ARCHIWUM MOTORYZACJI
1, pp. 79-90 (2008)

Problems of cooperation of a two-stroke combustion engine with an electric machine in the cogeneration system

WOJCIECH MAREK

Cracow University of Technology

The paper presents the problem of charge formation process and charge exchange in a two-stroke SI engine. This problem is connected with the application of two-stroke engine with pneumatic fuel injection (according to the conception of Prof. St. Jarnuszkiewicz) in cogeneration systems. There are presented the results of simulating and laboratory investigation connected with the cooperation of combustion engine and electric machine. The machine was equipped with an electronic controller that enables electric machine protection against overload by load current limitation, charge current limitation, output voltage limitation to full battery charge voltage. Another controller tasks are: protection against battery - generator reverse current, generator protection against too high temperature, keeping required voltages, and currents apart from ambient temperature. The problems of process of fuel charge exchange was presented as the main question. It was considered from the one hand as an energetic demand of arrangement extorting, and from the other hand as obtainments of minimise exhaust gases toxic components amount.

Nomenclature
m_0 - *mass of charge in combustion chamber before fuel injection phase*
m_{str} - *mass of exhaust gas - fuel mixture stream, which forces the pneumatic injection*
m_k - *mass of air and exhaust gases charged into combustion chamber*
m_c - *mass of air and the rest of exhaust gases in the cylinder*
r_k - *mean radius of combustion chamber*
V_i - *volume above the piston in next phases of crank shaft turn*
ρ_i - *density of mixture in next phases of crank shaft turn*
ϖ_1 - *solution angular velocity*
tw - *injection timing value*

1. Introduction

Two-stroke combustion engine with distributorless pneumatic fuel injection system by means of highly active exhaust gas directly to cylinders and the ignition and fuel dose electronic controller, under optimized operating parameters, can cooperate with typical energy exchange systems in arrangement of cogeneration system. Ignition and injection timing as well as keeping fuel dose is maintained by electronic control which gives possibilities to optimize the cogeneration process. The driving shaft of electric machine is directly connected with the driving shaft of combustion engine

without use of a transmission which simplifies the construction and improves the reliability.

2. Electric machine electronic control system

The electric machine operating as a starter motor (during combustion engine starting) or as a generator is controlled by electronic management system (Fig.1). The electronic control system assures automatic switching the electric machine from starter to generator condition and inversely. The electric machine was designed in a special form, as a separately excited one. The machine nominal parameters are: power - 5,5 kW, nominal current - 80A.

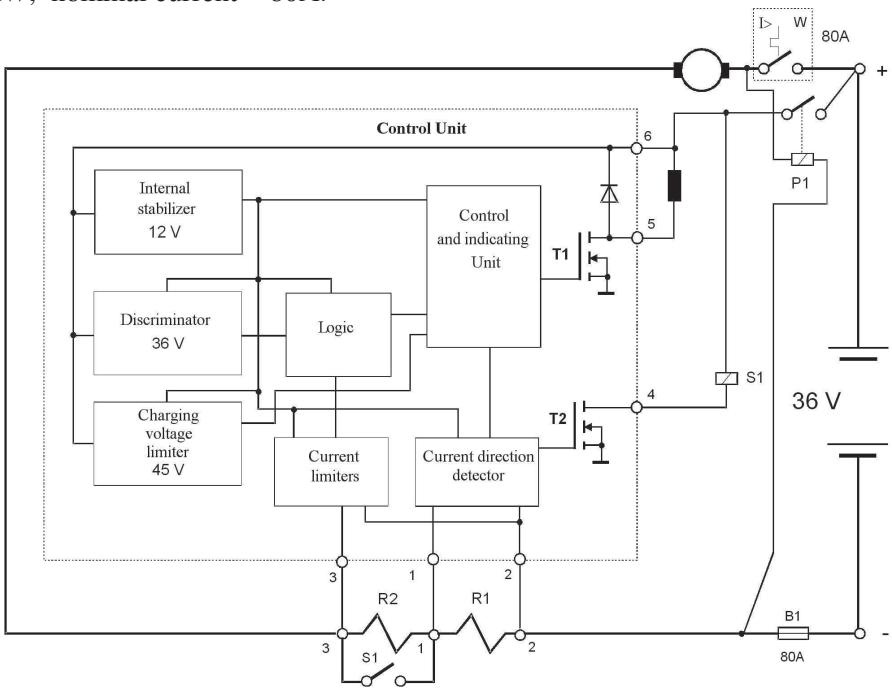

Fig. 1. Schematic diagram of electric machine control unit.
Rys. 1. Schemat układu sterowania pracą maszyny elektrycznej.

