SIMULATION ANALYSIS OF THE EFFECT OF INLET PIPE DIAMETER ON THE CYLINDER FILLING OF COMPRESSION-IGNITION ENGINE

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Abstract

In this paper are presented the basic problems accompanying the flow of charge through compression-ignition engine inlet system. The flow of medium through inlet pipe to cylinder was assumed to be one-dimensional and described with mass, momentum and angular momentum conservation equations and energy conservation law. On the other hand, a zero-dimensional model described by mass-energy conservation equations was used to determine parameters of the medium in total cylinder volume. Simulations were carried out with the use of numerical programme in which McCormack's explicit method was applied to solve differential equations describing the flow of charge in inlet system. The McCormack explicit technique was used due to the fact that an earlier attempt at solving a numerical problem with the method of lines had not yielded desired effects in view of high instability. Simulation aimed at determining the mass of air remaining in engine cylinder after inlet valve closure. Simulation results are presented in the form of in-cylinder air mass versus crank angle characteristics for different inlet pipe diameters. The study results are also presented in the form of a summary table and a summary diagram, the latter enabling a comparison of results for different inlet pipe diameters.

***Keywords**: inlet pipe diameter, in-cylinder mass, numerical model, inlet system, simulation, combustion engines*

1. Introduction

Charge exchange process in piston combustion engine is an important part of the whole thermal cycle of engine. It consists in the removal of residues after combustion process in previous thermal cycle from engine cylinder and the delivery of fresh charge for next cycle to a cylinder. Part of charge exchange process during which fresh charge is being delivered to a cylinder is usually called the filling process [1-3].

The flow of charge from the surroundings into engine cylinder is possible owing to negative pressure in a cylinder created due to piston motion from top dead centre (TDC) to bottom dead centre (BDC).

The occurrence of a number of phenomena, unfavourable in most cases, inducing smaller cylinder filling with fresh charge than it would result from displacement volume is connected with charge flow. Unfavourable phenomena decreasing the cylinder filling may include throttling in inlet system throats, charge warming up in inlet system and flow resistance in engine inlet system [4, 8].

Inlet system construction solution, dimensions of its components, workmanship accuracy, surface state and arrangement of system's elements may crucially affect the filling of engine cylinder and, what is connected with this, its operation parameters such as power and torque, and in particular its dependence versus rotational speed, fuel consumptions and engine response. The aforementioned engine operation parameters have a significant effect on motor-car traction properties and driving economy. Inlet system may also affect the emission of toxic exhaust gas components such as carbon and nitrogen oxides, hydrocarbons and particulate solids [5, 6].

The value of filling efficiency can be roughly determined having available the data referring to the value of temperatures and pressures characteristic for respective points of the theoretical diagram of filling process course. By carrying out an appropriate analysis, the following equation is being obtained [8]:

$$\eta_v = \frac{T_0}{p_0(\varepsilon - 1)}\left(\frac{\varepsilon \cdot p_s}{T_s} - \frac{p_r}{T_r}\right), \tag{1}$$

where:
T_0 - ambient temperature,
p_0 - ambient pressure,
T_s - temperature of medium in inlet valve passage,
P_s - pressure of charge in inlet valve passage,
T_r - temperature of exhaust gas residues,
P_r - exhaust gas pressure in exhaust valve passage,
ε - compression ratio.

When analysing the above dependence, it is possible to state that the filling efficiency is greatly affected by the inlet side of engine expressed by parameters p_s and t_s than the exhaust one expressed by parameters p_r and T_r [8].

2. Numerical model of inlet system

The flow of air in inlet system may be described by basic equations which were formulated for a general fluid model and resulting from three fundamental principles of mechanics [9, 10], namely:
- mass conservation law,
- momentum and angular momentum conservation law, and
- energy conservation law.

In the process of cylinder filling with air, two groups of physical phenomena can be distinguished. In the first group, one-dimensional elastic flow through inlet pipe is being analysed, whereas in the second one the in-cylinder phenomena being described by zero-dimensional models which ignore displacement of charge in the cylinder space. Hence, momentum and angular momentum conservation law is not being taken into consideration when describing the in-cylinder phenomena [11, 12].

