TEMPERATURE DYNAMIC
OF HEAT EXCHANGERS IN BOILERS

Lukáš Hubka

Technical university of Liberec,
Faculty of Mechatronics, Informatics and Interdisciplinary Studies,
461 17 Liberec, Studentská 2, Czech Republic

lukas.hubka@tul.cz

Abstract

This paper presents the possibilities of developing the temperature dynamic simulation models of water heating, steam generation and steam superheating in boilers of power plants. There are two basic methods considered for solving the problem of temperature dynamics. The first method is built on the lumped parameters, the second on global balances of non-isothermal system. The description of differential equation creating process is presented in this paper. The result is the nonlinear dynamic simulation model, which is prepared to work in wide range of operation parameters. Some results from both method models are shown, too. The comparison of model and real system dynamic behavior is demonstrated.

Keywords: steam generator, superheating, temperature model.

Presenting Author's biography

Lukáš Hubka graduated from Technical university of Liberec with his Master's of Science degree in Automatic Control and Computer Engineering. Currently, he is pursuing his PhD degree in Technical Cybernetics at Technical University of Liberec. His research interests lie in advanced technologies, control systems and modeling with the special focus on the power engineering. His current interests are in energetic/energy area and he is participating in monitoring and remote control of a biomass power plant and in developing a model and control system of temperatures in power plant rebuilding program. Lukáš Hubka is working closely with major energy producers in the country and is giving lectures at the Technical university of Liberec.

1 Introduction

Modeling is an extensive problem in the energetic plants and it can have many different forms and aims. The expected aims can be mechanical properties, steady state behavior, training simulator etc. Models focused on the temperature control area (dynamic behavior) are important part, too [1], [2]. The paper is aimed at this area and it is focused on the development of the model of the basic once-through boiler components. The steady state and dynamic behavior of these components, the ability of their simulation in wide range of use and the possibility to connect the control circuit are all considered to be the main aims and properties of new developed models. Consequently, the possibility to connect several subsystems together into one simulation model has to exist.

It is possible to imagine the once-through boiler as one tube, in which there is steam generated from water. Next, the steam is superheated to obtain the effective power energy.

The approach based on the combination of two methods was chosen. In order to realize the main aim, the dynamic nonlinear simulation model of high pressure part steam temperatures in once-through boiler, was chosen. The basis is the description of lumped parameters. The concept of lumped parameters in the case of heating exchangers can be simplified into the description of medium in dependence of time and one position coordinate. As the most significant system nonlinearity, the steam properties are directly implemented in final equations. Steam properties must be continually changing during the simulation process by the actual operating state. Therefore, the right implementation of steam tables is used for obtaining data. However, another method is used for solving the given problem. The latter method is based on the global non-isothermal equations for the medium on the tubes [2], [3]. The heat exchanger (tube) is divided into several parts in a row. It is possible to obtain the transfer function of higher order in a way, which is typical for heat exchangers [4]. This method became crucial for achieving necessary and accurate process of modeling of the temperature in the steam heat exchangers.

The use of steam tables appears to be very useful in the simulations. The steam tables (IAPWS [5]) exist in many different forms. For the simulation model it is necessary to choose the right implementation called Matlab-Simulink. The tables implemented into XSteam [6] and FluidProp [7] were discovered as optimal for this purpose. The solution is quick and accurate in every computation step.

2 Temperature dynamic model

2.1 General equations

The Euler equations for non-isothermal system [3], [8] are used as basis for construction the model. We can use these equations in the following form:

$$\frac{\partial \rho}{\partial t} + \frac{1}{F} \cdot \frac{\partial \dot{m}}{\partial z} = 0, \tag{1}$$

$$\frac{\partial \dot{m}}{\partial t} + \frac{\partial}{\partial z}\left[p \cdot F + \frac{\dot{m}^2}{\rho} \cdot \frac{1}{F} \right] = -\frac{1}{2} \cdot \zeta \cdot \frac{|\dot{m}| \cdot \dot{m}}{\rho} \cdot \frac{1}{V}, \tag{2}$$

$$\frac{\partial u \cdot \rho}{\partial t} + \frac{1}{F} \cdot \frac{\partial \dot{m} \cdot h}{\partial z} = \frac{1}{F} \cdot \frac{\partial \dot{Q}}{\partial z}. \tag{3}$$

Where ρ - density, m – flow rate, p – pressure, u – inner energy, h – enthalpy, Q – input heat, F – cross-section area, V – inner tube dimension, ζ – friction coefficient, t – time, z – space coordinate.

