CFD modelling of diesel engine at partial load

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Summary. The paper presents the results of CFD modelling of thermal cycle of compression ignition IC engine. The turbocharged 6CT107 engine powered by diesel oil was the object of investigations. The results of model validation at partial load are presented. Model of CI engine was used to the optimization of thermal cycle of the test engine. The 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.
Key words: diesel engine, combustion, modelling, mesh, CFD.

INTRODUCTION

CFD modelling of internal combustion engines has been greatly developing along with the increasing computational power that allows modelling flow processes, combustion processes, emissions and injection fuel using any computational meshes. To model a complete engine cycle with intake and exhaust stroke, some commercial programs are most often used. Internal combustion engine is such a complex object of modelling that building its model including all the important processes is becoming very difficult. Early models of the thermal cycle of the compression ignition engine appeared in the twenties of the last century. In 1926 Schweitzer published a model of heat release in the compression ignition engine. In the sixties, the development of computation models followed. At the beginning, the models were single-zone, and later they were extended to multi-zone direct-injection models. Models based on the fuel injection characteristics were created. One of the first advanced multi-dimension combustion process models for engines with liquid fuel injection to the combustion chamber was the CONCHAS-SPRAY model developed at the Los Alamos Scientific Laboratory in the USA . Since the seventies this model has been modified and complemented with a number of sub-models and has become a basis for creating the KIVA program [2, 3, 5, 6, 7, 8, 9, 10, 11, 12, 13]. The KIVA program was used by the author to model the thermal and flow processes with combustion and fuel injection into the combustion chamber. During the model researches the commercial programme was used. For the last few years the modelling of thermal cycle of IC engine is performed using AVL Fire program. This program allows the modelling of flows and thermal processes occuring in the intake and exhaust manifold and in combustion chamber of internal combustion engine. It also allows modelling the transport phenomena, mixing, ignition and turbulent combustion in internal combustion engine. Homogeneous and inhomogeneous combustion mixtures in spark ignition and compression ignition engine can be modelled using this software, as well. Kinetics of chemical reactions phenomena is described by combustion models which take oxidation processes in high temperature into consideration. Several models apply to modelling auto ignition processes. This program allows for creating three-dimensional computational mesh, describing boundary conditions of surfaces as well as the initial conditions of simulation [14, 15, 16, 17, 18, 19].

Modelling is one of the most effective and readily used research methods. Advanced numerical models allow for 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, it should be verified experimentally. Renganathan et. al. in their work presented reacting flow simulations performed in a single cylinder direct injection diesel engine with an improved version of the ECFM-3Z (extended coherent flame model – 3 Zones) model using ES-ICE and STAR-CD codes. Combustion and emission characteristics were studied in the sector of engine cylinder, which eliminates the tedious experimental task with conservation in resources and time. It was found that higher NOx emissions occur at peak temperatures, while soot and CO emissions occur at peak pressures. Additionaly, they stated that the numerical modelling of the combustion and emissions give clear understanding in the heat release and formation of reactant species in the direct injection Diesel engine [20, 21, 22, 23, 24, 25, 26].

Binesh et. al. present 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. Hélie and Trouvé presented 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 modelling 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.

This paper presents results of thermal cycle modelling of turbocharged internal combustion diesel engine at full and partial loads [27, 28, 29, 30].

OBJECT OF INVESTIGATION

Contemporary engines are designed to minimize exhaust emissions while maximizing power and economy. Emissions can be reduced by equipping the engine with advanced exhaust after-treatment systems or by controlling the combustion process occurring in the cylinder of internal combustion engine. In order to improve the performance of the engine, research and optimization of the combustion process are carried out. This is dictated by growing concern about decreasing energy resources and environmental protection. Therefore, intensive research is being carried out towards development of internal combustion engine systems. It involves improving the combustion process, introduction of a new fuel such as hydrogen as well as the optimization of engine parameters. The engine should operate with the greatest efficiency possible and with the least toxic compounds emissions [31, 32, 33].

