DEMONSTRATION OF A GREY-BOX APPROACH TOWARDS THE DIAGNOSTICS OF A FEEDWATER HEATER (PART I) – MODEL DEVELOPMENT

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Summary

Work related to the first-principle modeling of a boiler feedwater heater operating in a 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 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. The first objective has been addressed in this paper while the remaining objectives are dealt with in the second paper. In the first part of 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 approximate to the real-time was achieved on a regular PC workstation for a series of six low- and high-pressure heaters.

Key words: power plant, feedwater heater, modeling, system identification.

Streszczenie

Artykuł przedstawia proces modelowania podgrzewacza regeneracyjnego pracującego w systemie bloku energetycznego z wykorzystaniem równań fizycznych. Artykuł zawiera dyskusje dotycząca uproszeń struktury modelu, estymacji jego parametrów oraz walidacji. Celami pracy jest: (i) sformułowanie umiarkowanie złożonego modelu wymiennika odtwarzającego dane pomiarowe w rzeczywistej skali czasu, (ii) przedstawienie metody strojenia modelu, (iii) zaproponowanie wskaźników użyteczności podgrzewacza na podstawie podejścia wspartego modelem, oraz (iv) automatyzacja procesu wyznaczania tych wskaźników. Pierwszy z celów zawiera się w tej części pracy, a pozostałe w części drugiej. Pierwsza cześć pracy opisuje proces tworzenia modelu z uwzględnieniem koniecznych uproszczeń w celu podniesienia jego użyteczności oraz funkcjonalności. W wyniku zaproponowanych uproszczeń, uzyskano użyteczność zbliżoną do czasu rzeczywistego na stacji roboczej klasy PC, dla modelu obejmującego zespół sześciu wymienników nisko i wysokociśnieniowych.

Słowa kluczowe: elektrownia, podgrzewacz regeneracyjny, modelowanie, identyfikacja systemów.

INTRODUCTION

Virtual simulation plays an important role in reducing the time, cost and technical risk of developing new power plant systems and installations [1-2]. In 2005, the national research project “DIADYN”, founded from European Union structural funds, was initiated by a consortium of Polish technical universities and research institutes. The project name stands for “Integrated dynamic systems of risk assessment, diagnostics and control of structures and technological process” involving 40 research teams in its realization. Within the project, the Virtual Power Plant (VPP) modeling environment was proposed in [3] as an innovative approach towards reconstructing operation of a power plant unit based on a model and recorded process data. The VPP, described in detail in [4], provides a framework for integrating the range of models, data management system and visualization methods into a single simulation environment. The VPP consists of a number of computers and software modules, interconnected by a fast computer network that is designed to allow plug-and-play functionality. The VPP was included in the project as a practical laboratory, which facilitates validation of diagnostic methodology implemented as software algorithms or available in the form of hardware solutions, like a controller with embedded fault detection and isolation functionality. The novelty of the proposed simulation environment is strong support for model-based diagnostics, which is one of the most potential and fast developing technologies. A comprehensive survey presenting model-based theory and practice can be found in [5]. Despite numerous analytical and experimental studies, the feasibility of the model-based approach is limited when it is applied to large, industrial installations, such as power plant units. Key problems in practical implementations are twofold. The first group of problems are connected to the development of the model; in most cases, even if the underlying physical equations are known, the most important obstacle is to obtain the correct parameters, and thus – correct model behavior; on the other hand, when the “black-box” system identification approach is chosen – it is extremely important to acquire sufficient data, covering the operating range of the object. The second group is the lack of a flexible work environment; the process of model development and next – diagnostics requires the efficient cooperation of specialists from different fields: power plant staff should deliver data and technical documentation, diagnostic experts are responsible for modeling the process and drawing conclusions, results should be presented in a comprehensible way for the power plant experts and management; in practice, those tasks are executed with a set of heterogeneous tools, making the whole process hard to manage and inefficient. Therefore, the VPP environment is tailored for proper emulation of power unit functionality including: (i) a flexible structure enabling multiple configurations, (ii) the ability to import real data acquired at the object, (iii) the possibility to store models of a single component in different versions, (iv) the ability to achieve performance close to real time if model complexity is moderate, (v) the ability to present results in advanced and simplified forms, for experts and operational staff of power plants, respectively.

