THERMAL STATE SIMULATION FOR CYLINDER OF MARINE TWO-STROKE ENGINE

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Abstract

This paper deals with the modelling of heat flow through cylinder liner structural components of a two-stroke engine. Especially, I paid attention to simulating temperature distribution for the wet cylinder liner. Multidimensional equations for the transient heat conduction with the Dirichlet and Fourier boundary conditions have been applied. In particularly, local values for the convective and radiative heat transfer coefficients using the Fourier boundary conditions determined in space of cylinder volume are applied. In order to determine the temperature distribution for the considered space, the radiosity method is used. Simulation results have been presented in the form of a temperature field for cylinder liner structural components depending on the crankshaft position angle. Application of the iterative calculation method for solving differential equations of energy balance allowed me to use software easy to get. I carried out all iterative computations using MSEXCEL spreadsheet. This way, I could decrease the simulation cost significantly. The simplicity of such an approach allowed me to apply the obtained results for visualization of the conduction heat transfer phenomena occurring in a cylinder of working marine engine. The presented approach could be used for the development of ship machinery simulators as well.

Keywords: *marine two stroke engines, modelling, radiosity, heat transfer,*

1. Introduction

Most important propulsions used in marine transportation are diesel engines. For this reason, we can observe constant interests in increasing their heat effectiveness. The main source of energy losses in the piston engines is a thermal energy flux transferred to the liquid coolant from a cylinder volume. Many research results show us that approximately 25 % of the energy produced by a combustion process is lost to the engine cooling system [1]. In order to increase the piston engine heat effectiveness we should recognize the heat-releasing phenomenon from the engine cylinder volume. The results of such an approach could be used for the development of ship engine simulators also. The heat flow in a cylinder volume occurring during a combustion process is very complex. It is necessary to express that working features of piston engines, for example cyclic changes of combustion process parameters depending on crankshaft positions, can cause additional difficulties in describing such a process in an appropriate quantitative and qualitative way. Therefore, in order to identify the quantity of heat lost, we should determine not only parameters occurring in the combustion chamber and cooling volume, but also geometrical features (sizes and shapes) of a cylinder.

This paper deals with the building of a multidimensional model for heat conduction within the structural components of an engine cylinder by using the radiosity method [2]. Such a model has been applied to visualize temperature distribution of the mentioned components depended on load of the laboratory two-stroke engine installed in Gdynia Maritime University.

2. Modelling of the heat transfer coefficient

The heat transfer rate from the combustion gases to the combustion chamber wall is usually expressed by the Newton's law [3]. It uses an experimental heat transfer coefficient. Many models proposed various ways to obtain such a coefficient. They assumed that the heat flux is the same for all surfaces in construction elements of the engine combustion chamber.

First of them is Nusselt's model [4]. It assumed steady state heat transfer and shows dependence of amount of the heat transfer coefficient on the mean piston speed, the temperature difference, the cylinder gas pressure and a gas and the wall emissivity. Based on the same parameters, Sitkei [5] proposed a different experimental equation for four-stroke, indirect injection diesel engines. Eichelberg [4] proposed very simple model for two and four-stroke engines and Hohenberg studied the heat transfer for six engine types [6]. Wiebe's proposal presented in [7], makes a value of the overall heat-transfer coefficient dependent on geometrical sizes of a cylinder volume, average speed of a piston, and stage parameters of a mixture averaging for the total volume of a cylinder space. A value of such a coefficient changes according to the crankshaft position. Woschni proposes the similar dependency in [8]. He developed it adding variable coefficients, which vary with different phases of the engine cycle. Annand, cited in [9], determined amount of heat transferred to cylinder walls by means of adding heat conveying trough the convection and radiation phenomena. However, he made the accuracy of results received with using of this method dependent on the correct determination of so-called calibrating constants. As a rule, these constants can be obtained by means of laboratory tests.

The presented correlation dependencies can be helpful in setting up the total energy balance of piston engines because they require only a small number of the input data. Nevertheless, the accuracy of results received in this way depends on calibrations made every time for the specific engines. The comparison of values of the heat-transfer coefficients determined for a laboratory engine by using the described methods is presented in [10]. Results received by these methods have the sufficient discrepancy, reaching almost 80% for beginning and throughout of a combustion process. However, the mentioned dependencies describe only the overall heat quantity transferred from the cylinder liner or the whole engine and they are not able to describe the local and transient flow of heat [11].