Modelling of the thermal cycle of an auto-ignition internal combustion supercharged engine was carried out within the study. The object of investigation was a 6CT107 internal combustion engine powered by diesel oil, installed on an ANDORIA-MOT 100 kVA/ 80 kW power generating set. The engine was equipped with pressure sensors in each cylinder. The measurements results were used to the model validation. Based on the recorded results of indication, thermodynamic analysis of the engine was performed. Inter alia, the mean cylinder pressure and efficiency of the test engine were determined. It should be noted that in this engine the peak pressures in all the six cylinders are not significantly different from one another.

Table 1. Engine specification

Parameters	Value	
displacement	6.54	dm^3
rotational speed	1500	rpm
stroke	120	mm
cylinder bore	107	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

MODEL ASSUMPTIONS

The Fire program contains many submodels which are necessary to solve processes occurring in the combustion chamber of IC engine. The most important submodel is combustion model. The ECFM (Extended Coherent Flame Model) model was developed especially for modelling the combustion process in a compression ignition engine. The ECFM-3Z model belongs to the group of advanced combustion process models in a compression ignition engine. For several years ECFM-3Z combustion model has been successfully used, constantly modified and improved by many researchers. 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 was added, which describes the process of mixing fuel to be injected to the combustion chamber. 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. 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 . 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 . The ECFM makes use of the 2-stage fuel oxidation mechanism ($C_{13}H_{23}$). The fuel oxidation occurs in two stages: the first oxidation stage leads to the formation of large amounts of CO and CO2 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 CO2.The combustion model for the auto-ignition engine was 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, the second stage in the mixture zone exhaust gas, the previously formed CO is oxidized to CO_2. The reaction of formation of CO and H2 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 O2, N2, CO2, CO, H2, H2O, O, H, N, OH and NO are also solved. The concept of the injected fuel and air mixing model relies on the characteristic timescale 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. It is additionally assumed that the composition of gas, fuel + EGR is identical both in the mixture zone and in the zone which remins unmixed. The mixture auto-ignition delay is calculated from the empirical correlation. The combustion model for the auto-ignition engine was complemented with the unburned product zone. The exhaust gas contains unburned fuel and O2, N2, CO2, H2O, H2, NO, CO.

Table 2. Modelling parameters

Parameters	Value		
Load	100% (80kW)	70% (58kW)	45% (37kW)
Initial pressure	0.164 MPa	0.137 MPa	0.124 MPa
Initial temperature	317 K	314 K	311 K
Injection angle	-9 deg BTDC		
Fuel temperature	330 K		

Table 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	Dukowich
Breakup model	Wave

The above-mentioned submodels were used during modelling. The parameters shown in Table 1 were taken from the experiment and then these were used as input values for modelling.

RESULTS OF MODELLING

Modelling of the thermal cycle of the test supercharged compression ignition engine using the FIRE software was conducted. The object of research was the internal combustion test engine 6CT107, operated at constant rotational speed equal to 1500 rpm. The researches were conducted for three loads. The initial parameters were taken from the previous experiment. The boundary conditions such as temperature of combustion chamber parts, valves and ports were taken from literature .

