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Henryk RUSINOWSKI*, Grzegorz SZAPAJKO*, Wojciech STANEK*

HYBRID MODEL OF THE CONVENTIONAL POWER UNIT

SUMMARY

The paper presents the hybrid model of the conventional power unit. The model contains following fragmentary models: model of the boiler, turbine and regenerative heat exchangers. Fragmentary models have been elaborated with the application of the analytical modelling methods, regression and neural networks. Hybrid model of the boiler contains balance model compatible with the German norm DIN and regressive models describing content of the unburnt combustibles in the slag and in the dust and the neural model describing the flue gases temperature at the outlet of the boiler. Model of the turbine contains balance model and theoretical-empirical model of the steam expansion line in the turbine. Models of the heat exchangers contain the balance and empirical models describing the heat transfer.
The paper presents the exemplary calculation results and their comparison with the measurements.

Keywords: *hybrid model, conventional power unit, regression, neural networks*

MODEL HYBRYDOWY BLOKU ENERGETYCZNEGO

W pracy przedstawiono model hybrydowy bloku energetycznego. Model obejmuje modele cząstkowe: kotła, turbiny i regeneracyjnych wymienników ciepła. Modele cząstkowe opracowano, wykorzystując metody modelowania analitycznego, regresyjnego i neuronowego. Model hybrydowy kotła obejmuje model bilansowy zgodny z normą DIN oraz modele regresyjne opisujące zawartość części palnych w żużlu i w pyle oraz model neuronowy opisujący temperaturę spalin odpływających z kotła. Model turbiny obejmuje model bilansowy i model teoretyczno-empiryczny linii rozprężania pary w turbinie. Modele wymienników obejmują modele bilansowe i modele empiryczne opisujące przepływ ciepła.
W pracy przedstawiono przykładowe wyniki obliczeń i ich porównanie z wynikami pomiarów.

Słowa kluczowe: *model hybrydowy, blok energetyczny, regresja, sieci neuronowe*

Nomenclature

$\dot{E}$ – energy flux, kW
$\dot{G}$ – mass flow, kg/s (t/h)
h – specific enthalpy, kJ/kg
l – work, J
$\dot{m}$ – mass flow, t/h (kg/s)
N – electric power, kW
$\dot{Q}$ – thermal flux, kW (MW)
p – pressure, kPa (MPa)
S – relative energy losses,
T – temperature, K
t – temperature, ^{o}C
v – specific volume, m^3/kg
W_d – lower calorific value, kJ/kg

1. INTRODUCTION

Contemporary thermal diagnosis systems of the power units require mathematical models of the thermal processes. Mathematical models are used for calculating reference parameters and calculation of the influence of the operating parameters deviation from reference values on unitary fuel consumption, and for optimisation of the operation. Models are constructed on the basis of physics laws with auxiliary regression and neural models.

Presented simulation model of the power unit comprises:
– balance model of the turbine,
– balance model of the regenerative heat exchangers,
– balance model of the condenser,
– model of the steam expansion line in the turbine,
– model of the heat transfer in the heat exchangers.

Conventional power unit is a complex energy system. It is assembled by many elements: steam generator, turbine, condenser, regenerative heat exchangers and cooling tower (Fig. 1).

The elements which are parts of the system are connected and interdependent. The effective tool for constructing a model is integration of the analytical modelling and intelligent techniques in a hybrid model (Rusinowski *et al.* 2008, Stanek and Rusinowski 2008). Figure 2 presents diagram of the elaborated hybrid model of the power unit.

Data for calculations contain two groups:

A) Data from measurement system after validation by compensatory calculus $\hat{\mathbf{U}}$.

B) Set of values of operating parameters $\mathbf{X}_{BL}$, for which simulation calculations can be conducted:
   – set of parameters for a steam generator $\mathbf{X}_{K}$,
   – set of parameters for a cycle $\mathbf{X}_{OB}$,
   – set of parameters for a cooling tower $\mathbf{X}_{CH}$.

