



**HAL**  
open science

# A Feedwater Heater Model Intended For Model-Based Diagnostics

Tomasz Barszcz, Piotr Czop

► **To cite this version:**

Tomasz Barszcz, Piotr Czop. A Feedwater Heater Model Intended For Model-Based Diagnostics. Applied Thermal Engineering, 2011, 31 (8-9), pp.1357. 10.1016/j.applthermaleng.2010.12.012 . hal-00730302

**HAL Id: hal-00730302**

**<https://hal.science/hal-00730302>**

Submitted on 9 Sep 2012

**HAL** is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.

# Accepted Manuscript

Title: A Feedwater Heater Model Intended For Model-Based Diagnostics

Authors: Tomasz Barszcz, Piotr Czop

PII: S1359-4311(10)00528-4

DOI: [10.1016/j.applthermaleng.2010.12.012](https://doi.org/10.1016/j.applthermaleng.2010.12.012)

Reference: ATE 3336

To appear in: *Applied Thermal Engineering*

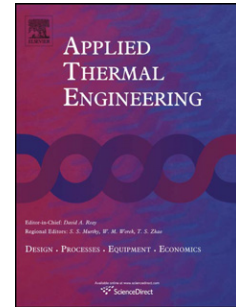
Received Date: 23 August 2009

Revised Date: 1 November 2010

Accepted Date: 4 December 2010

Please cite this article as: T. Barszcz, P. Czop. A Feedwater Heater Model Intended For Model-Based Diagnostics, *Applied Thermal Engineering* (2010), doi: 10.1016/j.applthermaleng.2010.12.012

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.



**Full title:** A Feedwater Heater Model Intended For Model-Based Diagnostics.

**Authors:** Tomasz Barszcz, Piotr Czop

**Authors' affiliation:**

University: AGH University of Science and Technology, Department of Robotics and Mechatronics,  
Street: Al. Mickiewicza 30,  
Postal code: 30-059,  
City: Cracow,  
Country: Poland.

**Corresponding author:** Piotr Czop

**Contact:**

Street: Lesna 2A,  
Postal code: 42-624,  
City: Ossy,  
Country: Poland.  
E-mail: piotr.czop@labmod.com  
Tel. (+48) 502026501  
Fax (+48) 2840877

**Abstract :**

Work related to the first-principle modeling of a feedwater heater operating in a coal-fired power unit is presented, along with theoretical discussion concerning its structural simplifications, parameter estimation, and dynamical validation. The objectives of this work are as follows: (i) formulate and deploy a moderately complex first-principle model of a feedwater heater to reproduce operational measurements in real-time simulations, (ii) develop a tuning method for this model, (iii) propose key indicators of heater performance using a model-based approach, and finally (iv) automate the calculation process of the indicators. A feedwater heater model is the main contributor to the performance of the entire simulation model of a power unit. In this work, the development process of such a model is presented, including necessary simplifications for improving its performance and functionality. As a result of the proposed simplifications, performance which allows operational data to be tracked by means of continuously updating model parameters in real-time mode was achieved on a regular PC workstation for a series of six low- and high-pressure heaters. The model variables (e.g. variability of the power rate of energy exchange) and estimated parameter values were used to formulate key performance indicators intended for a model-based diagnostics approach. Validation was successfully performed using operational data from a 225MW coal-fired power unit.

**Keywords:** power plant, feedwater heater, modeling, system identification

**Research highlights:**

The development process of the first-principle feedwater heater model is presented. > The tuning and validation were conducted for a 225MW coal-fired power unit.> The model reproduces operational measurements for purpose of model-based diagnostics.> The key performance indicators were proposed based on the model parameters.> A parametric representation of the indicators facilitates a fault detection process.>

## A Feedwater Heater Model Intended for Model-Based Diagnostics of Power Plant Installations

### Abstract

Work related to the first-principle modeling of a feedwater heater operating in a coal-fired power unit is presented, along with theoretical discussion concerning its structural simplifications, parameter estimation, and dynamical validation. The objectives of this work are as follows: (i) formulate and deploy a moderately complex first-principle model of a feedwater heater to reproduce operational measurements in real-time simulations, (ii) develop a tuning method for this model, (iii) propose key indicators of heater performance using a model-based approach, and finally (iv) automate the calculation process of the indicators.

A feedwater heater model is the main contributor to the performance of the entire simulation model of a power unit. In this work, the development process of such a model is presented, including necessary simplifications for improving its performance and functionality. As a result of the proposed simplifications, performance which allows operational data to be tracked by means of continuously updating model parameters in real-time mode was achieved on a regular PC workstation for a series of six low- and high-pressure heaters. The model variables (e.g. variability of the power rate of energy exchange) and estimated parameter values were used to formulate key performance indicators intended for a model-based diagnostics approach. Validation was successfully performed using operational data from a 225MW coal-fired power unit.

Keywords: power plant, feedwater heater, modeling, system identification

## Nomenclature

$\dot{m}$  - mass flux [kg/s]

$\dot{Q}$  - energy flux [J/s]

$h$  - enthalpy [J/kg]

$H$  - internal energy [J]

$\rho$  - density [kg/m<sup>3</sup>]

$p$  - pressure [Pa]

$T$  - temperature [K]

$V$  - chamber volume [m<sup>3</sup>]

$k$  - heat exchange coefficient [W·m<sup>-2</sup>·K<sup>-1</sup>]

$A$  - heat exchange area [m<sup>2</sup>]

$a, b$  - enthalpy saturation thresholds for the vapor and liquid phase

$h_v$  - enthalpy of saturated vapor [J/kg]

$h_L$  - enthalpy of saturated liquid [J/kg]

$c_p$  - specific isobaric heat [J·kg<sup>-1</sup>·K<sup>-1</sup>]

$h_v, T_v$  - saturated parameters of the vapor phase of the steam-water mixture

$h_L, T_L$  - saturated parameters of the liquid phase of the steam-water mixture

## 1. Introduction

The methodology proposed herein allows physical characteristics of a feedwater heater [1] to be reconstructed in order to analyze the performance of a heater using key process indicators. The power of this approach lies in tracking key process indicators by means of instantaneously adjusting, based on process data, parameters of the first-principle model.

The approach considered in this work is not intended to detect severe faults, which activate the safety systems of a power unit [2-4]; but aims at detecting relatively slow, i.e. of hours or days, changes in processes, e.g. internal leakage through a cracked pipe. A key process indicator, namely the power rate of energy exchange in a component reflects such a fault, enabling first-level analysis and indicating deviation from the targeted efficiency. The second-level analysis, including utilization of the engineering expertise and technical indicators, which are reconstructed parameters of a first-principle model, e.g. heat transfer coefficients, energy heat exchange rates and enthalpies, is performed. Investigations of this kind can be supported by process and control data, e.g. a tendency of the system to deviate from a required set-point of a controller. Among the greatest challenges, though beyond the scope of this paper, is definition, e.g. by means of 2D/3D graphs reflecting relationships among critical variables of patterns of key process indicators corresponding to a healthy system. These graphs require statistical bounds defining confidentiality range and involving process uncertainty to be imposed.

The structure of this paper is as follows. In the second section, the relevant aspects of constructing feedwater heaters are discussed together with functionalities of the model's underlying assumptions and simplifications. The third section describes the underlying assumptions as well as limitations of six-volume and four-volume heater models. The last section is the summary of the modeling work.

## 2. A combined first-principle and data-driven model of a feedwater heater

Investigation of power plant dynamics requires detailed models comprising sub-models which represent particular components of a plant. These models are based on first-principle equations (e.g. mass, momentum and energy balance) that involve phenomenological correlations, such as heat transfer coefficients [5-8]. Such models are commonly utilized to gain an understanding of physical processes as well as in process efficiency optimization [9-10]. These models are knowledge models, thanks to which process dynamics can be understood. The complexity of these models may differ depending on the modeling purpose, ranging from compact, lumped-parameter models capturing only the first-cut dynamics, through moderately complex ones, up to complex, large-scale, distributed-parameter models. In this context, a feedwater heater, as one of the components of a power plant, requires at least a moderately complex model to capture its fundamental thermodynamic processes. The proposed model applies three categories of parameters, geometrical, physical and phenomenological.

The approach proposed herein assumes that the number of tunable parameters is small compared to the number of known parameters. This, however, affects the correctness of the first-principle approach since, for example, the heat transfer phenomenon is treated using a combined coefficient which covers conduction, convection, and radiation phenomena. It is believed that the smaller the number of parameters, the more accurate the model, and the faster the convergence of the algorithm used for model adjustment. The models presented here can serve for both types of heaters, i.e. low-pressure and high-pressure ones. From a physical point of view, a feedwater heater is a heat exchanger which transfers thermal energy from three-phase liquid (i.e. water, wet and dry steam) to one-phase liquid (water feeding a boiler). Feedwater heaters are typically designed as two zones or three zones with a condensing section, desuperheater and integral subcooler. A mathematical description of the heat exchange process between two- or three-phase fluids is given in [5] and [11]. In turn, extended taxonomy of heat exchangers and a description of the heat exchanger design process, along with related engineering and constructional details are given by Shah and Sekulic [6] and Kuppan [7]. The role in power plant installations, as well as a description of constructional details, of feedwater heaters is given by Shah et al. [8] and Drbal et al. [1]. Numerical reliability aspects of modeling and simulation are raised by Henrik and Olsson in [12]. Furthermore, a simplified method of calculating the heat flow through a two-phase heat exchanger is described in [13]. The application of system identification techniques in estimating parameters of a heat exchanger working with a liquid medium is described in [14-15]. Bonivento et al. [16] discuss aspects of the predictive control vs. the PID control of an industrial heat exchanger. Focusing only on heaters operating in power plants, one can study an analysis of the influence of feedwater heaters on the operational costs of a steam power plant in [17-19]. Additionally, heater maintenance and typical operational malfunctions are described, for instance, by Andreone and Yokell in [20]. An engineering case-driven example of an implemented model of a steam-to-water heater is given by Hiltbrand and Choe [21], where a simulation model of the heater draining system is proposed towards improving the plant reliability. Strategies of condensate level control are discussed in [22-23].