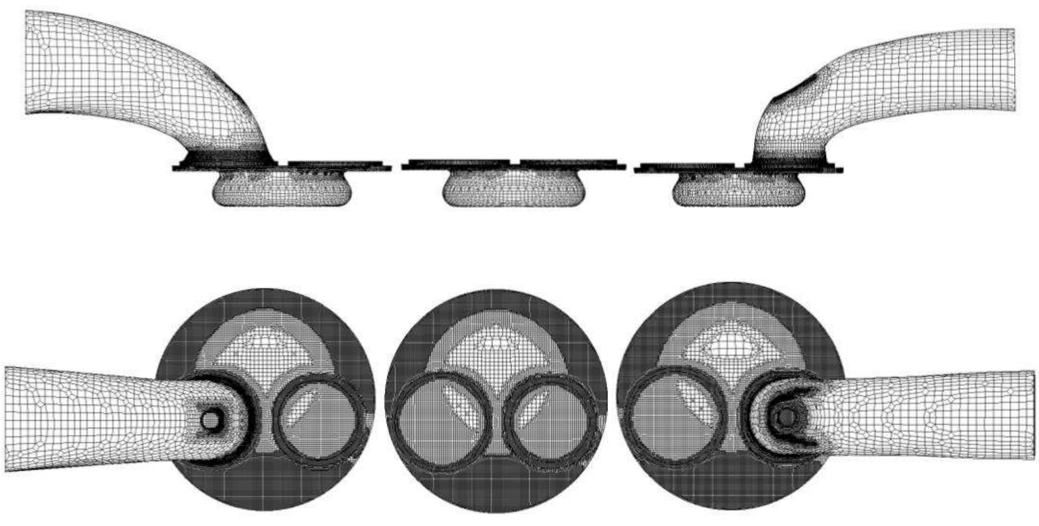

Fig. 1. Computational mesh domains

Figure 1 shows the computational domains for, respectively, intake stroke, compression and work stroke and exhaust stroke. During the calculation the program uses them consecutively.

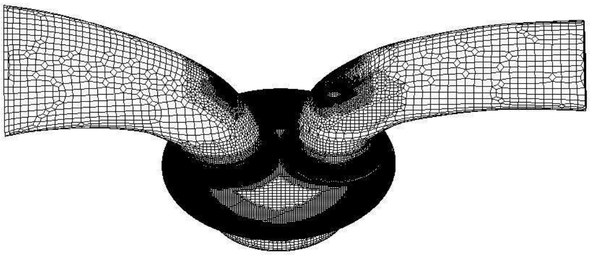

Fig. 2. Computational mesh which was created as a combination of three computational domains

Figure 2 shows computational domain for complete engine work space connected during calculation. The presented domain consists of 350000 cells and the density of this mesh was optimized.

Figure 3 shows the pressure and heat release rate courses. The courses for real test engine were taken on the basis of engine indication process and these courses were compared with courses achieved by modelling. Also, the satisfactory consistency of results was obtained. Of course, there are some differences occurring, but considering the complexity of the processes taking place in the cylinder, the results of the verification of the model can be considered as satisfactory.

Figure 4 shows indicated mean effective pressure and break thermal efficiency calculated on the basis of indication results of six cylinders of the test engine compared

