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Mathematical modelling of steam-water cycle with auxiliary empirical functions application

GRZEGORZ SZAPAJKO*
HENRYK RUSINOWSKI

Silesian University of Technology, Institute of Thermal Technology, Konarskiego 22, 44-100 Gliwice, Poland

Abstract Research oriented on identification of operating states variations with the application of mathematical models of thermal processes has been developed in the field of energy processes diagnostics. Simple models, characterised by short calculation time, are necessary for thermal diagnostics needs. Such models can be obtained using empirical modelling methods. Good results brings the construction of analytical model with auxiliary empirical built-in functions. The paper presents a mathematical model of a steam-water cycle containing mass and energy balances and semi-empirical models of steam expansion line in turbine as well as heat transfer in exchangers. A model of steam expansion line in a turbine is worked out with the application of a steam flow capacity equation and an internal efficiency of process equation for each group of stages for the analysed turbine. A model of a heat exchanger contains energy balance and the relation describing heat transfer in an exchanger, proposed by Beckman. Estimation of empirical equations coefficients was realised with the application of special and reliable measurements. Estimation criterion was a weighted relative sum of the remainder squares. There are exemplary calculations results presented in the final part of paper.

Keywords: CHP unit; Steam-water cycle; Bleed-condensing turbine; Semi-empirical model

*Corresponding author. E-mail address: grzegorz.szapajko@polsl.pl

Nomenclature

A	–	empirical coefficient
B	–	empirical coefficient
c	–	specific heat capacity, J/(kg K)
C	–	empirical coefficient
D	–	empirical coefficient
E	–	empirical coefficient
$\dot{G}$	–	mass flow rate, kg/s
i	–	specific enthalpy, kJ/kg
N	–	power, MW
p	–	pressure, MPa
$\dot{Q}$	–	thermal power, MW
s	–	specific entropy, kJ/(kg K)
t	–	temperature, °C
T	–	temperature, K
v	–	specific volume, m^3/kg
x	–	operating parameter

Greek symbols

α	–	empirical coefficient
β	–	empirical coefficient
δ	–	losses
Φ	–	load factor
η	–	efficiency

Subscripts

bl	–	bleed
c	–	condensate
cal	–	calculated
el	–	electrical
g	–	applies to gland
G	–	generator
he	–	applies to heat exchanger
i	–	internal
i	–	number of special measurements
in	–	inlet
m	–	mechanical
me	–	electromechanical
mea	–	measured
nom	–	nominal
out	–	outlet
s	–	saturation
T	–	applies to turbine
v	–	applies to valve
w	–	water

1 Introduction

There are factors describing operation of a CHP unit (Cogeneration – also Combined Heat and Power), such as specific fuel chemical energy consumption, energy efficiency and specific heat consumption in a turbine's cycle, applied to evaluate energy consumption in heat and electricity generation process. For each load of a CHP unit, there are values of operating parameters for which energy losses and specific fuel energy consumption are minimal. During actual operation of a CHP unit, values of operating parameters differ from values for which minimal energy losses are obtained. Thus, factors describing energy evaluation of operation differ from optimal.

For the operation and maintenance services (O&M) important is information about the influence of operating parameters which varies on specific energy consumption, specific heat consumption or energy efficiency of a CHP unit. Such information can be determined on the basis of boiler's energy characteristic and correction curves or a mathematical model. More information about the influence of operating parameters on deviations of factors describing operation of a CHP unit brings the model. It allows to analyse the influence of a bigger number of operating parameters than it is possible with the application of the correction curves and boiler's energy characteristic.

The paper presents a mathematical model of a steam-water cycle of a CHP unit with a bleed-condensing turbine. Analytical modelling techniques are combined there with empirical modelling. Basic equations of this model are mass and energy balances. A steam expansion line in a turbine and heat transfer in exchangers are described with the relations where unknown parameters occur. These unknown parameters were estimated using a regression analysis method. Exemplary calculation results are presented. They apply to the influence of selected operating parameters on specific energy consumption and generated electricity in a CHP unit with a bleed-condensing turbine in a condensing mode.

2 Mathematical model of steam-water cycle

A mathematical model can be formulated twofold: with the application of conservation laws – analytical model is constructed or with the application of measurements – empirical model is determined. Empirical models, comparing to the analytical ones, are easier to elaborate, however their application is limited to the range of operation, for which model was calibrated. They also don't explain physical issues of the process, because some parameters don't have physical interpretation. However, advantages for development of empirical models prevail, when analytical models are difficult to build or when there are the requirements about the on-line optimisation of the process parameters [1,6,8,14,17].

Figure 1 presents a diagram of the analysed CHP unit. Typical points of the cycle are denoted. Denotations of typical points are in accordance with the ones contained in special and reliable measurements [17,20].

