Temperature Dynamic Model of Once-through Boiler Based on Flue Gases Heat Transports

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Abstract: This paper presents a model of the temperature dynamics of a heat exchanger unit and it shows how it can be used as building block of a simulation model of a coal fired once-through boiler. The heat exchanger model is based on balances of non-isothermal system in the form of partial differential equations. The heat exchanger model solves the complex problem of heat energy transfer from flue gases to the water/steam. The main emphasis is on the use of the model for control design and optimization. The model developed in the paper can be used in a wide range of operation conditions and it covers all standard operation states and a huge set of possible disturbances. The model has been developed in the design phase of a power plant and its verification based on real measured data was performed using the measured data from another power plant with similar boiler design. This comparison is included in the paper. The complete model of a once-through boiler is now used to test and develop control algorithms for one power plant in its design phase and one power plant with similar technology that is already in operation. Some results were used to design control structures for the rejection of certain specific disturbances.

Keywords: heat exchanger model, once-through boiler, power plant.

1 INTRODUCTION

This paper is focused on the field of modelling the components and subsystems of power plant boilers and it develops further the preliminary ideas that were outlined in Hubka (2009, 2010a, 2010b), Hlava (2009). Similarly as in it is done e.g. in Lu (1999) and Ordys et al. (1994) the main point of interest is the temperature dynamics. The boiler, as a model object, is very complex and specific system. The problem of building the models of subsystems from the boilers has been in the centre of view for many years. It is a logical result of the complexity and also the continuous technological improvement. The main interesting problem solved in this paper is the temperature dynamics of the steam generator, superheating unit and reheating unit. The big accent is focused on the ability of simulation in the wide operation range of use and on the possibility to connect the control system.

Let shortly describe the main problem. The whole paper is focused on once-through boiler (double flue with dry-bottom furnace with direct blow of coal dust to burner sections, see Fig. 1). But almost all ideas and equations can also be used in the drum boiler. The once-through boiler can basically be regarded as one tube, in which steam is generated from water. Some more effective constructions of Rankine cycle use the steam reheating to improve the power energy and mainly the efficiency of the boiler. The purpose of reheater is to raise and to stabilize the temperature of the steam entering the intermediate pressure stage of the turbine.

2 TEMPERATURE DYNAMICS MODEL

2.1 Main ideas and procedures

This model includes two important sources of nonlinearity. The first source is primarily due to the nonlinear properties of steam. To achieve reliable wide range model, the steam properties must be changed continually during the simulation according to the current operating point. Therefore, the steam tables in a software form are used to obtain relevant steam properties (such as density, etc.). The steam tables are described by international society (IAPWS 1997) and they exist in many different forms. The presented model is constructed in Matlab/Simulink and good options for implementation of the steam tables are XSteam (Magnus 2007) and FluidProp (Colonna et al. 2010). The second important source of nonlinearity is the heat transfer coefficient that varies with the operating point of the boiler.
This relation must also be solved on-line during simulation using some acceptable method (the look-up table, etc.). The basis of the presented approach is the description of the heat exchanger in distributed parameters (PDE – partial differential equation description). The concept of distributed parameters can be simplified into the description of medium in dependence on time and one position coordinate in the case of heat exchangers. Some simplifications based on the global balances for the medium in tubes are used. Just input and output variables from the exchanger are considered (e.g. Ordys et al. (1994), Lienhard et al. (2008)) in these equations.

The heat energy is transported from flue gases to the steam through the mass of tubes. There are three main subsystems – media, in each heat exchanger, whose dynamic behaviour is solved separately, but the energy transfer still exists. The heat exchanger (tube) is divided into several control volumes. This step allows achieving acceptable accuracy in dynamic responses and it is also applicable in the wide range.

### 2.2. Theoretical basis for model construction

The PDE method is used to describe the process of heat exchanges. Equations (Euler’s) in the form (1)-(3) e.g. Logan (1999) are used.

\[
\frac{\partial p}{\partial t} + \frac{1}{F} \frac{\partial m}{\partial z} = 0, \tag{1}
\]

\[
\frac{\partial m}{\partial t} + \frac{1}{\rho} \frac{\partial (\rho \cdot F)}{\partial z} = -\frac{1}{2} \zeta \frac{\partial m}{\rho} - \frac{1}{V} \tag{2}
\]

\[
\frac{\partial u}{\partial t} + \frac{1}{F} \frac{\partial (\rho \cdot h)}{\partial z} = \frac{1}{F} \frac{\partial \dot{Q}}{\partial z} \tag{3}
\]

Where \( \rho \) - density, \( m \) – flow rate, \( u \) – inner energy, \( h \) – enthalpy, \( Q \) – input heat, \( F \) – cross-section area, \( V \) – inner tube volume, \( \zeta \) – friction coefficient, \( t \) – time, \( z \) – space coordinate.