Therefore, they are not suitable to describe thermal stages of structural components of an engine cylinder. In last years, models describing of the multidimensional flow of heat through cylinder walls have been developed according to the progress made in the field of the turbulent combustion of fuel. It is obvious that these models take into consideration the changeable conditions of a combustion process in various points of a cylinder volume.

3. Heat transfer through construction elements of the cylinder

Construction elements of the cylinder of large marine two-stroke engine have usually cylindrically shapes. Therefore, in this case we used the cylindrical arrangement of equations. The heat flow balance for an elementary geometrical area of engine cylinder structural element (Fig.1) can be presented as energy balance and according to Fourier's law of heat transfer, the heat flux is proportional to the local temperature gradient in any direction (Eq. 1 and 2).

$$Q_V + \sum_{i=1}^{n} Q_i = 0, \tag{1}$$

$$Q = -\lambda \cdot \nabla T, \tag{2}$$

where:

Q_V – internal heat source [J],

Q_i – amount of heat flow from a neighbouring elementary geometrical area [J],

n – number of neighbouring areas [-],

λ – thermal conductivity of material [W/(m·K)],

∇T – temperature gradient [m·K].

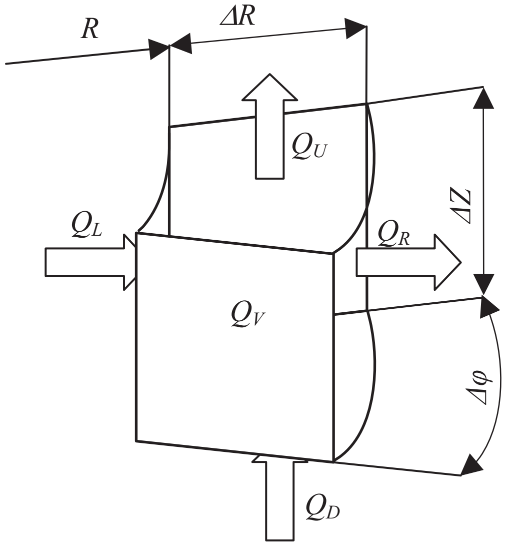

Fig. 1. An elementary volume of a cylinder liner structural component

Assuming that properties of structural material are isotropic in all directions, we can state that there is no heat source inside the considered area and the heat exchange process is stationary. Taking into account that $\Delta Z = \Delta R = X$, we obtained the relation determined the temperature in structural component as follow:

$$T_V = \frac{1}{4}T_U + \frac{1}{4}T_D + \left(\frac{R+X}{4R+2X}\right)T_R + \left(\frac{R}{4R+2X}\right)T_L, \tag{3}$$

where:

T – temperature [K],

$_{L,\ R,\ U,\ D}$ – indexes of neighbouring areas temperatures (see Fig. 1).

Thermal energy accumulation in the cylinder liner structural components complements such a kind of phenomena. In this case, I converted the Eq. (3) to the following form:

$$T_V^{+1} = T_V\left(1 - 4\Delta Fo\right) + \Delta Fo\left(T_U + T_D + T_R \frac{R+X}{R+0.5\cdot X} + T_L \frac{R}{R+0.5\cdot X}\right), \tag{4}$$

In Eq. (4) ΔFo is the Fourier discrete number for two-dimensional heat flow [12] inside elementary area:

$$\Delta Fo = \frac{\lambda \cdot \Delta t}{c_p \cdot \rho \cdot X^2}, \tag{5}$$

where:

T_V^{+1} – temperature after Δt time [K],

c_p – specific heat at constant pressure [J/(kg·K)],

ρ – density of the elementary control volume [kg/m^3],

Δt – considered time interval [s].

In the iterative solutions, the steady-state condition should be met. It can be expressed by the following inequality:

$$\Delta Fo > \frac{1}{4}. \tag{6}$$

More disquisition extraction of (3) and (4) formulas was presented in [13].