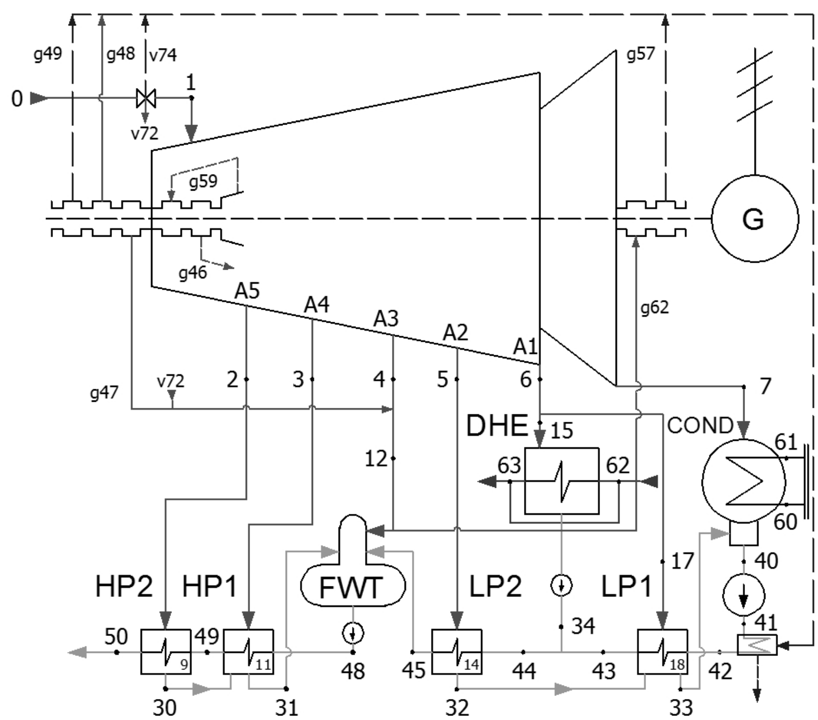

Figure 1. Diagram of the 70 MW_{el} CHP unit with bleed-condensing turbine.

Analysed steam-water cycle comprises turbine (T), condenser (COND), district heat exchanger (DHE), high- and low-pressure heat regeneration system (including heat exchangers HP2, HP1, LP2 and LP1), feed-water tank (FWT) with degasifier, vapour cooler and pumps. Mathematical model contains balance model, model of steam expansion line in turbine and empirical model of heat transfer in exchangers.

2.1 A balance model

The balance model of the steam-water cycle contains mass and energy balances for turbine, district heat exchanger, high- and low-pressure heat regeneration system, feed-water tank and condenser.

Turbine (Fig. 2):

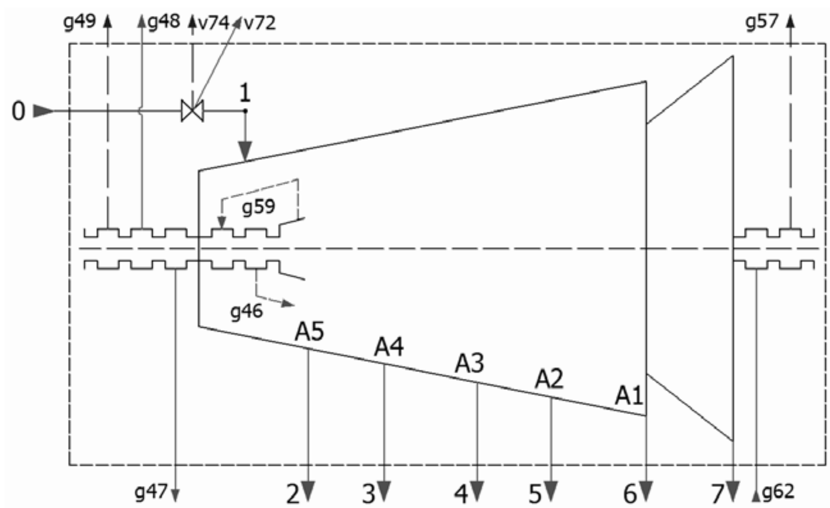

Figure 2. Diagram of a turbine.

$$\dot{G}_2+\dot{G}_3+\dot{G}_4+\dot{G}_5+\dot{G}_6+\dot{G}_7+\dot{G}_{g47}+\dot{G}_{g48}+ \\ +\dot{G}_{g49}+\dot{G}_{g57}+\dot{G}_{v72}+\dot{G}_{v74}=\dot{G}_0+\dot{G}_{g62}\ , \tag{1}$$

$$\dot{G}_{1-2}(i_1-i_2)+\dot{G}_{2-3}(i_2-i_3)+\dot{G}_{3-4}(i_3-i_4)+ \\ +\dot{G}_{4-5}(i_4-i_5)+\dot{G}_{5-6}(i_5-i_6)+\dot{G}_{6-7}(i_6-i_7)=N_{el}/\eta_{me}\ , \tag{2}$$

where steam mass flows in particular groups of stages are calculated from relations:

$$\dot{G}_{1-2}=\dot{G}_1-\dot{G}_{g46}-\dot{G}_{g47}+\dot{G}_{g48}+\dot{G}_{g49}\ , \tag{3}$$

$$\dot{G}_1=\dot{G}_0-\dot{G}_{v72}-\dot{G}_{v74}\ , \tag{4}$$

$$\dot{G}_{2-3}=\dot{G}_{1-2}-(\dot{G}_2-\dot{G}_{g46})\ , \tag{5}$$

$$\dot{G}_{3-4} = \dot{G}_{2-3} - \dot{G}_3 \ , \tag{6}$$

$$\dot{G}_{4-5} = \dot{G}_{3-4} - \dot{G}_4 \ , \tag{7}$$

$$\dot{G}_{5-6} = \dot{G}_{4-5} - \dot{G}_5 \ , \tag{8}$$

$$\dot{G}_{6-7} = \dot{G}_{5-6} - \dot{G}_6 \tag{9}$$

and electromechanical efficiency of turbine η_{me} from:

$$\eta_{me} = \eta_{mT}\eta_G \ . \tag{10}$$

It is assumed, that electromechanical efficiency of turbine in the whole range of the analysed operation is constant $\eta_{me} = 96.8\%$ [17,20].

Leaks from valves $\dot{G}_{v72}$, $\dot{G}_{v74}$ and steam and vapour mass flows from the glands $\dot{G}_{g46}$, $\dot{G}_{g47}$, $\dot{G}_{g48}$, $\dot{G}_{g49}$, $\dot{G}_{g57}$, $\dot{G}_{g59}$, $\dot{G}_{g62}$ are calculated on the basis of data provided by a turbine's producer [17]. They are approximated using a linear relation in the form:

$$\dot{G}_i = A_1 + A_2\dot{G}_0 \ , \tag{11}$$

where: A_1, A_2 – coefficients provided by the producer.

It is assumed that specific enthalpy decreases between a bleed A2 (superheated steam) and turbine outlet (wet steam). It can be determined using a linear relation in the form:

$$i = A_3 + A_4 s \ , \tag{12}$$

where: A_3, A_4 – constant coefficients, unknown in a set of equations. Value of specific enthalpy in bleed A1 – i_6 is assigned on the basis of intersection between a steam expansion line and an adequate isobar using iterative procedure.

District heat exchanger (Fig. 3):

$$\dot{Q} = \dot{G}_{15}(i_6 - i_{34}) = \dot{G}_{62}c_w(t_{63} - t_{62}) \ , \tag{13}$$

$$\dot{G}_{15} = \dot{G}_{34} \ , \tag{14}$$

$$\dot{G}_{15} + \dot{G}_{17} = \dot{G}_6 \ . \tag{15}$$

Other equations are the dependences which result from the equal flows.

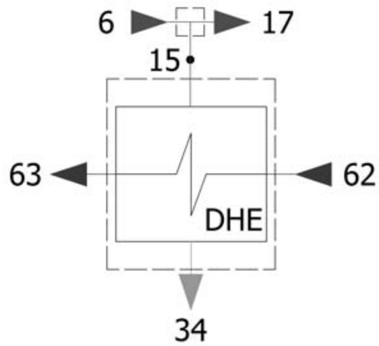

Figure 3. Diagram of a district heat exchanger.

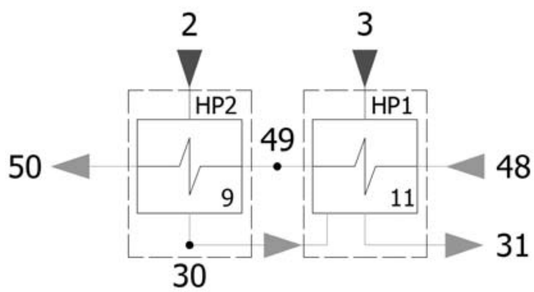

Figure 4. Diagram of a high-pressure heat regeneration system.

High-pressure heat regeneration system (Fig. 4):

$$\dot{G}_{50}i_{50} - \dot{G}_{49}i_{49} = \dot{G}_2 i_2 - \dot{G}_{30}i_{30} \, , \tag{16}$$

$$\dot{G}_{49}i_{49} - \dot{G}_{48}i_{48} = \dot{G}_3 i_3 + \dot{G}_{30}i_{30} - \dot{G}_{31}i_{31} \, , \tag{17}$$

$$\dot{G}_{30} + \dot{G}_3 = \dot{G}_{31} \, . \tag{18}$$

Low-pressure heat regeneration system (Fig. 5):

$$\dot{G}_{45}i_{45} - \dot{G}_{44}i_{44} = \dot{G}_5 i_5 - \dot{G}_{32}i_{32} \, , \tag{19}$$

$$\dot{G}_{43}i_{43} - \dot{G}_{42}i_{42} = \dot{G}_{17}i_6 + \dot{G}_{32}i_{32} - \dot{G}_{33}i_{33} \, , \tag{20}$$