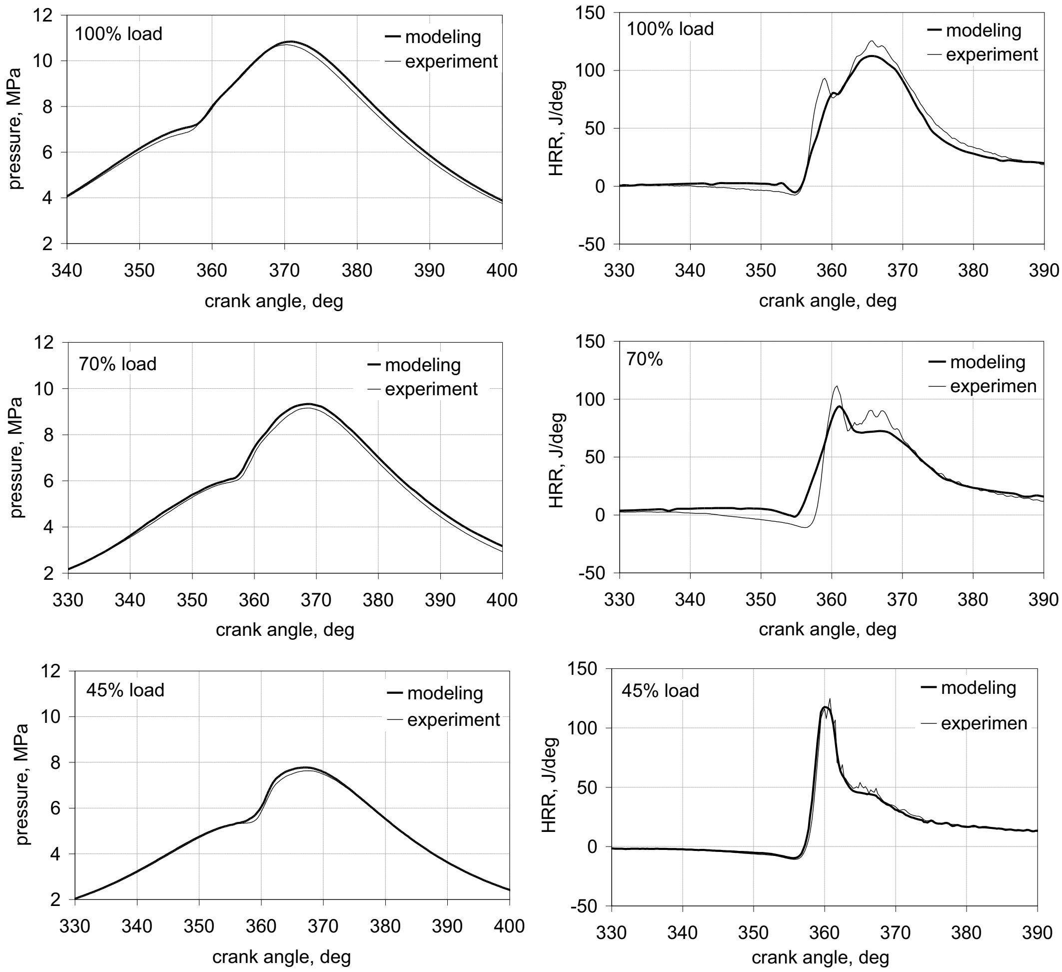

Fig. 3. Pressure and heat release rate courses

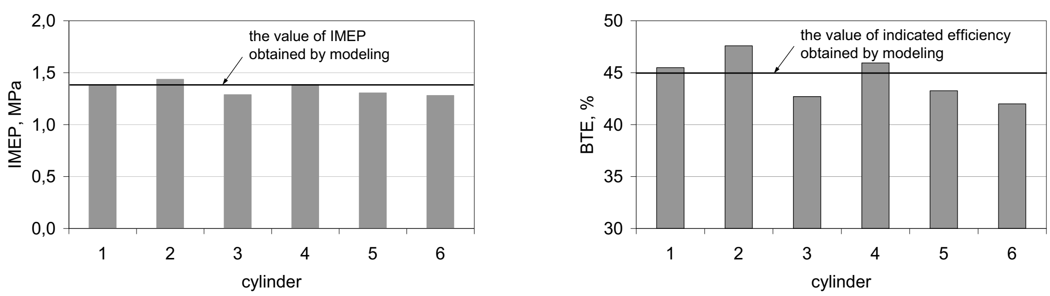

Fig. 4. Indicated mean effective pressure and break thermal efficiency

with the values obtained by the modelling of engine at full load (100%). In the case of modelling the indicated mean effective pressure was equal 1.4 MPa and break thermal efficiency was equal to 45%. The real engine parameters are varied in each cylinder. This could be due to different degrees of filling cylinders and other phenomena. At partial loads the similar correlations were obtained.

CONCLUSIONS

The paper presents results of modelling of supercharged IC compression ignited test engine at partial loads. The calculation was carried out using the Fire software . Pressure, temperature, heat release rate and other parameters in function of crank angle as well as spatial distribution of the above mentioned quantities at selected crank angles were determined. The biggest problem during modelling is to generate adequate mesh which will not affect the results. Obtaining the independence of the results on the density of the mesh is a key issue here. Local and temporary densification of the mesh was used in order to improve the results while reducing cost of calculations. The created model of diesel engine was successfully verified. The resulting differences are acceptable taking into account the complexity of the problem. The results of modelling allows the analysis of engine operation both in terms of thermodynamic and flow. Such verified engine model can be used to model and optimize the engine thermal cycle.