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

1. FIRST-PRINCIPLE BUT DATA-DRIVEN MODEL OF A FEEDWATER HEATER

Investigation of power plant dynamics requires detailed models comprising sub-models representing particular components of a plant. These models are based on first-principle equations (e.g. mass, momentum and energy balance) that involve phenomenological correlations, like heat transfer coefficients. Such models are commonly utilized to gain an understanding of physical processes and also in process efficiency optimization. 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. Geometrical parameters are deduced from construction or operational documentation. Nevertheless, models with deduced parameters are always biased, to some degree, by imprecision caused by the fact that a lumped-geometry model is used instead of a distributed-geometry one. The level of inaccuracy that is acceptable depends on the modeling purpose, available geometrical data and user preferences. Physical parameters can also be defined based on available documentation and, similarly to geometrical parameters, are also prone to the same error type during aggregation of a distributed-parameter representation into a lumped-parameter representation, e.g. a spatially distributed mass of a heater construction. The third category, the phenomenological parameters, describe physical processes, such as transfer or loss of energy, and are typically functions of functions of other sub-parameters, such as type of heat conduction surface, type of fluid, its density and velocity of the fluid flow. First-principle modeling is an excellent tool for understanding the physical phenomenon, however, it is insufficient for the fault detection purpose [5]. The reason for this is the lack of any formal approach allowing the model parameters to be updated according to the operational data. The authors noticed those limitations during the implementation of a large-scale model of a power unit [4]. First-principle models are frequently adjusted by trial-and-error, which can lead to non-optimal results. In order to avoid deficiencies of the trial-and-error approach, a formalized mathematical method using optimization techniques to minimize the error criterion, and find optimal values of tunable model parameters of a heater model, was developed and is described in the second part of this work. 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. To this end, the Fault Detection and Isolation (FDI) approach, based on the first-principle models, enables physics-based residua, which have direct interpretation like trends in values of the heat transfer coefficients, to be generated

2. LITERATURE SURVEY

Heaters are key elements of the feedwater regeneration process, which is essential for the efficiency of a power plant. Heaters are shell-and-tube type recuperators installed in a power station plant between the condenser and the boiler, and serve to preheat the feedwater in a steam-water circuit by transferring energy from the steam to the feedwater [6-7]. The steam flows from the low- and high-pressure sections of the turbine and is directed into the heaters, where condensation occurs on the tubes and the heat is transferred to the feedwater [8]. 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 [9], and [10]. In turn, extended taxonomy of heat exchangers and a description of the heat exchanger design process, along with related engineering and constructional details is given by Shah and Sekulic [11] and Kuppan [12]. The role in power plant installations and a description of constructional details of feedwater heaters is given by Shah et al. [13] and Drbal et al. [6]. Numerical reliability aspects of modeling and simulation are raised by Henrik and Olsson in [14]. Furthermore, a simplified method of calculating the heat flow through a two-phase heat exchanger is described in [15]. Application of system identification techniques in estimating parameters of a heat exchanger working with a liquid medium is described in [16]. Bonivento et al. [17] 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 [18-19]. 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. A simulation study of a condensate level control was presented in [7] where aspects of a feed-forward vs. a feedback control were highlighted. Heater models are also discussed as aspects of modeling power units; an example is a modular system consisting of numerous components utilized in [22] in building large-scale models of a power unit. From this perspective, an advanced heater model is also discussed by Alessandri et al. [23]. 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 and 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. [23]. 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 by 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 nonlinear 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+4. Two simplifications of a steam circuit model are considered. The first simplification not discussed in this paper involves reduction of the number of equations in the initial six-volume model. The second simplification of the model, presented in more detail in Section 4, is a four-volume model.

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.

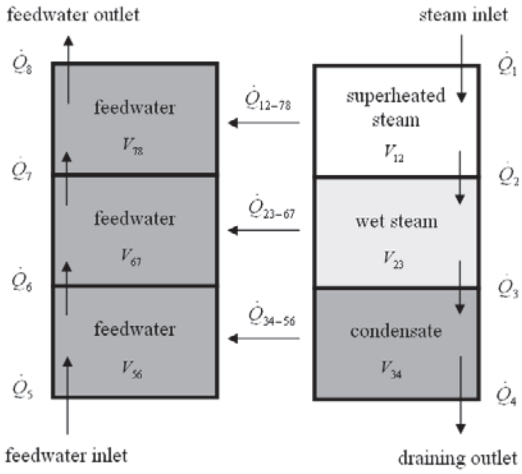

Fig. 1. Schematic representation of heat energy transfer in the six-volume heater model