$$\dot{G}_{17} + \dot{G}_{32} = \dot{G}_{33} \, , \tag{21}$$

$$\dot{G}_{34} + \dot{G}_{43} = \dot{G}_{44} = \dot{G}_{45} \, . \tag{22}$$

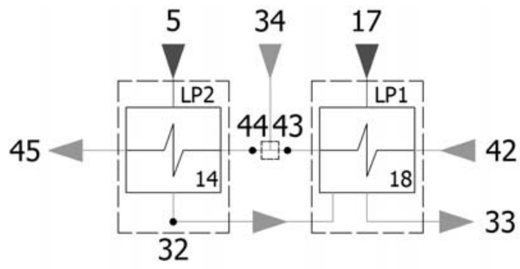

Figure 5. Diagram of a low-pressure heat regeneration system.

Feed water tank (Fig. 6):

$$\dot{G}_{47} = \dot{G}_{31} + \dot{G}_{12} + \dot{G}_{45} \ , \tag{23}$$

$$\dot{G}_{12} = \dot{G}_{4} + \dot{G}_{g45} + \dot{G}_{v72} - \dot{G}_{g62} \ . \tag{24}$$

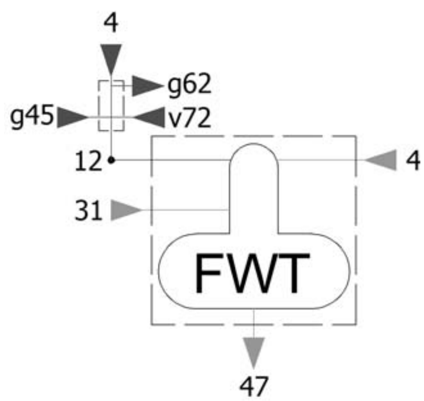

Figure 6. Diagram of a feed-water tank.

Condenser (Fig. 7):

$$\dot{G}_{33} + \dot{G}_{7} + \dot{G}_{vap} - \dot{G}_{inj} = \dot{G}_{40} = \dot{G}_{41} = \dot{G}_{42} = \dot{G}_{43} \ , \tag{25}$$

where:

$$\dot{G}_{vap} = \dot{G}_{g48} + \dot{G}_{g49} + \dot{G}_{v74} + \dot{G}_{g57} \ , \tag{26}$$

$$\dot{G}_{inj} = \dot{G}_{0} - \dot{G}_{47} \ . \tag{27}$$

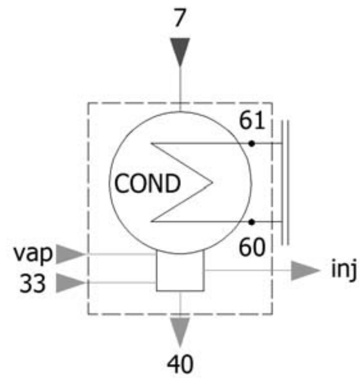

Figure 7. Diagram of a condenser.

A balance model of a steam-water cycle for the 70 MW_{el} CHP unit with a bleed-condensing turbine allows to analyse operation of a power unit on the basis of registered measurements, to determine non-measured flows and to identify a steam expansion line in a turbine.

2.2 A model of the steam expansion line

There are methods using flow modelling or methods basing on steam flow capacity and efficiency of process equations applied for evaluation of a steam expansion line in a turbine. Combining these methods is also possible. However, flow computations demand the knowledge of the flow system geometry. Such computations are time-consuming and require complex models. The computations on the basis of steam flow capacity and efficiency of process equations are simpler and less time-consuming.

The equations for a model of the steam expansion line in a turbine were formulated for particular groups of stages. Different forms of a steam flow capacity equation and an internal efficiency equation [4,5,7,10,11,13–17,19] were analysed. Good modelling results were obtained for a steam flow capacity equation [17]:

$$\dot{G}^2 \frac{v_{in}}{p_{in}} = B_1 + B_2 \left[1 - \left(\frac{p_{out}}{p_{in}} \right)^2 \right] , \tag{28}$$

and an internal efficiency equation:

$$\eta_i = B_3 + B_4 \frac{p_{out}}{p_{in}} + B_5 \left(\frac{p_{out}}{p_{in}} \right)^2 , \tag{29}$$

where: $\dot{G}$ – steam mass flow in a group of stages, p_{in}, v_{in} – steam pressure and specific volume at the inlet of the group of stages, p_{out} – steam pressure at the outlet of the group of stages, B_1, B_2, B_3, B_4, B_5 – empirical coefficients.

2.3 A model of heat exchangers

Figure 8. presents a diagram of a heat exchanger.