Heater models are also discussed as aspects of modeling power units; an example is a modular system consisting of numerous components utilized in [2] in building large-scale models of a power unit. From this perspective, an advanced heater model is discussed by Alessandri et al. in [2]. In this model, the cavity is divided into three control volumes corresponding to (i) the desuperheating area, (ii) the condensing area, and (iii) the subcooling area. In the desuperheating area, the superheated steam is cooled down through heat exchange with the feedwater flowing in the tube-bundle until it reaches the saturated steam condition. In the condensing area the saturated steam condenses, i.e. the transition of vapor to liquid occurs, while in the subcooling area the condensed steam and the drain coming from the downstream heaters undergo a process of heat exchange with the feedwater. The model involves an assumption that the heat exchange surface between the cavity, the fluid and the tube bundle is fixed in the desuperheating area, with the heat exchange magnitude depending on the

condensing and subcooling areas. The heater model discussed in this paper has a structure similar to the one considered in Alessandri et al. in [2]. However, the version considered in this paper was formulated without the assumption that the heat exchange area for the desuperheating volume is fixed. The proposed model describes the behavior of the three-phase fluid inside the heater cavity using the equations for the conservation of the mass of drain water, the conservation of the mass of water and steam, and the conservation of the energy of subcooled water. Moreover, the model describes the behavior of the fluid in the tube-bundle during heat exchange in the desuperheating, condensing and subcooling volumes. The heater model consists of two separate circuits, i.e. steam-condensate circuit and feedwater circuit. The steam circuit captures a highly non-linear and coupled process of mass and energy accumulation in steam and water. On the other hand, the feedwater circuit only considers the energy accumulation process for water, and thus the differential equations have a simple linear form, which does not require significant computational power. Therefore, model reduction, a topic of this paper, is focused on the steam circuit.

The process of model development is described in detail in Sections 3.1-3.3. Two simplifications of a steam circuit model are considered. The first simplification discussed in Section 3.2 involves reduction of the number of differential equations in the initial six-volume model. The number of equations for the steam circuit and the feedwater circuit was reduced thanks to averaging and assuming linearized steam density properties. The second simplification of the model, presented in more detail in Section 3.3, is a four-volume model.



### 3. Model Development

This section discusses the methodology and development stages in the formulation and simplification of the feedwater heater model. The development was initiated with a six-volume model and ended up with a four-volume one. A schematic representation of two versions of the heater models discussed is shown in Figures 1-2, respectively.

The particular control volumes are defined by input and output variables, namely mass flow rate, pressure and enthalpy. For instance, the control volume  $V_{12}$  is characterized by the input enthalpy  $h_1$  and output enthalpy  $h_2$  as indicated by the arrows in Figures 1-2. Heat energy leaving the control volume is calculated as a product of the enthalpy and mass flow rate

$$\dot{Q}_2 = h_2 \cdot \dot{m}_2 \quad (1)$$

The transfer of heat energy between the corresponding control volumes  $V_{12}$  of the steam circuit and the control volume  $V_{78}$  of the feedwater circuit is described by the formula (2). This uses logarithmic means temperature difference for counterflow conditions under the assumption of uniform physical properties of the tube-bundle metal and longitudinal heat conduction in both the pipe metal and the fluid

$$\dot{Q}_{12-78} = k_{12-78} \cdot A_{12-78} \cdot \frac{(T_1 - T_8) - (T_2 - T_7)}{\ln\left(\frac{T_1 - T_8}{T_2 - T_7}\right)} \quad (2)$$

where the heat exchange area is a non-linear function of the heater height (volume of the heater cavity). The assumption of uniform and average enthalpy distribution within the control volumes of the heater is constituted by the following equation of internal energy in particular control volumes, as follows

$$H_{12} = m_{12} \cdot (h_1 - h_2) \quad (3)$$

The same formulation of equations should be repeated for the remaining control volumes in both cases of the six- and four-volume heater models.

The computation of a heater model requires many steam property evaluations at each step of the solver, which integrates the differential equations in an iterative manner. It is necessary for one or more properties to be evaluated from different couples of entry variables, typically (p, h), and (p, T), where p is the pressure, h enthalpy, and T temperature. In some cases, viscosity, conductivity and thermodynamic partial derivatives, such as specific heats at the constant pressure  $c_p$  or line derivatives along the saturation curve, are required. These water-steam fluid properties are evaluated using look-up tables based on empirical formulas which are the implementation of the IAPWS IF97 standard [25]. The look-up tables provide accurate data for water, steam and mixtures of water and steam for the pressure range of 0-100 MPa and for the temperature range of 0-2000°C.

#### 3.1. Initial six-volume model

The feedwater heater model consists of two separate flow circuits for steam-condensate and feedwater respectively. Equations (4-8) describe the steam circuit model while Equations (9-11) describe the feedwater circuit model. In the steam-condensate circuit model, Equation (4) is formulated for the conservation of the energy in the draining volume. This equation follows from the assumption concerning uniform distribution of the water density(condensate). The volume  $V_{34}$  of the drain chamber is obtained from the difference between the total volume of the shell cavity

and the space allocated by the steam. Equation (4) for the conservation of the energy in the subcooling volume  $V_{34}$  gives

$$\frac{dH_{34}}{dt} = \dot{Q}_3 - \dot{Q}_4 + V_{34} \frac{dp_{34}}{dt} - \dot{Q}_{34-56} \quad (4)$$

The term  $\dot{Q}_4$  represents the outgoing energy rate of the condensate from the actual heater to the upstream heater corrected by the term of the incoming energy and mass rate of the condensate from the downstream heater. Equations (5-6) for the conservation of the energy in the desuperheating and condensing volumes are formulated separately for the steam volumes  $V_{12}$  and  $V_{23}$  respectively.

$$\frac{dH_{12}}{dt} = \dot{Q}_1 - \dot{Q}_2 + V_{12} \frac{dp_{12}}{dt} - \dot{Q}_{12-56} \quad (5)$$

$$\frac{dH_{23}}{dt} = \dot{Q}_2 - \dot{Q}_3 + V_{23} \frac{dp_{23}}{dt} - \dot{Q}_{23-67} \quad (6)$$

where  $p_{34}=p_{23}=p_{12}$  is equal to the inlet steam pressure  $p_1$ . Equations (7-8) for the conservation of the mass in the desuperheating and condensing volumes are formulated separately for  $m_{12}$  and  $m_{23}$  respectively.

$$\frac{dm_{12}}{dt} = \dot{m}_2 - \dot{m}_1 \quad (7)$$

$$\frac{dm_{23}}{dt} = \dot{m}_3 - \dot{m}_2 \quad (8)$$

In the feedwater circuit model (Fig. 1), the following equations

$$\frac{dH_{56}}{dt} = \dot{Q}_6 - \dot{Q}_5 + V_{56} \frac{dp_{56}}{dt} + \dot{Q}_{34-56} \quad (9)$$

$$\frac{dH_{67}}{dt} = \dot{Q}_7 - \dot{Q}_6 + V_{67} \frac{dp_{67}}{dt} + \dot{Q}_{23-67} \quad (10)$$

$$\frac{dH_{78}}{dt} = \dot{Q}_8 - \dot{Q}_7 + V_{78} \frac{dp_{78}}{dt} + \dot{Q}_{12-78} \quad (11)$$

where  $p_{78}=p_{67}=p_{56}$  equal to the inlet feedwater pressure  $p_5$  are formulated for the conservation of the energy in the volumes corresponding to volumes of the steam circuit model, i.e. draining, condensing, and superheating volumes. The level of the condensate inside the heater is calculated using the following formula

$$x = \frac{V_{34} - V_{340}}{A_{con}} \quad (12)$$

where  $A_{con}$  is the area of a condensate surface in a heater cavity and  $V_{340}$  is the nominal height of the condensate volume.

As shown in Equations (4-8), considering three steam fractions separately leads to a system of five highly-coupled differential equations for a steam circuit. Nonlinear properties of the steam-water fluid are major contributors to the complexity of the nonlinear form of the model.