When making a number of essential simplifying assumptions, the equations of mass, momentum and angular momentum and energy conservation for the flow in inlet pipe assume the following form:

$$\begin{aligned} \frac{\partial \rho}{\partial t} &= -u\frac{\partial \rho}{\partial x} - \rho\frac{\partial u}{\partial x}, \\ \frac{\partial u}{\partial t} &= -\frac{1}{\rho}\frac{\partial \rho}{\partial x} - u\frac{\partial u}{\partial x} - u \cdot k_t, \\ \frac{\partial p}{\partial t} &= u^2 \cdot k_t \cdot \rho(\chi - 1) - u\frac{\partial p}{\partial x} - \chi \cdot p\frac{\partial u}{\partial x}. \end{aligned} \tag{2}$$

When modelling the phenomena taking place in engine cylinder during the filling process, displacement of air particles inside cylinder in time and space are usually ignored using a zero-dimensional model to describe the in-cylinder phenomena. Furthermore, it is being assumed that in-cylinder pressure is equal in any point of space and that it is a scalar value. It results from this assumption as well as from the assumption that air is a semi-ideal gas that it is possible to ignore momentum and angular momentum conservation equation and that all in-cylinder phenomena can be described by mass and energy conservation equations and the Clapeyron equation of semi-ideal gas:

$$\frac{dm}{dt} = \frac{dm_d}{dt} + \frac{dm_w}{dt} + \frac{dm_p}{dt}, \tag{3}$$

$$dQ_{sp} + dQ_{ws} + dI_d = dI_w + dI_p + dU + pdV, \tag{4}$$

$$pV = mRT, \tag{5}$$

where:
m - in-cylinder mass of charge,
m_d - mass of charge flowing through inlet valve,
m_w - mass of charge flowing through exhaust valve,
m_p - mass of charge flowing through cylinder leaks.
Q_{sp} - heat released in the combustion process,
Q_{ws} - heat exchanged with cylinder walls,
I_d - enthalpy of charge flowing through suction valve,
I_p - enthalpy of charge blown through cylinder leaks,
U - in-cylinder internal energy of charge,
V - cylinder volume,
p - mean in-cylinder momentary pressure,

$$\frac{dm}{dt} = \mu A p_0 \sqrt{\frac{2}{RT_0}\frac{\chi}{\chi-1}\left[\left(\frac{p}{p_0}\right)^{\frac{2}{\chi}} - \left(\frac{p}{p_0}\right)^{\frac{\chi+1}{\chi}}\right]},$$

μ - flow ratio,
A - momentary flow surface of valve,
p - pressure behind flow surface,
p_0 - pressure before flow surface,
T_0 - temperature before flow surface,

$$\chi = \frac{c_p}{c_v},$$

$$\frac{dQ_{ws}}{dt} = h_s A_s (T_s - T),$$

h_s - heat exchange coefficient,
A_s - heat exchange surface,
T_s - mean temperature of cylinder walls,
T - momentary in-cylinder charge temperature.

3. Results of model tests

Analysis of the charge flow through inlet system (Fig. 1 and 2) should be started with examination of the course of pressures in inlet pipe and engine cylinder. It should be remarked that the amount of charge which will remain in engine cylinder after completion of the filling process will depend not only on the pressure course in inlet pipe but also on the moment when inlet valve is going to be closed. Because inlet valve is being usually closed after crossing the bottom dead centre by piston, the initial stage of compression stroke should be also taken into consideration in the analysis of in-cylinder pressure course.

In the initial stage of filling process (range 0 – approx. 70° CRA), a negative pressure increases fairly quickly together with the increase of crankshaft rotation angle because fresh charge inflow through inlet valve, which starts to open, is very small. In the subsequent stage of induction (range approx. 70 – approx. 150° CRA), fresh charge inflow increases together with the increase of inlet valve lift, inducing negative pressure decrease. Negative pressure decrease occurs in spite of the total cylinder volume increasing more and more quickly (piston speed increases from TDC up to the middle of piston travel). Therefore, fresh charge inflow through valve is quicker than the increase of total cylinder volume. The course of pressure within this range results from charge mass acceleration to large speed induced by piston motion [4].

The charge is being delivered to cylinder until the moment when the inertia of flowing charge is able to overcome the effect of charge induced by piston motion towards TDC. Inlet valve closure angle increase is unfavourable due to the phenomenon of pushing out the charge by piston through which the filling efficiency decreases. It also results from the carried out analysis that the speed of charge flowing to cylinder differs at different engine crankshaft rotational speeds and therefore its inertia and place where inertial forces equalise with the forces pushing out the charge from cylinder are different as well.