Equation (1) is the mass balance, (2) is the momentum equilibrium and (3) describes the energy balance. These complete set of equations together with the basic equation (4) for the enthalpy, which is known from thermodynamics, define the basic dynamics of the heat transfer process.

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

It is possible and also necessary to make some simplifications. The reason is because of the dimensions (small diameter and big length of the tube), the computing demand, system complexity, etc. Firstly, it is possible to consider only one axis dimension as it is shown in equations (1)-(3). The pressure is also an interesting part of the model, but a significant simplification is implemented to the pressure, too. The pressure drop depends on the velocity of the steam through the tube, on the density of the steam and on the coordinate parameter according to the following equation

\[
\Delta p = \int_{0}^{L} \left( \lambda \cdot \frac{L}{d} + \sum \xi \right) \frac{\rho \cdot v^2}{2} \, dz \tag{5}
\]

where \( \Delta p \) – pressure drop, \( v \) – medium velocity, \( \lambda \) – friction factor, \( \xi \) – pressure loss.

By further analysis, this equation can be reduced into a form where the pressure drop depends only on flow rate and on coordinates. So it is possible to find elementary function, which approximates the function form (5) as in (6).

\[
\Delta p = k_r \left( \frac{Q}{m} \right) \cdot m^\zeta \cdot z \tag{6}
\]

The \( k_r \) coefficient can be calculated from steady state behaviour or searched in some tables. All simplifications are made mainly with the aim of less computer demand, this mean shorter computation time.

### 2.3. Steam temperature dynamic in distributed parameters

Let us consider the heat exchanger represented by one of its tubes (see Fig. 2). Generally, it is possible to make the same description for both main media – water/steam and flue gas. The only difference will be in some parameters. The description is based on an appropriate rearrangement of Euler equations (1)-(3).

![Fig. 2: Heat exchanger – tube and heated medium](image)

Firstly the steam dynamics is presented. Let’s suppose \( \dot{Q}(t) \) is the input heat into the medium. The diameter of the tube is small enough that the temperature of the heated medium will be constant in the whole cross-section. The inner energy \( u \) from (4) is established into (3) and the final form is (7).

\[
\frac{\partial (\rho \cdot h)}{\partial t} + \frac{1}{F} \frac{\partial (\rho \cdot h)}{\partial z} = \frac{1}{F} \frac{\partial \dot{Q}}{\partial z} \tag{7}
\]

The equations of mass and energy balance have to be valid every time. Substituting the time derivative of density from (1) into (3) together with a new presumption that the time derivative of pressure is negligible brings the time change of enthalpy in the equation (8). In fact, the pressure is the parameter on which almost all steam properties depends. But (6) together with zero time derivatives allows this simplification.

\[
\frac{\partial h}{\partial t} = -\frac{m}{F \cdot \rho} \frac{\partial h}{\partial z} + \frac{1}{F \cdot \rho} \frac{\partial \dot{Q}}{\partial z} \tag{8}
\]

The equation (8) is re-sampled in the axis coordinate for \( \partial z = \Delta z = L/n \) and solved for the whole tube \( (i = 1...n) \). Constant flow rate inside the tube is considered \( (\partial m/\partial z = 0) \).

The equation system is after rearrangement in vector/matrix form

\[
\frac{d}{dt} = \frac{m}{N} \cdot \mathbf{h} + V \cdot \Omega + \frac{m}{N} \frac{h_i}{\rho_i} \tag{9}
\]
\[
\begin{bmatrix}
-1/\rho_1 & 0 & \ldots & 0 \\
1/\rho_2 & -1/\rho_2 & 0 & \ldots \\
0 & \ldots & 1/\rho_n & -1/\rho_n
\end{bmatrix} + \Delta V = F \cdot \Delta z
\]

Where \( \Gamma = \) and \( \Psi = \text{diag} \left( \frac{1}{\rho_1}, \frac{1}{\rho_2}, \ldots, \frac{1}{\rho_n} \right) \) and \( \Omega = [1 \ 0 \ 0]^T \).

\[\partial h = c_p \cdot \partial T \]

where \( c_p \) is the heat capacity coefficient by the constant pressure. Eq. (10) holds for water sections and for steam sections. The two-phase region (water evaporating) is described by the equation (11)

\[ h = c_p \cdot T + h_{\text{e}} \]

where \( h_{\text{e}} \) is evaporation heat of the water. But the simulation model does not need the equation (11). The reason is that the heat capacity coefficient is found directly in steam tables and the step change of this parameter in the evaporating phase is implemented. The evaporating process is shown on fig. 5. There is a noticeable area, where the water evaporates without any increase in the temperature.