### 3.2. Reduced six-volume model

Linearization of the density of a steam-water liquid was proposed to reduce the complexity of the steam circuit model. Linearization of the inverse density function results in the mass and internal energy relations treating the superheating and the condensing chambers jointly. Linearization of the inverse density function is performed, obtaining three linear segments which are restricted by the following coefficients:  $a_{12}$ ,  $a_{23}$ ,  $a_{34}$ ,  $b_{12}$ ,  $b_{23}$ ,  $b_{34}$ . These coefficients correspond to the enthalpy saturation thresholds of the vapor and liquid phase. This simplification can be derived based on the assumption that the steam mass in the chamber is equal to the average density and chamber volume, as shown in the equation below

$$m_{12} = \rho_{12} \cdot V_{12} = \frac{V_{12}}{h_1 - h_2} \int_{h_1}^{h_2} \rho_{12}(h) dh = \frac{V_{12}}{h_1 - h_2} \int_{h_1}^{h_2} \frac{1}{v_{12}(h)} dh \quad (13)$$

If the inverse linearized density  $v$  function is substituted in place of the density function, then Equation (13) is integrated within the boundary enthalpy conditions  $h_1$  and  $h_2$ . As a result of integration, the steam mass is described by the linear function of the  $a_{12}$ ,  $b_{12}$  coefficients.

$$\begin{aligned} m_{12} &= \frac{V_{12}}{h_1 - h_2} \int_{h_1}^{h_2} \frac{1}{a_{12}h + b_{12}} dh = \frac{V_{12}}{h_1 - h_2} \left[ \frac{1}{a_{12}} \ln|a_{12}h + b_{12}| \right]_{h_2}^{h_1} = \dots \\ &\dots = \frac{V_{12}}{h_1 - h_2} \cdot \frac{1}{a_{12}} \cdot \ln \left| \frac{a_{12}h_1 + b_{12}}{a_{12}h_2 + b_{12}} \right| \end{aligned} \quad (14)$$

Internal energy of the chamber can be derived in a similar manner and, after inverse density function substitution and integration, the equation is as follows

$$\begin{aligned} H_{12} &= \frac{V_{12}}{h_2 - h_1} \int_{h_1}^{h_2} \frac{h}{a_{12}h + b_{12}} dh = \frac{V_{12}}{h_2 - h_1} \left[ \frac{1}{a_{12}} (h_2 - h_1) - \frac{b_{12}}{a_{12}} \cdot \frac{1}{a_{12}} \ln \left| \frac{a_{12}h_2 + b_{12}}{a_{12}h_1 + b_{12}} \right| \right] = \dots \\ &\dots = \frac{V_{12}}{a_{12}} - \frac{V_{12}}{h_2 - h_1} \cdot \frac{b_{12}}{a_{12}^2} \cdot \ln \left| \frac{a_{12}h_2 + b_{12}}{a_{12}h_1 + b_{12}} \right| \end{aligned} \quad (15)$$

The chamber volume is obtained after conversion

$$H_{12} = \frac{V_{12}}{a_{12}} - \frac{b_{12}}{a_{12}} \cdot m_{12} \Rightarrow V_{12} = H_{12} \cdot a_{12} + b_{12} \cdot m_{12} \quad (16)$$

The system of equations allows the chamber volumes to be determined based on the known linearized inverse density functions. The final system of equations may be written as follows

$$\begin{cases} H = H_{12} + H_{23} = V_{12} \left( \frac{1}{a_{12}} - \frac{b_{12}}{a_{12}} \cdot L_{12} \right) + V_{23} \left( \frac{1}{a_{23}} - \frac{b_{23}}{a_{23}} \cdot L_{23} \right) \\ m = m_{12} + m_{23} = V_{12} \cdot L_{12} + V_{23} \cdot L_{23} \end{cases} \quad (17)$$

where

$$L_{12} = \frac{1}{(h_1 - h_2)a_{12}} \cdot \ln \left| \frac{a_{12}h_1 + b_{12}}{a_{12}h_2 + b_{12}} \right|, \quad L_{23} = \frac{1}{(h_2 - h_3)a_{23}} \cdot \ln \left| \frac{a_{23}h_2 + b_{23}}{a_{23}h_3 + b_{23}} \right| \quad (18)$$

$V_{23\max}$  maximal condensing volume and its internal energy can be determined for the steam of mass  $m$  using a saturated enthalpy function for vapor  $h_2=h_v$  and liquid phase  $h_3=h_L$  as follows

$$V_{23\max} = \frac{m}{L_{23}}, \quad H_{23\max} = V_{23\max} \left( \frac{1}{a_{23}} - \frac{b_{23}}{a_{23}} \cdot L_{23} \right) \quad (19)$$

The volume  $V_{23}^*$  can be determined for the steam of mass  $m$  using inlet steam enthalpy  $h_1$  and saturated liquid enthalpy  $h_3=h_L$  as follows

$$V_{23}^* = \frac{m}{L_{23}^*}, \quad \text{where } L_{23}^* = \frac{1}{(h_1 - h_3)a_{12}} \cdot \ln \left| \frac{a_{23}h_2 + b_{23}}{a_{23}h_3 + b_{23}} \right| \quad (20)$$

The following boundary conditions can be applied if the superheating and condensing volumes are greater than zero

$$V_{23\max} < V_{23}^* \rightarrow \begin{cases} V_{12} = \frac{m - V_{23} \cdot L_{23}}{L_{12}} \\ V_{23} = \frac{\frac{H \cdot a_{12}}{1 - b_{12}L_{12}} - \frac{m}{L_{12}}}{\frac{a_{12}(1 - b_{23}L_{23})}{a_{23}(1 - b_{12}L_{12})} - \frac{L_{23}}{L_{12}}} \end{cases} \quad (21)$$

and if the superheating volume is equal to zero and the condensing volume is greater than zero

$$V_{23\max} \geq V_{23}^* \rightarrow \begin{cases} V_{12} = 0 \\ V_{23} = H \cdot a_{23} + m \cdot b_{23} \end{cases} \quad (22)$$

The introduced simplifications in the steam circuit allow the number of differential equations to be reduced from five to the following three equations

$$\frac{dH_{34}}{dt} = \dot{Q}_3 - \dot{Q}_4 + V_{34} \frac{dp}{dt} - \dot{Q}_{34-56} \quad (23)$$

$$\frac{d(H_{12} + H_{23})}{dt} = \dot{Q}_1 - \dot{Q}_3 + (V_{12} + V_{23}) \frac{dp}{dt} - (\dot{Q}_{12-78} + \dot{Q}_{23-67}) \quad (24)$$

$$\frac{d(m_{12} + m_{23})}{dt} = \frac{1}{(h_3 - h_1)} \left( \frac{d(H_{12} + H_{23})}{dt} - (m_{12} + m_{23}) \frac{d(h_3 - h_1)}{dt} \right) \quad (25)$$

Equation (23) is formulated for the conservation of the energy in the draining volume. Equation (24) for the conservation of the energy in the desuperheating and condensing volumes is formulated for the total steam volume,  $V_{12}$  and  $V_{23}$  respectively. Equation (25) for the conservation of the mass

in the desuperheating and condensing volumes is formulated for the total mass,  $m_{12}$  and  $m_{23}$  respectively. The three equations of energy conservation in the feedwater circuit remain the same as in Sec. 3.1. This implies that the six-volume model uses three heat transfer coefficients as stated in the initial heater model, see Equations 9-11.

The numerical performance of the model was slightly improved so that it runs almost in real time when a single heater is simulated. However, the efficiency and numerical stability are still not sufficient to perform a simulation of an entire power plant including six feedwater heaters. On the other hand, in this model, the condensate outlet flow rate can not be controlled as it is in reality, since it is calculated in the model as an output variable while the steam inlet flow rate is given as an input variable. In turn, a model of the control system of the condensate level can not be applied here. The range of applications of such a model is therefore limited to steady-state conditions when the feedwater level is fixed around a specific operating level.

### 3.3. Four-volume heater model

The four-volume model of a heater involves further simplifications regarding the steam circuit model of the six-volume model (Fig. 2). In the four-volume model, the desuperheating zone is neglected thanks to the assumptions that the steam, after entering the heater cavity, immediately turns into the condensing phase. This assumption implies that the volume  $V_{12}$  is negligible so that the heat exchange area between the desuperheating and the corresponding feedwater section approaches zero. In turn, the heat transfer rate  $\dot{Q}_{12-78}$  between the volumes  $V_{12}$  and  $V_{78}$  also approaches zero (see Fig. 1). Nevertheless, the energy rate of the incoming steam is taken into account entirely by the energy balance of the condensing zone  $V_{23}$ . The assumption is valid when the mass of the superheated steam is relatively small in comparison with the entire mass of the steam in the heater cavity. Such conditions are true for typical power unit installations equipped with low- and high-pressure lines of feedwater heaters.

The model formulated in this section constitutes the form arising from the discussed assumptions. In the new four-volume model, that zone is replaced by the condensing zone of the same nomenclature of indexes. Equation (26), describing conservation of the energy in the condensing and subcooling volumes, is formulated separately for the steam volumes  $V_{12}$  and  $V_{23}$ , respectively. Equation (26), describing conservation of the mass in the condensing and subcooling volumes, is formulated separately for the mass of steam  $m_{12}$  and mass of water (condensate)  $m_{23}$ , respectively.

$$\begin{bmatrix} \frac{dm_{12}}{dt} \\ \frac{dH_{12}}{dt} \end{bmatrix} = \begin{bmatrix} \frac{\partial m_{12}}{\partial V_{12}} & \frac{\partial m_{12}}{\partial p_{12}} \\ \frac{\partial H_{12}}{\partial V_{12}} & \frac{\partial H_{12}}{\partial p_{12}} \end{bmatrix} \cdot \begin{bmatrix} \frac{dV_{12}}{dt} \\ \frac{dp_{12}}{dt} \end{bmatrix} = \begin{bmatrix} e_{11} & e_{12} \\ e_{21} & e_{22} \end{bmatrix} \cdot \begin{bmatrix} \frac{dV_{12}}{dt} \\ \frac{dp_{12}}{dt} \end{bmatrix} \quad (26)$$