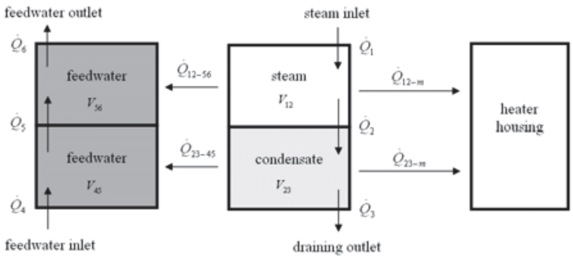

Fig. 2. Schematic representation of heat energy transfer in the four-volume heater model

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

$$\dot{Q}_2 = h_2 \cdot \dot{m}_2 \tag{1}$$

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

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

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:

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

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. SIX-VOLUME HEATER 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

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

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.

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

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

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.

$$\frac{dm_{12}}{dt} = \dot{m}_2 - \dot{m}_1 \tag{7}$$

$$\frac{dm_{23}}{dt} = \dot{m}_3 - \dot{m}_2 \tag{8}$$

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

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

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

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

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

$$x = \frac{V_{34} - V_{340}}{A_{con}} \tag{12}$$

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.

4. 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 (13), 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 (13), 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.

$$\begin{bmatrix} \frac{dm_{12}}{dt} \\ \frac{dH_{12}}{dt} \end{bmatrix} = \begin{bmatrix} \frac{\partial m_{12}}{\partial V_{12}} & \frac{\partial m_{12}}{\partial p_{12}} \\ \frac{\partial H_{12}}{\partial V_{12}} & \frac{\partial H_{12}}{\partial p_{12}} \end{bmatrix} \cdot \begin{bmatrix} \frac{dV_{12}}{dt} \\ \frac{dp_{12}}{dt} \end{bmatrix} = \begin{bmatrix} e_{11} & e_{12} \\ e_{21} & e_{22} \end{bmatrix} \cdot \begin{bmatrix} \frac{dV_{12}}{dt} \\ \frac{dp_{12}}{dt} \end{bmatrix} \tag{13}$$

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 (13) can be rearranged in the following form

$$\begin{aligned} e_{11}\frac{dV_{12}}{dt} + e_{12}\frac{dp_{12}}{dt} &= \dot{m}_1 - \dot{m}_2 \\ e_{21}\frac{dV_{12}}{dt} + e_{22}\frac{dp_{12}}{dt} &= \dot{Q}_1 - \dot{Q}_2 - \dot{Q}_{12-56} - \dot{Q}_{23-45} \end{aligned} \tag{14}$$

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

$$\begin{aligned}
e_{11} &= \frac{\partial m_{12}}{\partial V_{12}} = \frac{\partial(\rho_{12} \cdot V_{12})}{\partial V_{12}} = \\
&= \rho_{12} + \frac{\partial \rho_{12}}{\partial V_{12}} V_{12} = \rho_{12} \\
e_{12} &= \frac{\partial m_{12}}{\partial p_{12}} = \frac{\partial \rho_{12}}{\partial p_{12}} \cdot V_{12} + \frac{\partial V_{12}}{\partial p_{12}} \cdot \rho_{12} = \\
&= \frac{\partial \rho_{12}}{\partial p_{12}} \cdot V_{12} \\
e_{21} &= \frac{\partial H_{12}}{\partial V_{12}} = \frac{\partial(\rho_{12} V_{12} h_{12})}{\partial V_{12}} = \\
&= \rho_{12} h_{12} + \frac{\partial \rho_{12}}{\partial V_{12}} h_{12} V_{12} + \frac{\partial h_{12}}{\partial V_{12}} \rho_{12} V_{12} = \\
&= \rho_{12} h_{12} - p_{12} \\
e_{22} &= \frac{\partial H_{12}}{\partial p_{12}} = \frac{\partial(\rho_{12} V_{12} h_{12})}{\partial p_{12}} = \frac{\partial \rho_{12}}{\partial p_{12}} V_{12} h_{12} + \\
&+ \frac{\partial V_{12}}{\partial p_{12}} \rho_{12} h_{12} + \frac{\partial h_{12}}{\partial p_{12}} \rho_{12} V_{12} - V_{12} = \\
&= V_{12}\left(h_{12} \frac{\partial \rho_{12}}{\partial p_{12}} + \rho_{12} \frac{\partial h_{12}}{\partial p_{12}} \right) - V_{12}
\end{aligned} \tag{15}$$