* Institute of Thermal Technology, The Silesian University of Technology

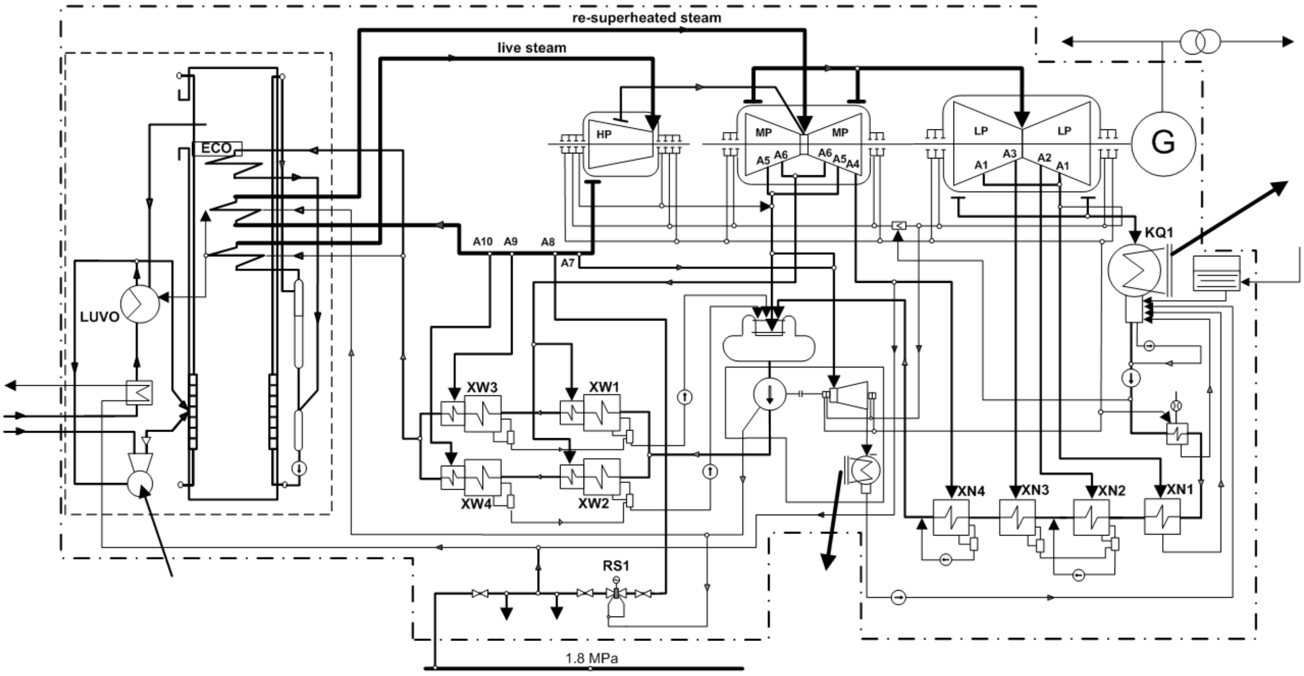

Fig. 1. Diagram of the conventional power unit

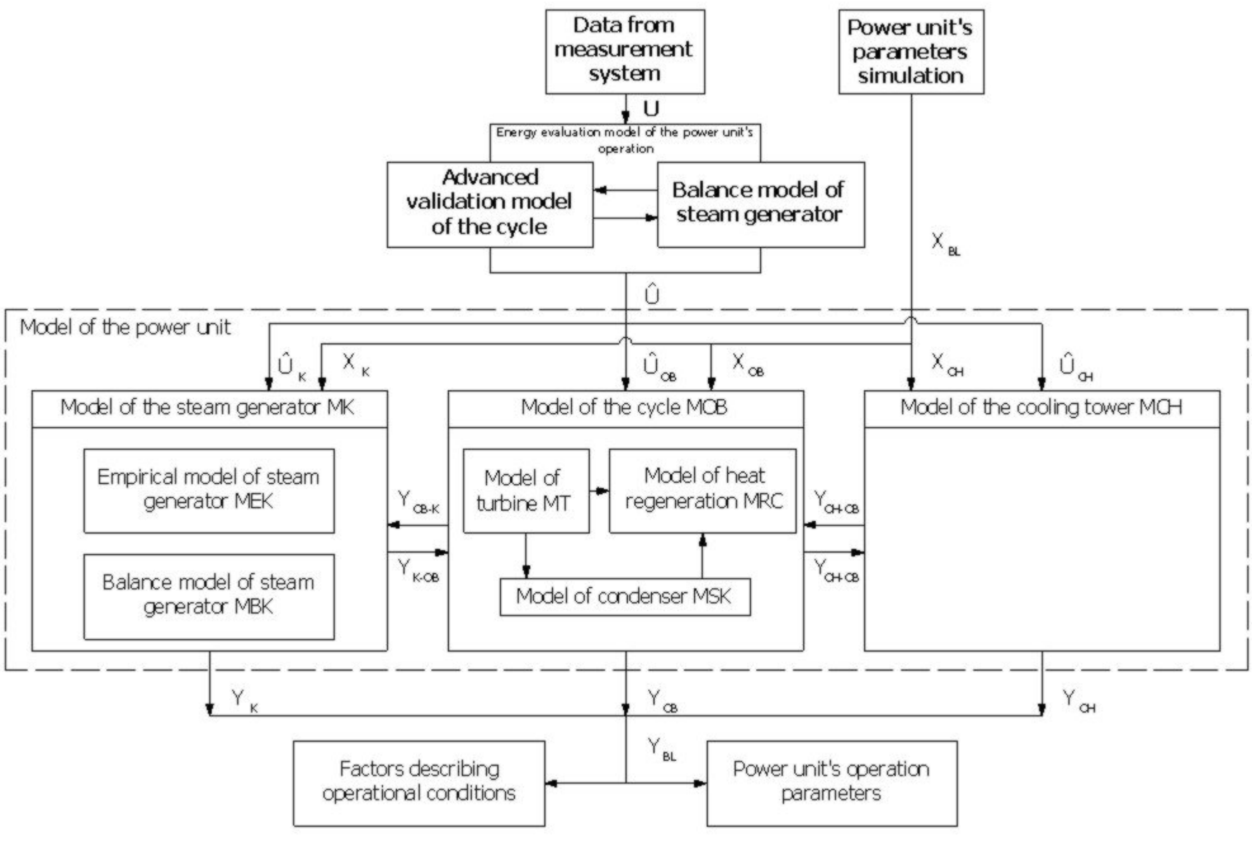

Fig. 2. Diagram of the hybrid model of the power unit

Calculation results contain: mass flows and thermal parameters in the typical points of the cycle $\mathbf{Y}_{OB}$, parameters of the agents at the outlet of the boiler $\mathbf{Y}_{K}$ and cooling water temperature at the outlet of the cooling tower $\mathbf{Y}_{CH}$.

2. THEORETICAL-EMPIRICAL MODEL OF THE STEAM-WATER CYCLE

Fundamental influence for energy evaluation of the operation has the accuracy of determination parameters of the turbine's operation in various operating conditions. One of the basic parameter for evaluation of the effectiveness of the turbine's operation is a course of the steam expansion line. It determines the energy efficiency of the transformation process in which the steam energy changes into mechanical energy. To point the steam expansion line in a turbine, there are two methods to use: modelling of the flow through groups of stages and a method based on the steam flow capacity equation and the efficiency of the process equation. Both methods combination is also possible. Performing flow calculations requires knowledge of the turbine's geometry. These calculations demand complex models and are time-consuming. Calculations on the basis of the steam flow capacity and efficiency of the process equations demand only the knowledge of the empirical parameters of these functions.

Regarding the purpose, which is constructing the model of the power unit for the operating control system needs, the paper presents a method to determine the steam expansion line. The method on the basis of the steam flow capacity and the efficiency of the process equations for the following groups of the turbine's stages has been used.

Regenerative heat exchangers in which multidimensional heat transfer from condensing steam to heated water is realized are important parts of the power unit. Apart from using the complex mathematical description, simpler empirical functions could be used. Such regenerative heat exchangers modelling method is presented below.

2.1. Model of the steam expansion line

In a turbine, operating with the constant rotational speed, the steam expansion process is determined by the inlet parameters (pressure p_{in}, temperature T_{in}) and the steam mass flow in the analysed group of the stages $\dot{m}$. For each turbine operating with constant rotational speed, there is an exact relation between p_{in}, T_{in} and outlet pressure of the group of stages p_{out}, called steam flow capacity of the turbine relation (Chmielniak 1993, Perycz 1992):

$$\dot{m} = f\left(p_{in}, T_{in}, p_{out}\right) \tag{1}$$

The steam flow capacity equation is written down for two states: actual state and assumed reference state, which could be a state, for which maximal efficiency of the turbine is obtained. One of the most common form of the steam flow capacity equation is Stodola-Flügel's equation (Perycz 1992):

$$\frac{\dot{m}}{\dot{m}^0} = \frac{\mu}{\mu^0}\frac{p_{in}}{p_{in}^0}\sqrt{\frac{p_{in}^0 v_{in}^0}{p_{in} v_{in}}}\sqrt{\left[\frac{1-\left(\frac{p_{out}}{p_{in}}\right)^{\frac{n+1}{n}}}{1-\left(\frac{p_{out}^0}{p_{in}^0}\right)^{\frac{n+1}{n}}}\right]} \tag{2}$$

where:

n – polytropic exponent,
μ – outlay coefficient and index 0 distinguish reference state.