3. Analysis of charge rotating in cylinder

During the following phases of pneumatic fuel injection the certain doses of exhaust gas - fuel mixture move in a gas duct joining the proper cylinders. Due to their mass and velocity, the mixture particles have kinetic energy in the cylinder inlet orifice. This energy causes rotation of mixture particles.

The combustion chamber geometry, stream kinetic energy and rotational speed of engine decide of swirl intensity. The coefficient of swirl intensity is described as the

relation of the charge rotational speed in rotating stream to the rotational speed of engine.

If the particle of stream has kinetic energy in cylinder inlet orifice, then some part of this energy will be use for charge rotating increase. There the particle of the injection's stream has the velocity vector in every moment. The vector contains three components:

- radial component of velocity guided perpendicularly and radialy to the cylinder axis,
- contiguous component of velocity guided contiguous to the cylinder section circle,
- axial component of velocity guided parallel to the cylinder axis.

Axial component causes longitudinal swirl, and the contiguous component causes transversal swirl. The location of the axe of the orifice gas duct does not transmit the stream of charge to the side of the chamber. The value of effective kinetic energy causing the mixture swirl depends on, the value of the contiguous component of velocity to the side of the chamber. Optimal location of the inlet orifice axis depends on individual constructional solution. In this analysis was made an assumption that velocity contiguous component to a combustion chamber wall is equal to a half of the value of the stream velocity in outlet from gas duct.

The swirl analysis includes two stages:

1) from beginning to the end of fuel injection process, when the full injection's stream is delivered to the chamber,
2) after the end of fuel injection and the close of the distribution ports, in the compression phase the pressure charging occurs.

The effective kinetic energy of stream will be defined:

$$E_{ke} = \frac{I_1 \cdot \varpi_1^2}{2} \tag{1}$$

where:

I_1 – The moment of inertia of solution contained in combustion chamber after the end of fuel injection phase,

ϖ_1 – solution angular velocity.

$$I_1 = (m_0 + m_{str}) \cdot R_z^2 \tag{2}$$

where:

m_0 – The mass of charge in combustion chamber before fuel injection phase,

m_{str} – The total mass of exhaust gas - fuel mixture stream, which forces the pneumatic injection,

R_z – The supplementary radius of combustion chamber from equation:

$$R_z^2 = 0{,}5 r_k^2 \tag{3}$$

where:

r_k – The mean radius of combustion chamber.

Transforming the equation of kinetic energy the formula of angular velocity of swirl was received:

$$\varpi_1 = \sqrt{\frac{2E_{ke}}{I_1}} \tag{4}$$

To calculate the angular velocity of swirl movement after pressure charging the air to the combustion chamber in compression stroke, the quantity of charge delivered to the combustion chamber during crankshaft turn should be described in advance.

As the density of the charge in every place in the cylinder in certain time is the same, calculation of the quantity of the air flow into the chamber can be performed from the equation of density above the piston, and in the combustion chamber:

$$\Delta m_k = m_{k2} - m_{k1} = \rho_2 V_k - \rho_1 V_k \tag{5}$$

$$\Delta m_k = V_k(\rho_2 - \rho_1)$$

$$\Delta m_k = V_k\left(\frac{m_c}{V_2} - \frac{m_c}{V_1}\right)$$

$$\Delta m_k = \frac{m_c \cdot V_k}{V_1 \cdot V_2}(V_1 - V_2) \tag{6}$$

where:

Δm_k – The mass of air and exhaust gases charged into combustion chamber,

m_c – The total mass of air and the rest of exhaust gases in the cylinder,

ρ_1, ρ_2 – The density of mixture in next phases of crank shaft turn,

V_1, V_2 – The volume above the piston in next phases of crank shaft turn.

The volume above the piston was calculated from formula:

$$V = V_S\left(\frac{1}{\varepsilon_t - 1} + \frac{1}{2}\psi(\varphi)\right) \tag{7}$$

The calculations were performed from 100° CA taking into account, that the air during fuel injection does not get to the combustion chamber. Velocity distribution in the inlet port to the combustion chamber qualification is necessary.