The volume  $V_{12}$  and the pressure  $p_{12}$  of the condensate inside the steam cavity were selected as the state variables and are related to the mass and the energy flow rates via the matrix of partial derivatives. Equation (26) can be rearranged in the following form

$$\begin{aligned} e_{11} \frac{dV_{12}}{dt} + e_{12} \frac{dp_{12}}{dt} &= \dot{m}_1 - \dot{m}_2 \\ e_{21} \frac{dV_{12}}{dt} + e_{22} \frac{dp_{12}}{dt} &= \dot{Q}_1 - \dot{Q}_2 - \dot{Q}_{12-56} - \dot{Q}_{23-45} \end{aligned} \quad (27)$$

where particular elements of the partial derivative matrix, including the assumption that  $\frac{\partial \rho_{12}}{\partial V_{12}} \equiv 0$ ,

$$\frac{\partial V_{12}}{\partial p_{12}} \equiv 0, \text{ yield}$$

$$\begin{aligned} e_{11} &= \frac{\partial m_{12}}{\partial V_{12}} = \frac{\partial(\rho_{12} \cdot V_{12})}{\partial V_{12}} = \rho_{12} + \frac{\partial \rho_{12}}{\partial V_{12}} V_{12} = \rho_{12} \\ e_{12} &= \frac{\partial m_{12}}{\partial p_{12}} = \frac{\partial \rho_{12}}{\partial p_{12}} \cdot V_{12} + \frac{\partial V_{12}}{\partial p_{12}} \cdot \rho_{12} = \frac{\partial \rho_{12}}{\partial p_{12}} \cdot V_{12} \\ e_{21} &= \frac{\partial H_{12}}{\partial V_{12}} = \frac{\partial(\rho_{12} V_{12} h_{12})}{\partial V_{12}} = \rho_{12} h_{12} + \frac{\partial \rho_{12}}{\partial V_{12}} h_{12} V_{12} + \frac{\partial h_{12}}{\partial V_{12}} \rho_{12} V_{12} = \rho_{12} h_{12} - p_{12} \\ e_{22} &= \frac{\partial H_{12}}{\partial p_{12}} = \frac{\partial(\rho_{12} V_{12} h_{12})}{\partial p_{12}} = \frac{\partial \rho_{12}}{\partial p_{12}} V_{12} h_{12} + \frac{\partial V_{12}}{\partial p_{12}} \rho_{12} h_{12} + \frac{\partial h_{12}}{\partial p_{12}} \rho_{12} V_{12} - V_{12} = \\ &= V_{12} \left( h_{12} \frac{\partial \rho_{12}}{\partial p_{12}} + \rho_{12} \frac{\partial h_{12}}{\partial p_{12}} \right) - V_{12} \end{aligned} \quad (28)$$

and, additionally,

$$V_{12} = V_{total} - V_{23} \text{ then } dV_{12} = -dV_{23} \quad (29)$$

The mass of the water in the condensate cavity is determined from the assumption written as follows

$$\text{if } \frac{dm_{12}}{dt} = \dot{m}_1 - \dot{m}_2, \text{ and } \frac{dm_{23}}{dt} = \dot{m}_2 - \dot{m}_3, \text{ then } \frac{dm_{23}}{dt} = \dot{m}_1 - \dot{m}_3 - \frac{dm_{12}}{dt} \quad (30)$$

Variables obtained from (30) are substituted to Equations (27)

$$\begin{aligned} \dot{m}_3 - \dot{m}_2 &= e_{11} \frac{dV_{12}}{dt} + e_{12} \frac{dp_{12}}{dt} \\ \dot{m}_3 h_3 - \dot{m}_2 h_2 - \dot{Q}_{12-m} - \dot{Q}_{23-m} &= e_{21} \frac{dV_{12}}{dt} + e_{22} \frac{dp_{12}}{dt} \end{aligned} \quad (31)$$

Unknowns are determined as follows

$$\begin{aligned} dp_{12} &= \frac{m_3(h_3 - h_2) - (e_{21} - e_{11}h_2) \frac{dV_{12}}{dt} - \dot{Q}_{12-m} - \dot{Q}_{23-m}}{(e_{22} - e_{12}h_2)} \\ \dot{m}_2 &= \frac{(e_{12}e_{21} - e_{11}e_{22}) \frac{dV_{12}}{dt} + e_{12}(\dot{Q}_{12-m} - \dot{Q}_{23-m}) - (e_{12}h_3 - e_{22})\dot{m}_3}{(e_{22} - e_{12}h_2)} \end{aligned} \quad (32)$$

Equations (33-34) are formulated for the conservation of the energy in the draining volumes, assuming uniform density of the feedwater.

$$\frac{dH_{45}}{dt} = \dot{Q}_4 - \dot{Q}_5 + \dot{Q}_{23-45} \quad (33)$$

$$\frac{dH_{56}}{dt} = \dot{Q}_5 - \dot{Q}_6 + \dot{Q}_{12-56} \quad (34)$$

The level of the condensate inside the heater is calculated using the following formula

$$x = \frac{V_{23} - V_{230}}{A_{con}}, \quad (35)$$

where  $A_{con}$  is the area of the condensate surface in a heater cavity and  $V_{230}$  is the nominal height of the condensate volume. The heat transfer from the heater cavity to the metal of the heater was additionally taken into account since the numerical performance of the model was significantly improved. The heat transfer is formulated using conservation of energy as follows

$$\frac{dH_m}{dt} = \dot{Q}_{12-m} + \dot{Q}_{23-m} \quad (36)$$

where

$$H_m = m_m \cdot c_{pm} \cdot T_m \quad (37)$$

A model of the controller uses the feedback from the condensate level sensor to control the opening of the condensate outlet drain. A controller uses a group of gain controls, i.e. proportional (P), integral (I), and derivative (D).

### 3.4. Exemplary implementation of a Feedwater Heater Model in Simulink

Equations of the feedwater model were implemented in a convention required by Simulink [25] and discussed in the previous section. Equations have the form of algebraic expressions grouped into blocks rather than a network of connections among elementary blocks, which allows the avoidance of “spaghetti code” caused by a large number of elementary blocks (e.g. gains, adders) and mutual links. The underlying idea is to facilitate conversion of the Simulink code to other modeling languages, e.g. Modelica [26]. The model requires conditional operators which were also coded in m-files for transparency of the model topography. Look-up tables with derivatives of steam properties were smoothed by means of interpolation to remove discontinuities or thresholds. Blocks were masked so that only selected parameters are transferred to these blocks as Matlab data structures. The topography of the feedwater model implementing Equation 32 is depicted in Fig. 3.

## 4. Data-driven tuning of the four-volume heater model

### 4.1. Results of Benchmark of Heater Models

The benchmark of heater models is presented in Table 1, where three versions of the heater model were compared regarding simulation time. Most of geometrical and physical parameters of heater models were extracted from the operational documentation. The remained parameters (heat exchange coefficients) were roughly adjusted using trial-and-error approach. The performances were compared using a data set with transient operation of the power unit in the range of 140-225MW (see Fig. 4). The models were run and tested on a PC with an Intel Pentium 2.8GHz CPU and 4 GB RAM under Microsoft Windows XP Professional x64 Edition. Matlab version 7.2 (R2006a) was used.

The simulation speed was rated versus available real time and expressed in a percentage scale (Table 1). The achieved performance for the four-volume model is enough to repeat the simulation a few times in a loop, as is required by an optimization routine adjusting model parameters, and still to be within real time for an installation consisting of seven heaters. Data-driven tuning of this model and its detail parameters are discussed further in the next sub-section.

### 4.2. Adjustment of Model Parameters Based on Operational Data

A physical model can conveniently be represented as a set of non-linear state-space equations formulated in the continuous-time domain. The objective of the estimation is to minimize the error function between the measurement signals and model responses by means of an iterative numerical technique constituting “the non-linear least-squares problem” [27]. The function describing the error has to be a positive and decreasing function. The procedure of model tuning consists of two in-a-loop phases: (i) simulation of a model by solving differential equations numerically in Simulink [25], and (ii) numerical minimization in the parameter space with respect to an error-related criterion function using Matlab Optimization Toolbox [25]. The interested reader may find more information concerning the available methods and algorithms that support identification of first-principle models in [27-28]. The simulation and optimization settings used in the parameter adjustment process are presented in Table 3. The Newton-Gauss method, `lsqnonlin(.)` routine implemented in the Optimization Toolbox of Matlab [25], was used to minimize the function describing the error in the measurement signals and model responses.

Geometrical and physical parameters of the heater model were extracted from the operational documentation and were assumed to be known. A high-pressure heater denoted as XW1 was used as a reference system characterized by the operational and constructional data presented in Table 2. The heater is part of a feedwater regeneration circuit in which feed pumps pass the condensed steam (feedwater) from a condenser through heater banks, heated by the steam extracted from the high, intermediate and low-pressure sections of a steam turbine. The condensate is pumped to the deaerator, through the bank of low-pressure heaters XN1-5, and further, from the deaerator to the steam generator (boiler) through the bank of high-pressure heaters XW1-3. The draining system of the feedwater heater consists of a drain removal path from each heater. The normal drain flow path is cascaded to the next lower stage heater and the alternate path is diverted to the condenser. The heaters XN1 and XN2, assembled in the condensers, are in continuous operation with the condensers. The simulation model considered in this section consists of a heater model and a model equivalent to a control system installed in a power plant. The control system could not be directly reconstructed in the simulation, due to its complexity and limited relevance to the functionality required in the model (e.g. trip logic). Hence, the module maintaining a constant level of the condensate inside the heater was simplified using a PID controller model.

Two heat transfer coefficients were identified and the two selected model responses are presented graphically in Fig. 4. The model reproduced the trend in the condensate and the feedwater



temperatures with good accuracy. Convergence trajectory plots (not presented here) show a stable trend towards constant values of the parameters, which correspond to a convergence towards the minimum of the criterion function, within less than 6 iterations.