The accuracy of the prediction on the basis of the relation (2) is not always satisfying (Bujalski and Lewandowski 2001). Thus, from the literature we selected the other form of the steam flow capacity of the turbine equation (Szapajko and Rusinowski 2008a, 2008b):

$$\dot{m}^2\frac{v_{in}}{p_{in}} = A\left[1-\left(\frac{p_{out}}{p_{in}}\right)^2\right] + B \tag{3}$$

where: A and B – empirical coefficients.

The internal efficiency of the turbine expresses the relation of the internal work to the theoretical work. On its basis the level of irreversibility of the adiabatic mechanical devices can be specified. The formula expresses this efficiency:

$$\eta_{i\ in\text{-}out} = \frac{l_i}{l_{th.}} = \frac{h_{in}-h_{out}}{h_{in}-h_{outs}} \tag{4}$$

where:

$\eta_{i\ in\text{-}out}$ – the internal efficiency of the turbine,
l_i – internal work,
l_{th} – theoretical work,
h_{in} – specific enthalpy of the steam at the inlet of the turbine,
h_{out} – specific enthalpy of the steam at the outlet of the turbine,
h_{outs} – specific enthalpy of the steam for the adiabatic reversible process.

Different empirical functions are used in the literature for expressing the internal efficiency of the turbine. The majority depend on an outlet pressure from the group of the turbine's stages or on the relation between outlet and inlet pressure.

After literature analysis the paper presents the following relation for prediction efficiency of the process for the individual group of stages of the turbine (Bujalski and Lewandowski 2001, Miller 1975):

$$\eta_i = \alpha + \beta\left(\frac{p_{out}}{p_{in}}\right)^{-1} + \gamma\left(\frac{p_{out}}{p_{in}}\right)^4 \tag{5}$$

where: α, β and γ are empirical coefficients.

Figure 3 presents diagram of the high-pressure (HP) part of the analyzed turbine.

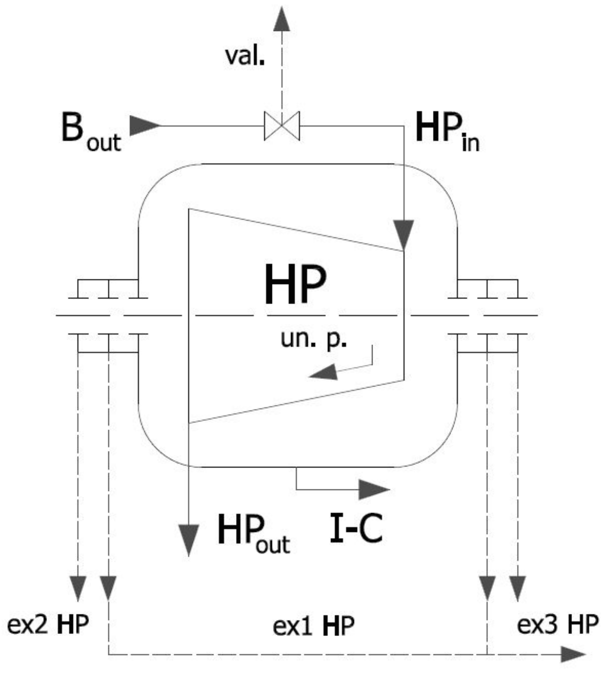

Fig. 3. Diagram of HP part of the turbine

There is a superheated steam in a whole analysed loading range of the power unit in the HP part of the turbine.

The steam mass flow at the inlet of the high-pressure part of the turbine $\dot{m}_{HP_{in}}$ is lesser than the mass flow fresh steam from the boiler $\dot{m}_{B_{out}}$ about leaks in valves seals $\dot{m}_{val.}$ and about the mass flow on unload piston $\dot{m}_{un.p.}$:

$$\dot{m}_{HP_{in}} = \dot{m}_{B_{out}} - \dot{m}_{val.} - \dot{m}_{un.p.} \tag{6}$$

The steam mass flow at the outlet of the high-pressure part of the turbine is lesser than mass flow of the steam at the inlet of the HP part of the turbine about mass flows of the steam from the external glands $\dot{m}_{ex1_{HP}}, \dot{m}_{exr2_{HP}}, \dot{m}_{ex3_{HP}}$ and about the mass flow of the inter-cylinder steam $\dot{m}_{I-C}$ (ABB Zamech Ltd.):

$$\dot{m}_{HP_{out}} = \dot{m}_{HP_{in}} - \dot{m}_{ex1_{HP}} - \dot{m}_{ex2_{HP}} - \dot{m}_{ex3_{HP}} - \dot{m}_{I-C} \tag{7}$$

The mass flows of the external glands and inter-cylinder steam, leaks in the valves seals and the mass flow on unload piston are approximated by the linear relation:

$$\dot{m}_i = D + C \cdot \dot{m}_{B_{out}} \tag{8}$$

where:

$i = ex1\,HP;\ ex2\,HP;\ ex3\,HP;\ ex1\,MP;\ ex2\,MP;\ ex1\,LP;$

$ex2\,LP;\ I-C;\ un.p.;\ val.$

Values of the coefficients from Eq. (8) were obtained on the basis of the producer's data (ABB Zamech Ltd.).

For the estimation of the parameters of the functions (3) and (5) the following estimation criterion is assumed:

$$F_{HP} = \sum_{i=1}^{M}\left[\left(\frac{t^m_{HP_{out}} - t^{cal}_{HP_{out}}}{t^m_{HP_{out}}}\right)^2 + \left(\frac{p^m_{HP_{out}} - p^{cal}_{HP_{out}}}{p^m_{HP_{out}}}\right)^2\right] \to \min \tag{9}$$

where:

i, M – number and total number of the special measurement(s),
index cal – means calculated value,
index m – means measured value.

Data for calculations is a results of 25 special measurements from wide load of the turbine range. On account of the slip control of turbine's power, steam pressure at the inlet of the HP part of the turbine (in kPa) was determined on the basis of the relation (Szapajko and Rusinowski 2008b):

$$p^m_{HP_{in}} = 8275 + 5.38e - 2N - 8e - 8N^2 \tag{10}$$

where N – electric power of the turbine, in kW.

The minimum of the function (9) has been found by Powell's method.

Figure 4 presents diagram of medium-pressure (MP) part of the analysed turbine.