The first equation Eq. (1) is mass balance, the second equation Eq. (2) is momentum equilibrium and the last equation Eq. (3) is energy balance. The complete set of equations is defined in combination with the basic equation for enthalpy Eq. (4), which is known from thermodynamic.

$$u = h - \frac{p}{\rho} \tag{4}$$

Because of the dimensions (small diameter and big length of the tube) and the way of heating the medium it is possible to consider a simplification of the whole description into one axis dimension as it is shown in equations Eq. (1)-Eq. (3). The model based on the lumped parameters in the full space (3D) would be much more complicated for the computation and the obtained information would be at the expense of the computation speed. The speed is an important criterion factor for usability. It is supposed that the input heat power to every part (every heat exchanger) is known in advance. It is almost always true, because the heat power can be solved separately as a steady state problem (constructor team made this computation). This presumption brings the restriction in the relative flow motion. We do not solve the problem of heat exchange combustion gas – steam, we solve only the problem of orthogonal flow [9].

2.2 Description of medium temperature dynamic in lumped parameters

Let us consider the heat exchanger represented by one of its tubes (see Fig. 1). The description is based on the right arrangement of Euler equations Eq. (1)-Eq. (3). First, we solve the process of transporting the heat energy from the tube into the medium (steam or water). Subsequently, we proceed to solve the temperature dynamic (energy balance) of the tube.

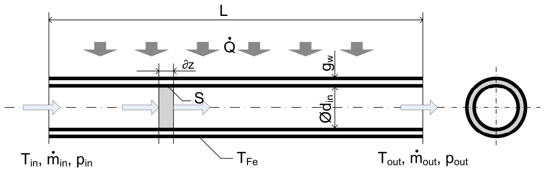

Fig. 1 Heat exchanger – tube and heated medium

Let's suppose $\dot{Q}(t)$ is the heat input into the medium. The diameter of the tube is small, therefore the temperature of the heated medium will be constant in the whole cross-section. We establish the inner energy *u* from Eq. (4) into Eq. (3) and the final form is Eq. (5) .

$$\frac{\partial(\rho\cdot h)}{\partial t}-\frac{dp}{dt}+\frac{1}{F}\frac{\partial(\dot{m}\cdot h)}{\partial z}=\frac{1}{F}\cdot\frac{\partial\dot{Q}}{\partial z} \tag{5}$$

The equations of mass and energy balance have to be valid every time. If the time derivative of density from Eq. (1) is substituted into Eq. (3), then a new presumption is made. This presumption states that the time derivative of pressure is negligible. The time change of enthalpy is in Eq (6).

$$\frac{\partial h}{\partial t}=-\frac{\dot{m}}{F\cdot\rho}\cdot\frac{\partial h}{\partial z}+\frac{1}{F\cdot\rho}\cdot\frac{\partial\dot{Q}}{\partial z} \tag{6}$$

