ECONTECHMOD. AN INTERNATIONAL QUARTERLY JOURNAL – 2012, Vol. 01, No. 2, 57–62

Influence of exhaust gas recirculation on the ignition delay in supercharged compression ignition test engine

W. Tutak

Czestochowa University of Technology, Institute of Thermal Machinery,
42-201 Czestochowa, tutak@imc.pcz.czest.pl

Received March 18.2012: accepted April 30.2012

Abstract. The results of analysis of thermal cycle of the test engine are presented in the paper. The study focused on determining the ignition delay in compression ignition engine. The correlations available in literature, Hardenberg and Hase, Wolfer and Watson and Assanis were used to determine ignition delay. With the increase of the EGR the ignition delay has increased. It turned out that very often it is necessary to determine own ignition delay correlation.

Key words: ignition delay, combustion, modeling, engine

INTRODUCTION

Ignition delay is one of the most important parameters of diesel engines which will directly affect the performance, emissions and combustion. A number of investigations have been conducted to study the ignition delay of diesel fuel. The results showed that the ignition delay depends on fuel parameters and pressure, temperature and excess air fuel ratio.

Rodríguez and all [3, 2, 9] in their work the results of engine tests of biodiesels obtained by transesterification of palm oil and rapeseed oil and with fossil diesel fuel as a reference have presented. Palm oil and rapeseed oil biodiesel gave shorter ignition delay than fossil diesel fuel due to the higher cetane number for the biodiesels. The ignition delay data were correlated as a function of the equivalence ratio, the mean cylinder pressure and mean temperature over the ignition delay interval. A comparison was made with other available correlations [3]. In their study the correlation for predicting the ignition delay of two biodiesels in a direct injection diesel engine was developed. The start of combustion was estimated using the pressure rise curves. At each condition, ignition delay was determined as the difference between start of injection and start of combustion. The new proposed correlations for biodiesels have been compared against the Watson and Assanis correlations. The comparison of results showed that the new correlations predict ignition delay for biofuels better than the available correlations for diesel fuel. It is therefore concluded that the new correlations significantly improve the ignition delay predictiveness for biodiesels in a wide range of parameters such as the cylinder pressures and temperatures at injection, the equivalence ratio, and the engine load and speed conditions [3]. Alkhulaifi and Hamdalla [4] in they work results of studies of ignition delay are presented. Watson, Assanis, Hardenberg and Hase correlations have been developed based on experimental data of diesel engines. However, they showed limited predictive ability of ignition delay when compared to experimental results. The objective of the study was to investigate the dependency of ignition delay time on engine brake power. An experimental investigation of the effect of automotive diesel and water diesel emulsion fuels on ignition delay under steady state conditions of a direct injection diesel engine was conducted [4, 12]. The ignition delay experimental data were compared with predictions of Assanis and Watson ignition delay correlations. The results of the experimental investigation were then used to develop a new ignition delay correlation. The newly developed ignition delay correlation has shown a better agreement with the experimental data than Assanis and Watson when using automotive diesel and water diesel emulsion fuels especially at low to medium engine speeds at both loads. In addition, the second derivative of cylinder pressure which was the most widely used method in determining the start of combustion was investigated [4, 16, 17]. Zou and all in their work the ignition delay of a dual fuel engine operating with methanol ignited by pilot diesel have investigated. The experimental results showed that the polytrophic index of compression process of the dual fuel engine decreases linearly while the ignition delay increases with the increase in methanol mass fraction. Compared with the conventional diesel engine, the ignition delay increment of the dual fuel engine was about 1.5° at a methanol mass fraction of 62%, an engine speed of 1600 r/min, and full engine load [11]. With the elevation of the intake charge temperature from 20°C to 40°C and then to 60°C, the ignition delay of the dual fuel engine decreases and was more obvious at high temperature. Moreover, with the increase in engine speed, the ignition delay of the dual fuel engine by time scale (ms) decreased clearly under all engine operating conditions. However, the ignition delay of the dual fuel engine increased remarkably by advancing the delivery timing of pilot diesel, especially at light engine loads [11, 24, 26]. Liu and Karim [13] changes in the physical and chemical processes during the ignition delay period of a gas-fueled diesel engine (dual-fuel engine) due to the increased admission of the gaseous fuels and diluents has examined. The extension to the chemical aspects of the ignition delay with the added gaseous fuels and the diluents into the cylinder charge was evaluated using detailed reaction kinetics for the oxidation of dual-fuel mixtures at an adiabatic constant volume process while employing n-heptane as a representative of the main components of the diesel fuel. The extension to the chemical process of the ignition delay, which results from the chemical interactions between the diesel and gaseous fuels, was the main rate-controlling process during the delay period of the dual-fuel engine. The extent of the extension to the ignition delay period depends strongly on the type of the gaseous fuel used and its concentration in the cylinder charge [13]. In spark ignition engines, under defined conditions, a self-ignition of the air-fuel mixture can occur, too [15, 18]. This phenomenon is known as knock combustion [19, 22].