ACCEPTED MANUSCRIPT

## 5. Performance indicators for a feedwater heater

Two types of indicators, efficiency and technical ones, have been proposed in order to assess the technical state of a heater. Such indicators take the form of a scalar value (e.g. amount of transferred energy) or a characteristic (e.g. power rate vs. amount of transferred energy) and allow a pattern of values corresponding to different regimes of operation (e.g. low vs. high power rate) to be defined. Bounds imposed on the pattern of normal operation of a heater define the tolerance range beyond which the performance is unacceptable.

The diagnostic process proposed in this paper consists of two stages (i) fault detection, and (ii) fault recognition. Firstly, symptoms of a malfunction are detected based on variation of an efficiency indicator, i.e. by detecting the efficiency indicator crossing tolerance bounds. Secondly, a technical indicator enables a problem to be addressed more precisely. The methodology proposed herein does not eliminate the need for specialists and experts to contribute to the fault recognition process, as their role is to interpret trends in indicators. The method is an extension of available symptom indicators to provide new early warning indicators of physical meaning.

Execution of the procedure for numerical adjustment of the model parameters allowed values of these parameters that assure the heater model that best fits the data to be found. The model was tested on the same PC configurations as presented in the previous section. Parameters of the feedwater heater model were updated according to the flowchart presented in Fig. 5 for low- and high-pressure heaters, designated as XN4 and XW1 respectively. The results for models of both heaters are qualitatively the same, so only results for XW1 will be presented in this section. The Newton-Gauss method, `lsqnonlin(.)` routine implemented in the Optimization Toolbox of Matlab, used to update model parameters is efficient enough, as shown in Fig. 6, to follow the operational data in the real-time mode. Values of updated parameters are used as initial guess conditions in an algorithm adjusting model parameters for the next data sequence. As a result, the minimization algorithm has a better starting point and so a smaller number of iterations is required in each sequence. The value of an objective function error and the number of iterations are used as stopping criteria for the parameter updating process.

Fig. 7 presents an example of an efficiency indicator based on an operational curve, i.e. a relation of the electrical power rate to corresponding overall energy transfer rate from the steam to the feedwater (see the left plot in Fig. 7). The overall energy transfer rate can be split into the steam-to-feedwater and the condensate-to-feedwater transfer, for the upper and lower volume of a heater respectively (see the middle and right plots in Fig. 7). These two energy transfer rates are governed by respective heat transfer coefficients present in the heater model. As shown in Fig. 7, data points approximately lay along a line (solid line) and are bounded by 95% confidence intervals (dashed lines). An example of technical indicators is presented in Fig. 8 where the hysteresis phenomenon in the energy transfer in the steam-to-feedwater and the energy transfer in the condensate-to-feedwater, corresponding to two heat transfer coefficients used in the heater model, is shown. The size of a hysteresis loop, among others, is an indication of heat accumulation in the heater jacket.

Another option for constructing a diagnostic tool is to investigate trends in heat transfer parameters versus operational time. Values of these parameters vary depending on the operating point of the power unit, however, and thus yet another possibility of constructing a technical indicator is considered, namely the relationship between the value of the heat transfer coefficient and the temperature of the feedwater leaving the heater (Fig. 9).

## 6. Summary and conclusions

This paper focuses on the tuning and validation process of the first-principle feedwater heater model intended for model-based diagnostics as part of the entire model of a power unit. Moreover, the paper proposes key performance indicators which reflect operational changes in the process of heating the feedwater versus assumed statistical bounds. The summary addresses the objectives stated in the abstract of the paper and provides further development perspectives. The scope of the proposed methodology is limited to power plant systems with modern data acquisition systems. Such systems should be capable of gathering required input-output data with a sampling frequency to capture relevant heat transfer and fluid flow dynamics. This paper presents a representative case study where data are gathered with a sampling frequency of 10 seconds. This resolution is sufficient when compared to the normal operation of a power plant. On the other hand, the application scope is limited by assumptions of the model listed in Sections 3.1-3.3.

The first objective of this work was to formulate and deploy a moderately complex first-principle model of a feedwater heater to reproduce operational measurements in real-time simulations. The development process of such a model is presented in three steps through Sections 3.1-3.3. In the first step, the initial advanced six-volume model was used as the starting point for the simplification process. Such a model allows the thermodynamics of the heat exchange process to be correctly captured, however, the model complexity does not enable model parameters to be adjusted in the on-line mode. In the second step, the six-volume model was linearized with respect to the steam properties to reduce numerical complexity, yet at the same time not dramatically sacrificing accuracy. The goal behind model simplification was to develop a model capable of achieving performance that provides enough time capacity to allow self-adjustment of the model to operational data. The four-volume model was proposed in the last step. The proposed simplification has provided the greatest improvement towards numerical stability of a power unit model and significantly shorter computation time (Table 1). The advantage of the four-volume model is that the model is characterized by only two adjustable coefficients instead of three for the six-volume model. The first one describes the thermal energy transfer (conduction, convection, and radiation) between the condensate and the feedwater, while the second one describes the thermal energy transfer between the steam (the mixture of superheated and wet steam) and the feedwater. Taking advantage of the better numerical performance of the four-volume model, heat accumulation in the heater jacket was implemented to allow simulation of a start-up operation. The model was deployed within the Virtual Power Plant environment [29-30] using Simulink software.

The second objective was to develop a tuning method for the moderately complex first-principle model. Such a method is advocated for industrial conditions when the values of physical and geometrical parameters are known, while the values of phenomenological ones have to be adjusted as only their rough pre-calculated initial values are available. Measurement data from a 225MW coal-fired unit were used to validate the model's accuracy. The validation process presented in the paper indicates that the performance in steady and transient conditions is good, achieving a correlation between the simulations and measurements at a level of 60-90%. This proves that the model can be used in further studies and the development of techniques related to model-driven diagnostics.

To complete the third objective of this paper, the efficiency and technical performance indicators were formulated using a statistical approach to facilitate the recognition of specific patterns in data. Pattern-based analysis was proposed as the most suitable form of analysis because of the availability of a large amount of operational data. Pattern analysis allows a few scenarios, represented by different patterns which correspond to the sequential operation of power units, to be created. A power unit can be in a few operational states corresponding to its rotational speed expressed in rpm; these states usually are: idle ( $\text{rpm} = 0$ ), turning gear ( $0 < \text{rpm} < 200$ ), transient ( $201 < \text{rpm} < 2990$ ) and synchronized ( $2991 < \text{rpm} < 3010$ ). Sequential operation of a power unit enables two groups of patterns, belonging to transient and steady operation, to be obtained. Typically, the indicators

(measures) introduced in this paper reflect non-linear relationships and are therefore represented by first- or second-order trend curves.

The fourth objective was to automate the calculation process of the indicators. In this respect, a parametric representation of the performance indicators was proposed to allow boundary conditions to be easily imposed. These boundaries can be automatically detected and, as such, are able to be utilized in an early warning malfunction notification function. Moreover, such parametric representation facilitates the quantification of the uncertainty of the diagnosis. There are numerous statistical methods supporting the decision-making process which are based on sets of uncertain and inconsistent data [4]. Such methods should be considered to reject false alarms.

Future investigations are planned to focus on the repeatability and reproducibility of the system identification results separately, based on a number of data sets measured in similar operational conditions. Repeatability and reproducibility indicators are important from a diagnostic point of view since these indicators directly yield confidence intervals for adjusted parameters and confirm, statistically, the correctness of the proposed approach.

## References

- [1] L. Drbal, K. Westra, P. Boston, *Power Plant Engineering*, Springer 1995.
- [2] D. Flynn (eds.), *Thermal Power Plants - Simulation and Control*, The Institution of Electrical Engineers, London 2000.
- [3] M.A. Antar, and S.M. Zubair, The Impact of Fouling on Performance Evaluation of Multi-Zone Feedwater Heaters, *Applied Thermal Engineering*, 27 (2007) 2505-2513.
- [4] L. Ma , Y. Ma1 and J. Ma1, Fault Diagnosis for the Feedwater Heater System of a 300MW Coal-Fired Power Generating Unit Based on RBF Neural Network, *Lecture Notes in Computer Science*, Springer Berlin / Heidelberg 3930 (2006) 832-841.
- [5] J.H. Lienhard IV, J.H. Lienhard V, *A heat transfer textbook*, Phlogiston Press, 2004, US.
- [6] R.K. Shah., D.P. Sekulic, *Fundamentals of Heat Exchanger Design*, John Wiley and Sons 2003.
- [7] T. Kuppan, *Heat Exchanger Design Handbook*, Mechanical Engineering, Marcell Dekker Inc 2000.
- [8] K. Bamardouf, D.A. McNeil, Experimental and numerical investigation of two-phase pressure drop in vertical cross-flow over a horizontal tube bundle, *Applied Thermal Engineering*, 29 (2009) 1356–1365.
- [9] H. van Putten , P. Colonna, Dynamic Modeling of Steam Power Cycles. Part I – Simulation of a Small Simple Rankine Cycle System, *Applied Thermal Engineering* 27 (2007) 467-480.
- [10] J. Dipama, A. Teysseidou, F. Aubé, L. Lizon-A-Lugrin, A grid based multi-objective evolutionary algorithm for the optimization of power plants, *Applied Thermal Engineering* 30 (2010) 807–816.
- [11] G.F. Naterer, C.H. Lam, Transient Response of Two-Phase Heat Exchanger With Varying Convection Coefficients, *Journal of Heat Transfer*, 128 (2006) 953-962.
- [12] K. Henrik, A. Olsson, *Model Order Reduction of a Heat Exchanger Model*, Numerical Analysis and Computer Science (NADA) Royal Institute of Technology (KTH), Stockholm, Sweden 2005.
- [13] Y.G. Yohanis, O.S. Popel, SE. Frid, A Simplified Method of Calculating Heat Flow Through a Two-Phase Heat Exchanger, *Applied Thermal Engineering*, 25 (2005) 2321–2329.
- [14] A. Alessandri, T. Parisini, R. Zoppoli, Sliding–window neural state estimation in a power plant heater line, *International Journal Of Adaptive Control and Signal Processing*, 15 (8) (2001) 815:836