The Eq. (6) is re-sampled in the axis coordinate for $\partial z=\Delta z=L/n$ and solved for the whole tube (i = 1...n). The result is an equation system of n equations, the unknown parameter is the enthalpy h. We suppose the constant flow rate inside the tube. It means $\partial\dot{m}/\partial z=0$. The equation system is after rearrangement in vector form

$$\frac{d\mathbf{h}}{dt}=\frac{\dot{m}}{\Delta V}\cdot\mathbf{\Gamma}\cdot\mathbf{h}+\frac{L}{\Delta V}\cdot\mathbf{\Psi}\cdot\dot{\mathbf{q}}+\mathbf{\Omega}\cdot\frac{\dot{m}}{\Delta V}\cdot\frac{h_0}{\rho_1}, \tag{7}$$

where

$$\mathbf{\Gamma}=\begin{bmatrix}-\frac{1}{\rho_1} & 0 & . & . & 0\\ \frac{1}{\rho_2} & -\frac{1}{\rho_2} & 0 & . & 0\\ 0 & . & . & . & .\\ . & . & . & . & 0\\ 0 & . & 0 & \frac{1}{\rho_n} & -\frac{1}{\rho_n}\end{bmatrix},\ \Delta V=F\cdot\Delta z$$

$\mathbf{\Psi}=diag\left(\frac{1}{\rho_1}\quad\frac{1}{\rho_2}\quad ..\quad\frac{1}{\rho_n}\right)$ and $\mathbf{\Omega}=\begin{bmatrix}1 & 0 & .. & 0\end{bmatrix}^{\mathrm{T}}$.

The last part of Eq. (7) $\mathbf{\Omega}\cdot\frac{\dot{m}}{\Delta V}\cdot\frac{h_0}{\rho_1}$ is the boundary condition of the process. The boundary condition is a parameter of input medium (steam) that enters into heat exchanger.

The enthalpy is known in every time step (as solution of equation system) and in every cross-section cut. The temperature of the medium can be computed from steam tables in every cross-section cut, resp. in the end, because $T = T(h, p, z)$. Sometime it is useful to compute the temperature directly. The relation between enthalpy and temperature can be written as shown in Eq. (8).

$$\partial h=c_p\cdot\partial T \tag{8}$$

Where c_p is the heat capacity coefficient by the constant pressure.

The Eq. (8) is known as a consequence of the first law of thermodynamic. This equation can be put into the Eq. (7). Then the temperature is directly in the position of a computed state value (see Eq. (9))

$$\frac{d\mathbf{T}}{dt}=\frac{\dot{m}}{\Delta V}\cdot\mathbf{\Gamma}\cdot\mathbf{T}+\frac{L}{\Delta V}\cdot\mathbf{\Psi}_c\cdot\dot{\mathbf{q}}+\mathbf{\Omega}\cdot\frac{\dot{m}\cdot T_0}{\Delta V\cdot\rho_1} \tag{9}$$

Where $\mathbf{\Psi}_c=diag\left(\frac{1}{\rho_1\cdot c_{p1}}\quad\frac{1}{\rho_2\cdot c_{p2}}\quad\dots\quad\frac{1}{\rho_n\cdot c_{pn}}\right)$

and $\mathbf{T}=\begin{bmatrix}T_1 & T_2 & \dots & T_n\end{bmatrix}^{\mathrm{T}}$.

$\mathbf{T}$ is the temperature vector of medium (steam) temperatures in cross-section parts.

2.3 Temperature dynamic of the barrier

The tube (barrier) is the next important part of the whole heat exchanger system. The material and dimensions of the tube have the dominant impact on the speed of temperature changes of flow medium (water/steam), because the tube has the dominant temperature capacity effect.

The temperature dynamic of barrier can be written in the vector form as in Eq. (10).

$$\frac{d\mathbf{T}_{Fe}}{dt}=\frac{1}{m_{Fe}\cdot c_{Fe}}\cdot\left(\dot{\mathbf{Q}}_{input}-\dot{\mathbf{Q}}_{output}\right) \tag{10}$$

The input heat power is known in advance and it is equal to constant in every tube element (closely like a cross section flow). The output heat power depends on temperatures of the medium (steam) and the tube, the heat exchange area S and the heat exchange coefficient α according to the Eq. (11).

$$\dot{\mathbf{Q}}_{input}=\alpha\cdot S\cdot\Delta\mathbf{T}=\alpha\cdot S\cdot\left(\mathbf{T}_{Fe}-\mathbf{T}\right) \tag{11}$$