4. Proposed model of temperature in the cylinder construction elements

The input data for a thermal state model are the boundary conditions on walls of the cylinder liner structural components and initial conditions determined by means of the experimental measurements or modelling of a combustion process in the cylinder volume. Both the boundary and the initial conditions determine a thermal state for structural component boundaries. All calculations were carried out for the one-cylinder, two-stroke, crosshead laboratory engine with the loop scavenging. The fresh water flowing through the wet cylinder liner cools its cylinder volume. The main parameters of this engine are; the piston stroke 350mm, a diameter of the cylinder 220mm and the actual rotational speed 200 rpm. The structural similarity to the large-size marine engines was the reason of its selection as the modelling item. Using design documentation of the mentioned engine, we have divided the cylinder structural components into the elementary components with a size 2 mm of their sides (parameter X in Eq. 3 and 4). Taking into mind axial-symmetrical shapes of the modelled elements, we limited our modelling to the two-dimensional areas located in the longitudinal section. These elementary control patches allow us to describe the temperature field for the following components: the engine piston together with its rings considered as the uniform structural component, the cylinder liner, the cylinder block with under-piston chamber and the cylinder head together with its exhaust valve and injector.

For describing the heat exchange phenomena, the following forms of boundary conditions were applied:

– Dirichlet's conditions for a surface between the structural component walls and the surrounding air, which were obtained by determination of the wall temperature equals to the air temperature measured during the experimental study,
– Dirichlet's conditions for a surface of under-piston chamber which were obtained by calculation of compression in this area during the engine working [14],
– Fourier's condition for a surface between the structural component walls and the cooling volume,
– Fourier's condition for a surface between the structural component walls and the combustion chamber,
– IV'th gender condition (contact of elements) without a heat resistance for connections of structural components of the engine cylinder.

The Fourier boundary condition was obtained with the use of heat transfer coefficients, which are calculated individually for local conditions:

$$\alpha_C = 0{,}023 \cdot \left(\frac{C \cdot X}{\nu} \right)^{0,8} \cdot \left(\frac{c_p \cdot \nu}{\lambda} \right)^{0,4} \cdot \frac{\lambda}{X}, \tag{7}$$

$$\alpha_{Ri} = \frac{\varepsilon \cdot C_C}{T_i - T_V} \left(T_i^4 - T_V^{\;4} \right), \tag{8}$$

where:

α_C - convective heat transfer coefficient [W/(m^2·K)],
C - speed of gas in a combustion chamber assumed as a mean piston speed or mean speed of water in a cooling volume, calculated by Poiseuille formula [m/s],
ν - kinematic viscosity [m^2/s], assumed $\nu = 6 \cdot 10^{-11} \cdot T^2 + 7 \cdot 10^{-8} \cdot T - 1 \cdot 10^{-5}$ for gas in a cylinder and $8{,}60 \cdot 10^{-5}$ m^2/s for the cooling water,

λ - thermal conductivity of material [W/(m·K)], assumed $\lambda = -2 \cdot 10^{-8} \cdot T^2 + 8 \cdot 10^{-5} \cdot T + 0{,}0037$ for gas in a cylinder and 0,612 W/m·K for the cooling water,

c_p - specific heat at constant pressure [J/(kg·K)], assumed $c_p = 3 \cdot 10^{-7} \cdot T^2 + 0{,}1974 \cdot T + 938{,}14$ for gas in a cylinder and 4178 J/kgK for the cooling water,

α_{Ri} - radiative heat transfer coefficient determined on the basis of Newton's and Stefan -Boltzmann's laws [W/(m^2·K)],

ε - relative emissivity determined for „grey" flame and „lusterless" surface of cylinder walls equals ε=0,79,

C_C - Stefan-Boltzmann's constant, equals Cc= 5,67·10^{-8} W/(m^2·K^4),

T_i - temperature of a combustion chamber and a local temperature in the cooling volume [K].

The sum of convective and radiative heat transfer coefficient was taking into account to describe Fourier's condition for surfaces. The correct value of T_V temperature of elementary area was obtained by using the iterative method, whereas the input data regarding temperature T_i in a combustion chamber were taken from the combustion process model presented in [14]. Calculations were started at crankshaft position of 8º before top dead centre of the piston with using of a formula (3). Calculations were continued with 8º intervals of the crankshaft position to 120º after TDC with using the formula (4). Set intervals of the crankshaft position gave the time interval of 6 [ms] at the set rotational speed (200 rpm). Temperatures T_V in this formula were taken from results from the earlier crankshaft position.