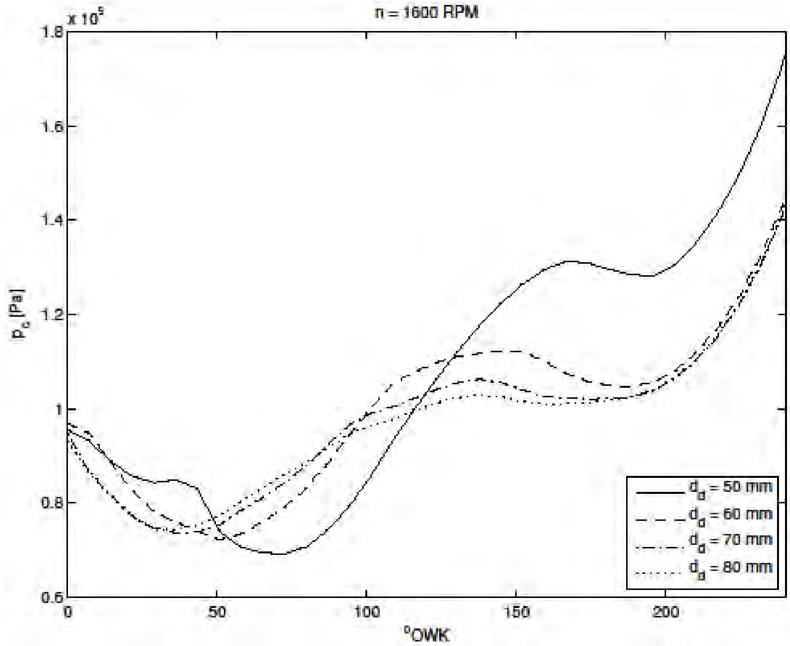

Fig. 1. Dependence of in-cylinder pressure in the function of crankshaft rotation angle at rotational speed of 1600 1/min for different inlet pipe diameters

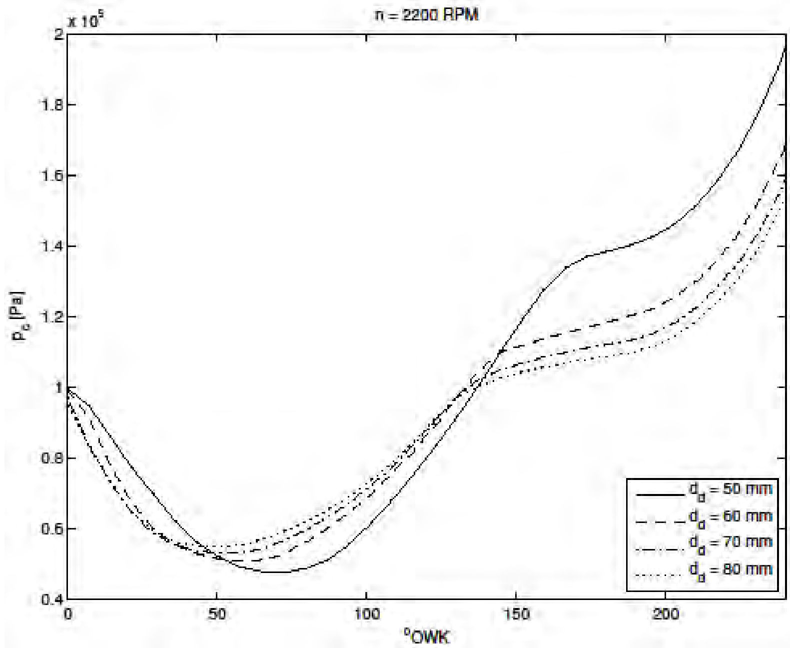

Fig. 2. Dependence of in-cylinder pressure in the function of crankshaft rotation angle at rotational speed of 2200 1/min for different inlet pipe diameters

During model tests, simulations were carried out of the effect of inlet pipe with a length of 0.843 m on charge pressure value in engine cylinder during its filling at rotational speeds ranging from 1000 to 2200 1/min, every 200 1/min.