Finally, after establishing the Eq. (11) into Eq. (10), the temperature dynamic of the barrier has the form of Eq. (12).

$$\frac{d\mathbf{T}_{Fe}}{dt}=\frac{1}{m_{Fe}\cdot c_{Fe}}\cdot\left(\dot{\mathbf{Q}}_{input}-\alpha\cdot S\cdot\left(\mathbf{T}_{Fe}-\mathbf{T}\right)\right) \tag{12}$$

2.4 Temperature model of heat exchanger

The complete model of the heat exchanger is constructed from Eq. (12) and Eq. (9). The last step

(Eq. (13)) is to institute the value of input heat power from tube into the enthalpy dynamic of medium (steam) Eq. (9).

$$\frac{d\mathbf{T}}{dt}=\frac{\dot{m}}{\Delta V}\cdot\mathbf{\Gamma}\cdot\mathbf{T}+\frac{S}{V}\cdot\mathbf{\Psi}_c\cdot\left(\mathbf{T}_{Fe}-\mathbf{T}\right)+\mathbf{\Omega}\cdot\frac{\dot{m}}{\Delta V}\cdot\frac{T_0}{\rho_1} \quad (13)$$

Moreover, this equation system for the heat exchanger is independent of the medium phase and can be used to compute the evaporator temperature dynamic with water on the inlet and steam on the outlet side.

2.5 Series of global equations method – partially distributed parameters

The description in the lumped parameters has some disadvantages. The main weakness is the computing intensity. The lumped parameter method is based on a huge number of differential equations and in addition it is in every solving step necessary to access into steam tables for steam parameters (density, etc.).

Let the presumptions be the same like in the case of lumped parameters system description. The whole heat exchanger can be divided into separate parallel tubes (all tube are same with the same behavior) as is constructed. Than it is sufficient solving only one tube. This tube is sectionalized into a number of *n* parts with the same length. The sought-after information is the outlet steam temperature, more specifically, the temperature of the last part of intersected tube. The rearranged global balances of nonisothermical system are possible to be induced from basic Euler equations by the integration of one tube in heat exchanger over the full length (see Eq. (14) and Eq. (15)). The density and the flow rate are now considered constant along the whole tube length.

$$V\overline{\rho}\frac{dh}{dt}=\dot{m}_{in}\left(h_{in}-h_{out}\right)+\alpha S\left(T_{Fe}-T_{out}\right) \quad (14)$$

$$m_{Fe}\cdot c_{Fe}\frac{dT_{Fe}}{dt}=\dot{Q}_{dodané}-\alpha\cdot S\cdot\left(T_{Fe}-T_{out}\right) \quad (15)$$

The complete equation system of temperature dynamic of the medium and the tube is based on the idea of constant steam parameters in the tube. It brings the simplification in the solution, because instead of computing the density (and heat capacity) in every cross-section part, the density is computed only in the outlet. For example, if the tube is divided into 50 parts in the lumped parameters, we have to look up fifty times into steam tables. However, with the new method introduced in this paper it is necessary to find only two parameters in every computation solving step. The density and the heat capacity of the steam are written in Eq. (16).

$$\overline{\rho}=\frac{1}{2}\cdot\left(\rho_{in}+\rho_{out}\right),\ \overline{c}_p=\frac{1}{2}\cdot\left(c_{pin}+c_{pout}\right) \quad (16)$$

