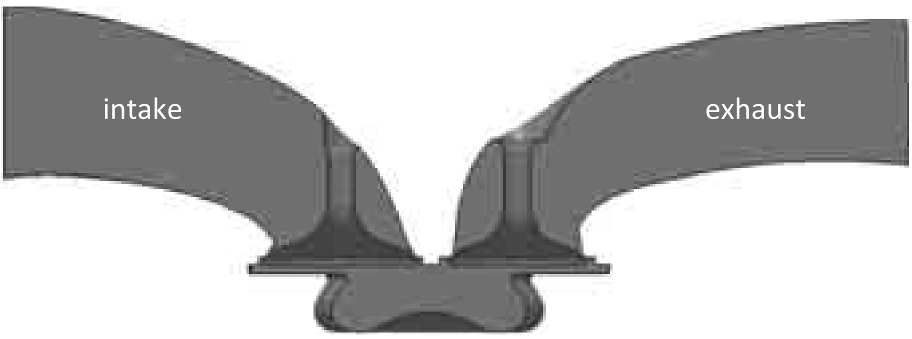

Fig. 2. Geometry in TDC of modelled test engine in CAD

Figure 3 shows three computational domains created in CAD software, which were used to mesh generation process.

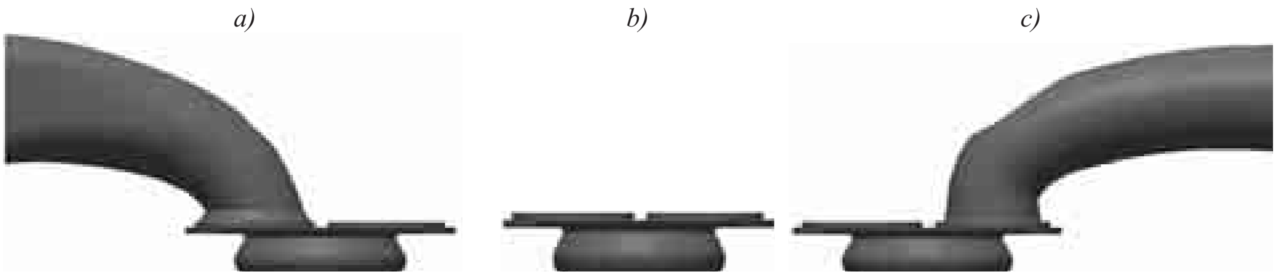

Fig. 3. Geometry of computational domains in CAD: a) intake stroke (EVC), b) compression and power stroke, c) exhaust stroke (IVC)

The modelling process began with an analysis of the mesh quality. The all construction details of combustion chamber, intake and exhaust ducts and valves were included by drawing process. This forced the appropriate scaling of mesh. The computational mesh consisted of 186403 cells and 178260 nodes.

Both local and temporary densification of the mesh was used. This led to more efficiently use of computing power. For example, when the valve is opening mesh has many times smaller cells in this area, because at this time the flow processes occur most intensively.

a) b) c)

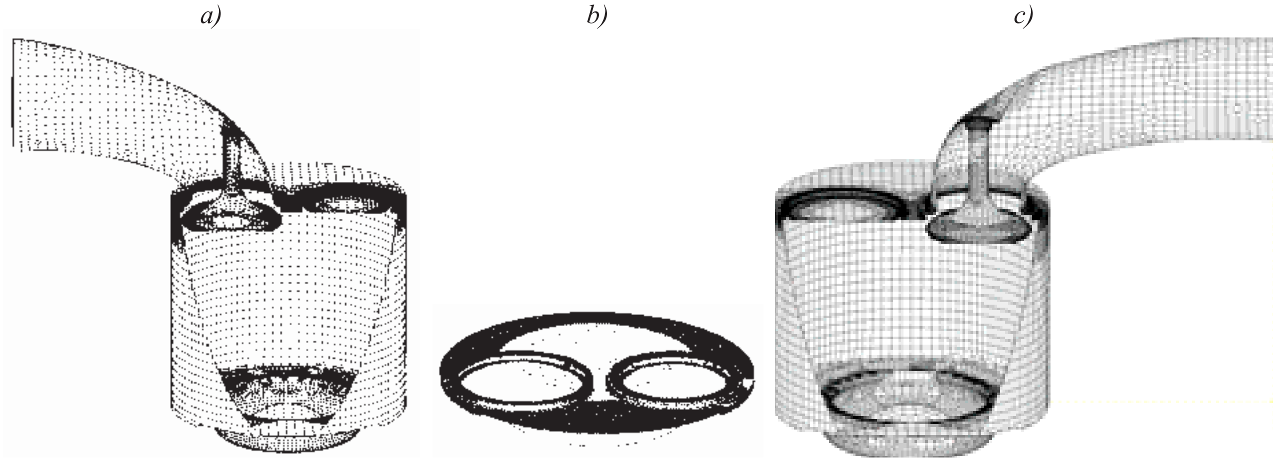

Fig. 4. Computational domains: a) intake stroke (EVC), b) compression and power stroke (TDC), c) exhaust stroke (IVC)

a) b)