In the intersection of the port connecting the combustion chamber with cylinder, the molecules have both axial and radial velocity components.

The value of velocity calculations were performed taking into account the following assumptions:

– The pressures in the combustion chamber and in the cylinder are the same,

– The mass density in the combustion chamber and in the space over the piston is the same,

– The elliptic form of the port connecting the cylinder with the combustion chamber was approximated by the circle even profile to the real profile, which area is equal to the real area and which centre covers the cylinder axis.

If the compression charge phase was divided into limited sections, and the value of axial velocity and quantity of expressed charge are known, than the kinetic energy value could be calculated:

$$E_{kpi} = 0{,}5\Delta m_{ki} w_0^2 \tag{8}$$

Total value of kinetic energy, coming from the charge delivered to the combustion chamber charge after the end of expression, will be the sum of kinetic energy values in certain sections:

$$E_{kp} = \sum_{i=1}^{n} E_{kpi} \tag{9}$$

Total kinetic energy coming from the charge expression and effective kinetic energy of the exhaust gas - fuel mixture stream, will be the basis for angular velocity calculation, that is the coefficient of charge swirl intensity after the end of the process of fuel injection and the charge expression to the combustion chamber.

$$E_{k2} = E_{ke} + E_{kp} \tag{10}$$

The mass of charge in combustion chamber will be:

$$m_2 = m_0 + m_{str} + \sum \Delta m_{ki} \tag{11}$$

The moment of inertia:

$$I_2 = m_2 R_z^{\ 2} \tag{12}$$

The angular velocity will carry out:

$$\omega_2 = \sqrt{\frac{2 \cdot E_{k2}}{I_2}} \tag{13}$$

The gas contained in the cylinder is a fluid carrier in a used model. The gas consists of a residual exhaust gas and fresh air. Liquid fuel injected to the cylinder shortly before the exhaust port closure produces dispersed fluid. The amount of fresh air with lower value of enthalpy in the cylinder than that of residual gas after scavenges process depends on the ports timing and rotational velocity. This phenomenon was observed by use of Phoenics CFD program in cylindrical coordinates. It is shown in Figs 2 - 5. Geometry of the cylinder was simplified by change of space of the cylinder head to the cylinder of the same volume. However, such procedure enabled moving of the mesh of whole space according to the piston movement. The charge was treated as one phase medium, but additional concentration of different species could be predicted such as: residual exhaust gas, fresh air and fuel.

On the base of known thermodynamic parameters of the fuel mixture in the chamber, the fuel distribution in the cylinder was obtained after opening pneumatic valve at the beginning of compression process.

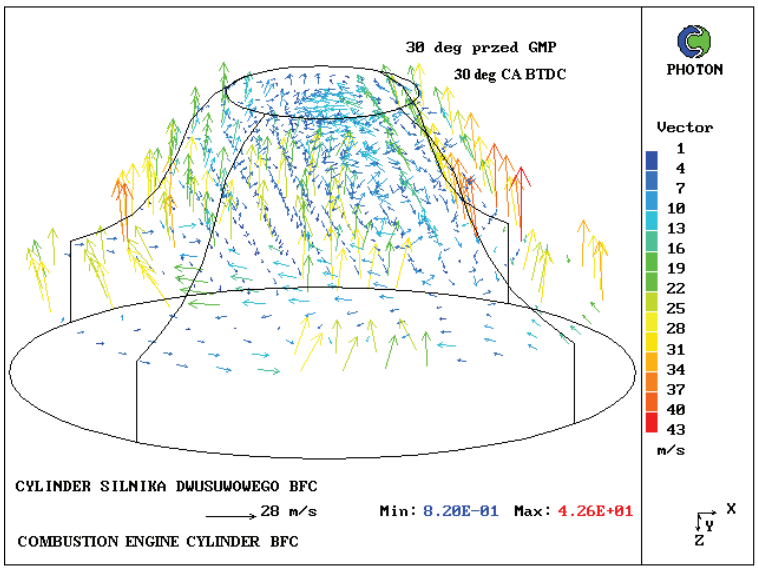

Fig. 2. The charge velocity vectors at piston position 30 deg BTDC.
Rys. 2. Wektory prędkości ładunku przy położeniu tłoka 30 stop. OWK przed GMP.