Finally, the final equation for temperature dynamics of inner medium (water/steam) is obtained from the equation (10) and some rearrangement of (9) in the form of (12).

\[
\frac{dT}{dt} = \frac{m}{\Delta V} \cdot \Gamma \cdot T + \frac{L}{\Delta V} \cdot \Psi \cdot q + \Omega \cdot \frac{m \cdot T_0}{\Delta V \cdot \rho_1} \]

(12)

Where \( \Psi = \text{diag} \left( \frac{1}{\rho_1 \cdot c_{p1}}, \frac{1}{\rho_2 \cdot c_{p2}}, \ldots, \frac{1}{\rho_n \cdot c_{pn}} \right) \) and \( T = [T_1 \ T_2 \ \ldots \ T_n]^T \).

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

The water/steam is always heated from the tube (not directly from flue gases). This means, heat exchange is always convective. The input heat can be computed as

\[ Q = \alpha_s \cdot S_{\text{out}} \cdot (T_{\text{tube}} - T_t) \]

(13)

where \( \alpha_s \) is the heat transfer coefficient (HTC). HTC can be found in tables e.g. Cerny (1975) or it can be computed in some range from the (14). There are some empirical graphs and formulas which describe the actual value of HTC for some states, tube lay-outs in dependence on the variable temperature, pressure, etc.

\[ \frac{\alpha_u}{\alpha_r} \sim \left( \frac{m}{m_{0}} \right)^{0.8} \]

(14)

The zero’s index means the property value in some initial condition. The computation of HTC brings two new problems. First, there is the problem of the initial value \( \alpha_u \). I used a pre-computed steady state values and tables to find this initial value. The second problem is trickier. The question is how to cover all variations of HTC as a function of the current state of the medium. The HTC estimation process is easier in areas where with only one medium (only water, only steam). The equation (14) holds in these areas. Most difficult part is the evaporating area. There is made a simplification. But the model allows separating the part with water, wet steam and steam. The HTC is estimated in every area separately. The keys for the right HTC estimation are in simplifications and in the right implementation of the empirical tables.

2.4. Flue gas – temperature dynamic and properties

The problem of flue gas dynamic and the computation of all flue gas properties can be very complicated. But, the main purpose of the presented model is to obtain a useful tool to develop a control algorithm. Consequently, it is possible to make some basic simplifications. The same equation for the flue gas as for the steam (12) can be used. The only, but important, difference is in the heat transfer. The character of heat transfer depends on the position of heat exchanger in the boiler and it can be both convective and radiant. The (15) is mainly used for radiant heat transfer.

\[ Q = \alpha_s' \cdot S_{\text{out}} \cdot (T_{\text{tube}}' - T_{\text{med}}) \]

(15)

The problem is in the heat coefficient. This coefficient is very small and it is not easy to find the right value. Maybe the easiest way is to use once more some tables and empirical formulas. But the problem with radiant heat transfer stays open for a future development. The standard equation for convection heat transfer was used in the model. The results from simulations show, that it is possible to make so large simplification like this.

The density and the heat capacity are the last unknown parameters for flue gases equations. There exist many sophisticated equations for the computation of these values e.g. Logan (1999), but some approximation e.g. Budaj (1992) is easier to use. Generally it is also possible to use standard stoichiometric equation. Let the flue gases composition is known in advance. The density and the heat capacity for defined temperature range can be computed in advance and finally express as a common function of the temperature. For example, the density is

\[ \rho_{\text{flue}}(T) = -3.1 \times 10^{-3} \cdot T^3 + 2.1 \times 10^{-6} \cdot T^2 - 0.00164 \cdot T + 1.005 \]

(16)

for the flue gas of 66% N\(_2\), 17% H\(_2\)O, 12% CO\(_2\), 3.5% O\(_2\), 0.34% SO\(_2\) in the full range 200 °C – 1500 °C. The equation
(16) is a result of an interpolation of several density values from the specific range. The density for one temperature can be computed by help of gas-state equation and with using of molecular weight of all gas components.

