Journal of Polish CIMAC
Gdansk University of Technology
The Faculty of Ocean Engineering and Ship Technology

TEMPERATURE AS SYMPTOM OF THERMAL PERFORMANCE IN PISTON-CONNECTING ROD CORRECTNESS OF COMBUSTION ENGINE

Jan Rosłanowski Ph.D.Ch.(Eng.)

Gdyni Maritime Academy
Faculty of Marine Engineering
81-87 Morska str.
81-225 GdyniaPoland
e-mail: rosa@am.gdynia.pl

Abstract

The following article characterizes thermal operation in piston-connecting rod system of combustion engines in an energetic aspect, with regard to oil temperature in its bearings. It has been proposed to use, among others, oil temperature in slide bearings of piston-connecting rod system in combustion engines as symptom of thermal operation correctness. It results from energy exchange with its neighborhood whish is nothing else but gases contained in the combustion chamber and the elements of the piston-connecting rod system of combustion engines. This energy originates from chemical energy transformation of fuel into a thermal one and from friction in movable elements in the piston-connecting rod system of the engine. Special attention was also paid to the fact that the utilization of oil temperature as a symptom requires the knowledge of layer thickness changes, hydrodynamic pressure in the oil film and the load of the slide bearing.

Key words: *oil temperature in slide bearing, thermal operation of piston-connecting rod system in combustion engines, work conditions of slide bearings*

1. Introduction

Thermal performance of the piston-connecting rod system in combustion engines originates from energy exchange with its neighborhood. This exchange takes place in the form of work and heat. The size of such exchange can be determined on the grounds of the energetic balance of the open system during fixed condition. Thermal operation of the above mentioned system is caused by receiving heat from the gases through the piston – bottom and the heat caused by friction of cooperating elements whose surfaces are in direct contact. Such operation is not much desired and by means of adequate constructions, prevented but impossible to eliminate completely, which results from the second principle of thermodynamics [4,6].

Friction force between guiding part of the piston with its rings and walls of cylinder liner causes formation of the heat. The quantity of emitted heat depends on:

- piston material, rings and cylinder liner,
- value of normal force pressing the piston against cylinder walls,
- lubricating conditions.

Friction between the piston and the walls of cylinder is not of smooth character and partly takes place depends on medium speed of the piston and this, in turn, means friction dependence on revolution speed of the engine. Besides, the above mentioned friction is influenced by the pressure stage arising from constructional parameters of the combustion chamber. Because it determines the value of the pressure in the chamber [5].

Temperature of the gases in combustion chamber is one of the symptoms of thermal operation correctness. The second symptom is the temperature of the piston surface which in the case of the seizure increases excessively. The temperature of oil in the bearings of connecting rod system is another symptom.

Dynamic load of the bearings in piston-connecting rod system of the engine decides that only smooth friction in the bearings secures their durable and reliable work.

The hydrodynamic lift which develops in the oil film results from the oil wedge and the effect of squeezing oil out of the slot which tears off the pin from the bearing bush. In bearings working in rotary motion, the main role is played by the oil wedge contracting in the form of a slot around the pin [1,2,7].

When the slide bearing is working, hydrodynamic lift of the oil layer depends on the relative motion speed of the pin and the bearing bush. It means that at small speed of this motion, pressure generated in the oil layer is not able to equalize outside load which brings about disappearance of fluid friction. Such short- lived situations take place during starting of the engine. Long-lasting disappearance of fluid friction in the slide-bearing leads to a loss of the thermal balance, which causes an increase of temperature and, as a consequence, its destruction by seizing [2,7].

Bearing averages of piston-connecting rod system i.e.(crosshead, connecting rod and the main ones) constitute about 45% of general number of averages in an engine. Analysis of bearing averages, given by Kozłowiecki [2] indicates, that the main reason of these averages are exploitation mistakes, caused by the change of their work conditions.

Reliability evaluation of movable slide bearings loaded dynamically is carried out on the grounds of the courses:

- loading,
- thickness change of oil film,
- hydrodynamic pressure in oil film,
- medium temperature of oil film [7].

2. Work conditions of slide bearings in piston-connecting rod system of combustion engine

Interaction dependence between parameters characterizing work condition of the bearings, do not concern only mass forces loading and combustion pressure. They have to take into account the following conditions:

- engine operation as a whole,
- its technical state,
- the way of exploitation.