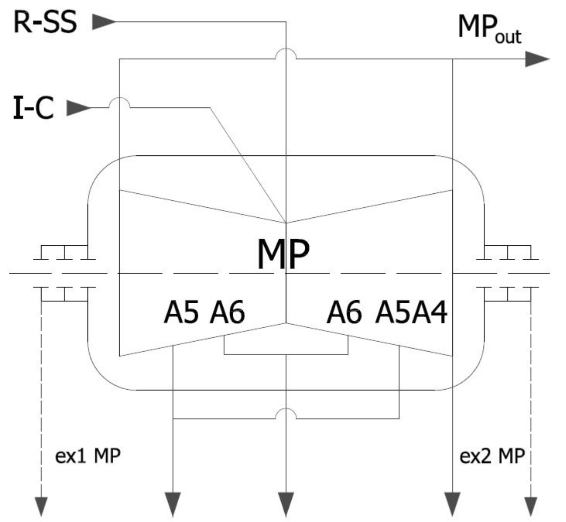

Fig. 4. Diagram of MP part of the turbine

There is a superheated steam in whole analysed loading range of the power unit in the MP part of the turbine.

Steam's symmetric spill has been assumed. The mass flows of the external glands have been omitted. Three groups of stages has been distinguished between:

- inlet to the MP part and A6 bleed: index MP_{in} –$A6$,
- bleeds A6 and A5: index $A6 - A5$,
- bleed A5 and outlet of the MP part (bleed A4): index $A5 - A4$.

The steam mass flows in the following groups of stages are determined on the basis of the mass balance (Fig. 4):

$$\dot{m}_{MP_{in}-A6} = \frac{1}{2}\left(\dot{m}_{R-SS} + \dot{m}_{I-C}\right) \tag{11}$$

$$\dot{m}_{A6-A5} = \frac{1}{2}\left(\dot{m}_{R-SS} + \dot{m}_{I-C} - \dot{m}_{A6}\right) \tag{12}$$

$$\dot{m}_{A5-A4} = \frac{1}{2}\left(\dot{m}_{R-SS} + \dot{m}_{I-C} - \dot{m}_{A6} - \dot{m}_{A5}\right) \tag{13}$$

where: index $R-SS$ means re-superheated steam from the boiler.

For the estimation of the parameters of the functions (3) and (5) the following estimation criterion is assumed:

$$F_{MP} = \sum_{j=1}^{3}\sum_{i=1}^{M}\left[\left(\frac{t_{i,j}^{m} - t_{i,j}^{cal}}{t_{i,j}^{m}}\right)^2 + \left(\frac{p_{i,j}^{m} - p_{i,j}^{cal}}{p_{i,j}^{m}}\right)^2\right] \to \min \tag{14}$$

where: j – outlet from the group of turbine's stages, $j = A6$; $A5$; $A4$.

Data for calculations is a results of 25 special measurements from wide load of the turbine range. The minimum of the function (14) has been found by Powell's method.

Figure 5 presents a diagram of the low-pressure (LP) part of the analysed turbine, a condenser and a system of its cooling.

In whole analysed loading range of the power unit in the LP part of the turbine steam from the bleed A3 is superheated and steam from the bleed A1 and at the outlet of the turbine is wet. Whereas steam from the bleed A2 is near the border of the saturation state. For majority of the loads it is superheated steam.

The steam's symmetric spill of has been assumed. The mass flows of the external glands have been omitted. The following groups of stages are distinguished between:

- inlet to the LP part and A3 bleed: index $LP_{in} - A3$,
- inlet to the LP part and A2 bleed: index $LP_{in} - A2$,
- bleeds A3 and A1: index $A3 - A1_L$,
- bleeds A2 and A1: index $A2 - A1_P$,
- A1 bleed and outlet of the LP part: index $A1 - LP_{out}$.

The steam mass flows in the following groups of stages are determined on the basis of the mass balance (Fig. 5):

$$\dot{m}_{LP_{in}-A3} = \dot{m}_{LP_{in}-A2} = \frac{1}{2}\dot{m}_{LP_{in}} \tag{15}$$

$$\dot{m}_{A3-A1_L} = \frac{1}{2}\dot{m}_{LP_{in}} - \dot{m}_{A3} \tag{16}$$

$$\dot{m}_{A2-A1_P} = \frac{1}{2}\dot{m}_{LP_{in}} - \dot{m}_{A2} \tag{17}$$

The steam pressure measurement from bleed A1 is located in the pipeline right after mixing steam from the left- and right-side pipelines. Thus, computations for the A3 and A2 bleeds run from the inlet to the LP part and for A1 bleed from the outlet of the turbine.

For the elaborated model of the steam expansion line in the medium-pressure part of the turbine, steam flow capacity equation (3) and efficiency of the process equation (5) have been investigated. As the pressure at the outlet of the turbine $p_{LP_{out}}$ depends not only on the course of the steam expansion line, but also on the cooling of the condenser conditions, for its prediction linear function of the steam

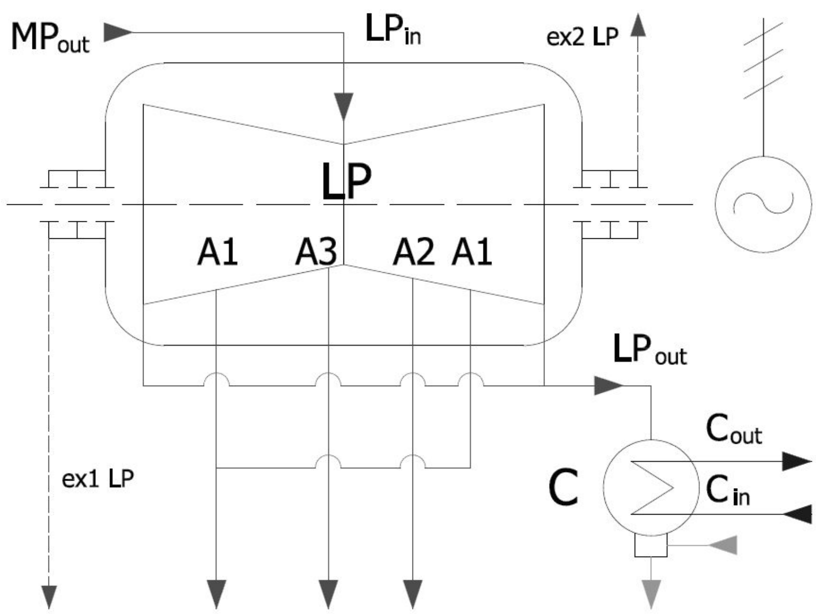

Fig. 5. Diagram of the low-pressure (LP) part of the turbine

mass flow at the outlet of the turbine $\dot{m}_{LP\ out}$ and temperature of the cooling water at the inlet of the condenser t_{Cin} has been used:

$$p_{LP\ out} = A_4 + B_4 \cdot \dot{m}_{LP\ out} + C_4 \cdot t_{C\ in} \tag{18}$$

For the estimation of the parameters of the functions (3) and (5) the following estimation criterion is assumed:

$$F_{LP} = \sum_{k=1}^{4}\sum_{i=1}^{M}\left[\left(\frac{h_{i,j}^{m} - h_{i,j}^{cal}}{h_{i,j}^{m}}\right)^2 + \left(\frac{p_{i,j}^{m} - p_{i,j}^{cal}}{p_{i,j}^{m}}\right)^2\right] \to \min \tag{19}$$

where: k – outlet from the group of turbine's stages, k = A3; A2; A1; LP_{out}.