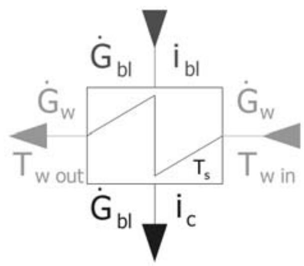

Figure 8. Diagram of a heat exchanger.

An analytical description of heat transfer phenomena in a real heat exchanger is complex. As the literature and calculation results [2,3,13,17,18] show, the analytical description can be replaced by the thermal efficiency method (the load factor method). Such a method allows to work out a mathematical model of a heat exchanger which in mathematical notation is simpler. Calculation time is shorten what is priceless for the developed model, especially when the results are satisfactory. The description of heat transfer phenomena, in a heat exchanger where steam condensation proceeds, is replaced by a load factor Φ:

$$\Phi = \frac{T_{w\,out} - T_{w\,in}}{T_s - T_{w\,in}} \,. \tag{30}$$

Beckman proposed, that a load factor Φ can be approximated using the power function dependent on operating parameters in a form [2]:

$$\Phi = C_1 \dot{G}_w^{\alpha_1} \dot{G}_{bl}^{\alpha_2} T_{w\,in}^{\alpha_3} T_s^{\alpha_4} \,, \tag{31}$$

where: C_1, α_1, α_2, α_3, α_4 – empirical coefficients.

Coefficients C_1, α_1, α_2, α_3, α_4 are determined on the basis of detailed information about geometry of exchanger and heat exchange in the reference state. Detailed data can be decreased by dividing the load factor Φ by a value Φ^{nom}, corresponding to a value in its nominal state. It allows to eliminate the coefficient C_1 leading to relation [2]:

$$\frac{\Phi}{\Phi^{nom}} = \left(\frac{\dot{G}_w}{\dot{G}_w^{nom}}\right)^{\beta_1} \left(\frac{\dot{G}_{bl}}{\dot{G}_{bl}^{nom}}\right)^{\beta_2} \left(\frac{T_{w\,in}}{T_{w\,in}^{nom}}\right)^{\beta_3} \left(\frac{T_s}{T_s^{nom}}\right)^{\beta_4} , \tag{32}$$

where: β_1, β_2, β_3, β_4 – empirical coefficients.

Constant value Φ^{nom} and exponents β_1, β_2, β_3, β_4 are determined on the basis of measurements. On the basis of research [2], Beckman states, that exponents β_2 and β_3 are close to zero, whereas the exponents β_1 and β_4 vary in a short range. Thus, the relation (32) can be written down in a simpler form:

$$\frac{\Phi}{\Phi^{nom}} = \left(\frac{\dot{G}_w}{\dot{G}_w^{nom}}\right)^{\beta_1} \left(\frac{T_s}{T_s^{nom}}\right)^{\beta_4} . \tag{33}$$

In regenerative heat exchangers, variations of condensing steam saturation temperature results from variations of the bleed steam pressure. These variations are not significant, so Eq. (33) can be rewritten:

$$\Phi = \Phi^{nom} \left(\frac{\dot{G}_w}{\dot{G}_w^{nom}}\right)^{\beta_1} . \tag{34}$$

In the presented model of the heat exchanger, linear dependence on heated water mass flow is proposed [17,18]:

$$\Phi = D_1 + D_2 \dot{G}_w , \tag{35}$$

where: D_1, D_2 – empirical coefficients.

Energy balance for the heat exchanger presented in Fig. 8 has a form:

$$\dot{G}_{bl}(i_{bl} - i_c)\eta = \dot{G}_w c_w (T_{w\,out} - T_{w\,in}) . \tag{36}$$

It is assumed here that the efficiency of a heat exchanger η is 99% [9].

Condensate specific enthalpy can be determined from the relation:

$$i_c = i(p_{he}, T_s - \Delta T) , \tag{37}$$

where: p_{he} – pressure in the exchanger, ΔT – condensate subcooling.

Condensate temperature T_c at the outlet of heat exchanger is lower than saturation temperature T_s for pressure in an exchanger p_{he} about a so-called condensate subcooling:

$$\Delta T = T_s - T_c . \tag{38}$$

A linear empirical relation describing a condensate subcooling as a function of thermal power of an exchanger $\dot{Q}$ is applied [17,18]:

$$\Delta T = E_1 + E_2\dot{Q} \; , \tag{39}$$

where: E_1, E_2 – empirical coefficients.

Pressure p_{he} occurring in Eq. (37) can be determined from a relation:

$$p_{he} = p_{bl} - \delta p \; , \tag{40}$$

where: δp – pressure losses between a bleed and an exchanger.

Pressure losses in a short pipeline δp can be calculated using the relation:

$$\delta p = \frac{1}{2}\lambda_f \frac{R\bar{T}}{p_{bl}} \frac{L}{d} \left(\frac{\dot{G}_{bl}}{F}\right)^2 \; , \tag{41}$$

where: λ_f – friction number, R – individual gas constant for steam, $\bar{T}$ – mean steam temperature, L, D, F – length, diameter and cross-section area of a pipeline. Assuming $\lambda_f = idem$, the dependence (41) can be written as:

$$\delta p = E_3 v_{bl} \dot{G}_{bl}^2 \; , \tag{42}$$

where: E_3 – an empirical coefficient. An empirical model of the heat exchanger enables determination of a bleed-steam mass flow $\dot{G}_{bl}$ and heated water temperature $T_{w\,out}$.