In bearings loaded dynamically, thermal flux caused by friction and carried away from slide surfaces is the function:

- variable loading in course of time,
- variable of the effective angle speed of the pin,
- intensity of oil flow through the bearing,
- piston of the pin middle in relation to the bearing bush.

It is the reason why thermal condition of the bearing is estimated on the grounds of mean temperature of the oil film or mean temperature of oil flowing from the oil slot [2,7]. Determination of thermal balance of the bearing follows after definite time, which is the reason why mean values of thermal streams calculated for variable loads and positions of the pin middle in the bearing bush, are put into the equation of thermal balance.

Periodical changes of relative eccentricity "ε" in dynamically loaded bearings of the piston-connecting rod system of combustion engines, cause periodical changes of the oil intensity flow [2].

High temperature of the inside of crankshaft casing allows to omit taking up the heat by the bearing body to the neighborhood as a result of convection. It accounts for analyzing work condition of the bearings as adiabatic. It means that heat generated in the bearing as a result of friction, and being developed by cutting down oil layers, is carried away by lubricating oil [7].

3. Oil temperature in slide bearings of piston-connecting rod system as correctness symptom of their thermal operation

Transfer of thermal energy in lubricating slot of the bearing takes place by means of mechanical work by cutting down of oil layers and is described by the equation of energy (1). This equation results from the law of energy conservation in fixed conditions of bearing work. It presents volume energy balance of oil element under balance screen. Internal element of oil is delivered to the element volume and then carried away by means of convection and mechanical work carried out by the surface stresses and mass forces of oil time unit. After taking into account the equation of flow continuity and Navier-Stokes and omission of mass forces, the equation of energy in the case of adiabatic model of the bearing, can be written down in the following way:

$$\rho \cdot c \cdot u_{śr} \cdot \frac{\partial t}{\partial x} + \rho \cdot c \cdot w_{śr} \frac{\partial t}{\partial z} = \eta \cdot \left[\left(\frac{\partial u}{\partial y} \right)^2_{śr} + \left(\frac{\partial w}{\partial y} \right)^2_{śr} \right] \qquad (1)$$

where:

ρ [kg/m^3] – oil density,
c [J/kgK] – specific heat of oil,
$u_{śr}$ [m/s] – average tangential velocity of oil,
$w_{śr}$ [m/s] – average axial speed of oil,
t [°C] – temperature of oil.

The equation (1) in transverse calculations of slide bearings, as given by J. Kiciński in work [1], can be still more simplified, omitting convection in the axial direction of the bearing of (the axis z), i.e. acceptance of constant temperature along the breadth of the bearing $\left(\frac{\partial t}{\partial z} = 0 \right)$. Fig.1 presents coordinate system "z" and designation of oil flow speed and the way of dimensioning of the oil wedge in a cylindrical slide bearing.

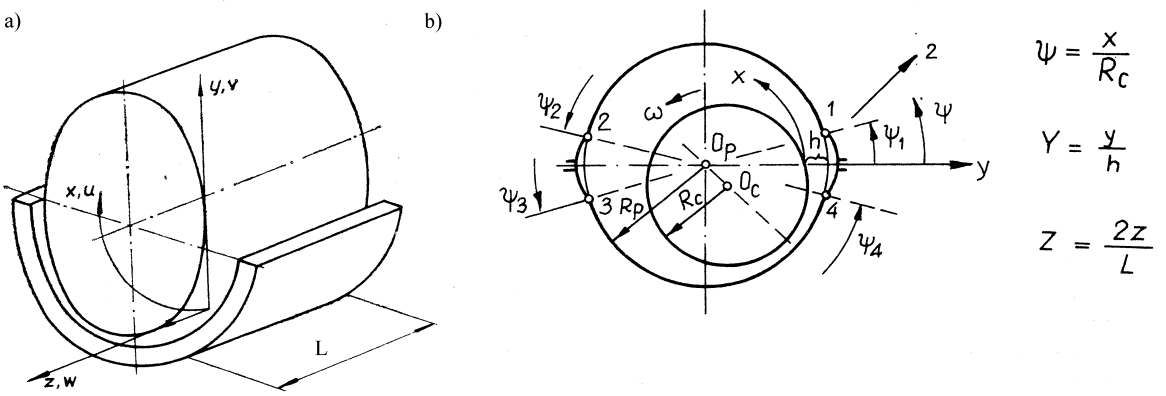

Fig.1. Dimensioning of cylindrical bearing [1,2]; a) coordinate system with designated speeds of grease flow, b) the way of dimensioning of grease wedge