2.5. The tube temperature dynamic

The last part of the heat exchanger subsystem is the tube. The tube intermediates the heat transfer between the flue gases and the water/steam. The material properties, construction size, amount of mass have the dominant impact to the steam temperature dynamic. It is primarily because of large temperature capacity of the tube material.

The temperature dynamic of barrier can be written in the vector form as in (17).

$$\frac{dT_{\text{Fe}}}{dt} = \frac{1}{m_{\text{Fe}} \cdot c_{\text{Fe}}} (\dot{Q}_{\text{input}} - \dot{Q}_{\text{output}})$$  \hspace{1cm} (17)

The input and output heat are common variables for both equations (12) of flue gases and steam.

2.6. Final heat exchanger temperature model

The final model is a set of equations for both media and the tube (18)-(20).

The steam balances equations

$$\frac{dT_{\text{e},i}(t)}{dt} = \frac{m_{\text{e},i}(t) \cdot n}{V_{\text{e},i} \cdot \rho_{\text{e},i}} \left( T_{\text{e},i}(t) - T_{\text{e}}(t) \right) + \frac{\alpha_{\text{e},i} \cdot S_{\text{e}}}{V_{\text{e},i} \cdot \rho_{\text{e},i} \cdot c_{\text{e},i}} \left( T_{\text{tube},i}(t) - T_{\text{e},i}(t) \right)$$  \hspace{1cm} (18)

The flue gas equations

$$\frac{dT_{\text{g},i}(t)}{dt} = \frac{m_{\text{g},i}(t) \cdot n}{V_{\text{g},i} \cdot \rho_{\text{g},i}} \left( T_{\text{g},i}(t) - T_{\text{g},i}(t) \right) + \frac{\alpha_{\text{g},i} \cdot S_{\text{g}}}{V_{\text{g},i} \cdot \rho_{\text{g},i} \cdot c_{\text{g},i}} \left( T_{\text{tube},i}(t) - T_{\text{g},i}(t) \right)$$  \hspace{1cm} (19)

The tube temperature dynamics

$$\frac{dT_{\text{tube},i}(t)}{dt} = \frac{\alpha_{\text{g},i} \cdot S_{\text{g}}}{m_{\text{tube}} \cdot c_{\text{tube}}} \left( T_{\text{g},i}(t) - T_{\text{tube},i}(t) \right) + \frac{\alpha_{\text{e},i} \cdot S_{\text{e}}}{m_{\text{e},i} \cdot c_{\text{e}}} \left( T_{\text{e},i}(t) - T_{\text{tube},i}(t) \right)$$  \hspace{1cm} (20)

This equation set, with the combination of interactive steam tables and some simplifications, is almost independent on the water phase (liquid vs. vapour) and can be used to compute the evaporator temperature dynamic with water on the inlet and steam on the outlet side, too.

The described presentation in the distributed parameters has some disadvantages, mainly it is computationally demanding. The reason is a huge number of differential equations for one heat exchanger (computed as a matrix problem) and a very frequent search in steam tables. One easy solution to this problem can be considered. It is possible to use a set of equations in lumped parameters instead of fully distributed parameters and to consider steady values of some significant parameters in the whole heat exchanger (for all i in eq. 18-20). For example it is possible to set

$$\rho = \frac{1}{2} (\rho_{\text{in}} + \rho_{\text{out}}) \cdot \tau_{p} = \frac{1}{2} (c_{\text{in}} + c_{\text{out}})$$  \hspace{1cm} (21)

The application of this assumption leads to significant simplification and restricts searching in steam tables. The second possible simplification is to optimize (lower) the number of computed cross-section parts. It means to find the minimal number of control volumes for every heat exchanger. This simplification can be done, but it has effect on the accuracy. So it is necessary to find the right number of control volumes, where the accuracy stays on high level and the number of cross-section parts is minimal (Fig. 3).

Fig. 3: The influence of number of cross-section parts on accuracy

The main equation (12) is then in the form

$$\frac{dT}{dt} = -\frac{m}{\Delta V \rho} \cdot \Gamma \cdot r + \alpha \cdot \frac{S}{\sqrt{\rho c_{p}}} \cdot (T_{\text{Fe}} - T) + \Omega \cdot \frac{m}{\Delta V \rho} \cdot \frac{T_{\text{in}}}{\rho_{\text{in}}}$$  \hspace{1cm} (22)

where

$$
\begin{bmatrix}
1 & 0 & . & . & 0 \\
-1 & 1 & 0 & . & 0 \\
. & . & . & . & . \\
0 & 0 & -1 & 1 \\
\end{bmatrix}

3 SIMULATION RESULTS

3.1. Point of interest

All simulations were made with the described simulation model and some remarkable results are presented. The point of the interest is the coal fuelled once-through boiler. The boiler specification is: double flue with dry-bottom furnace with direct blow of coal dust to burner sections. The nominal electric power output is 250 MW, feed water 187.4 kg/s output steam 575/580 °C and 18.3 MPa, the efficiency 42.75 % gross.