The full equation system for the heat exchanger is in the form of Eq. (17) and Eq. (18)

$$\frac{d\mathbf{T}_{Fe}}{dt}=\frac{1}{m_{Fe}\cdot c_{Fe}}\cdot\left(\dot{\mathbf{Q}}_{input}-\alpha\cdot S\cdot\left(\mathbf{T}_{Fe}-\mathbf{T}\right)\right) \quad (17)$$

$$\begin{aligned}\frac{d\mathbf{T}}{dt}=&-\frac{\dot{m}}{\Delta V\overline{\rho}}\cdot\mathbf{\Gamma}^*\cdot\mathbf{T}+\alpha\frac{S}{V\overline{\rho}\overline{c}_p}\cdot\left(\mathbf{T}_{Fe}-\mathbf{T}\right)+\\&+\mathbf{\Omega}\cdot\frac{\dot{m}}{\Delta V}\cdot\frac{T_{in}}{\rho_{in}}\end{aligned} \quad (18)$$

where $\mathbf{\Gamma}^*=\begin{bmatrix}1 & 0 & . & . & 0\\ -1 & 1 & 0 & . & 0\\ 0 & . & . & . & .\\ . & . & . & . & 0\\ 0 & . & 0 & -1 & 1\end{bmatrix}$.

The only problem is the accuracy of this solution. It was found that for standard heat exchangers in power plants, where any change is on the phase of the medium, the error is very small both in the steady state, and in the dynamic. Moreover the steady state error can be easily compensated by the simple look up table.

3 The steam generation and superheating in the once-through boiler

3.1 Superheater technology model structure

The whole technological part of steam superheating and steam reheating consist of several partial subsystems, which can be described and modeled separately. Suppose the technological system has the superheating part as shown on the scheme (see Fig. 2). The same heat exchangers are repeating in other technological constructions of once-through boiler and therefore it is possible to use the same simulation models as absolutely universal.

It is possible to separate the steam generation and superheating into several parts. The first part is without primary temperature control by some valves and is set up from the economizer, the steam generator and first part of superheating. The next part, in this special configuration, is the counter-current heat exchanger between high pressure and middle pressure steam. The reaction of this system is relatively fast, in the comparison with other heat exchangers [10]. The last, and very important, part is the steam superheating. It is realized by three separate superheaters. Every superheater is likely to control the outlet steam temperature by the control valve on the input. The input valve is the spray attemperator and it brings the injection water to the inlet side of every superheater. The steam reheating is separate part [11].

The basic technological parts show the need to construct the water injection model and steam mixing model.

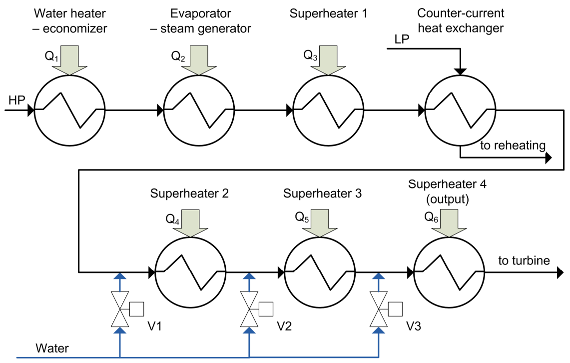

Fig. 2: Feasible architecture of the high pressure part of the once-through boiler

3.2 Modeling method

The simulation input to every heat exchanger is the steam (water) with three significant parameters: the temperature, the pressure and the flow rate. The last important input is the input heating power. These four parameters together create the heating level and all parameters are defined on several heating levels (in the range 50 – 100%).

The models of separate parts of boiler are built from equations in the lumped parameters or as a series of global method. The steady state and dynamic behavior of separate component and the full high pressure part were discovered in several experiments. The simulations were made in open and closed loop. The closed loop was working with standard controller, which was implemented on the real system.

4 Simulation results

4.1 Steam generator (evaporator)

The main problem with the evaporator dynamic model construction can be the fact, that there is the phase change from liquid to vapor in this exchanger. The significant changes of flowed medium physical properties (density, enthalpy, heat capacity) are connected together by this change. On the other hand, the temperature change is small, in some area there is no change at all. It is clear, that the use of global balance equations is problematic and the results can have a big error, mainly in the dynamic responses. The better possibility is to describe the heat exchanger in the lumped parameters. This description should lead to the best steady states and dynamic responses of this heat exchanger. The other advantage is the possibility of viewing the evaporating area. The disadvantage is in the computing demands, because it is necessary to find some important physical parameters in every computation step. If we have only one (or two) heat exchangers constructed this way in the whole system, there is no problem with the solving speed.