REFERENCES

1. **Schweitzer P, 1926**: The tangent method of analysis of indicator cards of internal combustion engines. Bulletin nr 35, Pennsylvania State University, September
2. **Heywood J. B., 1988**: Internal combustion engine fundamentals. McGraw-Hill.
3. **Rychter T., Teodorczyk A., 1990**: Modelowanie matematyczne roboczego cyklu silnika tłokowego. PWN, Warsaw.
4. **Amsden A.A., O'Rourke P.J., Butler T.D., 1989**: KIVA-II, A computer program for Chemically Reactive Flows with Sprays. Los Alamos National Laboratory LA-11560-MS.
5. **Jamrozik A., Tutak W., Kociszewski A., Sosnowski M., 2013**: Numerical simulation of two-stage combustion in SI engine with prechamber. Applied Mathematical Modelling, Volume 37, Issue 5, 2961–2982.
6. **Szwaja S., Jamrozik A., Tutak W., 2012**: A Two-Stage Combustion System for Burning Lean Gasoline Mixtures in a Stationary Spark Ignited Engine. Applied Energy, 105 (2013), 271-281, 2013.
7. **Tutak W., 2012**: An analysis of EGR impact on combustion process in the SI test engine. Combustion Engines, Vol 148, No. 1/2012, 11-16.
8. **Tutak W., 2011**: Modelling and analysis of some parameters of thermal cycle of IC engine with EGR. Combustion Engines 4/2011 (147), 43-49.
9. **Tutak W., 2011**: Numerical analysis of some parameters of SI internal combustion engine with exhaust gas recirculation. Teka Kom. Mot. i Energ. Roln. – OL PAN T. 11, 407-414.
10. **Tutak W., 2011**: Numerical analysis of the impact of EGR on the knock limit in SI test engine. Teka Kom. Mot. i Energ. Roln. – OL PAN T.11, 397-406.
11. Tutak W., 2011: Possibility to reduce knock combustion by EGR in the SI test engine. Journal of KONES, Powertrain and Transport, No 3, 485-492.
12. **Tutak W., 2008**: Thermal Cycle of Engine Modeling with Initial Swirl Proces Into Consideration, Combustion Engines, 1/2008 (132), 56-61.
13. **Tutak W., 2011**: *Modelling and analysis of some parameters of thermal cycle of IC engine with EGR.* Combustion Engines 4/2011 (147), 43-49.
14. **Tutak W., Jamrozik A., 2010**: *Modeling of thermal cycle of gas engine using AVL FIRE software.* Combustion Engines, R.49 nr 2 (141), 105-113.
15. C**upiał K., Tutak W., Jamrozik A., Kociszewski A., 2011**: The accuracy of modelling of the thermal cycle of a compression ignition engine. Combustion Engines, R.50 nr 1 (144), 37-48.
16. **Tutak W., Jamrozik A., Kociszewski A., Sosnowski M., 2007**: Numerical analysis of initial swirl profile influence on modelled piston engine work cycle parameters. Combustion Engines, 2007-SC2, 401-407.
17. **Tutak W., 2008**: Thermal Cycle of Engine Modeling with Initial Swirl Proces Into Consideration, Combustion Engines, 1/2008 (132), pp. 56-61.
18. **Renganathan Manimaran, Rajagopal Thundil Karuppa Raj and Senthil Kumar K., 2012**: Numerical Analysis of Direct Injection Diesel Engine Combustion using Extended Coherent Flame 3-Zone Model. Research Journal of Recent Sciences, Vol. 1(8), 1-9.
19. **Binesh A.R., Hossainpour S., 2008**: Three dimensional modeling of mixture formation and combustion in a direct injection heavy-duty diesel engine, World Academy of Science, Engineering and Technology 41, 207-212.
20. **Hélie J., Trouvé A., 2000**: A modified coherent flame model to describe turbulent flame propagation in mixtures with variable composition. Proceedings of the Combustion Institute, Volume 28, Issue 1, 193-201.
21. AVL FIRE, VERSION 2009ICE, Physics & Chemistry. Combustion, Emission, Spray, Wallfilm, 2009, Users Guide.
22. **Colin O., Benkenida A., 2004**: The 3-Zones Extended Coherent Flame Model (ECFM3Z) for Computing Premixed/Diffusion Combustion, Oil & Gas Science and Technology.
23. **Tatschl R., Priesching P., Ruetz J., 2007**: 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, Detroit, MI.
24. **Tutak W., Jamrozik A., Kociszewski A., 2011**: Three Dimensional Modelling of Combustion Process in SI Engine with Exhaust Gas Recirculation. International Conference on Heat Engines and Environmental Protection, 203-208.