The equation (1) can be presented in no dimensional form, assuming that $\left(\frac{\partial T}{\partial Z}=0\right)$ in the following way:

$$U_{śr}\frac{\partial T}{\partial \Psi}=\frac{M}{H^2}\left[\left(\frac{\partial U}{\partial Y}\right)^2_{śr}+\left(\frac{\partial W}{\partial Y}\right)^2_{śr}\right] \qquad (2)$$

or

$$\frac{\partial T}{\partial \Psi}=\frac{1}{U_{śr}}\cdot\Phi$$

where:

$T=\rho\cdot c\cdot\frac{\left(\frac{\Delta R}{R}\right)^2}{\eta_o\cdot\omega}\cdot t$ - no dimensional temperature of lube oil,

H = (1 + ε·cosΨ) – no dimensional thickness of the oil slot,

$M=\frac{\eta}{\eta_o}=e^{-0{,}045(t-t_o)}$ – no dimensional dynamical viscosity of oil,

t = 30ºC => η_o = 0,08 [Pa·s],

$U_{śr}=-\frac{1}{12}\cdot\frac{H^2}{M}\int_0^1\frac{\partial\pi}{\partial\Psi}dz+\frac{1}{2}=-\frac{1}{18}\varepsilon\cdot e^{0{,}045(t-t_o)}\left(\frac{L}{D}\right)^2\cdot\frac{\cos\Psi+\varepsilon+2\cdot\varepsilon\sin^2\Psi}{(1+\varepsilon\cos\Psi)^2}+\frac{1}{2}$,

$\left(\frac{L}{D}\right)$ – ratio of bearing breadth to its diameter,

$$Y=\frac{y}{h},\quad U=\frac{u}{\omega R},\quad W=\frac{w}{R\omega},\quad Z=\frac{2z}{L}.$$

Simplified differential equation of energy (formula 2) of adiabatic model of cylindrical slide bearing can be solved by means of the following differential expression:

$$\left(\frac{\partial T}{\partial \Psi}\right)_{i,k}=\frac{T_{i+1,k}-T_{i,k}}{\Delta\Psi}$$

or

$$T_{i+1,k}=T_{i,k}+\Delta\Psi\cdot\frac{1}{U_{śr}}\cdot\Phi \qquad (3)$$

where:

$T_{i,k}$ – no dimensional assumed temperature of oil on the inlet to the bearing,

Ψ_i= (i - 1) · ΔΨ – circumferential coordinate of oil slot,

$\left.\begin{matrix} i=1,\,2,\,3... \\ k=1,\,2,\,3...\end{matrix}\right\}$ – respective temperature nodes,

Ψ_{i+1}, -Ψ_i = ΔΨ = 1 [rad].

Parallelism of the pin and the bearing bush axis has been assumed in the adiabatic cylindrical slide bearing, that is, simmetricalness of pressure schedule in the axial direction. It allows to carry out calculations only up to the half of bearing breadth and multiply by two, receiving temperature for the whole breadth.

Assuming the slide surfaces are not deformed, the shape of the oil slot, will be the function only of the circumferential coordinate Ψ, i.e. H = H(Ψ).

Next, assuming constant temperature in the axial direction of the bearing, that is, temperature variability T = T(Ψ) and viscosity M = M(Ψ) only in circumferential direction Ψ we can obtain the expression:

$$T_{i+1,k} = T_{i,k} + \Delta\Psi \cdot \Phi(\Psi_i) \cdot \frac{1}{U_{śr}(\Psi_i)} \qquad (4)$$

where:

$$\Phi(\Psi_i) = \frac{2}{45} \cdot \varepsilon^2 \cdot \left(\frac{L}{D}\right)^4 \cdot e^{0,045(T_i - T_o)} \cdot \frac{(\cos\Psi_i + \varepsilon + 2 \cdot \varepsilon \cdot \sin^2\Psi_i)^2}{(1 + \varepsilon \cdot \cos\Psi_i)^6} + \frac{e^{-0,045(T_i - T_o)}}{(1 + \varepsilon \cdot \cos\Psi_i)^2} +$$

$$+ \frac{1}{9}\left(\frac{L}{D}\right)^2 \cdot e^{0,045(T_i - T_o)} \cdot \frac{\varepsilon^2 \sin^2\Psi_i}{(1 + \varepsilon \cdot \cos\Psi_i)^4},$$

$$U_{śr} = -\frac{1}{18}\varepsilon \cdot e^{0,045(T_i - T_O)} \cdot \left(\frac{L}{D}\right)^2 \cdot \frac{\cos\Psi_i + \varepsilon + 2\varepsilon \cdot \sin^2\Psi_i}{(1 + \varepsilon \cdot \cos\Psi_i)^2} + \frac{1}{2}.$$