When analysing the in-cylinder pressure courses for different inlet pipe diameters, it was found that the highest pressure at the time of inlet valve closure was obtained for inlet pipe diameter d_d = 50 mm. The same diameter has the entrance to inlet channel in the engine head. It should be remarked that the most favourable effect was obtained at rotational speeds of 1600 1/min and 2200 1/min, i.e. at the speeds of maximum torque and maximum power, respectively. Values of these pressures are larger than those for other diameters by approximately 15%. The effect of inlet pipe diameter at other rotational speeds from among the examined range was much smaller and did not exceed 5%. The given length of inlet pipe was selected based on earlier simulations and test bed examinations [5].

For the examined inlet pipe diameters, simulations were also made of the mass of air remained in cylinder after inlet valve closure.

Figures 3 and 4 present dependencies of the in-cylinder mass of air in the function crankshaft rotation angle for the examined inlet pipe diameters at rotational speeds 1600 and 2200 1/min. Two engine cycles (without combustion process) have been presented each. The straight section of the course, parallel to the horizontal axis, illustrates the mass of air which remained in cylinder after inlet valve closure. Within this section the mass of air is constant because there is no air inflow (due to closed inlet valve). Because exhaust is closed in this time, air does not flow out from cylinder either. It should be stated at the same time that charge losses resulting from leaks were ignored.

It is seen in the figures that the mass of air in engine cylinder at the bottom dead centre of piston is larger than that after inlet valve closure. This results from displacement volume decrease due to piston motion towards top dead centre. In this time, part of the air which has been already present in cylinder will be pushed out from it by piston.

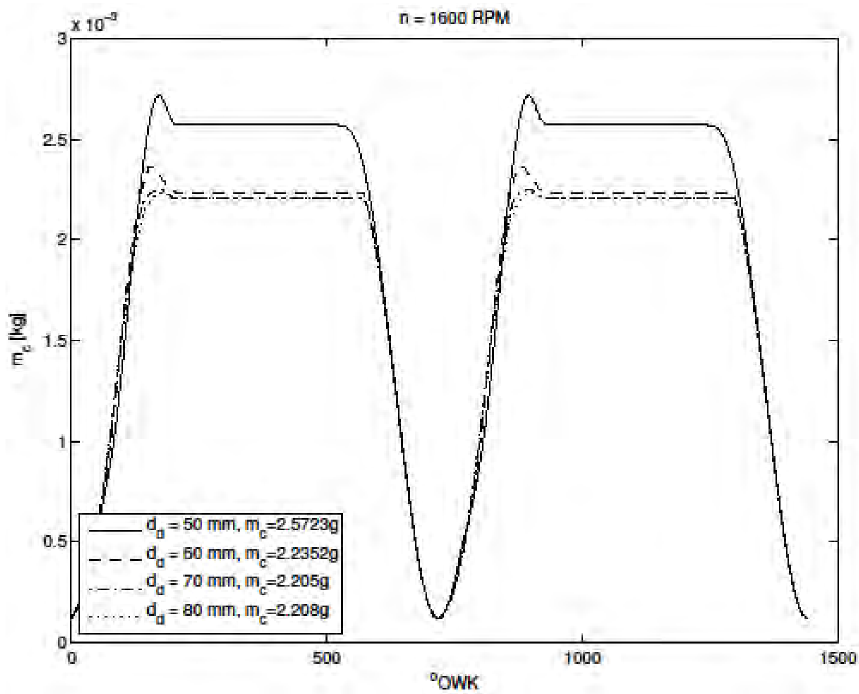

Fig. 3. In-cylinder mass of air in the function of crankshaft rotation angle for the examined inlet pipe diameters at rotational speed of 1600 1/min

Mass numerical values for all diameters and speeds are placed in Tab. 1 and presented in the form of diagram (Fig. 5).