- [15] K.Y. Kim Y.J. Lee, Fault Detection and Diagnosis of the Deaerator Level Control System in Nuclear Power Plants, *Journal Of the Korean Nuclear Society*, 36 (1) (2004) 73-82, available: [article.nuclear.or.kr/jknsfile/v36/JK0360073.pdf](http://article.nuclear.or.kr/jknsfile/v36/JK0360073.pdf)
- [16] C. Bonivento, P. Castaldi, D. Mirotta, Predictive Control vs PID Control of an Industrial Heat Exchanger, *The 9th Mediterranean Conference on Control and Automation*, Dubrovnik, Croatia 2001, June 27-29.
- [17] J. Szargut, Influence of Regenerative Feed Water Heaters on the Operational Costs of Steam Power Plants and HP Plants, *Int. J. of Thermodynamics*, 8 (3) (2005) 137-141.
- [18] K. Roth, V. Scherer, K. Behnke, Enhancing the dynamic performance of electricity production in steam power plants by the integration of transient waste heat sources into the feed-water pre-heating system". *Int. J. Energy Technology and Policy*, 3 (1/2) (2005) 50-56.
- [19] S. Farhad, M. Saffar-Avval, M. Younessi-Sinaki, Efficient design of feedwater heaters network in steam power plants using pinch technology and energy analysis, *International Journal of Energy Research*, 32 (1) (2007) 1 – 11.
- [20] C.F. Andreone, S. Yokell, *Tubular Heat Exchanger Inspection, Maintenance, and Repair*, McGraw Hill 1997.
- [21] D.W. Hiltbrand, G.W. Choe, Heater drains system redesign at Comansche Peak using the RETRAN-03 transient analysis computer code, *Heat Transfer, Proceeding of 11th IHTC*, vol. 6, Kyongju, Korea 1998.
- [22] C. Thanomsata, C.W. Taftb and A.M. Annaswamy, Level control in feedwater heater systems using nonlinear strategies, *ISA Transactions* 37 (4) (1998) 299-312.
- [23] G.V. Murphy, J.M. Bailey, LQG/LTR control system design for a low-pressure feedwater heater train with time delay, *Industrial Electronics Society, 1990. IECON '90., 16th Annual Conference of IEEE*.
- [24] M. Holmgren, *X Steam for Matlab*, 2006, available: [www.x-eng.com](http://www.x-eng.com).
- [25] MATLAB-SIMULINK documentation, Mathworks Inc., 2005. Available: [www.mathworks.com](http://www.mathworks.com)
- [26] Åkesson J. *Languages and Tools for Optimization of Large-Scale Systems*, PhD Thesis, Lund University, 2007. available: [www.control.lth.se/documents/2007/ake07dis.pdf](http://www.control.lth.se/documents/2007/ake07dis.pdf)
- [27] Ljung L. *System identification - Theory for the User*, Prentice-Hall, USA, 1999.
- [28] Bohlin T. *Practical grey-box process identification, Theory and Applications (Advances in Industrial Control)*, Springer-Verlag, London, 2006.
- [29] T. Barszcz, Virtual Power Plant in Condition Monitoring of Power Generation Unit, *Proceedings of the 20th International Congress on Condition Monitoring and Diagnostic Engineering Management*, Faro, Portugal 2007, June 13–15.

- [30] T. Barszcz, P. Czop, Methodologies and Applications of Virtual Power Plant - New Environment for Power Plant Elements Modeling, Institute of Sustainable Technologies, Radom 2007.

ACCEPTED MANUSCRIPT

## Figures

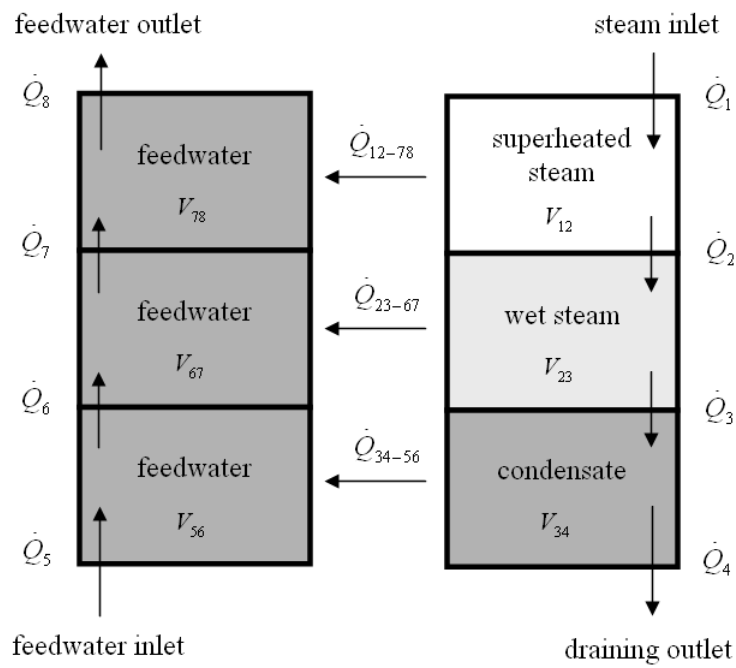


Fig. 1. Schematic representation of heat energy transfer in the six-volume heater model.



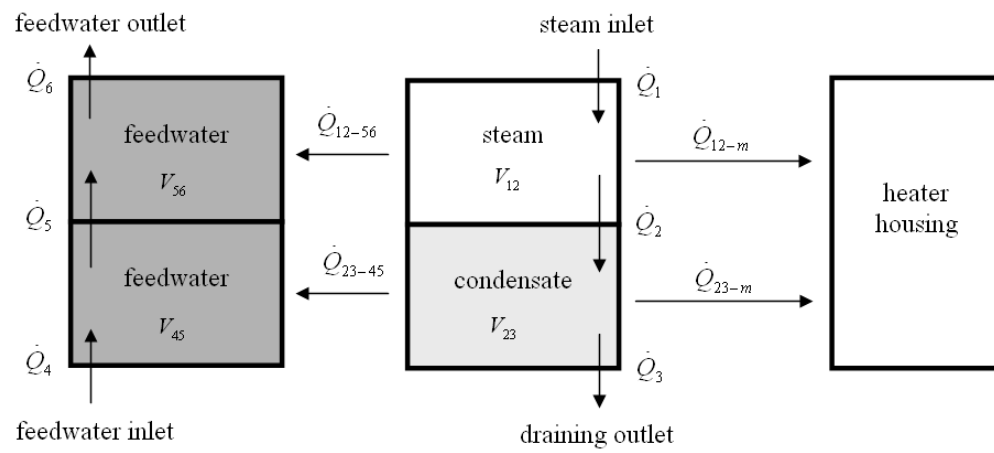


Fig. 2. Schematic representation of heat energy transfer in the four-volume heater model.

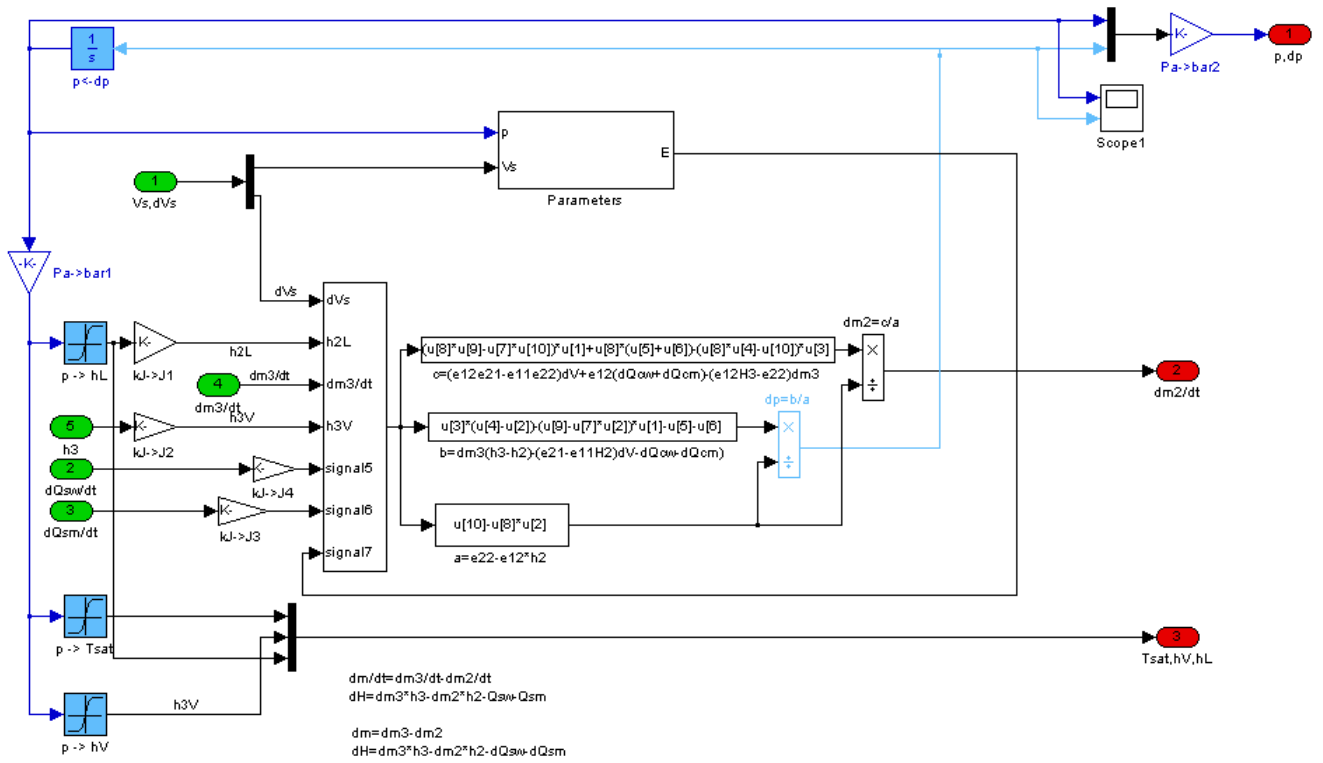


Fig.3. Topography of a model block implementing Equation 32.