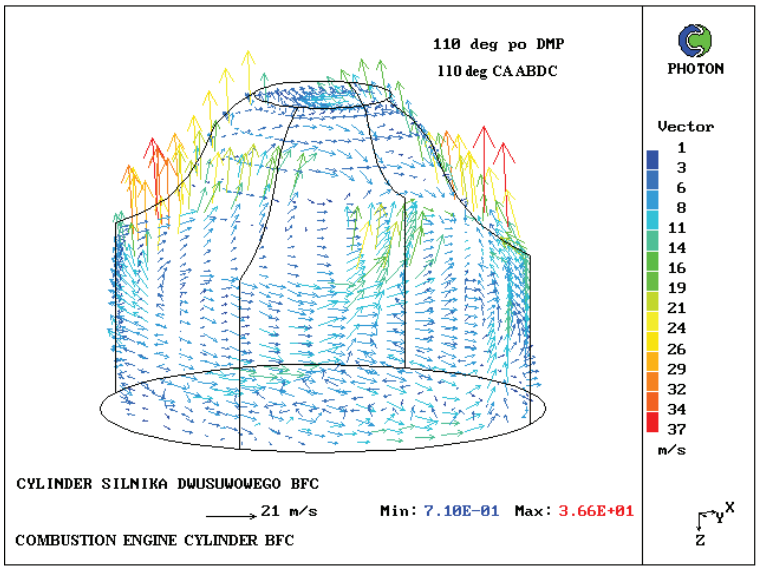

Fig. 3. The charge velocity vectors at piston position 110 deg ABDC.
Rys. 3. Wektory prędkości ładunku przy położeniu tłoka 110 stop. OWK po DMP.

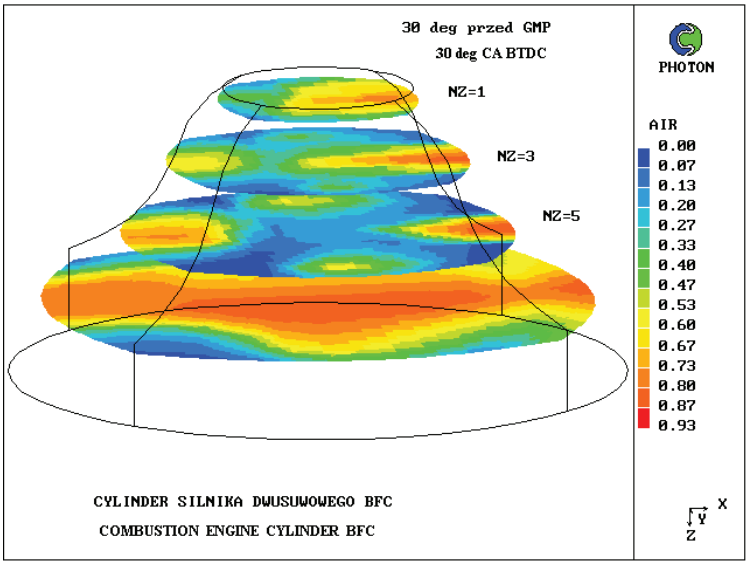

Fig. 4. Distribution of air in charge at piston position 30 deg BTDC.
Rys. 4. Koncentracja powietrza w ładunku przy położeniu tłoka 30 stop. OWK przed GMP.

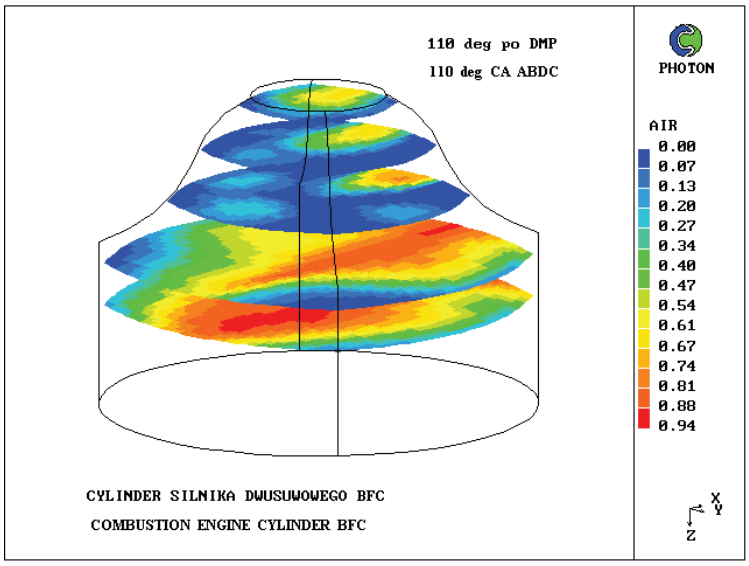

Fig. 5. Distribution of air in charge at piston position 110 deg ABDC.
Rys. 5. Koncentracja powietrza w ładunku przy położeniu tłoka 110 stop. OWK po DMP.