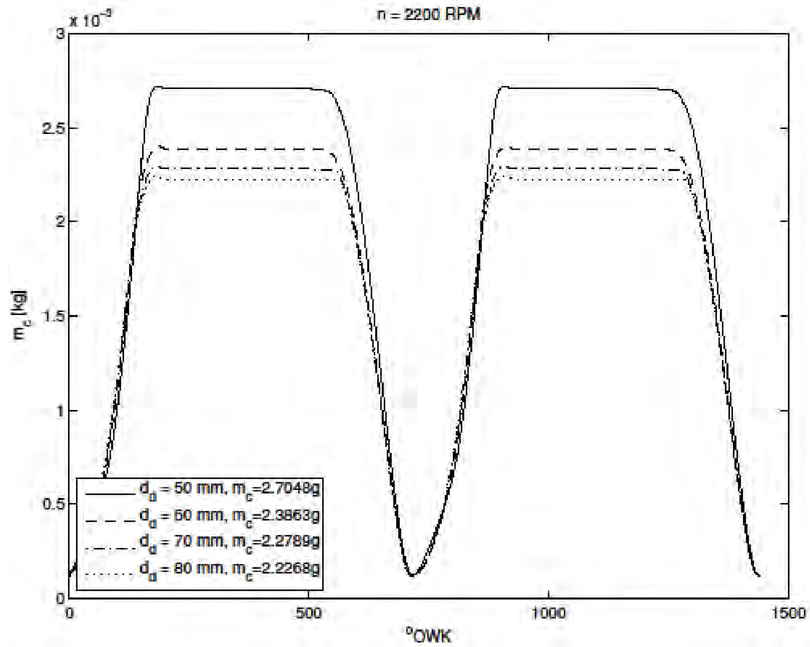

Fig. 4. In-cylinder mass of air in the function of crankshaft rotation angle for the examined inlet pipe diameters at rotational speed of 2200 1/min

Tab. 1. In-cylinder mass of air for the examined inlet pipe diameters

	Inlet pipe diameters [mm]							
	50		60		70		80	
n [1/min]	m_c	η_v	m_c	η_v	m_c	η_v	m_c	η_v
1000	2.2345	0.9855	2.1922	0.9668	2.1943	0.9677	2.2010	0.9707
1200	2.2930	1.0113	2.2679	1.0002	2.2515	0.9930	2.2404	0.9881
1400	2.3394	1.0317	2.2916	1.0106	2.2711	1.0016	2.2550	0.9945
1600	2.5723	1.1344	2.2352	0.9858	2.2050	0.9725	2.2080	0.9738
1800	2.4474	1.0794	2.2890	1.0095	2.2291	0.9831	2.2021	0.9712
2000	2.5197	1.1112	2.4451	1.0783	2.4128	1.0641	2.4096	1.0627
2200	2.7048	1.1929	2.3863	1.0524	2.2789	1.0050	2.2268	0.9821

Based on Fig. 5, it is possible to state that the largest in-cylinder mass within the whole range of rotational speed was obtained for inlet pipe with a diameter of 50 mm. The particularly large values of mass were obtained at rotational speeds 1600 1/min and 2200 1/min. It is also seen that the in-cylinder mass of air for that length of inlet pipe did not obtain its maximum yet which most probably will occur at higher rotational speed. Unfortunately, a speed of 2200 1/min is the maximum rotational speed for the examined engine. Results at other inlet pipe diameters were much worse, in particular at speeds close to the rated one. It should be also remarked that changes of mass at inlet pipe diameters larger than the diameter of inlet channel are of milder character, which suggests that the cylinder filling will be slightly smaller but within a larger range of rotational speeds.

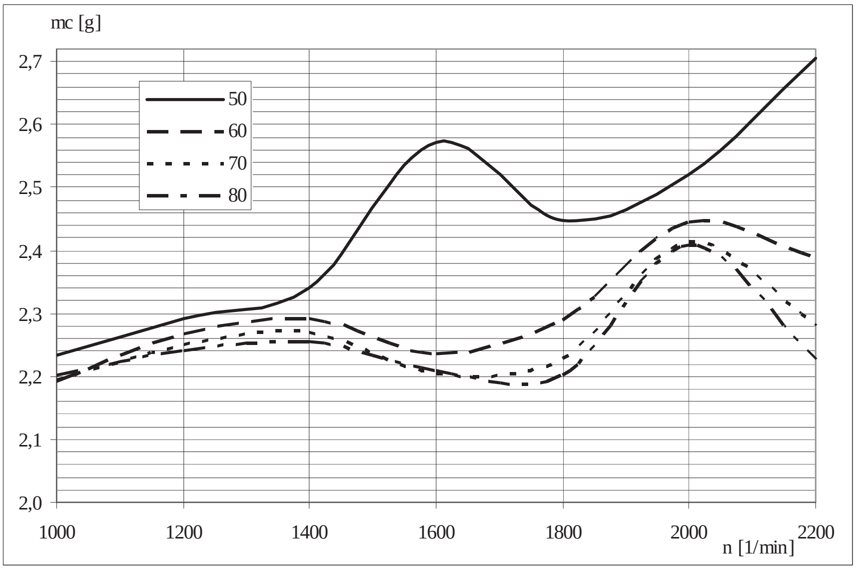

Fig. 5. Dependence of in-cylinder mass on rotational speed at examined inlet pipe diameters

4. Conclusions

Based on model calculations made by the author and earlier engine test bed examinations, it is possible to state that a number of analyses should be made referring to inlet system construction parameters and their effect on the cylinder filling and, what is connected with this, on engine operation parameters, such as power, torque or fuel consumption, when designing a new engine or rebuilding the existing one.