2.4 Estimation Group

Presented in Section 2.3 Eq. (28), (29), (35), (39) and (42) contain unknown coefficients B_1–B_5, D_1, D_2 and E_1–E_3. These coefficients were estimated using a least-squares method. The task of coefficient estimation is a nonlinear problem in mathematical notification. The *Engineering Equation Solver* (*EES*) software was used to calculate solutions.

Taking into account limitations of *EES* in extent of optimized parameters number, it was impossible to perform computations for all groups of stages and all heat exchangers together. Thus, estimation coefficients for equations describing the course of steam expansion line in a turbine and heat transfer in exchangers was performed for each group of turbine stages separately. Equations describing the course of steam expansion line and equations describing heat exchanger fed from the outlet of the analysed group were estimated together. The estimation criterion was assumed in the form:

$$\sum_{i=1}^{n}\left\{\kappa_i\left(\frac{i_i^{mea}-i_i^{cal}}{i_i^{mea}}\right)^2+\kappa_p\left(\frac{p_i^{mea}-p_i^{cal}}{p_i^{mea}}\right)^2+\right.$$
$$\left.+\kappa_T\left(\frac{T_{w\,out\,i}^{mea}-T_{w\,out\,i}^{cal}}{T_{w\,out\,i}^{mea}}\right)^2+\kappa_{\dot{G}}\left(\frac{\dot{G}_{bl\,i}^{mea}-\dot{G}_{bl\,i}^{cal}}{\dot{G}_{bl\,i}^{mea}}\right)^2\right\}\rightarrow \min\ , \qquad (43)$$

where κ is the influence weight for difference between measured and calculated values. Minimum of function (43) was found using the Powell's method [12].

For statistical evaluation of quality prediction for the elaborated model, a coefficient of determination R and a mean squared error δ for pressure and specific enthalpy at the outlet of the group of stages are used:

$$R=\frac{\sum_{i=1}^{n}(Y_i-\bar{Y})(\hat{Y}_i-\bar{Y})}{\sqrt{\sum_{i=1}^{n}(Y_i-\bar{Y})^2\sum_{i=1}^{n}(\hat{Y}_i-\bar{Y})^2}}\ , \qquad (44)$$

$$\delta=\sqrt{\frac{\sum_{i=1}^{n}(\hat{Y}_i-Y_i)^2}{n-m-1}}\ , \qquad (45)$$

where: Y_i – measurement, $\bar{Y}$ – mean value, $\hat{Y}_i$ – estimated value, n – number of measurements, m – number of estimated coefficients, $Y=\dot{G},T,p,i$.

3 Calculations results

3.1 Results of steam expansion line identification

Calculations for five operating states in a condensing mode and four operating states in the heating mode were performed with the application of the elaborated mathematical model of the steam-water cycle for 70 MW $_{el}$ CHP unit with a bleed-condensing turbine. Data for computations were carefully and reliable collected measurements. Figures 9 and 10 present exemplary courses of a steam expansion line in the condensing mode with N_{el}=72,3 MW (Fig. 9) and in the heating mode with $\dot{Q}$=73.8 MW and N_{el}=62.3 MW (Fig. 10). Solid lines represent calculated values, whereas points represent measurements. Denotations of typical points are in accordance with Fig. 1.

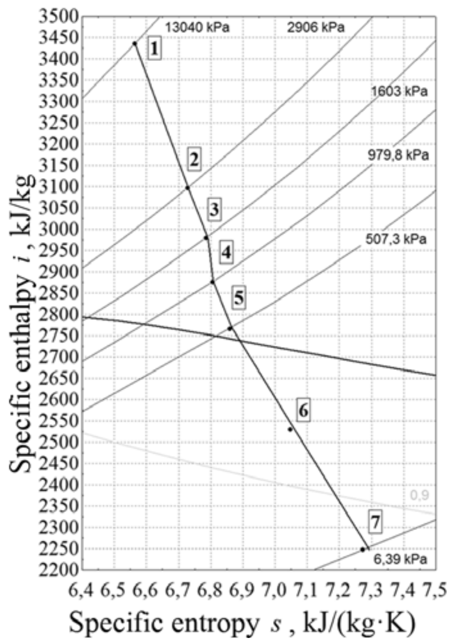

Figure 9. Course of the steam expansion line for selected operating state in a condensing mode.