The expression (4) is a numerical algorithm which allows to determine oil temperature in every point of lube slot circumference at well known pressure schedule in the adiabatic cylindrical slide bearing.

Fig.2 and 3 present temperature schedule in the oil slot on the circumference of the adiabatic slide bearing model, determined by means of numerical algorithm (formula4).

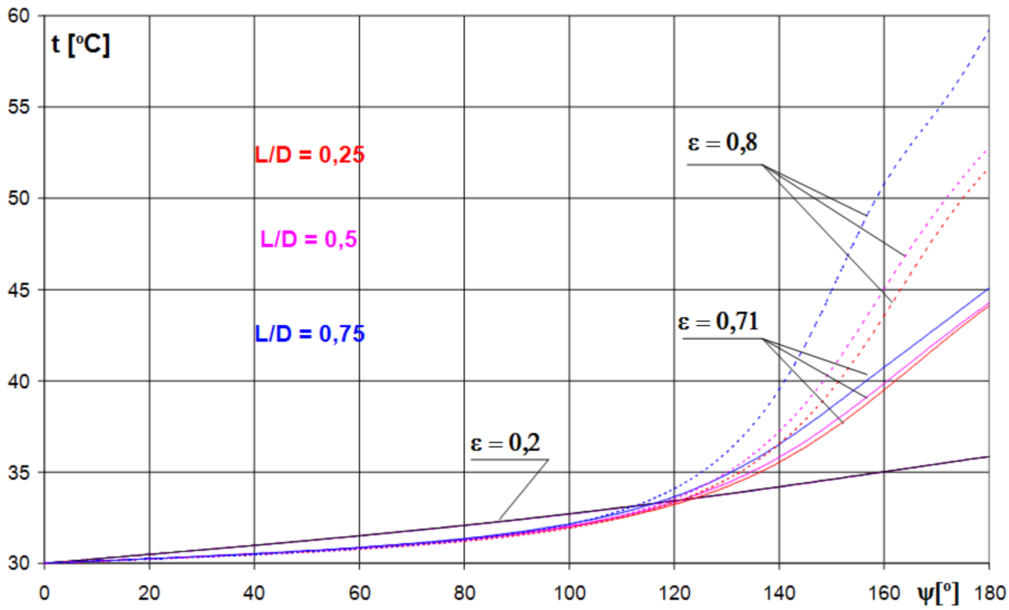

Fig.2. Changes of oil temperature along the circumference of lube slot in the presence of given values of relative eccentricity "ε" and the bearing breadth ratio to its diameter (L/D)

It follows from the diagrams that elementary volume of the oil flowing into the slide bearing will have, immediately after entering the bearing , a little higher temperature than before flowing into it. On the other hand, when leaving the bearing, the temperature of the elementary volume, violently increases.

It is caused by continual delivery of heat when flowing through the oil slot and undergoing the process of oil layers being cut down. It causes a violent increase of oil temperature on the outlet from the bearing.

Non linear increase of outlet oil temperature from the bearing depends firmly on an increase of its relative eccentricity and on the ratio of the bearing breadth to its diameter.

It follows from fig.3 that the least increase of oil temperature along the circumference of the oil slot, is obtained, in the presence of, relative eccentricity "ε = 0,71", which is, in accordance with the information, given in work [2].