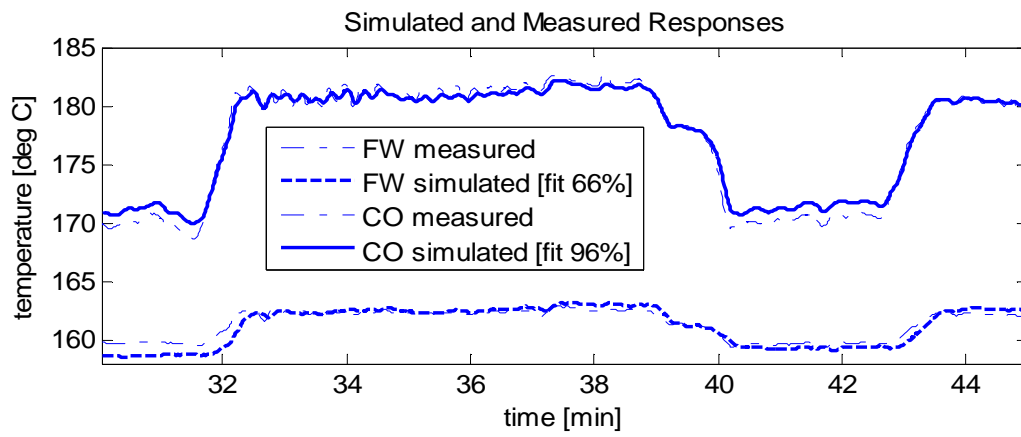


Fig. 4. Exemplary model response (FW – feedwater, CO – condensate).