and, additionally,

$$V_{12} = V_{total} - V_{23} \text{ then } dV_{12} = -dV_{23} \tag{16}$$

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

$$\begin{aligned}
&\text{if } \frac{dm_{12}}{dt} = \dot{m}_1 - \dot{m}_2 \text{, and } \frac{dm_{23}}{dt} = \dot{m}_2 - \dot{m}_3 \text{,} \\
&\text{then } \frac{dm_{23}}{dt} = \dot{m}_1 - \dot{m}_3 - \frac{dm_{12}}{dt}
\end{aligned} \tag{17}$$

Variables obtained from (17) are substituted to Equations (14)

$$\begin{aligned}
&\dot{m}_3 - \dot{m}_2 = e_{11} \frac{dV_{12}}{dt} + e_{12} \frac{dp_{12}}{dt} \\
&\dot{m}_3 h_3 - \dot{m}_2 h_2 - \dot{Q}_{12-m} - \dot{Q}_{23-m} = \\
&= e_{21} \frac{dV_{12}}{dt} + e_{22} \frac{dp_{12}}{dt}
\end{aligned} \tag{18}$$

Unknowns are determined as follows

$$\begin{aligned}
dp_{12} &= \frac{m_3 (h_3 - h_2) - (e_{21} - e_{11} h_2) \frac{dV_{12}}{dt}}{(e_{22} - e_{12} h_2)} \\
&+ \frac{-\dot{Q}_{12-m} - \dot{Q}_{23-m}}{(e_{22} - e_{12} h_2)} \\
\dot{m}_2 &= \frac{(e_{12} e_{21} - e_{11} e_{22}) \frac{dV_{12}}{dt}}{(e_{22} - e_{12} h_2)} \\
&\frac{e_{12} (\dot{Q}_{12-m} - \dot{Q}_{23-m}) - (e_{12} h_3 - e_{22}) \dot{m}_3}{(e_{22} - e_{12} h_2)}
\end{aligned} \tag{19}$$

Equations (20-21) are formulated for the conservation of the energy in the draining volumes, assuming uniform density of the feedwater.

$$\frac{dH_{45}}{dt} = \dot{Q}_4 - \dot{Q}_5 + \dot{Q}_{23-45} \tag{20}$$

$$\frac{dH_{56}}{dt} = \dot{Q}_5 - \dot{Q}_6 + \dot{Q}_{12-56} \tag{21}$$

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

$$x = \frac{V_{23} - V_{230}}{A_{con}} , \tag{22}$$

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

$$\frac{dH_m}{dt} = \dot{Q}_{12-m} + \dot{Q}_{23-m} \tag{23}$$

where

$$H_m = m_{m\cdot} \cdot c_{pm} \cdot T_m \tag{24}$$

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).

Equations of the feedwater model were implemented in a convention required by Simulink [25].

5. SUMMARY AND CONCLUSIONS

The paper presents a feedwater heater model intended for model-based diagnostics compromising performance and functionality, which is, in its advanced-physics form, very complex and deteriorates the execution time of a whole power plant model. An advanced heater model allows the thermodynamics of the heat exchange process to be correctly captured, however, the model complexity does not enable on-line simulation of a complete power block model. Thus, the simplified heater models were proposed to reduce numerical complexity and model tuning effort. The goal behind model simplification was to develop a model capable of achieving performance at the level permitting real-time simulation, yet at the same time not dramatically sacrificing accuracy. The proposed simplification of the heat exchanger model has provided the greatest improvement towards numerical stability of a power unit model and significantly shorter computation time.

The development process of a feedwater heater model is presented in three steps through Sections 3+4. The initial six-volume model was used as the starting point for the simplification process. The four-volume model described in this paper has the potential to increase the level of understanding of modeled processes and play a diagnostic role. Another 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 a four-volume model, heat accumulation in the heater jacket was implemented to allow simulation of a start-up operation.

The validation process and application of a feedwater heater model will be presented in the second part of this work.

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