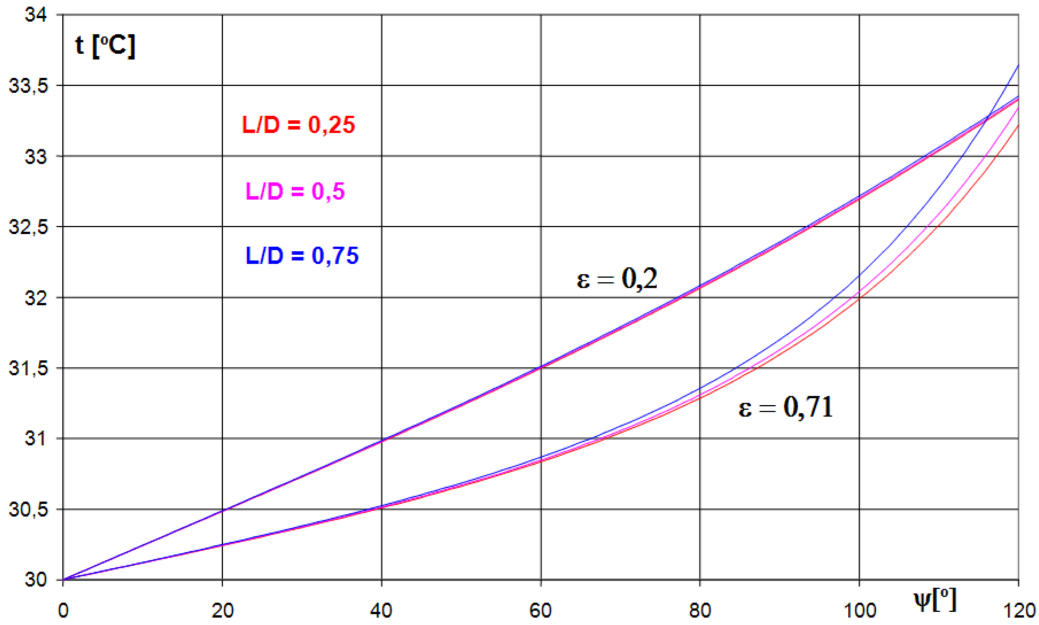

Fig.3. Changes of oil temperatures along the circumference of the lube slot in the presence of different relative eccentricity "ε" and the bearing breadth ratio to its diameter (L/D)

Temperature of oil is the sum of mean temperature and its fluctuation which can be described by trigonometric function. Mean temperature for the accepted adiabatic model of crosswise slide bearing can be determined on the grounds of the equation of energy balance and fluctuation parameters from experimental research. In the following way one can obtain a formula for oil temperature changes of the oil flowing through the lube slot in the function of time:

$$t(\tau) = y \cdot \tau^{z} + \Delta t \cdot \sin k \cdot \pi \cdot \tau, \qquad (5)$$

where:

τ[s] – determination time of oil temperature when the oil flows through the lube slot of slide bearing,

Δt[°C] – amplitude of oil temperature changes,

$y\left[\frac{^{o}C}{s^{2}}\right] = 1$ – dimensional coefficient of temperature signal intensification,

k[-] – multiple of fluctuation period of oil temperature, flowing through the bearing.

Fig.4 presents temperature changes of oil flowing through lube slot of crosswise slide bearing in function of time at its fluctuations of identical frequency and different amplitude. Amplitude of temperature fluctuation can be caused by short duration contact of the pin and bearing bush, causing its wear down. As a result one can observe intensive emission of heat in the slide bearing, collected by oil, which causes an increase of its temperature. After the pin separation from the bearing bush, the amount of heat, collected by oil, decreases and its temperature drops. The time of bearing bush and the pin contact is proportional to the amplitudes value of oil temperature changes. On the other hand, the frequency of oil temperature changes, depends on the rotational speed of the pin in the bearing.

Particularly dangerous are maximal values of oil temperature for example, over 80 [^{0}C]. They can cause bearing seizure.