The minimum of the function (19) has been found by Powell's method.

For the statistical evaluation of the model's quality, the correlation coefficient R^2 and the mean coefficient of the model's error δ are used.

The correlation coefficient is defined (Mańczak and Nahorski 1983) as follows:

$$R^2 = \frac{\left[\sum_{i=1}^{M}\left(Y_i^{cal} - \bar{Y}\right)\left(Y_i^{m} - \bar{Y}\right)\right]^2}{\sum_{i=1}^{M}\left(Y_i^{m} - \bar{Y}\right)^2 \sum_{i=1}^{M}\left(Y_i^{cal} - \bar{Y}\right)^2} \tag{20}$$

where:

- Y – analysed parameter (pressure, temperature or specific enthalpy);
- $\bar{Y}$ – mean value of the analysed parameter (on the basis of the measurements).

The mean coefficient of the model's error δ_Y is defined as:

$$\delta_Y = \frac{\sqrt{\sum_{i=1}^{M}\left(Y_i^{m} - Y_i^{cal}\right)^2}}{M} \tag{21}$$

Values of both coefficients have been calculated for each group of the turbine's stages.

Table 1 presents results of the statistical evaluation for the high-, medium-and low pressure part of the analyzed turbine.

Table 1. Statistical evaluation results of the turbine

No.	Group of the turbine's stages	Statistical evaluation of predicted:					
		pressure		temperature		specific enthalpy	
		δ_p, kPa	R^2, -	δ_t, °C	R^2, -	δ_i, kJ/kg	R^2, -
1.	HP part of the turbine: HP_{in}-HP_{out}	2	0,999	0,2	0,995	-	-
2.	MP part of the turbine: SP_{in}-A6 A6-A5 A5-SP_{out} (A4)	 0,4 0,4 0,4	 0,999 0,989 0,930	 0,5 1,4 0,8	 0,926 0,192 0,744	 - - -	 - - -
3.	LP part of the turbine: NP_{in}-A3 NP_{in}-A2 A1-LP_{out}	 0,4 0,6 0,3	 0,999 0,960 0,748	 - - -	 - - -	 0,5 1,2 1,8	 0,930 0,538 0,748

Figure 6 presents a courses of the steam expansion lines for the nominal and for the minimal (among considered) electrical power of the turbine. Plotted points are on the basis of the measurements and the lines are the results from the model.

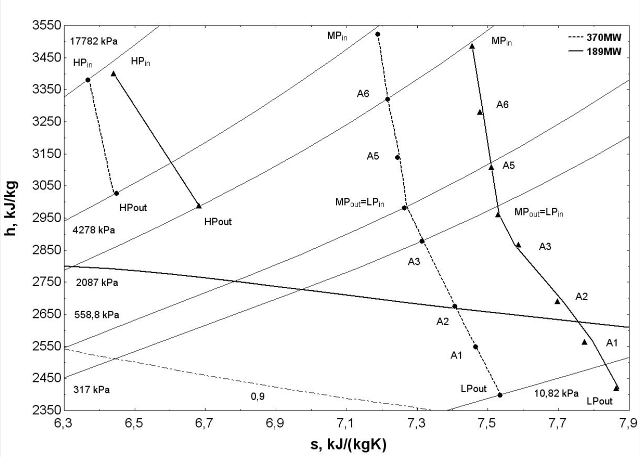

Fig. 6. The exemplary steam expansion lines

2.2. Model of the regenerative heat exchangers

Figure 7 presents a diagram of the regenerative heat exchanger of the power unit.

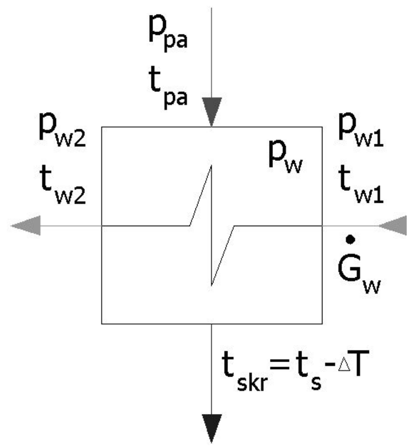

Fig. 7. Diagram of the regenerative heat exchanger: p_{pa}, t_{pa} – turbine's bleed steam pressure and temperature; p_{w1}, t_{w1}, p_{w2}, t_{w2} – pressure and temperature of the heated water at the inlet and at the outlet of regenerative heat exchanger; p_w – pressure in the r regenerative heat exchanger; t_{skr} – temperature of condensate; G_w – water mass flow; t_s – saturation temperature for pressure in regenerative heat exchanger; ΔT – supercooling of condensate

In real heat exchanger there is a combination of the directions of the agent's flow which complicates its mathematical model. In order to simplify the description of the heat transfer phenomena in exchanger thermal efficiency method could be applied (Beckman and Heil 1965; Bogusz *et al.* 2002). It allows to formulate mathematical model of the regenerative heat exchanger in a form which is easy to use in thermal diagnosis. Intensity of the heat transfer process in the regenerative heat exchanger is described with the application of the parameter called a load factor. The form of the relation for the regenerative heat exchanger, where the steam condensation proceeds (Fig. 7) is (Beckman and Heil 1965):

$$\Phi = \frac{T_{w2} - T_{w1}}{T_s - T_{w1}} \tag{22}$$

In the thermal efficiency method the load factor is approximated by the empirical function in which independent variables are steam mass flow, water mass flow, inlet temperature of the water, and saturation temperature for the pressure in the exchanger. Determination of the function in this form is complex and time-consuming. Thus, simpler relation to determinate the load factor as a function of the operating parameters has been searched. As a result of the analysis and calculations, the linear function, in which independent variable is water mass flow has been assumed (Bogusz *et al.* 2002; Stanek and Rusinowski 2008):

$$\Phi = A + B \cdot \dot{G}_w \tag{23}$$

where: A and B are empirical coefficients.