25. **Tutak W., 2009**: Modelling of flow processes in the combustion chamber of IC engine. Perspective Technologies and Methods in MEMS Design, MEMSTECH 2009. 45-48.
26. **Szwaja S., 2011**: Knock and combustion rate interaction in a hydrogen fuelled combustion engine. Journal of KONES Powertrain and Transport, T. 18, 431-438.
27. **Tutak W., 2012:** Influence of exhaust gas recirculation on the ignition delay in supercharged compression ignition test engine. ECONTECHMOD, An International Quartely Journal on Economics of Technology and Modelling Processes. Vol 1, No 2, 57-62.
28. **Jamrozik A. Tutak W., 2010**: Numerical analysis of some parameters of gas engine. Polish Academy of Science Branch in Lublin, TEKA, Commission of Motorization and Power Industry in Agriculture, Vol. X, 491-502.
29. **Szwaja S., Jamrozik A., Tutak W., 2013**: A Two-Stage Combustion System for Burning Lean Gasoline Mixtures in a Stationary Spark Ignited Engine. Applied Energy, 105 (2013), 271-281, 2013.
30. **Szwaja S., 2009**: Combustion Knock – Heat Release Rate Correlation of a Hydrogen Fueled IC Engine Work Cycles. 9th International Conference on Heat Engines and Environmental Protection. Proceedings. Balatonfured, Hungary.
31. **Szwaja S, Jamrozik A., 2009**: Analysis of Combustion Knock in the SI Engine. PTNSS KONGRES. International Congress of Combustion Engines. The Development of Combustion Engines.
32. **Jamrozik A., 2009**: Modelling of two-stage combustion process in SI engine with prechamber. MEMSTECH 2009, V-th International Conference PERSPECTIVE TECHNOLOGIES AND METHODS IN MEMS DESIGN, Lviv-Polyana, 22-24.
33. **Jamrozik A., 2008**: Analiza numeryczna procesu tworzenia i spalania mieszanki w silniku ZI z komorą wstępną. Teka Komisji Motoryzacji Polskiej Akademii Nauk oddział w Krakowie. Konstrukcja, Badania, Eksploatacja, Technologia Pojazdów Samochodowych i Silników Spalinowych, Zeszyt Nr 33-34.

MODELOWANIE CFD SILNIKA O ZAPŁONIE SAMOCZYNNYM PRZY OBCIĄŻENIACH CZĘŚCIOWYCH

Streszczenie. W pracy przedstawiono wyniki modelowania CFD obiegu cieplnego tłokowego silnika spalinowego o zapłonie samoczynnym. Przedstawiono siatkę obliczeniową modelowanego silnika oraz weryfikację przyjętego modelu dla trzech obciążeń silnika 100, 70 i 45%. Weryfikacja modelu polegała na porównaniu przebiegu ciśnienia w cylindrze silnika modelowanego z przebiegami uzyskanymi drogą eksperymentalną, porównano także przebiegi szybkości wydzielania ciepła. Szybkość wydzielania ciepła w silniku jest bardzo miarodajnym parametrem obiegu cieplnego silnika tłokowego. W prezentowanym modelowaniu uzyskano niezależność wyników od gęstości siatki, co jest zagadnieniem kluczowym w modelowaniu. Uzyskaną zgodność wyników uznano za satysfakcjonującą.
Słowa kluczowe: silnik tłokowy, spalanie, modelowanie, siatka obliczeniowa.

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