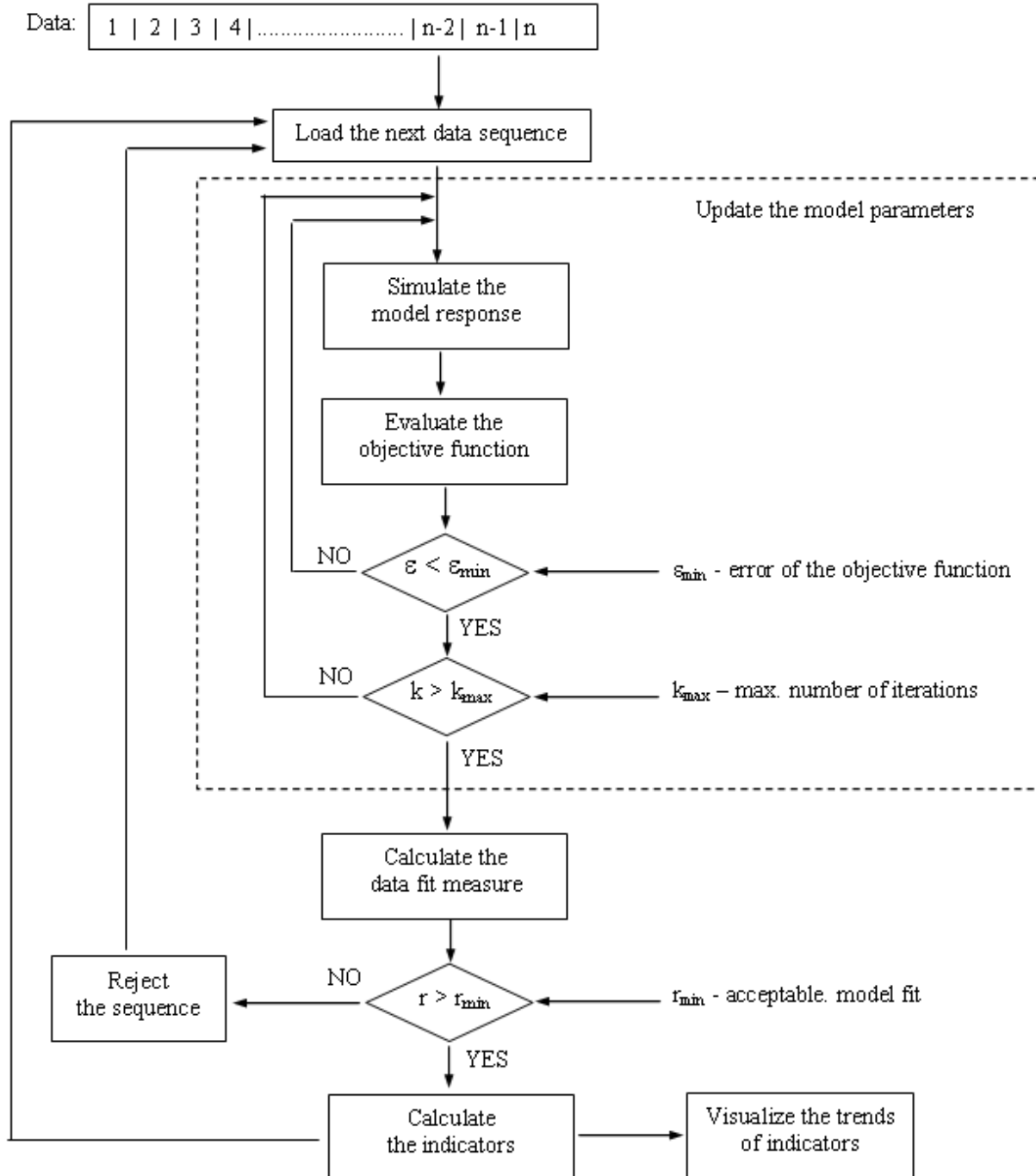


Fig. 5. Procedure of updating model parameters

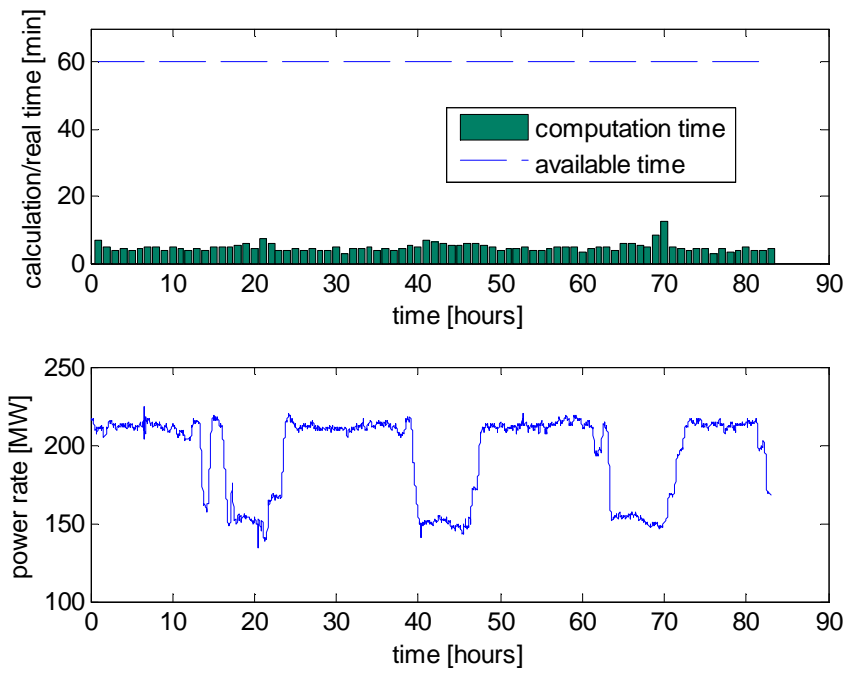


Fig. 6. Actual time vs. available real time for computations of a single heater

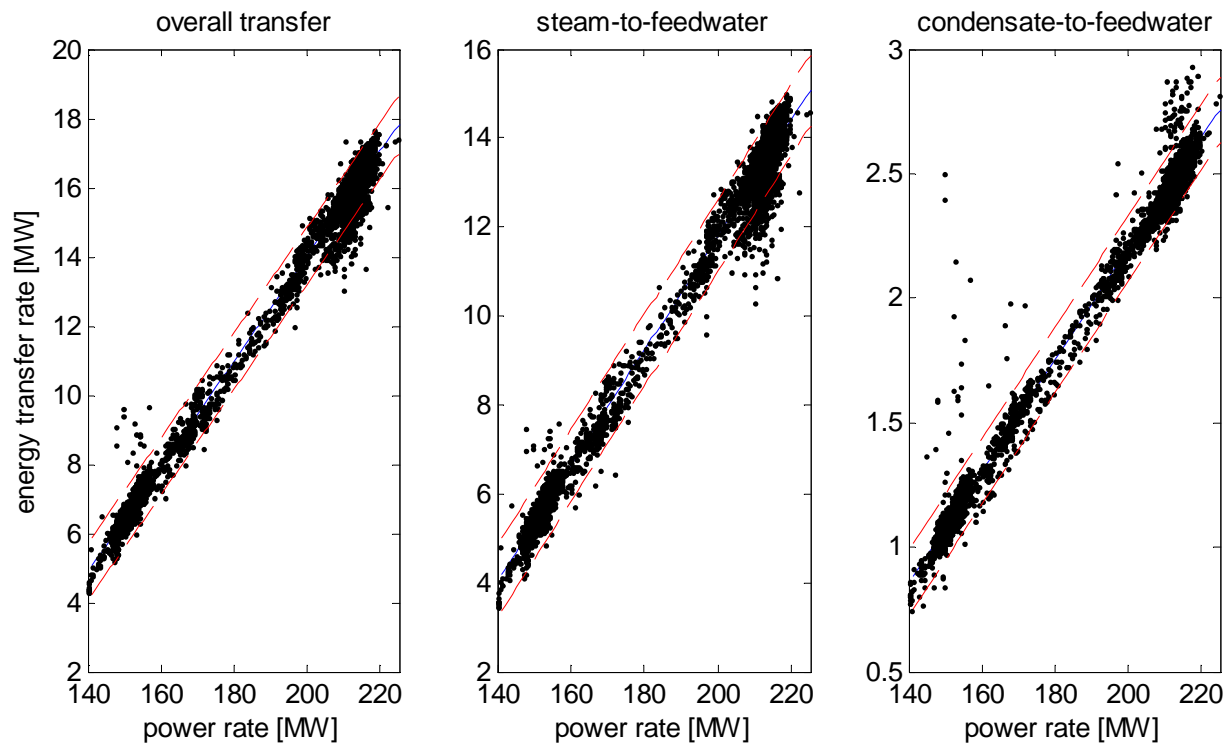


Fig. 7. An example of reconstructed steam-to-feedwater heat transfer rate as a function of measured electrical power rate (XW1 heater).

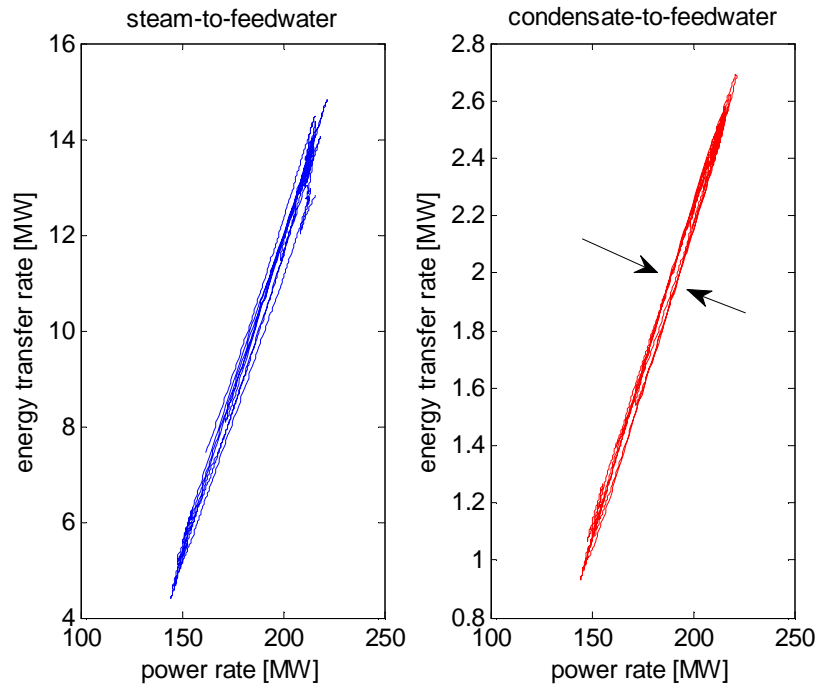


Fig.8. An example of a technical indicator enabling evaluation of the size of the hysteresis of energy transfer

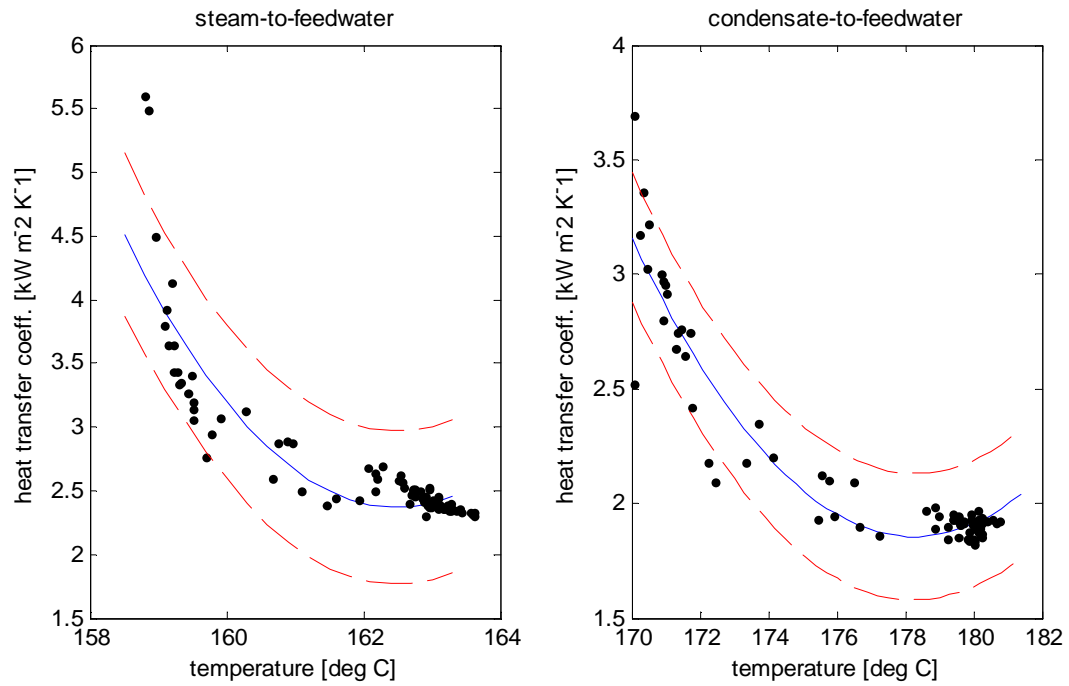


Fig. 9. Heat transfer coefficients vs. temperature of the feedwater leaving the heater.



## Tables

Table 1. Benchmark of developed heater models

	Initial six-volume model [simulation/real time]	Linearized six-volume model [simulation/real time]	Four-volume model [simulation/real time]
1 decoupled heater	123%	34%	3%
7 heaters in a series	760%	267%	22%

Table 2. Parameters of the high-pressure heater XW1 used in simulation.

Type of parameter	Parameter	Symbol	Unit	Value
Geometrical	Heat Exchange area – steam	$A_{12}$	$[\text{m}^2]$	$f_A(V_{12})$
	Heat Exchange area – condensate	$A_{23}$	$[\text{m}^2]$	$A_{tot} - f_A(V_{12})$
	Overall heat exchange area	$A_{tot}$	$[\text{m}^2]$	600
	Steam and condensate volume	$V_{12} + V_{23}$	$[\text{m}^3]$	2.9
	Feedwater volume	$V_{45} + V_{56}$	$[\text{m}^3]$	4
	Nominal (reference) height of condensate level	$x_0$	$[\text{m}]$	3.275
	Heater height	$x$	$[\text{m}]$	10
Physical	Mass of the metal of a heater	$m_m$	$[\text{kg}]$	35500
	Specific heat of a metal	$c_{pm}$	$[\text{J}/\text{kg}\cdot\text{K}]$	$500 \cdot 10^{-3}$
Phenomenological	Heat transfer coefficient steam to metal	$k_{12-m}$	$[\text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}]$	1.5
	Heat transfer coefficient condensate to metal	$k_{23-m}$	$[\text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}]$	0.6
PID-settings	Proportional	P	$[-]$	0.8
	Integration	I	$[\text{s}]$	53
	Derivative	D	$[\text{s}^{-1}]$	0

Table 3. Simulation and optimization settings

Simulation		Optimization (minimization)	
Option	Value	Option	Value
Solver	ode23tb (stiff/TR-BDF2)	Gradient type	basic
Max step size	auto	Algorithm	lsqnonlin
Min step size	auto	Cost type	SSE
Zero crossing control	disable all	DiffMaxChange	0.1
Relative tolerance	auto	DiffMinChange	1E-08
Absolute tolerance	auto	Large scale	true
		MaxIter	28
		RobustCost	False
		TolCon	1E-6
		TolFun	1E-6

List of captions of figures

Fig. 1. Schematic representation of heat energy transfer in the six-volume heater model.

Fig. 2. Schematic representation of heat energy transfer in the four-volume heater model.

Fig. 3. Topography of a model block implementing Equation 32.

Fig. 4. Exemplary model response (FW – feedwater, CO – condensate).

Fig. 5. Procedure of updating model parameters.

Fig. 6. Actual time vs. available real time for computations of a single heater.

Fig. 7. An example of reconstructed steam-to-feedwater heat transfer rate as a function of measured electrical power rate (XW1 heater).

Fig.8. An example of a technical indicator enabling evaluation of the size of the hysteresis of energy transfer.

Fig. 9. Heat transfer coefficients vs. temperature of the feedwater leaving the heater.

List of Captions of Tables

Table 1. Benchmark of developed heater models.

Table 2. Parameters of the high-pressure heater XW1 used in simulation.

Table 3. Simulation and optimization settings.

ACCEPTED MANUSCRIPT