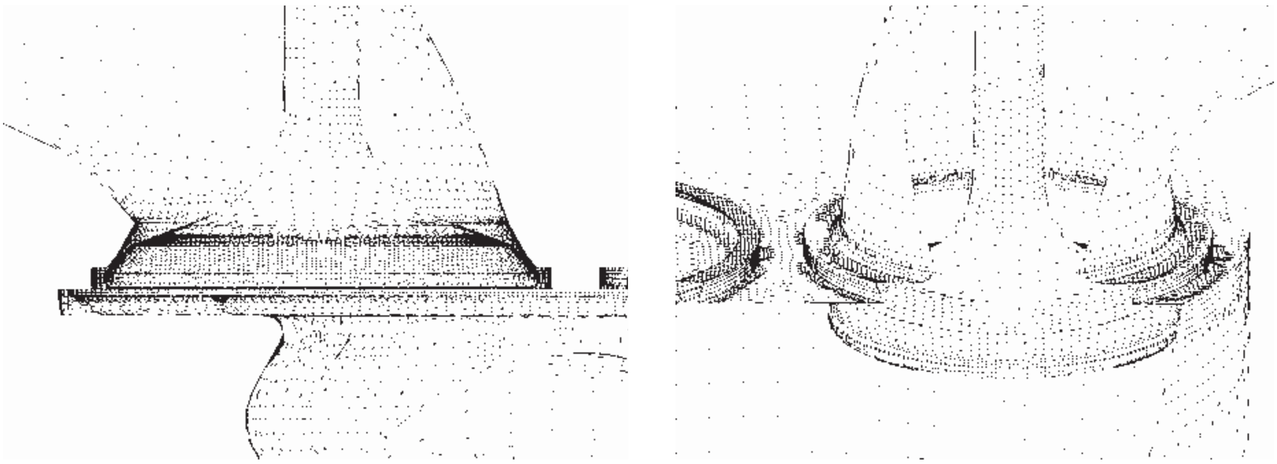

Fig. 5. View of the inlet valve closed (a) and partly opened exhaust valve (b)

Tab. 2. Modelling parameters

Parameters	Value		
Load	80 kW	65 kW	58 kW
Initial pressure	0.164 MPa	0.147 MPa	0.127 MPa
Initial temperature	317 K	313 K	314 K
Injection angle	–9 deg BTDC		
Fuel temperature	330 K		

Tab. 3. Submodels

Model	Name
Combustion model	ECFM-3Z
Turbulence model	k-zeta-f
NO formation model	Extended Zeldovich Model
Soot formation model	Lund Flamelet Model
Evaporation model	Dukowicz
Breakup model	Wave

The above-mentioned submodels were used during modelling. The parameters shown in Tab. 2 are taken from experiment and then these were used as input values for modelling.

3. Results

Modelling of the thermal cycle of the test supercharged compression ignition engine using the FIRE software was conducted. As the research object was taken internal combustion test engine 6CT107, operated at constant rotational speed equal to 1500 rpm. The researches were conducted for three loads. Initial parameters were taken from experiment. The boundary conditions such as temperature of combustion chamber parts, valves and ports were taken from literature [3]. In Fig. 6 results of model validation are presented.