5. Modelling results

Solving Eqs. (3) and (4) for each of elementary control patches, we can obtain a temperature field picture for cylinder components in cylindrical arrangement. All necessary calculations we performed for the described laboratory engine. The engine thermodynamic state varied according to its crankshaft position angle. It is represented by changes of temperature in a combustion chamber. Values of these temperatures are presented in [13] and [15].

In the left side of Fig. 2, the example of modelling results is presented in the form of the temperature field of engine structural components shown in axial cross-section. The results are presented for rotational speed of the engine equal 200 rpm, power output equal 14,6 kW and TDC crankshaft position. Due to the large number of elementary areas, the obtained modelling results are presented in the form of multi-colour map. Borders of particular colours correspond with the isotherms dividing the cylinder construction areas into the temperature intervals of 10 K. The central area of Fig. 2 presents the schematic axial cross-section of the engine cylinder construction applied in modelling the temperature field within particular structural components. In this section of figure black dots presents chosen elementary volumes. Results of calculations for these volumes in the cylinder liner material for 9 loads of the engine in 8º before TDC crankshaft position are presented in the right side of Fig. 2. Results for elementary volumes in piston area are shown in Fig. 3. Graph in Fig. 2 shown higher temperature in upper side of cylinder liner for all considered loads of the engine. Temperature in the middle side of the cylinder liner is stabilized due to cooling of this element by fresh water. The lower side of the cylinder liner is cooled by flowed round air and heated only during final stage of the combustion process. This side of the cylinder liner is much cold. Qualitatively similar calculation results we obtained for another considered crankshaft positions.

The left side of Fig. 3 presents calculated temperature of chosen elementary volumes in the piston material in 8º before TDC crankshaft position. Temperature of the piston is lower in bottom areas for all loads of the engine but these results may be treated only demonstrative, because construction of the engine piston is more complicate than modelled (cooling system in piston,

piston rings etc.). More detailed results for the 1 and 2 elementary volume in the piston material (see Fig. 2) for chosen load of the engine are presented in the right side of Fig. 3. Positions of elementary volumes 1 and 2 are accordingly 2mm and 16mm under the piston surface. According to this drawing temperature of the elementary volume 1 lowered during the combustion process and temperature volume 2 is not changed. Low temperature of both volumes in 120º after TDC of crankshaft position is caused starting of charging process.