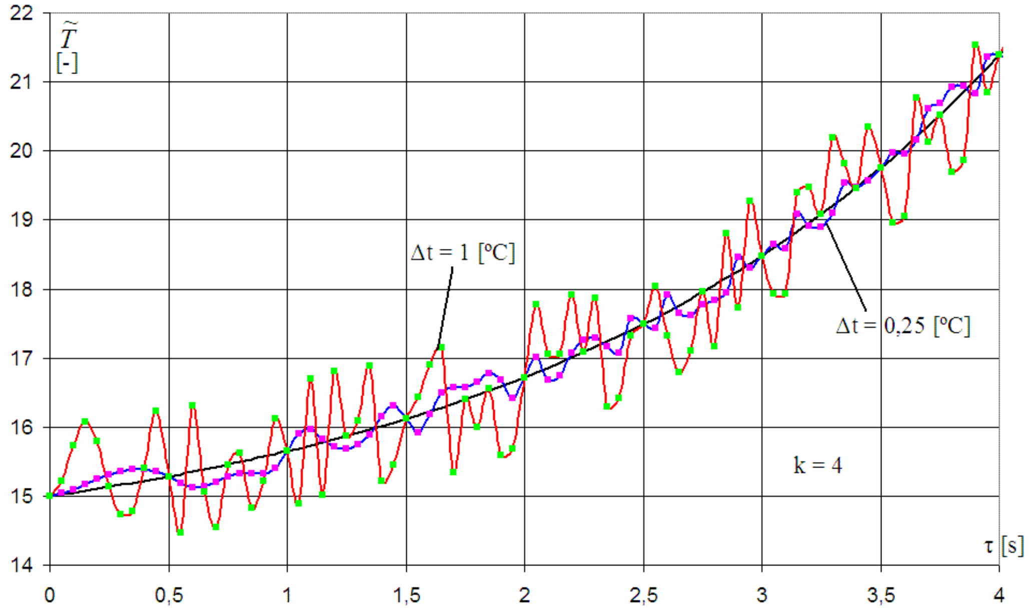

Fig.4. Temperature changes of oil flowing through the lube slot of crosswise slide bearing, caused by disturbances of different amplitudes and identical frequency in time function

Temperature of oil flowing from the slide bearing is a diagnostic symptom of its condition. If it exceeds critical value, then we observe, destruction of lubrication film, which is a symptom of the slide bearing seizing, according to temperature criterion by H.Blak [3,7]. According to Matwiejewski [3] critical temperature of lubrication oil which causes distinctive destruction, is within the range of 150 – 180 [^{0}C]. Permissible temperature increase of oil flowing through the bearing, which does not disturb the oil wedge, finds its place within the range of 3 27 [^{0}C].

4. Recapitulation

On the grounds of the analysis determining work conditions of slide bearings in the piston-connecting rod system of the engine, one can propose oil temperature as the symptom defining the correctness of heat operation.

Temperature of oil in the bearing includes information, concerning load conveyance abilities. Basing on the knowledge of temperature changes, one can conclude about the condition changes of bearing work and quickly identify its conditions and inefficiency as well.

Temperature of oil is not the only symptom but also a difference of temperatures in respective points of friction area.

However inlet temperature of oil to the bearing, according to, Kozłowiecki research [2] affects insignificantly the temperature in the friction area.

Designation of temperature increase of oil flowing through the bearing, does not give an exact information about the kind of friction, that takes place in the oil slot. Because the amount of oil flowing through the slot is not big when compared with the whole amount in the bearing [6].

Oil temperature increase, in connection with the knowledge of flow intensity, determines the size of friction, taking place in the bearing.

On the other hand, the temperature of oil under the surface of the bearing bush allows to define the heat state of slide layer and fix the limiting state of the bearing, warning against its seizing.

Areas of minimal oil slot are the places of intensive emission of heat and it is just in their presence, when the temperature of the bearing bush is the highest.

To make use of oil temperature as a correctness symptom of bearing heat operation, it is necessary to determine earlier:

- dependence between temperature measured under the surface of the bearing bush and the temperature of oil film,
- the zone of minimal thickness of oil film,
- dependence of hydrodynamic pressure in oil film on the temperature of bearing bush surface [2,7].

References

[1] Kiciński, J., *Theory and research of crosswise slide bearings,* Publishers Ossolineum PAN, Wrocław- Warszawa-Kraków 1994.

[2] Kozłowiecki, H., *Bearings of Piston Combustion Engines,* WKiŁ, Warszawa 1982.

[3] Szczerek, M., Tuszyński, W., *Tribology Research-Seizing,* Publishers ITE, Radom 2000.

[4] Girtler, J., *Energy-based aspect of operation of diesel engine,* COMBUSTION ENGINES No 2/2009 (137).

[5] Kozaczewski, W., *Construction of piston cylinder group of diesel engines,* WKŁ, Warszawa 2004.

[6] Rosłanowski, J., *Identification of ships propulsion engine operation by means of dimensional analysis*. Journal of POLISH CIMAC, Vol 4, No 1, 2009.

[7] Włodarski, J. K.,– *Piston combustion engines tribological processes,* WKŁ, Warszawa 1982.