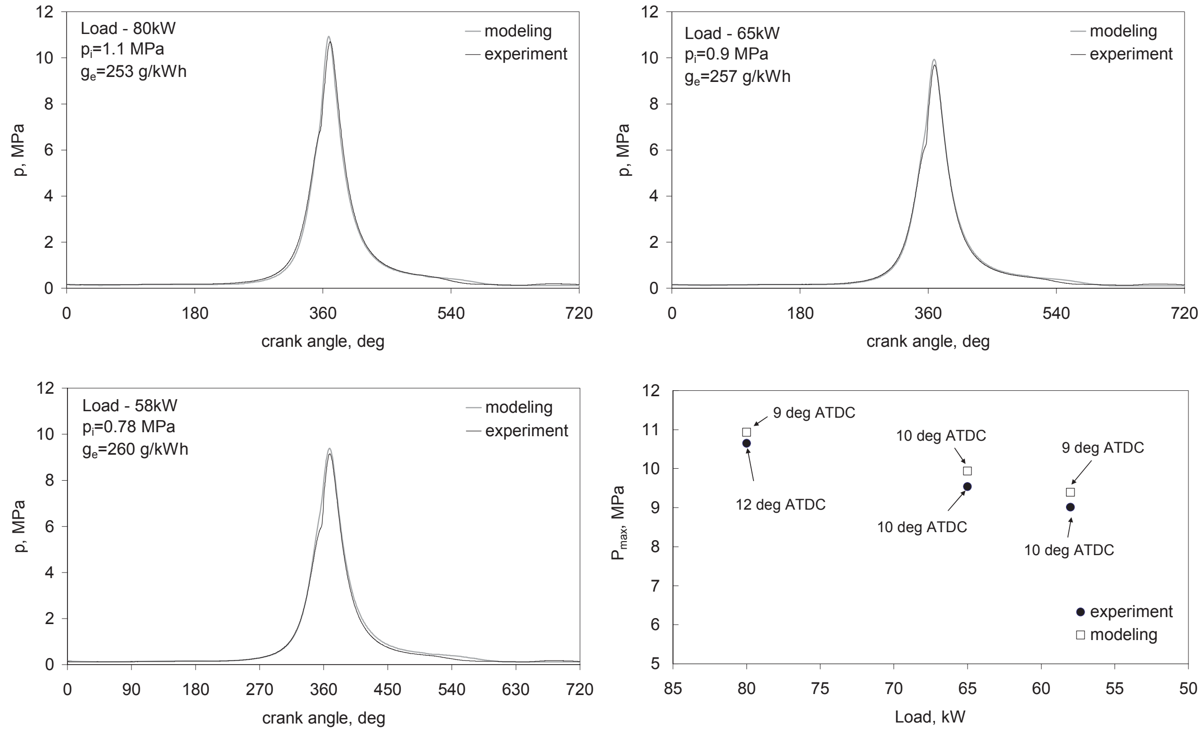

Fig. 6. Results of model validation. Comparison of pressure courses for three loads and comparison of peak pressures

For all three load the satisfactory results of validation was achieved and the results are presented above. In all three cases the biggest values of pressure pick was obtained for modelling.

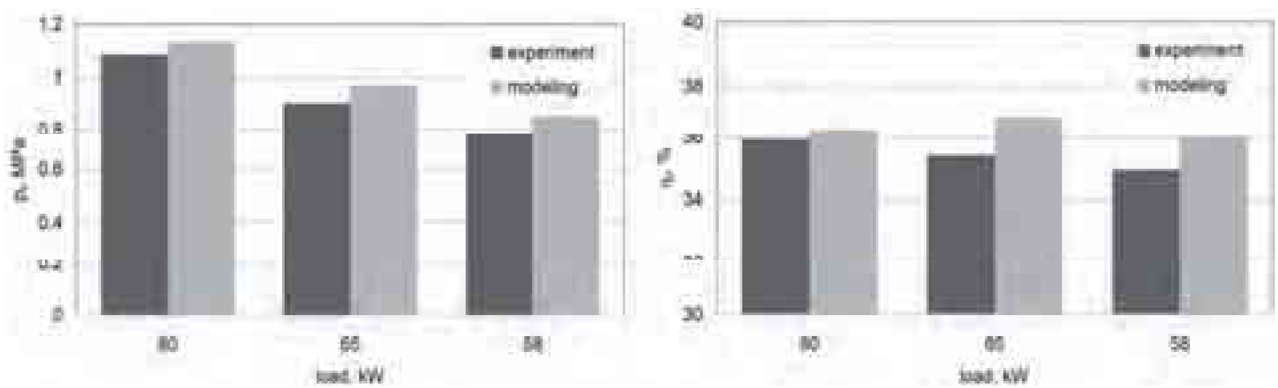

Fig. 7. Indicated pressure and efficiency

Figure 7 presents indicated pressure and efficiency of modelled thermal cycle of the test engine. In the case of model, the higher values of p_i and η_i was obtained. The biggest difference in p_i was in the case 58 kW and it was equal 0.08 MPa. However, for efficiency, the maximum difference was observed at a load of 65kW and the difference was near to 2%. In both cases, these parameters were determined in the same way.