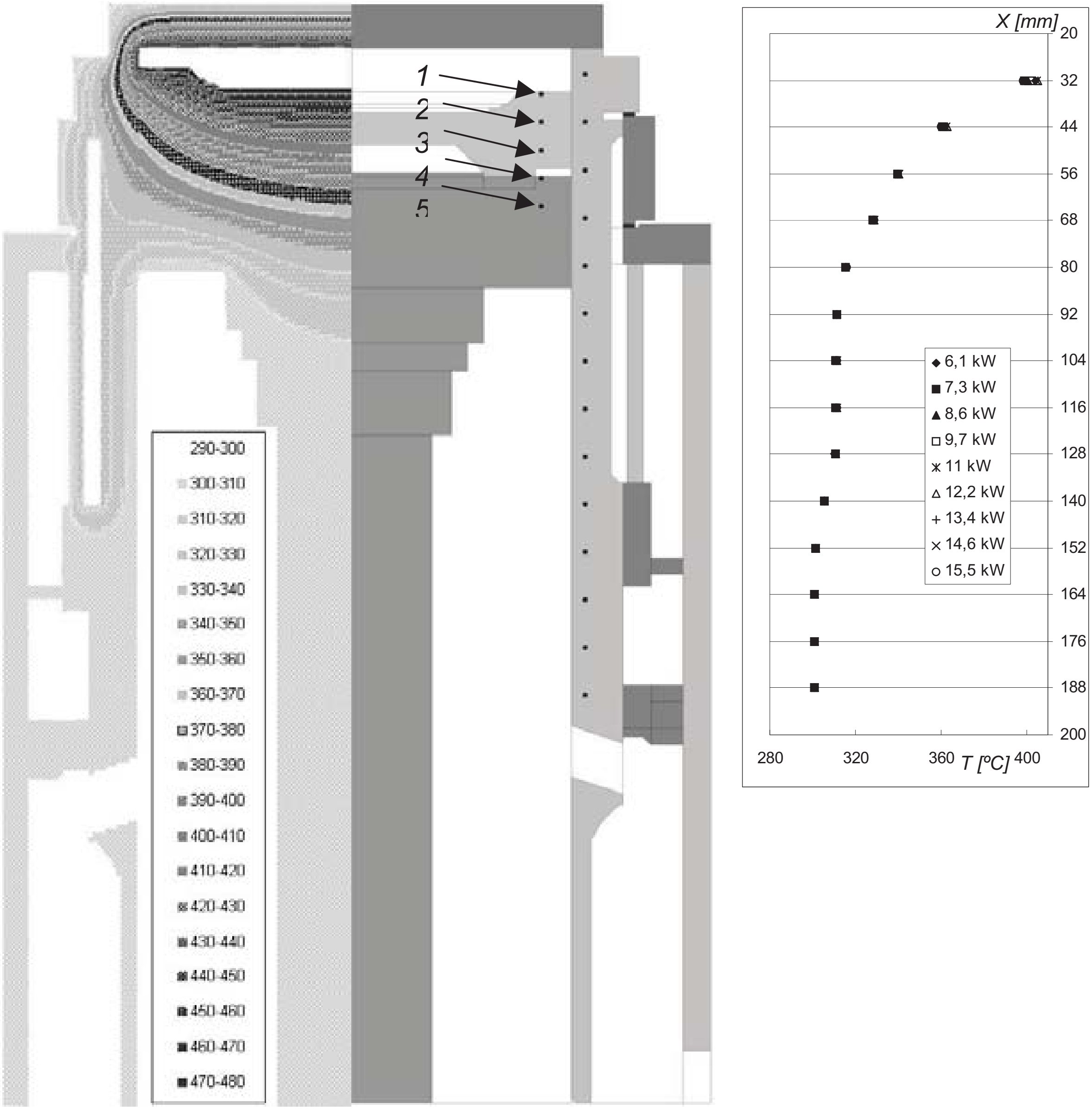

Fig. 2. Example of modelling results for rotational speed of the engine equal 200rpm, power output equal 14,6 kW and TDC crankshaft position (left side), schematic axial cross-section of engine cylinder construction (central area) and results of calculations for chosen elementary volumes in the cylinder liner material for all considered loads of the engine in 8º before TDC crankshaft position (right side)

6. Conclusions

The presented paper deals with a model of heat transfer through structural components of the engine cylinder. Results obtained during modelling allowed us for the qualitative estimation of a thermal state of the engine cylinder. Nevertheless, the lack of experimental verification does not permit to carry out the quantitative estimation of such a state. Moreover, simplification made in representation of a cylinder structure within the piston and the cylinder head makes impossible to obtain the high accuracy of such a modelling. Therefore, in order to increase the model adequacy, we should increase an accuracy of representation of a structure of the considered areas by means of modelling friction nodes: a piston - piston rings - a cylinder liner. It, in turn, entails the necessity to decrease geometrical sizes of elementary control patches.