Many researches around the world are involved in the modeling of the process of combustion in compression ignition engines [23, 28, 14, 18]. It is advanced computer programs that are used for this purpose, which serve for solving flows in combustion engine chambers of any geometry by numerical methods. These are programs belonging to the fluid mechanics field, where numerical methods are employed for solving CFD (Computational Fluids Dynamics) problems. One of them is AVL Fire.

THEORETICAL APPROACH

The ignition delay is the time between the start of injection and the start of combustion. It is widely accepted that the ignition delay has a physical and a chemical delay. The physical delay is the time required for fuel atomization, vaporization and mixing with the air, whereas the chemical delay is the pre-combustion reaction of fuel with air [1]. Ignition delay in diesel engines has a direct effect on engine efficiency, noise and exhaust emissions. A number of parameters directly affect the ignition delay period, among them cylinder pressure and temperature, swirl ratio and misfire. In addition to these effects the recent trend of changing fuel quality and types has a great effect on ignition delay. Experimentally, the start of ignition is mainly determined by the first appearance of visible flame on a high speed video recording [5], or sudden rise in cylinder pressure or temperature caused by the combustion [6, 25, 27].

In the literature there are many correlations for predicting ignition delay. They exist as a function of engine and charge parameters. This correlation can be written of the form:

$$\tau_{id} = Ap^{-n} \exp\left(\frac{E_A}{RT}\right) \tag{1}$$

where: E_A – apparent activation energy for the fuel auto ignition process, R – universal gas constant, A and n – constants dependent on the fuel.

The ignition delay is defined as the time between the start of fuel injection and the start of detectable heat release [1]. The ignition delay is a function of mixture pressure, temperature, excess air ratio. In an engine, pressure and temperature change during the delay period due to the compression resulting from piston motion. To account for the effect of changing conditions on the delay the following empirical integral relation is usually used [1]:

$$\int_{t_{si}}^{t_{si}+\tau_{id}} \left(\frac{1}{\tau}\right) dt = 1, \tag{2}$$

where: t_{si} – the time of start of injection, τ_{id} – the ignition delay period, τ - the ignition delay at the conditions pertaining at time t.

Well-known correlation describing the ignition delay is formula developed by Hardenberg and Hase for the duration of the ignition delay period in DI engines [2]:

$$\tau_{id} = \left(0,36 + 0,22c_m\right) \exp\left[E_A\left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21,2}{p-12,4}\right)^{0,63}\right], \tag{3}$$

where: τ_{id} – ignition delay in crank angle degrees (CA), T – temperature in K, p – pressure in bars. E_A – the apparent energy activation, c_m – the mean piston speed (m/s), R – the universal gas constant (8.3143 J/molK).

The activation energy is given by:

$$E_A = \frac{618840}{CN + 25}, \tag{4}$$

where: CN – the fuel cetane number,

Another correlation proposed Wolfer [9], Watson [10]:

$$\tau_{id} = 3,45 \frac{\exp\left(\frac{2100}{T}\right)}{p^{1,02}}, \tag{5}$$

where: p – pressure, T – temperature.