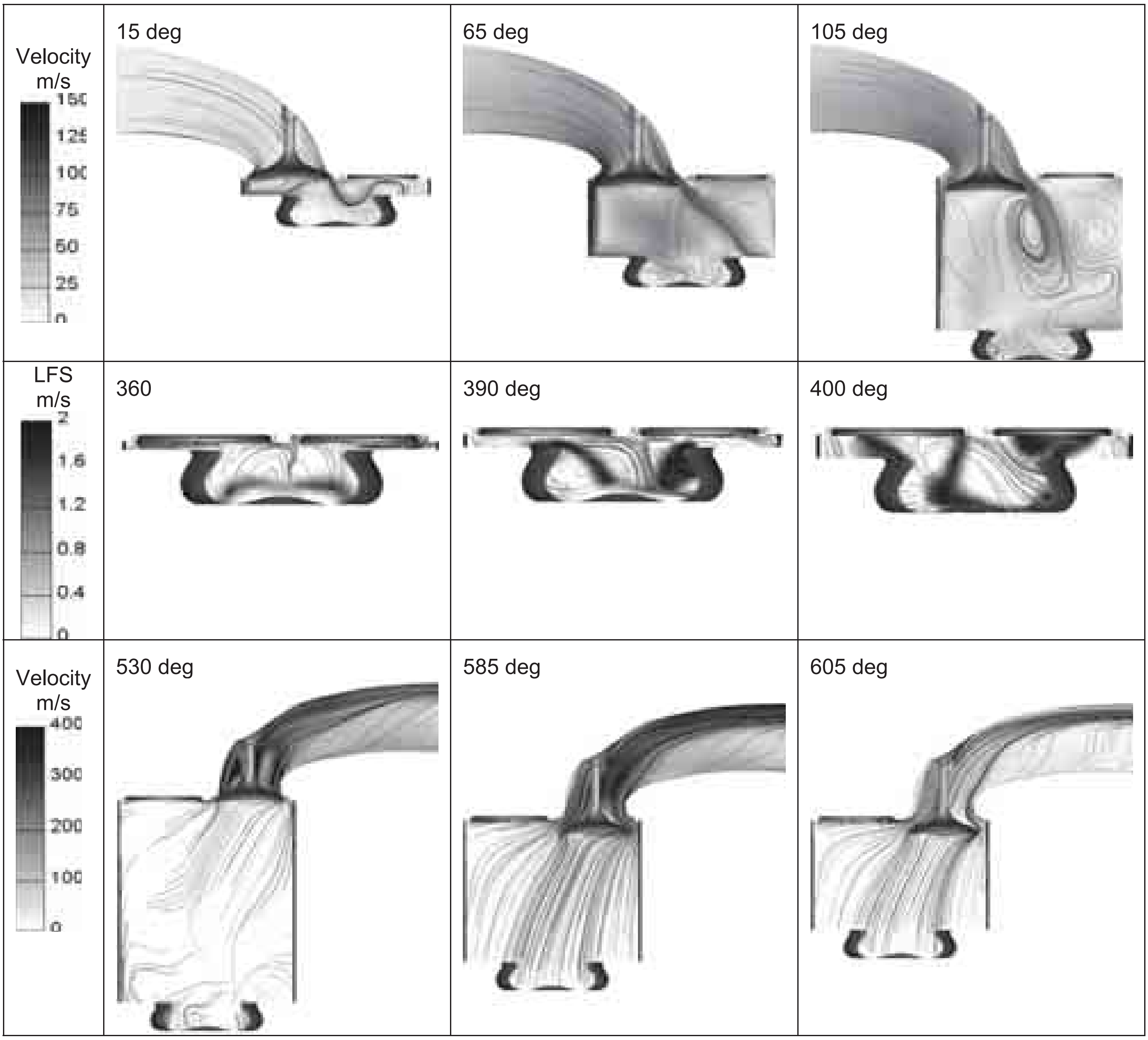

Fig. 8. Cross-sections of the engine cylinder, a) intake stroke – flow field with streamlines, b) LFS with streamlines during combustion process, c) temperature, d) exhaust stroke – flow field with streamlines

Figure 8 shows cross sections of the cylinder at different angles of the crankshaft position. The first three pictures show the process of filling the cylinder. Streamlines highlight the phenomenon of charge motion in the cylinder. The drawings of engine with the closed valves, shows the process of flame propagation illustrated by LFS (Laminar Flame Speed). The last three pictures represent the exhaust stroke.