The final model is realized by the equations Eq. (13) and Eq. (12).

The functional ability of the model in lumped parameters is shown on Fig. 3 and Fig. 4. The space distribution of the density in the evaporator is in the Fig. 3. The density is shown on several heating levels and it is possible to see the rapid change of density in small length of tube. The point of changing the phase from liquid to vapor (water to steam) is also nicely shown. The point of the beginning of this change is in the maximal break on the curves in fig. 3. The yellow line (for maximal heating power) does not have this significant point, because the process of evaporation has a high pressure, very close to critical pressure.

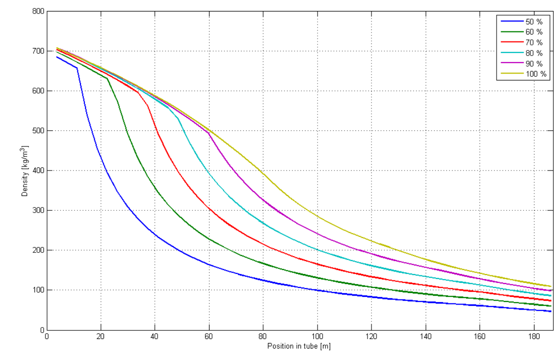

Fig. 3: The density space distribution in the evaporator model

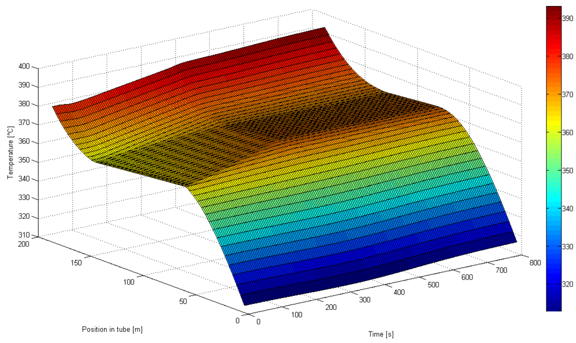

Fig. 4: The temperature fields in evaporator model by the heating level change

The temperature field in evaporator is in Fig. 4. This figure presents the temperature changes in space and time in the situation of changing the heating level from 90 to 100 % with the standard tendency 10 % per 5 min. Furthermore, it is possible to find the characteristic area of evaporating, which is represented by the place with any changes of temperature.

4.2 Water spray attemperator, steam/steam mixing

The control process of changing water into steam injection through the spray attemperator is the primary part of control circuits of all steam generating power plants. It is possible to say, that the temperature dynamic behind the spray attemperator is much faster than the temperature dynamic of whole heat exchanger (the outlet steam temperature). The temperature sensors dynamic, the dynamic of valve servo and, of course the temperature dynamic of the mixing pit have the significant impacts on the steam temperature behind the valve.

A simplified model of spray attemperator was created. A very simple description based on the algebraic equations of global balances was elected. This solution has many advantages. Mainly the output temperature of steam after water injection is absolutely correct in the steady state. The next positive thing is the possibility of absolutely free dynamic. The model is prepared to import the dynamic of the real spray attemperator, if it exists. If the dynamic is unknown, it is possible to use some approximation, for example, the first order system with time delay, and estimate the real properties from the construction parameters. The temperature of the outlet of the injection process (inlet of heat exchanger) can be described by the Eq. (19).

$$h_{out} = \left(\dot{m}_w h_w + \dot{m}_s h_s\right)\frac{1}{\dot{m}_{out}}$$
$$T_{out}\left(t\right) = T_{out}\left(t, h_{out}, p_{out}\right), \qquad (19)$$

where the index *out* is for the outlet side (after mixing), *w* are the properties of injection water and *s* are the properties of steam.

Another important advantage is that the same equation can be used for steam/steam mixing and simple water/steam mixing, too. This implies to the final spray attemperator solution. Every valve system is constructed as water/steam mixing with separate dynamic (see Fig. 5).