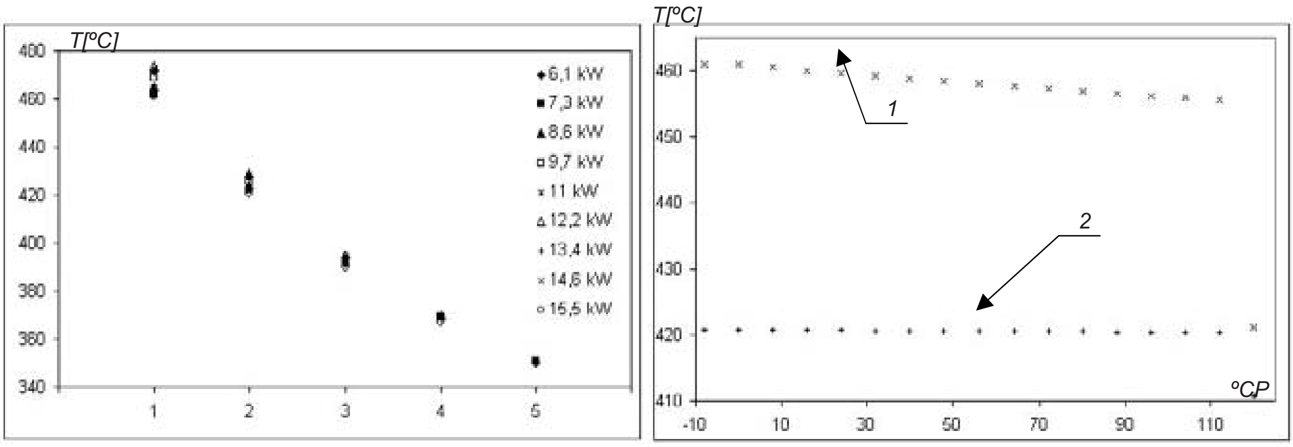

Fig. 3. Calculated temperature of chosen elementary volumes (presented in Fig. 2) in piston material in 8º before TDC crankshaft position for all considered loads (left side) and calculated temperature of 1 and 2 elementary volumes of the piston for all considered crankshaft positions (CP) and load 15,5kW (right side)

Obtained modelling results may contribute to an increase of modelling accuracy of the phenomena occurring within the engine combustion chamber, accompanied by a significant decrease of the modelling cost associated with using special computer software. Such a model makes possible to develop guidelines for designing of the engine cylinder structure taking into account decreasing of thermal stresses by optimization of temperature distribution in the cylinder components. As far as combustion process models are concerned, the modelling of temperature distribution within cylinder walls may effectively contribute to an increase of engine efficiency by decreasing of total heat flowing out to the engine cooling system and by possibility of forming the heat flow in complex areas of a cylinder space. Moreover, the obtained results may be used in teaching the subjects associated with combustion engines, i.e. analysis of thermal stresses and temperature distribution within engine elements.

References

[1] Heywood, J. B., *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.

[2] Nagórski Z., *Modelowanie przewodzenia ciepła za pomocą arkusza kalkulacyjnego*, Wydawnictwo Politechniki Warszawskiej, Warszawa, 2001.

[3] Incropera, F. P., DeWitt, D. P., *Fundamentals of Heat and Mass Transfer*, Wiley, 2001.

[4] Wu, Y-Y., Chen, B-Ch, Hsieh, F-Ch., *Heat transfer model for small-scale air-cooled spark-ignition four-stroke engines*, Int. J. Heat Mass Transfer Vol. 49, pp. 3895–3905, Elsevier Science, 2006.

[5] Borman, G., Nishiwaki, K., *Internal-combustion engine heat transfer*, Prog. Energy Combust. Sci, No. 13, pp. 1–46, Elsevier Science, 1986.

[6] Hohenberg, G. F., Advanced approaches for heat transfer calculations, Diesel Engine Thermal Loading, SAE SP-449, 1979, pp. 61–79.

[7] Nguien, H., *A simulation model of heat exchange in the ship diesel engine cylinder – environment system*, Polish Maritime Research, No 1(31), Vol. 9, pp. 9-14. Gdańsk, 2002.

[8] Woschni, G., *A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine*, SAE paper No. 670931, SAE Trans. Vol. 76, 1967.
[9] Heywood, J. B., Sher, E., *The two-stroke cycle engine, its development, operation and design*, Taylor & Francis, Philadelphia, 1999.
[10] Ghojel, J., Honnery, D., *Heat release model for combustion of diesel oil emulsions in DI diesel engines*, Applied Thermal Engineering Vol. 25, pp. 2072-2085. Elsevier Science, 2005.
[11] Rychter, T., Teodorczyk, A., *Modelowanie matematyczne roboczego cyklu silnika tłokowego*, PWN, Warszawa, 1990.
[12] Galindo, J., Lujan, J.M., Serrano, J.R., Dolz, V., Guilain, S., *Description of heat transfer model suitable to calculate transient processes of turbocharged diesel engines with one-dimensional gas-dynamic codes*, Applied Thermal Engineering, Vol. 26, pp. 66-76, Elsevier Science, 2006.
[13] Kowalski, J., Tarelko, W., *Heat transfer model for marine two stroke engine cylinder*, Explo-Diesel & Gas Turbine 2007 Conference, Gdańsk – Stockholm - Tumba., pp. 273-280.
[14] Kowalski, J., Tarelko, W., *Modeling aspects of nitric oxides emission from the two-stroke ship engine*, Explo-Diesel & Gas Turbine 2003, pp. 329-338, Miedzyzdroje-Lund, 2003.
[15] Kowalski, J., Tarelko, W., *The thermal state modelling of cylinder liner of marine two-stroke combustion engine*, Polish Maritime Research, No 2(48), pp. 15-20, Gdańsk 2006.