3.2. Steam generator (evaporator)

From the modelling viewpoint, the steam evaporator is one of the most difficult parts of the once-through boiler. The reason is that there are phase changes in this part. It brings rapid changes of density (see Fig. 4) and heat capacity and
relatively small changes of the output temperature (see Fig 5). The cause for so small changes in temperatures and so big changes in heat capacity is in (11), especially in evaporation heat. This is the main reason, why it is not possible to make so many simplifications in the description of the evaporating area. Mainly (21) cannot be used. The model must be solved as a fully distributed model. Fig. 4 describes the space distribution of density in the evaporator by the constant power level (show the legend) in a steady state (all output values as pressure or temperature are constant).

3.3. Once-through boiler model components

Standard once-through boiler has approximately 30 heat exchanges areas. This means that is necessary to construct about 30 models of heat exchangers plus some models for the attemperator sprays (in Hubka 2010b) and mixing zones. The complexity of the full model allows simulating all changes on heating levels, power levels, disturbances on flue gases, spray attemperator malfunctions, tests of control systems etc.

3.4. Model verification

The verification is an important part of the model development process. As the main motivation for this research was to model a power plant that is in its design phase, it was necessary to use measurement data from another plant that is already in operation and that has a similar boiler design. Tests on the functionality of spray attemperator and heat exchanger dynamic (Fig. 6) were realized, then the heating level changes were tested.

3.5. Model primary and advanced use – disturbances elimination

The model of full once-through boiler is primary used to optimize the control circuit of high pressure part steam temperature control and intermediate pressure part steam temperature control. The development of new control algorithms based on robust control, model predictive control and fuzzy control is another area where the model is used. Namely last two principles (MPC, fuzzy) are effective and under development see e.g. Hlava (2010).

The unmeasured disturbance rejection is one of the last simulation and control problems. The disturbance has big influence to the output steam temperature and this is the only place, where the disturbance effect is observable and detectable. The effect of the disturbance in a real operation is shown in Fig. 7. This figure shows the temperature oscillation of the power plant, which has to be limited. The task is: find, localize and identify the disturbance and eliminate the negative effect. The model based on the heat transfers from flue gasses to the steam became an ideal tool to fulfill this task. The possible place of disturbance genesis, the disturbance value and time range was discovered. The disturbance should come from the irregular and unstable burning and should be made by the flue gases temperature growth for a relatively short time (approx. till to 2 min). The reaction of the output heat exchanger is in Fig. 7 (outlet) and Fig. 8 (inlet). The output heat exchanger and its spray attemperator valve is the right place where the disturbance should and can be eliminated on the steam side. The elimination on the flue gasses side is not possible. The elimination by optimization of the feed water/fuel mass flow is impossible, because the effect of this disturbance is really short and it takes a long time to control the output steam temperature with water mass flow.

After localization of the source of this disturbance, the model was used to optimize the structure and tuning of the gain
scheduled PID cascade control system. Control responses before and after optimizations are shown in Fig. 9. The improvement is evident. The manipulated variables reach their limits.

Fig. 7: The comparison of the model and real data – disturbance effect on the output temperature

Fig. 8: The comparison of the model and real data – disturbance effect on the temperature on output superheater inlet

Fig. 9: The optimization result – output steam temperature before and after optimization

4 CONCLUSION

This model considerably simplifies development and testing of control algorithms because it allows simulating not only standard operation states, but also a wide range of disturbances. Currently it is used for this purpose with the objective to optimize the control structure and control system tuning of a power plant that is being completely refurbished.

The described boiler model has more over 2500 states and the standard simulation of the heating level changes takes about 1 hour (the simulated time is 1.5 hour). The complex model allows separating some heating areas – heat exchangers and simulates it separately. The computation speed is higher in this case and the same simulations take under 10 minutes.

The model was used to the first optimization of control system structure for the superheater control – output steam temperature and all changes should be implemented into real power plant control system. Some significant results from simulation and verification were presented in this paper. The main focus is now to the disturbance rejection and output steam temperature optimization.

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REFERENCES