4. Conclusions

Paper presents results of supercharged engine modelling using AVL Fire software. Pressure, temperature, heat release rate and other parameters in function of crank angle as well as spatial distribution of above-mentioned quantities at selected crank angles were determined. Creating an appropriate mesh was required for many simulations, in order to become independent of calculations on the density of the mesh. Local and temporary densification of the mesh was used. Created model of diesel engine was successfully verified. The resulting differences are acceptable.

The results of modelling allow analysis of engine operation both in terms of thermodynamic and flow. Detailed analysis of the modelled engine thermal cycle will be subject of the following publications.

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References

[1] AVL FIRE, VERSION 2009ICE, *Physics & Chemistry. Combustion, Emission, Spray, Wallfilm*, Users Guide, 2009.

[2] Colin O., Benkenida A., *The 3-Zones Extended Coherent Flame Model (ECFM3Z) for Computing Premixed/Diffusion Combustion*, Oil & Gas Science and Technology 2004.

[3] Tutak, W., *Modelling and analysis of some parameters of thermal cycle of IC engine with EGR.* Combustion Engines 4/2011 (147), pp. 43-49, 2011.

[4] Tutak, W., Jamrozik, A., *Modelling of thermal cycle of gas engine using AVL FIRE software*. Combustion Engines R. 49, No. 2 (141), pp. 105-113, 2010.

[5] Baumgarten, C., *Mixture Formation in Internal Combustion Engines,* Springer-Verlag Berlin Heidelberg 2006.

[6] Tatschl, R., Priesching, P., Ruetz, J., *Recent Advances in DI-Diesel Combustion Modeling in AVL FIRE – A Validation Study*. International Multidimensional Engine Modeling User's Group Meeting at the SAE Congress April 15, Detroit, MI 2007.

[7] Musculus, M. P., Rutland, C. J., *Coherent flamelet modeling of diesel engine combustion.* Combustion science and technology, Vol. 104, No. 4-6, pp. 295-337, 1995.

[8] Hélie, J., Trouvé, A., *A modified coherent flame model to describe turbulent flame propagation in mixtures with variable composition*. Proceedings of the Combustion Institute, Volume 28, Issue 1, pp. 193-201, Revue de l'Institut Français du Pétrole, 2000.

[9] Kusaka, J., Daisho, Y., *Simulating combustion and exhaust gas emissions in a DI diesel engine by using a CFD code combined with detailed chemistry*, Journal of KONES Internal Combustion Engines, Vol. 10, No. 1-2, 2003

[10] Binesh, A. R., Hossainpour, S., *Three dimensional modeling of mixture formation and combustion in a direct injection heavy-duty diesel engine*, World Academy of Science, Engineering and Technology 41, pp. 207-212, 2008.

[11] Cupiał, K., Tutak, W., Jamrozik, A., Kociszewski, A., *The accuracy of modelling of the thermal cycle of a compression ignition engine*. Combustion Engines, R. 50, No. 1 (144), pp. 37-48.

[12] Colin, O., Pires da Cruz, A., Jay, S., *Detailed Chemistry-Based Auto-Ignition Model Including Low Temperature Phenomena Applied to 3D Engine Calculations*, Proc. Combustion Institute 30, pp. 2649-2656, 2005.

[13] Tatschl, R., Priesching, P., Ruetz, J., Kammerdiener, Th., *DoE Based CFD Analysis of Diesel Combustion and Pollutant Formation*, SAE 2007-24-0048, 2007.

[14] Jaworski, P., Priesching, P., Teodorczyk, A., Bandel, W., *Validation of the Numerical Simulation of Iso-octane Auto-ignition Delay Time in Rapid Compression Machine with the Use of ECFM-3Z (Extended Coherent Flame Model - 3 Zones) Combustion Model Against Experimental Data*, Archivum Combustionis Vol. 30, No. 1-2, 2010.

[15] Priesching, P., Ramusch, G., Ruetz, J., Tatschl, R., *3D-CFD Modeling of Conventional and Alternative Diesel Combustion and Pollutant Formation – A Validation Study*, SAE 2007-01-1907d.

[16] Szwaja, S., *Knock and combustion rate interaction in a hydrogen fuelled combustion engine*. Journal of KONES, Vol. 18, No. 3, pp. 431-438, 2011.