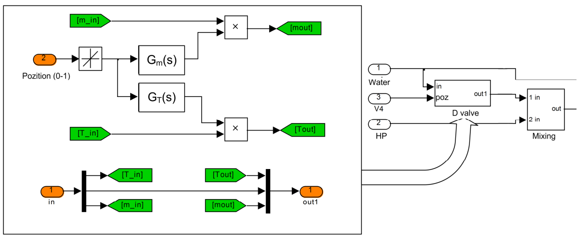

Fig. 5: The model of steam attemperator

The valve realized in the model has implemented the adjustment time 30 sec and linear steady state character.

4.3 Superheaters

The controlled superheater is mostly separated into some subsystems such as [spray attemperator, heat exchanger] or [three-way bypass valve, heat exchanger, mixing].

Let us solve, for example, the output heat exchanger. This is the last heat exchanger in the high pressure part. The steam produces a considerable amount of heat in this heat exchanger and the temperature markedly grows from the inlet (before the spray attemperator) value of 485 °C to final controller outlet temperature 575 °C. This high temperature changes are connected together with changes in the density and heat capacity. It could be considered to use lumped parameters model, but it is much better to find other way. The partially distributed model equations are used (Eq. (18) and Eq. (17)) in the case of superheaters. The density is considered as a mean of the inlet steam density and the outlet steam density. It was proved that the error is small and therefore it is possible to use this description. The most important proof is shown in Fig. 6, where there is the verification of the model with data from real measurement. There the response to the step change on the injection valve position on output superheater is shown. The response of the model is compared to the response of the real system. Both responses are nearly the same. The dynamic on the inlet side (before the superheater) is the same in model and in the real application. Moreover, the model is prepared to add some other dynamic, such as the dynamic of sensors, etc.

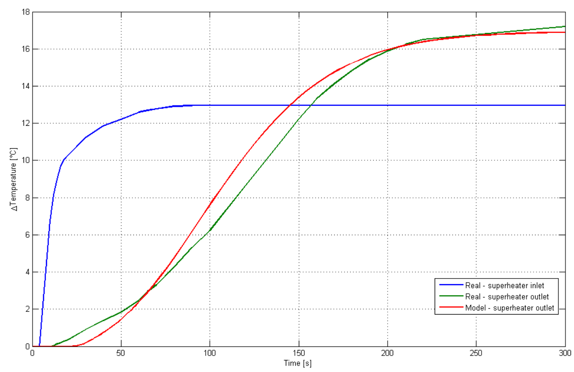

Fig. 6: The comparison of the model and real data

The Fig. 7 shows the simulation model of one heat exchanger (superheater) as is realized in Simulink.

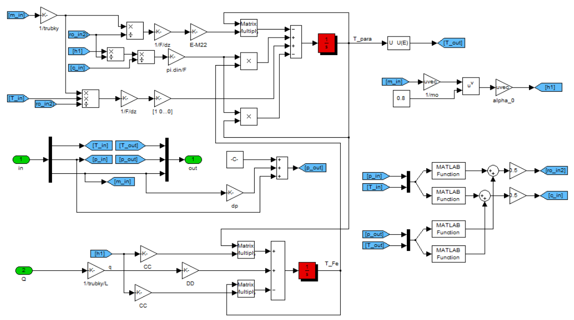

Fig. 7: The Simulink simulation model of output superheater

5 Conclusion

The method of building temperature models of boiler components (heat exchangers) was prepared, verified and tested. The presented algorithm and the structure of the model are prepared to include some other dynamics (the changing of technological parameters through the life cycle, the sensor dynamics, etc.). The most significant presumption on the known heat flux into heat exchanger can be replaced by the dynamic model of combustion gases. Probably will be possible to close the circuit of water/steam (the turbine, the condenser, the HRSG, etc.).

Some significant results from simulation and verification are presented. The model is actually used for testing the control system and the development of new possibilities in control.

Acknowledgement

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