The correlation proposed by Assanis [7] is a function of equivalence ratio, with a pre-exponential factor of 2.4

that considers the equivalence ratio variations (from 2.6 to 3.8 for the combined pre-exponential factor) as opposed to the Watson correlation where it is fixed at 3.45. This constant value used by Watson would translate into =0.116:

$$\tau_{id} = 2,4 \frac{\exp\left(\frac{2100}{T}\right)}{p^{1,02}\varphi^{0,2}}. \qquad (6)$$

In AVL Fire the ignition delay is calculated on the basis of correlation [8]:

$$\tau_{id} = 4,804 \cdot 10^{-8} \left(N^{u,M}_{O_2}\Big|_{u,M} \right)^{-0,53} \left(N^{u,M}_{Fu}\Big|_{u,M} \right)^{0,05} \left(\bar{\rho}^{u} \right)^{0,13} e^{\frac{5914}{T^{u}}}$$

where: mole fraction of species (O_2, fuel) is in mol/m^3, temperature in K, density $\bar{\rho}^u$ in kg/m^3

THE OBJECT OF INVESTIGATION

Modeling of the thermal cycle of an auto-ignition internal combustion engine in the AVL FIRE program was carried out within the study. The object of investigation was a 6CT107 turbocharged auto-ignition internal combustion engines fed with diesel oil, installed on an ANDORIA-MOT 100 kVA/ 80kW power generating set in a portable version.

Engine specification:

- CI 6-cylinder engine, supercharged,
- 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.

The computational grids used in modeling is presented in figure 1. Computations were conducted for the angle range from -180° CA before TDC to 180° CA after TDC.

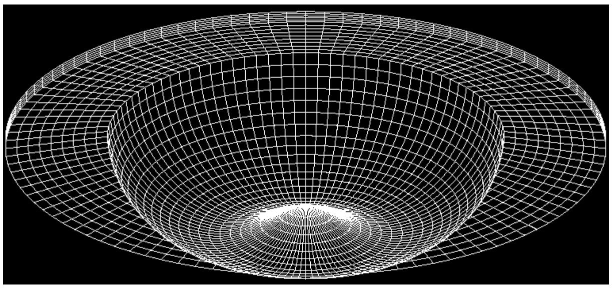

Fig. 1. The computational grid for combustion chamber modeling in TDC

The grid of the modeled combustion chamber of the 6CT107 test engine consisted of nearly 36000 computation cells. Two-layered wall boundary layer was considered.

Table 1. Modeling parameters

Engine rotational speeds	-	1500 rpm
Cylinder bore	-	107.19 mm
Crank throw	-	60.325 mm
Connecting-rod length	-	245 mm
Initial pressure for 180° CRA before TDC	-	0.16 MPa
Initial temperature for 180° CRA before TDC	-	310 K
Injection angle	-	-9° CA before TDC
Injected fuel mass	-	0.0735 g/cycle
Injection duration angle	-	20° CA
Fuel temperature	-	330 K
FIRE program's sub-models		
Turbulence model	-	k-zeta-f
Combustion model	-	Coherent Flame ECFM-3Z Model

The ECFM (Extended Coherent Flame Model) model [8] was developed specially for modeling the combustion process in a compression ignition engine. The CFM has been successfully used for modeling the process of combustion in spark ignition engines. The ECFM-3Z model belongs to a group of advanced models of the combustion process in a compression ignition engine. For several years it 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, and other sub-models, they constitute a useful tool for modeling and analysis of the thermal cycle of the compression ignition internal combustion engine. 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 [8]. The combustion model for the self-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 [20, 21].

RESULTS AND DISCUSSION

The paper presents results of 3D modeling of engine thermal cycle operating at a constant rotational speed.

The work investigates the influence of EGR on engine operating parameters, on heat release rate and ignition delay. The study was conducted for constant injection timing and load. Research engine is supercharged engine. After the validation process of the model were started modeling. Highly compatible modeling and experimentally obtained results (Fig. 2) was received. Calculated engine efficiency is the gross efficiency, the modeling does not include charge exchange loop.

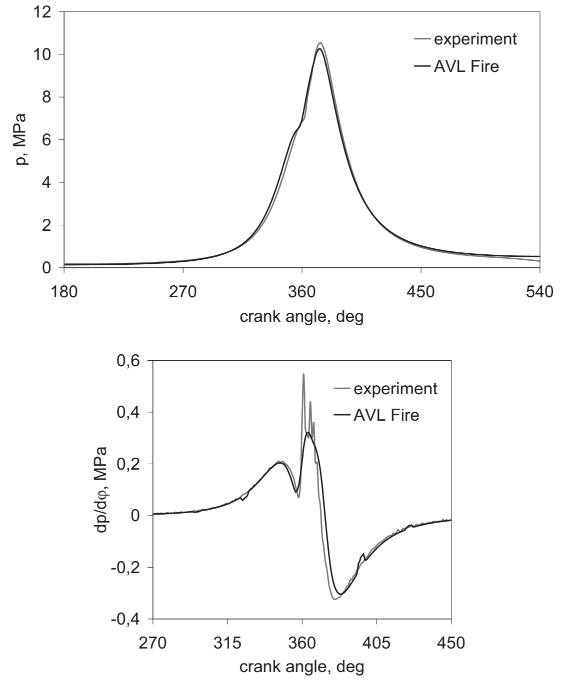

Fig 2. Result of model validation, traces of pressure and dp/dφ.