After estimation of the empirical coefficients from (23) it is possible to determinate load factor values depending on water mass flow. Knowledge of the load factor values allows to predict of the water temperature at the outlet of the regenerative heat exchanger based on (22) and steam mass flow based on energy balance.

Steam pressure losses on the way of flow from turbine's bleed to the regenerative heat exchanger is determined by the dependence arise from gas flow in the pipeline. For gas flow in a short pipeline the following relationship for pressure losses is set:

$$\delta p = \frac{1}{2} \lambda_f \frac{R T_m}{p_{pa}} \frac{L}{d} \left(\frac{\dot{G}_{pa}}{F} \right)^2 \tag{24}$$

where:
- δp – steam pressure losses in pipeline;
- λ_f – friction number;
- R – individual gas constant for steam;
- T_m – mean temperature of steam;
- L, d – length and diameter of the pipeline;
- F – cross-section area of pipeline;
- $\dot{G}_{pa}$ – bleed steam mass flow.

Assuming λ_f = *idem*, dependence (3) could be written as follows:

$$\delta p = C v_{pa} \dot{G}_{pa}^2 \tag{25}$$

where:
- C – empirical coefficient,
- v_{pa} – steam specific volume.

Coefficient C is a subject of identification on the basis of special measurements or validated operating measurements. The method of validation is the compensatory calculus.

For bleed steam pressure p_{pa}, steam pressure in an exchanger results from the relationship:

$$p_w = p_{pa} - \delta p \tag{26}$$

This pressure allows to calculate the saturation temperature presented in (22) and fluid saturation enthalpy.

Temperature of the outlet condensate is lower than saturation temperature for pressure in an exchanger:

$$\Delta T = t_s - t_{skr} \tag{27}$$

Empirical function describing variation of the condensate super-cooling as a function of thermal power of the heat exchanger has been worked out (Szapajko and Rusinowski 2008a):

$$\Delta T = D + E \cdot \dot{Q} \tag{28}$$

where: D and E are empirical coefficients.

Calculated specific enthalpy of the condensate is based on:

$$h_{skr} = h'(p_w) - c_w \Delta T \tag{29}$$

where:
- h_{skr} – condensate specific enthalpy,
- c_w – water specific heat capacity.

Steam mass flow from turbine's bleed, feeding heat exchanger, results from energy balance of the regenerative heat exchanger, which form is presented in Figure 1:

$$\dot{G}_{pa} \left(h_{pa} - h_{skr} \right) \eta_w = \dot{G}_{skr} c_w \left(t_{w2} - t_{w1} \right) \tag{30}$$

where:
- h_{pa} – bleed steam specific enthalpy;
- η_w – efficiency of the regenerative heat exchanger, taking into account environmental heat losses; such efficiency values are between 0.99–0.995.

As a result of calculations for the exemplary heat exchanger, the following estimators of the empirical coefficients were obtained: $\hat{A} = 1.111087$, $\hat{B} = -0.001324$, $\hat{C} = 2.007$, $\hat{D} = 3.94207$, $\hat{E} = -0.000116$. Full results have been presented in (Szapajko and Rusinowski 2008a).

Obtained results show high accordance with the measurements for temperature of the heated water. The results of steam pressure's losses in the pipelines and of the mass flow of the bleed steam have also sufficient accuracy however these values have less accordance to the measurements.

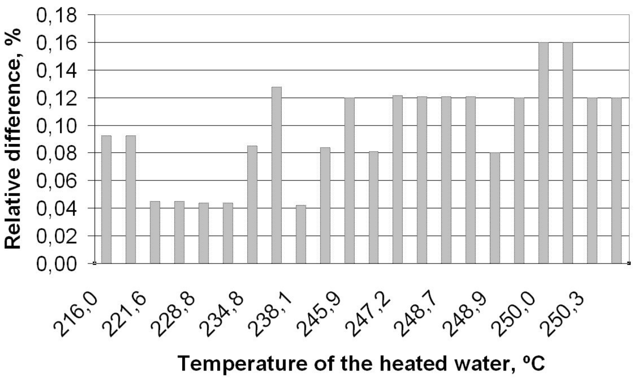

Fig. 8. Relative difference's of temperature of the heated water distribution

Figure 8 presents relative difference's of temperature of the heated water distribution calculated from the relation:

$$\delta t = \frac{t_{w2}^{m} - t_{w2}^{cal}}{t_{w2}^{m}} \cdot 100\% \tag{31}$$

Figure 8 shows that relative difference between calculated and measured values of the temperature of the heated water at the outlet of the regenerative heat exchanger is small, approximately about 0.12%.

3. HYBRID MODEL OF THE BOILER

Empirical model of a boiler comprising the dependence of the energy efficiency or energy losses on boiler operational parameters can be formulated as a set of relations between input and output parameters without the knowledge of the physical phenomenon proceeding within the model. Such a model is a so-called "black-box" model. Better results can be achieved building an analytical model in which only some auxiliary empirical functions are included.

Presented model of the boiler has been developed with the application of both analytical modelling and artificial intelligence method (Rusinowski and Stanek 2008). Such a models are classified as hybrid ones. Analytical part of the model includes balance equations compatible with the methodology of the DIN 1942 standard (Deutsche Norm 1994). Empirical models express the dependence of flue gas temperature and mass fraction of unburnt combustibles in solid combustion products on operational parameters of the boiler. Empirical models have been worked out with the application of neural and regression methods.

The block diagram of boiler hybrid model is presented in Figure 9.

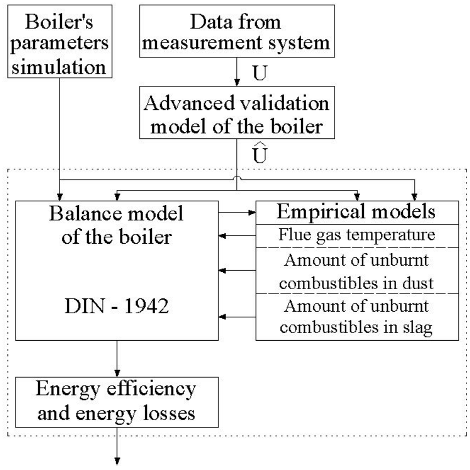

Fig. 9. Scheme of the boiler hybrid model

The energy balance equation for the boiler takes the following form:

$$\dot{E}_P + \dot{Q}_{XL} + N_{pom} = \dot{Q}_{u\dot{z}} + \dot{E}_w + \dot{E}_{\dot{z}} + \dot{E}_u + \dot{Q}_{zdm} + \dot{Q}_{ot} \tag{32}$$

where:

- $\dot{E}_P$ – flux of input energy of fuel proportional to the flux of fuel, kW,
- $\dot{Q}_{XL}$ – heat flux for steam air preheater, kW,
- N_{pom} – fans and mills driving power, kW,
- $\dot{Q}_{zdm}$ – heat flux for steam soot blower, kW,
- $\dot{Q}_{ot}$ – flux of heat losses, kW,
- $\dot{Q}_{u\dot{z}}$ – thermal power of the boiler, kW,
- $\dot{E}_u$ – losses due to enthalpy and unburned combustibles in flue dust, kW,
- $\dot{E}_w$ – flue gas energy losses, kW,
- $\dot{E}_{\dot{z}}$ – losses due to enthalpy and unburned combustibles in slag, kW,

Energy efficiency of the boiler could be determined by means of direct and indirect method. In the direct method the energy efficiency is defined as follows:

$$\eta_{Ek} = \frac{\dot{Q}_{u\dot{z}}}{\dot{P}W_d^* + \dot{Q}_{XL} + N_{pom}} = \frac{\dot{Q}_{u\dot{z}}}{\dot{E}_d} \tag{33}$$

where:

- $\dot{E}_d$ – boiler input energy, kW,
- $\dot{P}$ – fuel consumption, kW,
- W_d^* – corrected lower heating value of fuel (lower heating value of fuel with specific enthalpy of fuel and enthalpy of combustion air appended), kJ/kg,

whereas in indirect method it can be formulated as follows:

$$\eta_{Ek} = \frac{1 - \sum_i S_i^*}{1 + \dfrac{\dot{Q}_{zdm} + \dot{Q}_{ot} - \left(\dot{Q}_{XL} + N_{pom}\right)\sum_i S_i^*}{\dot{Q}_{u\dot{z}}}} \tag{34}$$

where:

S_i^* – energy losses related to the flux of fuel chemical energy,
$i = \mathrm{w}$ – flue gasses,
$\dot{z}$ – slag,
u – dust.

Due to uncertainty of the fuel consumption measurement, for the energy boilers the indirect method is used.

Basic parameters influencing the energy efficiency of the boiler are: specific amount and temperature of flue gasses as well as specific amount of solid combustion products and their composition (mass fraction of unburnt combustibles). Specific amount of flue gasses and solid combustion products can be determined by means of stoichiometry calculations. In this case the knowledge of substrates parameters of combustion process and measurement results of O_2 and CO content in flue gasses is enough. Flue gas temperature and mass fraction of combustibles in dust and slag are dependent on flow, combustion and heat transfer processes proceeding in the boiler. Mathematical description of these processes is very complicated. For this reason, the dependence of the flue gases temperature at the outlet of the boiler and contents of the unburnt combustibles in the flue gases as a function of the operating parameters, have been modelled by empirical modelling methods.

Neural model describing the flue gas temperature as a function of boiler operational parameters has been worked out (Rusinowski and Stanek 2007). Structure of the model (number of neural network layers, number of neural in layers) has been chosen in experimental way by observations of effectiveness of different network configurations. Sigmoid has been applied as activating function of neural. Different training parameters (η – learning factor, α – momentum factor) have been examined. The general structure of the neural model describing flue gas temperature is presented in Figure 10.

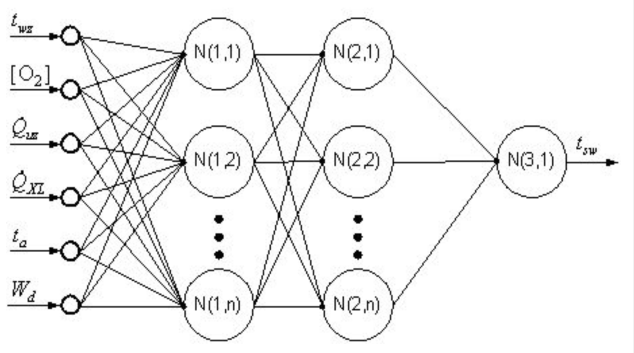

Fig. 10. Structure of neural model describing flue gas temperature

Independent variables of the model are as follows:

- feed water temperature t_{wz}, °C,
- oxygen content in flue gases $[O_2]$, %,
- boiler thermal power $\dot{Q}_{uz}$, MW,
- heat flux to steam air preheater $\dot{Q}_{XL}$, MW,
- air temperature t_a, °C,
- fuel lower heating value W_d, MJ/kg.

Flue gas temperature represents the dependent variable (output) of the neural model.

As the results of identification of neural model with the application of Lavenberg-Marquardt learning algorithm the neural network has been obtained which parameters are presented in Table 2.

Table 2. Parameters of neural network describing flue gas temperature

Number of neurons	
in input layer	6
in layer 1 and 2	9
In output layer	1
Learning factor η	0,05
Coefficient of activation function β	0,25

The network training has been carried out for 760 operational points (Fig. 11), while verification for 120 other operational points (Fig. 12).

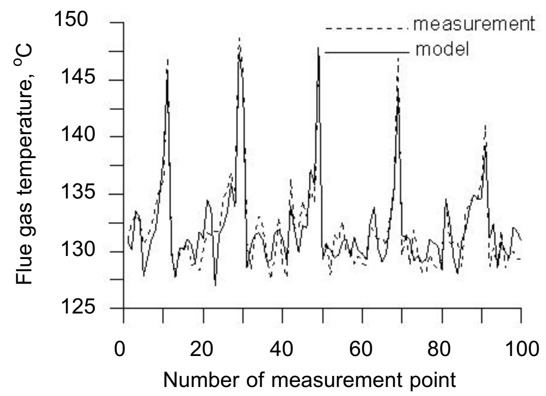

Fig. 11. Results of network training

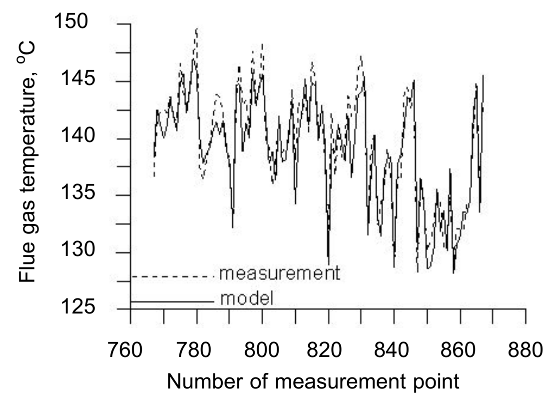

Fig. 12. Results of network verification

The differences between the network answer and expected value don't exceed 2 K. For this reason the developed model has been introduced into the hybrid model of the boiler.