4. Problem of elimination of scavenge loss

How already was mentioned, in presented system of combustion will be distributorless system of pneumatic injection by means of hot exhaust gas used. This solution characterizes with simplicity of design, however for the price of potential possibilities of pronouncement of occurrence of scavenge - loss, as injection of exhaust gas-fuel mixture begins in moment when exhaust port is yet open. Elimination these unfavorable occurrences one can execute by simultaneous realization of following requirements:

– exhaust gas-fuel mixture cannot penetrate of stream of gases leaving cylinder,
– from point of view of charge exchange it should be obtained a favorable course of pressures in exhaust system and optimum timing of exhaust port.

Well-known two main systems of limiting scavenge - losses are, giving in effect improvement of elasticity without simultaneous losses of maximum power. First relies on choosing optimum - values of free section of exhaust port flow, the second one obtainment in the latter part of process of charge exchange, height of pressure in parts of exhaust system being found behind exhaust port immediately before it closes.

First system, which does not introduce an asymmetry to the cylinder charge exchange phases, however in essential manner influences on qualitative and quantitative management of charge exchange geometrical parameters. Qualitative control is obtained by desirable selection of phase of opening and closing of port, and quantitative control is realized due to proportionality of timing and rotational speeds. Necessity of such a choice is a result of a fact, that in constructional systems of timing with symmetrical and unsymmetrical phases of charge exchange, and with constants angular-section, dependence of inverse proportionality of rotational speeds, so-called hyperbolically dependence of diminishing timing-section oneself with height of rotational speed exists. Because in a two-stroke engine the piston supports exchange of charge in minimum, this dependence influence in essential manner on work of this kind of engine. An introduction of optimization of geometrical parameters of charge exchange, will improve simultaneously a course of torque thanks to limitation of scavenge - loss.

Second system makes possible obtainment of proportional to rotational speed frequencies of natural oscillations of complicated system composted from resonance chamber and parts of exhaust system. The period of natural oscillations of the stream of the gases filling the exhaust system is defined in this manner. Profitable from charge exchange course of pressure in exhaust system point of view, is attained when period of natural oscillations of exhaust gases is approximately to equal period of scavenge process. Before closing exhaust port, close to the charge exchange process end, the pressure increase occurs. This occurrence makes difficult outflow of charge from cylinder. It is also possible withdrawing parts of recent charge from exhaust system to cylinder, which even more improves the filling.

To qualification the optimum parameters of charge exchange, from point of view the limitation of scavenge - loss, the model works was conducted (the simulations),

and the measurements on research stand. These investigations concentrated on obtainment, the profitable course of pressures from point of view of charge exchange in exhaust system. Simulations concentrated on the influence of geometrical parameters of exhaust system, and obtainment the growth of pressure before closing exhaust port. The works were realized by use the specialist software "Engine2s", designed to the simulation of thermodynamical cycle of two-stroke engine.

The computer programme calculates the thermodynamic parameters of charge in the whole engine from inlet and exhaust systems and points out the main parameters resulting from analysis of working cycle. Then, on basis of preliminary results of investigations, the suitable modifications of engine exhaust gases system could be executed, The measurements on research stand were conducted for final verification of possibility of obtainment the profitable course of pressures before exhaust port, from charge exchange point of view. Figs 6, 7, 8, show the results. In the index *"ts"* the state before modification was marked, and the *"tow"* after modification of exhaust system of engine.