Figure 2 shows the comparison of engine cylinder pressure obtained through the real engine indication and modeling. Satisfactory agreement of these curves (p and dp/dφ.) at this point of engine work was achieved.

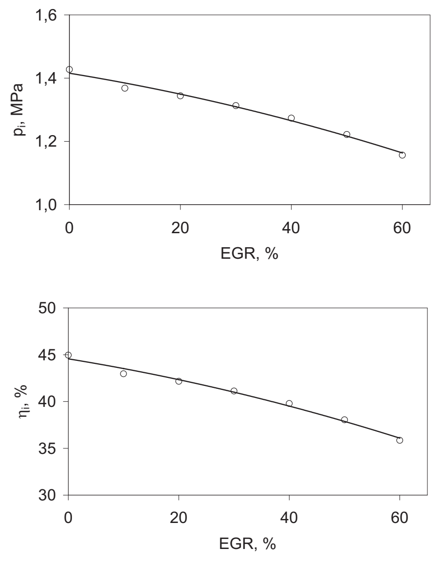

Fig. 3. Mean indicated pressure and indicated efficiency

Figure 3 shows the influence of EGR on the mean indicated pressure and indicated efficiency of the test engine. At the time of increase the participation of the EGR mass fraction of fuel injected into the cylinder was constant. At 60% share of the EGR the biggest drop in efficiency was received, and it reached a value of 36%. Similarly, the value of indicated pressure was decrease to value equal 1.16 MPa. There has not been optimization of the thermal cycle of the test engine.

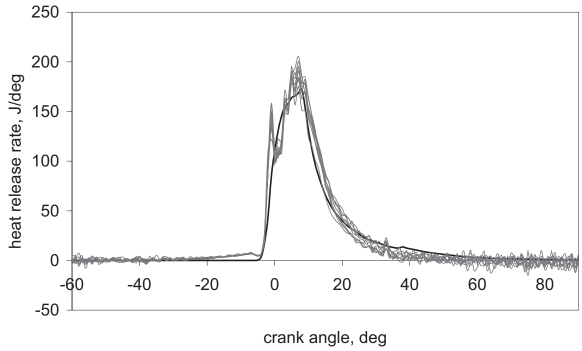

Fig. 4. Comparison of traces of heat release rate taken from real engine (set of red traces) and modeling (black)

Figure 4 shows a comparison of the rate of heat release curves. One line represents heat release rate of modeled engine and the others traces were obtained on the basis of experimental studies. The curves of heat release rate were used to determine the ignition delay. The algorithm for determining the ignition delay was presented in the literature review.

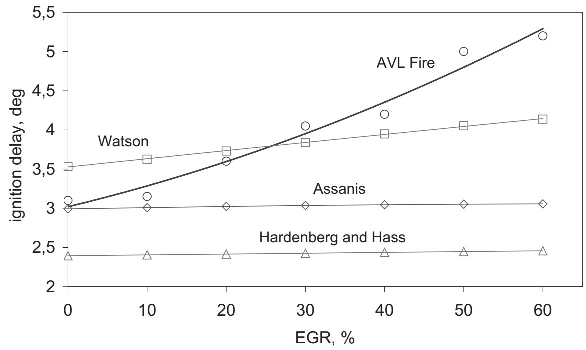

Fig. 5. Ignition delay lines calculated on the basis of correlations and determined on the basis of modeling results

Figure 4 shows the lines for ignition delay calculated using Hardenberg and Hase, Watson and Assanis correlations and determined on the basis of modeling results. In conditions without EGR, the closest value of ignition delay was obtained by using of the Assanis correlation. With the increasing of the recirculated exhaust gas participation, the ignition delay increased, which is consistent with results obtained by other authors. The results obtained with the use of Hardenberg and Hase and Assanis correlation did not give satisfactory results.