Contents of the unburnt combustibles in the slag and in the dust have been described by the regression model. The most general type of the linear model of Multi Input

Single Output (MISO) can be described by the following equation:

$$y = \alpha_0 X_0 + \alpha_1 X_1 + ... + \alpha_p X_p + \varepsilon \qquad (35)$$

where:

y – output variable of the model (dependent variable),

$\alpha_0, \alpha_1, \alpha_2, ..., \alpha_p$ – parameters of the model,

$X_0 = 1$ – blind variable,

$X_j = X_j(u_1, u_2, ..., u_k)$ – assumed function of input variables,

ε – prediction error.

Estimators of coefficients of linear regression model can be determined by means of the following formula:

$$\mathbf{a} = (\mathbf{X}^T \mathbf{X})^{-1} \mathbf{X}^T \mathbf{y} \qquad (36)$$

In order to determine the influence of boiler operational parameters on mass fraction of unburnt carbon in slag $c_{\dot{z}}$ and in dust c_u the regression model of the general structure described by Eq. (35) has been investigated. Using step-wise regression method for elimination of the statistically not important elements of function (33) the linear approximation has been finally accepted:

$$c_{\dot{z}} = 5.877 + 0.6729\,[O_2] \quad R^2 = 29\% \qquad (37)$$

$$c_u = 2.4837 + 0.1994\,[O_2] \quad R^2 = 48\% \qquad (38)$$

It is the result of rather small statistical influence of boiler operational parameters on content of unburnt combustibles in solid products of combustion process**.**

Figures 13 and 14 present carbon mass fraction in slag and in the dust.

4. CONCLUSION

Nowadays, the operating control systems require mathematical models for the processes that proceed in energetic machines and devices. However, in many cases these processes are so complex that the construction of analytical models is extremely difficult and time-consuming. Therefore, more useful is such a model which includes empirical models as a source of information about the process.

The authors present the application of neural and regression methods for constructing a hybrid model of the power unit. The hybrid model contains: the model of the boiler, the model of the steam expansion line in the turbine and the models of the regenerative heat exchangers. Obtained results allow to application the model for the simulation of the power unit's work and for the energy evaluation of the operation.

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The work is co-financed by the science budget resources in years 2007–2010 as the research project (Project N N512 1746 33).

References

ABB Zamech LTD. *Leaks curves of the seals of valves and the external glands – instruction*.

Beckman G., Heil G. 1965, *Mathematische Modelle für die Beurteilung vob Kraftwersprozessen.* EKM Mittellungen, 10.

Bogusz P., Kopczyński O., Lewandowski J. 2002, *Simplified mathematical model of the heat exchanger.* XVIIIth Congress of the Thermodynamics, Research Works – Conferences, vol. 22, Publishing House of the Warsaw University of Technology, pp. 129–135 (in Polish).

Bujalski W., Lewandowski J. 2001, *Identification of the mathematical model of the steam turbine using data from Distributed Control System.* Research Works – Mechanics, Vol. 190, Publishing House of the Warsaw University of Technology, Warsaw (in Polish).

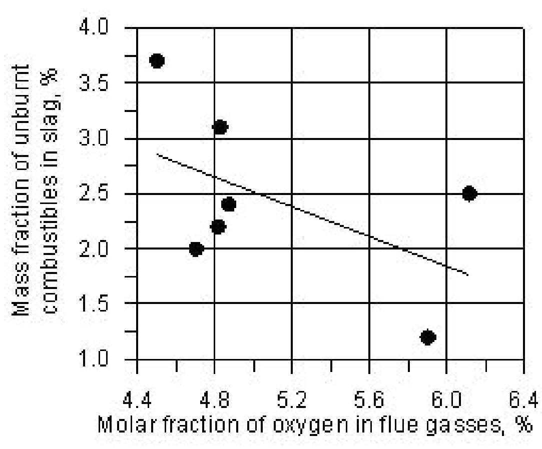

Fig. 13. Carbon mass fraction in slag

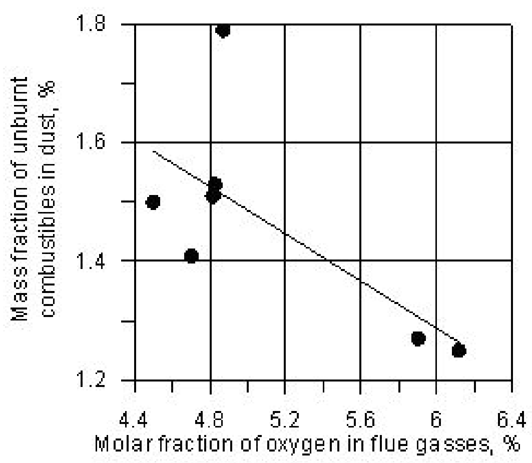

Fig. 14. Carbon mass fraction in dust

Chmielniak T. 1993, *Thermal turbines.* Gliwice: Publishing House of the Silesian University of Technology, script number 1737.

Deutsche Norm DIN 1942. *Acceptance testing of steam generators.* Edition 02/1994.

Mańczak K., Nahorski Z. 1983, *Computer identification of dynamical objects.* Warsaw, PWN (in Polish).

Miller A. 1975, *Determination of the efficiency of the turbine's group stages in the variable operating conditions.* Archives of the Machine Construction, 1(22); 55–66 (in Polish).

Perycz S. 1992, *Steam turbines.* Wrocław, Ossolineum (in Polish).

Rusinowski H., Stanek W. 2007, *Neural modelling of steam boilers*. Energy Conversion and Management 48, pp. 2802–2809.

Rusinowski H., Stanek W. 2008, *Hybrid model of the steam boiler.* Proceedings of the 21th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact on Energy Systems ECOS 2008, vol. II, Kraków, Poland, June 24–27, pp. 913–920.

Rusinowski H., Stanek W., Szapajko G. 2008, *Thermal diagnosis with application of empirical modelling.* Proceedings of 9th International Carpathian Control Conference ICCC 2008, Romania, Sinaia, May 25–28, pp. 563–566.

Stanek W., Rusinowski H. 2008, *Choice of the method for empirical modelling of steam boiler.* Proceedings of 9th International Carpathian Control Conference ICCC 2008, Romania, Sinaia, May 25–28, pp. 563–566.

Szapajko G., Rusinowski H. 2008a, *Empirical modelling of the regenerative heat exchangers for the operating control system of the conventional power unit*. Proceedings of 9th International Carpathian Control Conference ICCC 2008, Romania, Sinaia, May 25–28, pp. 671–674.

Szapajko G., Rusinowski H. 2008b, *The identification of the steam expansion line in the condensing turbine for the operating control system.* Proceedings of the 21th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact on Energy Systems ECOS 2008, vol. II, Kraków, Poland, June 24–27, pp. 1059–1066.