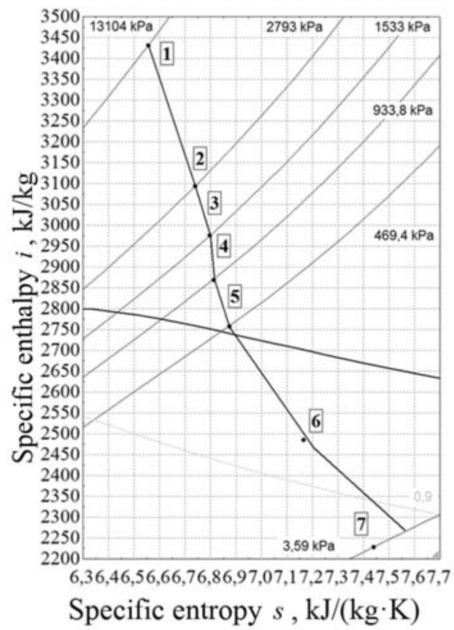

Figure 10. Course of the steam expansion line for selected operating state in a heating mode.

Calculated steam expansion lines show high consistency with the measurements in the area of superheated steam for both condensing and heating mode. In the area of wet steam, high consistency was also obtained in the condensing mode. However, in a heating mode results achieved on the basis of model have lesser agreement with measurements.

Quality of prediction for the steam expansion line is considerably better for condensing mode than for the heating one. It is caused, among the other, by over simplification in the district heat exchanger and condenser modelling. Application of the heat exchange model, which works for regenerative heat exchangers, where small load variations proceed properly, seems to be insufficient for a district heat exchanger. Moreover, the influence on quality of prediction have also a small number of special measurements available.

3.2 Calculations of selected operating parameters' influence on performance and energy consumption

Multivariate computations of selected operating parameters' influence on generated electricity, specific heat consumption in a turbine's cycle, specific fuel chemical energy consumption and energy efficiency of a CHP unit were performed using the elaborated model.

Figures 11–13 present the influence of selected operating parameters variations on variations of electricity generated in condensing mode of a CHP unit. Figure 11 presents the influence of live steam temperature's variations, Fig. 12 cooling water mass flow's variations and Fig. 13 cooling water temperature's at the inlet of a condenser.

Figures 14–16 present the influence of selected operating parameters variations on variations of specific energy consumption in condensing mode of a CHP unit. Figure 14 presents the influence of live steam temperature's variations, Fig. 15 cooling water mass flow's variations and Fig. 16 cooling water temperature's at the inlet of a condenser.

The influence of live steam temperature on electrical power and specific fuel chemical energy consumption has a course close to linear. In the analyzed variability range of live steam temperature (between 535 and 538 °C) electrical power changes about 0.4 MW, and specific energy consumption about 8 kJ/kWh. The influence of cooling water mass flow on power and specific energy consumption has more non-linear course. In the analyzed variability range (between 2700 and 3150 kg/s) electrical power changes slight (50 kW), and specific energy consumption about 7 kJ/kWh. The influence of cooling water temperature at the inlet of a condenser is important and has clearly a non-linear course. In the analyzed variability range (between 17 and 31 °C) electrical power changes by about 1.8 MW, and specific energy consumption by 250 kJ/kWh, respectively.

4 Conclusions

The elaborated mathematical model of the steam-water cycle for 70 MW_{el} CHP unit with a bleed-condensing turbine contains partial models of: a turbine, a district heat exchanger, a high- and low-pressure regeneration system, a feed-water tank with a degasifier and a condenser. The model was implemented in *Engineering Equation Solver* (*EES*). Data for calibration were carefully and reliable collected measurements. Taking into account limitations of *EES*, for each group of turbine's stages, estimation of empir-

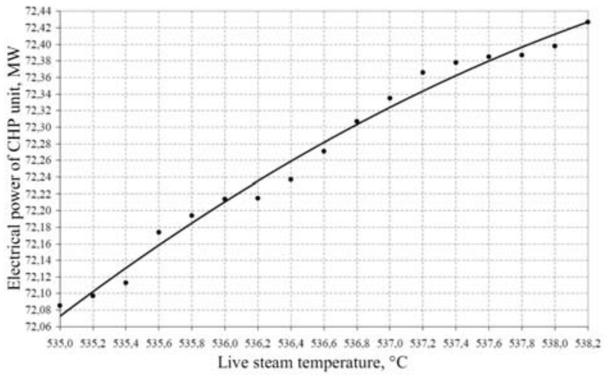

Figure 11. Influence of live steam temperature on electrical power for selected operating state in a condensing mode.

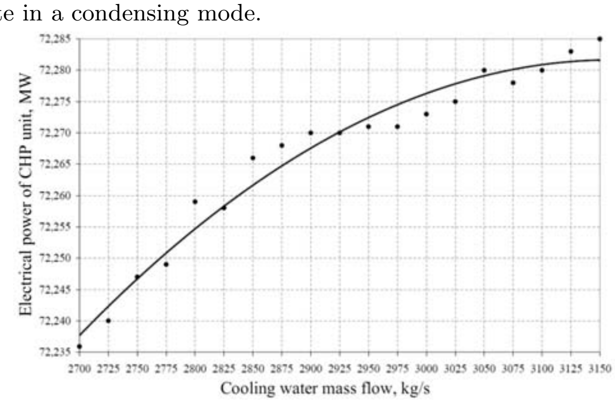

Figure 12. Influence of cooling water mass flow on electrical power for selected operating state in a condensing mode.