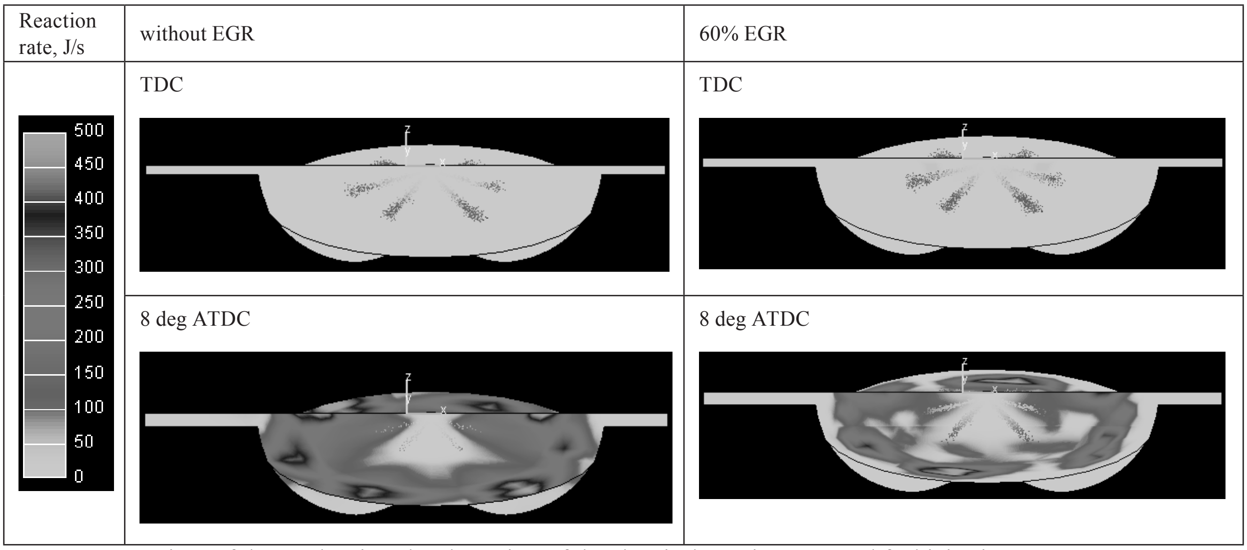

Fig 6. Cross sections of the combustion chamber. View of the chemical reactions rate and fuel injection process

With the increase of the EGR to 60% the ignition delay value has increased from 3.1 to 5.2 deg.

Figure 6 shows the cross sections, in two planes, of the combustion chamber. The reaction rates of combustion in the modeled combustion chamber are presented. It is clear that in the case with EGR the space covered by combustion process in the combustion chamber is smaller than in the case without recirculation. This is due, inter alia, by the ignition delay.

CONCLUSIONS

The results of analysis of thermal cycle of the test engine are presented in the paper. The study focused on determining the ignition delay in compression ignition engine. Ignition delay is one of the most important parameters of diesel engines which will directly affect the performance and emissions. In order to determine this parameter, the correlations available in literature were used. The ignition delay on the basis of modeling results was determined, too. It turned out that, in this case, these correlations did not give satisfactory results. With the increase of the EGR to 60%, the ignition delay value increased from 3.1 to 5.2 deg, in the model test engine.

Based on the literature study it can be said that other authors also stated that these universal correlations usually do not give the expected results. Often it is necessary to determine the correlation by defining ignition delay.