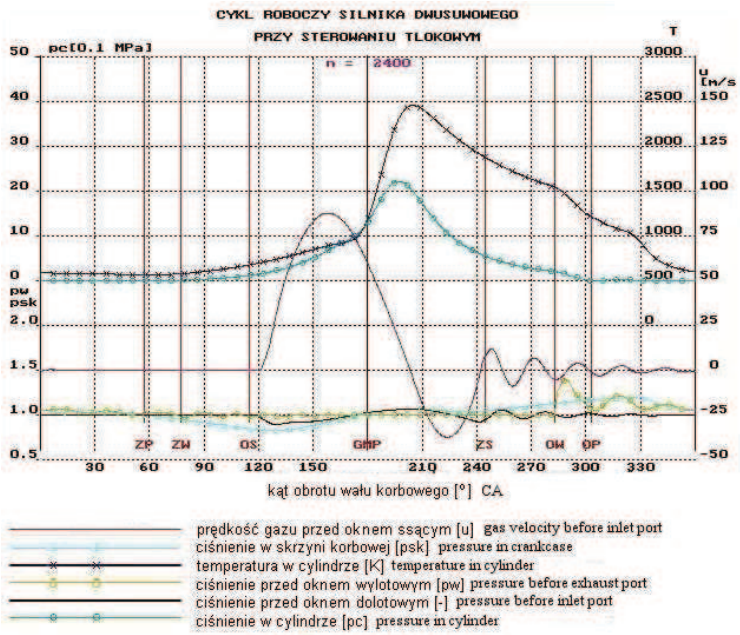

Fig. 6. Simulation of working cycle with exhaust system "ts".
Rys. 6. Symulacja cyklu roboczego z układem wydechowym ts.

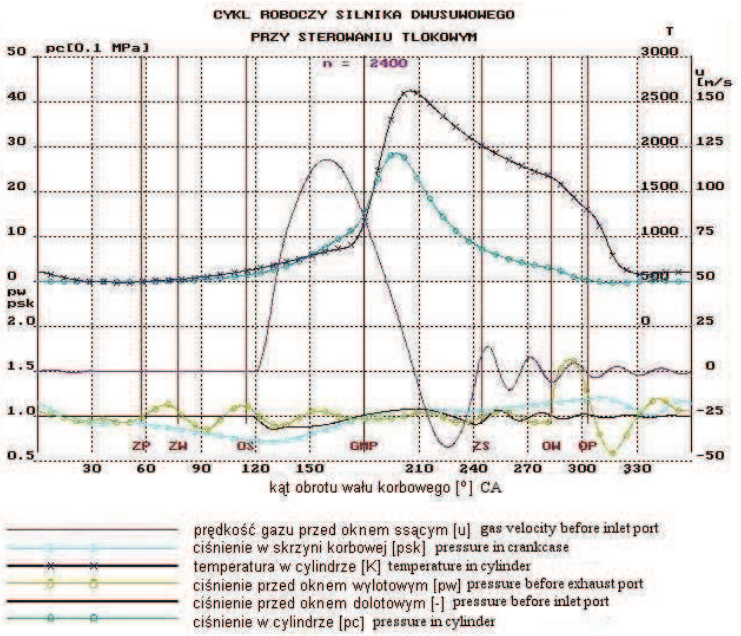

Fig. 7. Simulation of working cycle with exhaust system "tow".
Rys. 7. Symulacja cyklu roboczego z układem wydechowym tow.

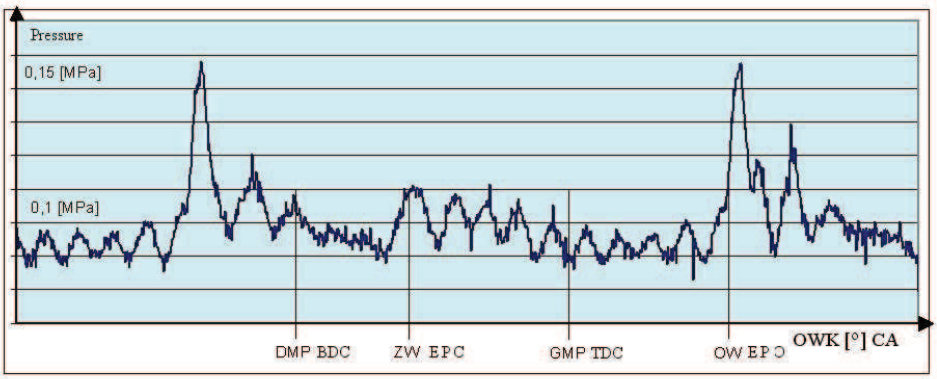

Fig. 8. Registration of exhaust gas pressure before exhaust port in exhaust system "tow".
Rys. 8. Rejestracja przebiegu ciśnienia przed oknem wylotowym z układem wylotowym tow.