Inlet system construction parameters can be determined based on experimental tests but a great number of parameters will require repeated examinations. Considering the fact that the likelihood of obtaining satisfactory results as early as in the first series of tests is practically equal to zero while the execution of all planned examinations for a number of changing parameters will be unusually time-consuming and expensive. Hence, the initial estimation of required parameters and the execution of tests confirming the accuracy of selection just after this seem to be purposeful.

The developed programme enables evaluation of the effect of basic inlet system parameters on the engine filling. Based on the carried out simulation and experimental test, it is possible to come to the following conclusions:

- inlet pipe diameter has a considerable effect on the in-cylinder pressure and mass of air after completion of the filling process,
- the use of inlet pipe with a diameter larger than that of inlet channel in the engine head induces a decrease in the in-cylinder mass and pressure of air,
- according to [9], the use of inlet pipe with a diameter smaller than that of inlet channel in the engine head induces an improvement of the filling process within a range of low and medium rotational speeds and its worsening within a range of higher speeds,
- the filling process is also affected by timing gear system parameters, in particular the moment of inlet valve closure – it is possible to limit the outflow of air mass from cylinder into inlet system by slightly accelerating the moment of inlet valve closure,
- numerical simulations allow shortening of time and reduction of the cost of introduction of inlet system construction modifications but can not entirely replace experimental examinations [5, 6].

During the performed tests and analyses of study results, the author found a necessity for carrying out in future the studies that will determine as follows:

- the effect of timing gear phase on the filling process,
- the effect of ambient conditions (temperature and pressure) which will allow application of numerical programme to evaluate the filling of engine with combined supercharging system.

References

[1] Ambrozik, A., *Wybrane zagadnienia procesów cieplnych w tłokowych silnikach spalinowych*, Monografie, Studia, rozprawy, Politechnika Świętokrzyska, Kielce 2003.
[2] Brandstetter, W. T., *Ahnlichkeltsgesetze des viertaktmotors*, MTZ, Nr 9, 1967.
[3] Broome, D., *Induction Ram*, Automobile Engineer, April, May, June 1969.
[4] Kordziński, Cz., Środulski, T., *Układy dolotowe silników spalinowych*, WKŁ, Warszawa 1968.
[5] Lisowski, M., *Impact analysis of dynamic supercharging on engine properties*, Teka Komisji Motoryzacji i Energetyki Rolnictwa/PAN Oddział w Lublinie, Vol. 7A, 2007.
[6] Lisowski, M., *Określenie możliwości zastosowania doładowania dynamicznego do silnika 359*, Motoryzacja i energetyka rolnictwa, 3 Międzynarodowa Konferencja Naukowo-Techniczna MOTROL, Wydawnictwo Akademii Rolniczej, T. 4, Lublin 2001.
[7] Mysłowski, J., *Doładowanie dynamiczne silnika SW-680*, WPT Nr 12, 1975.
[8] Mysłowski, J., *Studium wpływu parametrów geometrycznych układu dolotowego tłokowego silnika spalinowego na jego napełnienie i elastyczność*, Prace Naukowe Politechniki Szczecińskiej, Nr 307, Szczecin 1986.
[9] Prosnak, W. J., *Mechanika płynów*, WNT, T. 1, 2, 1971.
[10] Prosnak, W. J., *Wprowadzenie do numerycznej mechaniki płynów, Cz. A: Podstawowe metody numeryczne*, Ossolineum 1993.
[11] Seifert, H., *Ehrfahrungen mit einen mathematischen Modell zur Simulation von Arbeitsverfaheren in Verbrennungsmotoren*, MTZ, Nr 7, 8, 12, 1978.
[12] Sobieszczański, M., *Modelowanie procesów zasilania w silnikach spalinowych*, WKŁ, Warszawa 2000.