REFERENCES

1. **Heywood J. B. 1988.** *Internal combustion engine fundamentals.* McGraw-Hill.
2. **Hardenberg H. O. and Hase F. W. 1979.** *An Empirical Formula for Computing the Pressure Rise Delay of a Fuel from Its Cetane Number and from the Relevant Parameters of Direct-Injection Diesel Engines.* SAE Paper 790493, SAE Trans. Vol. 88, 1979, DOI: 10.4271/790493
3. **Rodríguez R. P., Sierens R. and Verhelst S. 2011**. *Ignition delay in a palm oil and rapeseed oil biodiesel fuelled engine and predictive correlations for the ignition delay period.* Fuel 90 (2011), 766-772.
4. **Alkhulaifi K. and Hamdalla M. 2011.** *Ignition Delay Correlation for a Direct Injection Diesel Engine Fuelled with Automotive Diesel and Water Diesel Emulsion.* World Academy of Science, Engineering and Technology 58.
5. **Lee, J.H. and Lida N. 2001.** *Combustion of diesel spray injected into reacting atmosphere of propane-air homogeneous mixture.* Int. J. Engine Res., 2(1): 69-80.
6. **Aligrot, C., J.C. Champoussin and N. Guerrassi, 1997.** *A correlative model to predict auto ignition delay of diesel fuels.* SAE transactions, 106(3): 958-963.
7. **Assanis, D.N., Z.S. Filipi and S.B. Fiveland, 2003.** *A predictive ignition delay correlation under steady-state and transient operation of a direct injection diesel engine.* J. Eng. Gas Turbines Power, 125: 450.
8. **Colin O. and Benkenida A. 2004.** *The 3-Zones Extended Coherent Flame Model (ECFM3Z) for Computing Premixed/Diffusion Combustion.* Oil & Gas Science and Technology.
9. **Wolfer H.H. 1938.** *Ignition lag in diesel engines.* VDI-Forschungsheft, 392: p. 621-436.047
10. **Watson N., Pilley A. D. and Marzouk M. 1980.** *A Combustion Correlation for Diesel Engine Simulation.* SAE 800029.
11. **Zou H., Wang L., Liu S. and Li Y. 2008.** *Ignition delay of dual fuel engine oprating with methanol ignited by pilot diesel.* Front. Energy Power Eng. China, 2(3): 285-290, DOI: 10.1007/s11708-008-0060-z
12. **Asad U. and Zheng M. 2008.** *Fast Heat Release Characterization of Modern Diesel Engines*, International Journal of Thermal Sciences, Vol. 47, Issue 12, 1688-1700, doi:10.1016/j.ijthermalsci.2008.01.009.
13. **Liu Z. and Karim G. A. 1998.** *An Examination of the Ignition Delay Period in Gas-Fueled Diesel Engines.* Transaction of the ASME Journal of Engineering for Gas Turbines and Power, 120, January, 225-231, January 01.
14. **Cupiał K., Tutak W., Jamrozik A. and Kociszewski A. 2011.** *The accuracy of modelling of the thermal cycle of a compression ignition engine.* Combustion Engines.
15. **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, Warszawa.

16. **Tutak W. 2011.** *Numerical analysis of the impact of EGR on the knock limit in SI test engine*. TEKA PAN, 397-406, T11.
17. **Tutak W. 2011.** *Numerical analysis of some parameters of SI internal combustion engine with exhaust gas recirculation*. TEKA PAN, 407-414. T11.
18. **Jamrozik J. and 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, Lublin.
19. **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, 83-88, Hangary.
20. **Szwaja S. and Naber J.D. 2009.** *Combustion of N-Butanol in a Spark-Ignition IC Engine*, Fuel, 2009.
21. **Szwaja S. 2009.** *Hydrogen Rich Gases Combustion in the IC Engine*, Journal of KONES Powertrain and Transport Vol.16 nr 4, 447-454.
22. **Szwaja S. 2009.** *Time-Frequency Representation of Combustion Knock in an Internal Combustion Engine*, Silniki Spalinowe R.48 nr SC2, 306-315, 2009.
23. **Jamrozik A. 2006.** *Modelowanie procesu tworzenia tlenku azotu w komorze spalania gazowego silnika ZI*. VII Międzynarodowa Konferencja Naukowa SILNIKI GAZOWE 2006, Zeszyty Naukowe Politechniki Częstochowskiej 162, Mechanika 26, 348-359.
24. **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, Zeszyt Nr 33-34, Kraków, 143-150.
25. **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, UKRAINE, 13-16.
26. **Jamrozik A. 2011.** *Analysis of indication errors of the SI gas engine with a prechamber*. TEKA PAN. Teka Commission of Motorization and Power Industry in Agriculture. Volume XI, 2011, 143-156.
27. **Jamrozik A. 2011.** *Numerical optimization of ignition in the internal combustion engines*. Teka PAN, Teka Commission of Motorization and Power Industry in Agriculture. Volume XI, 2011, 157-165.
28. **Jamrozik A. 2011.** *Numerical study of EGR effects on the combustion process parameters in HCCI engines*. Combustion Engines, No. 4/2011 (147), 2011, 50-61.

Acknowledgements. The author would like to express his gratitude to AVL LIST GmbH for providing a AVL Fire software under the University Partnership Program.