Table 1. Exhaust gas after catalytic converter components concentrations.
Tabela 1. Stężenia składników spalin za konwerterem katalitycznym.

Measurement	CO	CO_2	O_2	HC
condition	% vol.	% vol.	% vol.	ppm
tw = 1,63 ms, n = 2400 1/min	0	9,9	7,7	25
tw = 1,63 ms, n = 2400 1/min	0	10,2	7,1	23

5. Conclusions

On the basis of the performed experimental works and analyses of the results of investigations the following conclusions can be formulated:

- At lower loads, that is most of engine operating life, the requirements in relation assurances of fuel dosage repeatability are realized with reserve,
- Mixture formation at direct fuel injection in a two-stroke engine must be investigated also as an effect of gas swirl during scavenge process. Mixture movement is caused by velocity of continuous phase after transfer port closing,
- From point of view of elimination scavenge loss, much more profitable nature of the course of pressures in front of the exhaust port,
- The profitable course of pressures occurs when the value of rotational speed of engine was n = 2400 1 / min,
- Correct co-operation with electric machine.

In further works of engine combustion and control system, the above value of rotational speed was accepted as the decisive parameter.

References

[1] CARLISLE W.H., FREW R.W., MILLS J.R., ARADI A.A., AVERY N.L., *The Effect of Fuel Composition and Additive Content on Injector Deposits and Performance of an Air-Assisted Direct Injection Spark Ignition (DISI) Research Engine*, SAE Paper 2001-01-2030.

[2] FOX J.W., MIN K.D., CHENG W.K., HEYWOOD J.B., *Mixture preparation in a SI engine with port fuel injection during starting and warm-up*, SAE Paper 922170.

[3] HAYAKAWA K., SHIGERU Y., SHINTARO T., HIDEKATSU O., YANG K. C., WORTH D., *125cc Small Engine Fuel Injection System with Low Emissions Solutions*, SAE Paper 2004-32-0094 / 20044381.

[4] JUDA Z., MAREK W., TUTAJ J., *Problems of co-operation the internal combustion engine with electric motor in hybrid propulsion system*, Journal of KONES' 2001.

[5] MAJOR G. *Learning from experiences with small-scale cogeneration*. CADDET Analyses Series No. 1. Sitard, Netherlands 1993.

[6] MAREK W., MITIANIEC W., *Study of mixture preparation in a two-stroke engine with fuel injection*, Journal of KONES' 2003.

[7] PSTRZEDNIK S.; ŻMUDKA Z., *Termodynamiczne oraz ekologiczne uwarunkowania eksploatacji tłokowych silników spalinowych*, Wydawnictwo Politechniki Śląskiej, Gliwice 2007.

[8] POGOREUTZ M.: *Economical and technological comparison of small-scale CHP on the basis of biomass*. Papers 1st World Conference on Biomass for Energy and Industry. Sevilla 2000.

Problematyka współpracy dwusuwowego silnika spalinowego z maszyną elektryczną w układzie kogeneracyjnym

S t r e s z c z e n i e

Referat obejmuje zagadnienia kształtowania procesu wymiany ładunku w dwusuwowym silniku ZI. Problematykę tę powiązano z możliwością aplikacji silnika dwusuwowego z pneumatycznym wtryskiem paliwa (wg koncepcji prof. St. Jarnuszkiewicza) w układach kogeneracyjnych. Zaprezentowano wyniki badań symulacyjnych i laboratoryjnych, związanych z zagadnieniem współpracy silnika spalinowego i maszyny elektrycznej, wyposażonej w elektroniczny regulator napięcia prądnicy, umożliwiający: zabezpieczenie maszyny przed przeciążeniem poprzez ograniczenie wartości prądu obciążenia, ograniczenie prądu do wartości maksymalnej prądu ładowania, ograniczenie napięcia wyjściowego do wartości napięcia końcowego naładowania baterii akumulatorów, zabezpieczenie przed przepływem prądu zwrotnego od baterii akumulatorów do prądnicy, ochronę prądnicy przed przekroczeniem temperatury dopuszczalnej, utrzymywanie zadanych wartości napięć i prądów niezależnie od temperatury otoczenia. Jako główne zagadnienie zaprezentowano problematykę procesu wymiany ładunku, w której uwzględniono z jednej strony energetyczne zapotrzebowanie układu wymuszającego, z drugiej zaś strony zapewnienie minimalizacji zawartości toksycznych składników w spalinach.