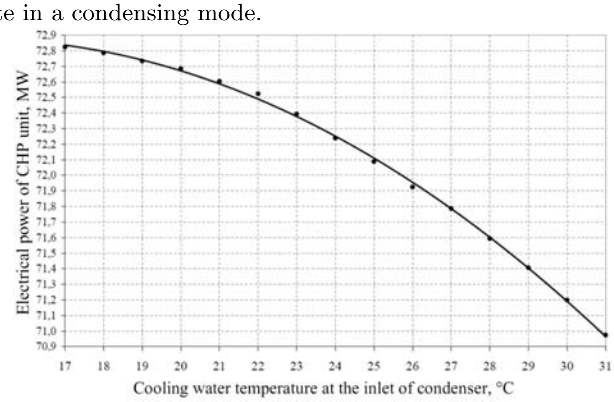

Figure 13. Influence of cooling water temperature on electrical power for selected operating state in a condensing mode.

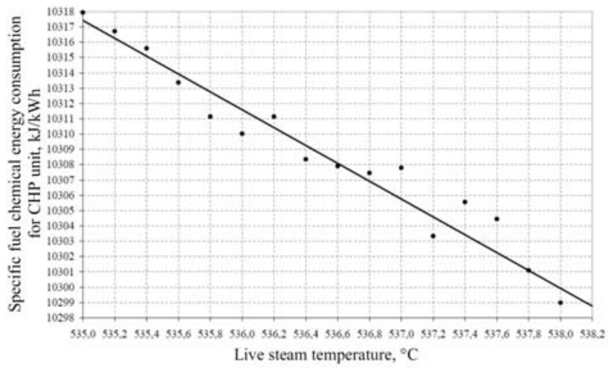

Figure 14. Influence of live steam temperature on specific energy consumption for selected operating state in a condensing mode.

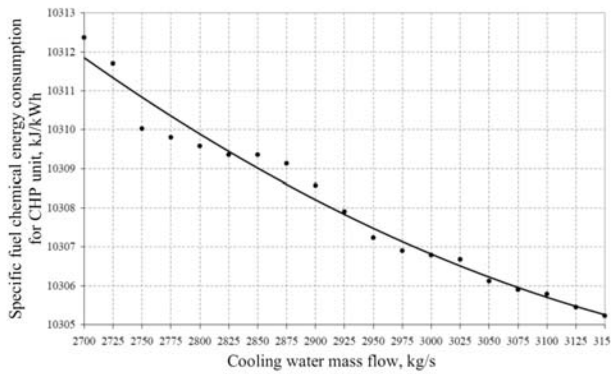

Figure 15. Influence of cooling water mass flow on specific energy consumption for selected operating state in a condensing mode.

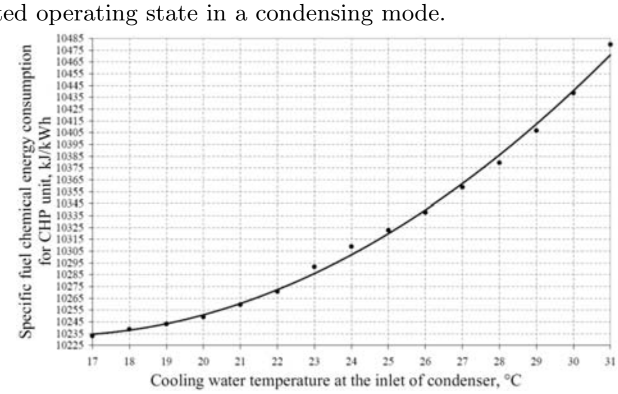

Figure 16. Influence of cooling water temperature on specific energy consumption for selected operating state in a condensing mode.

ical coefficients occurring in equations describing the course of the steam expansion line was performed together with relations describing heat transfer in the exchanger fed from the bleed.

Computations of condensing and heating modes show high accordance with measurements in the condensing mode and lower in the heating one. It applies to steam expansion line identification especially. The method of identification based on steam flow capacity equation and internal efficiency of the process equation do not bring satisfying results in the area of wet steam. Especially when steam mass flow in last group of stages is low. Therefore, other modelling techniques should be applied in a heating mode for bleed-condensing turbines.

Presented exemplary results of calculations for the influence of selected operating parameters on the factors describing the CHP unit performance in the condensing mode show a high accuracy of the model. Short calculation time allow to perform multivariate analyses, which brings a worked out model as a useful tool in the operation of control systems.

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