A Generalized Dynamic Water Side Model for a Once-Through Benson Boiler

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Abstract: This paper presents a dynamic model of a supercritical circulating fluidized bed (CFB) Benson type once-through utility (OTU) boiler. The water-steam cycle of the boiler is modelled using generalized model blocks able to handle thermodynamics of water, saturated water-steam mixture, and superheated steam under sub- and supercritical pressures. The generalized water model block was developed for the modelling of different type boilers. In the Benson type boiler a certain part of the boiler can act as an evaporator or superheater depending on the boiler load. This set challenges to the modelling work, because heat transfer and pressure drop correlations are phase-specific. The model block consists of two sections able to handle both evaporation and superheating. In the block where the heat transfer mode is changed from evaporation to superheating, the location of the boundary is determined during every simulation step according to the energy balance and corresponding heat transfer areas. The location of the boundary between different phases is an important factor from the point of view of modelling. This paper presents how the location of the boundary can be determined in the model.

Keywords: Circulating Fluidized Bed Boiler, Steam, Dynamic models, Modelling, Simulation

1. INTRODUCTION

A steam power plant is a complex system comprised of numerous components. The search for the optimal performance of the complex process under different operating conditions is not simple, because there are several variables that affect the dynamic behaviour of a system. However, the effective control of a power plant is in an important role from the point of view of safety, economical and environment aspects, Diaz (2003), Bracco et al.(2009).

The dynamic behaviour of the process can be studied by test runs. However, the test runs are not always possible for economic, productive, or safety reasons. For example, the test runs may require additional instrumentation, which can be installed only when the process is shut down.

Simulation is an alternative method to analyze the dynamic behaviour. Simulation has some advantages. The process can be analyzed in depth. Simulation gives possibilities to observe process variables and parameters which are not measurable. As well, the process can be studied safely under extreme conditions far away from the normal operating conditions. Simulation is also an efficient way to search for optimal operating conditions. Simulation can also be used for training purposes, Diaz (2003). Furthermore, simulation is an important tool for process development and control design.

In literature, there are found many mathematical models for simulating the dynamic behaviour of steam power plants. For example, Elmegaard (1999), Diaz (2003) and Habib et al. (2010) present dynamic models. In this work, the aim is to develop the dynamic model of the supercritical circulating fluidized bed (CFB) Benson type once-through utility (OTU) boiler. The goal is that the model can be used for several purposes such as control design, process development and operator training.

The presented model is based on the earlier CFB drum boiler model by Majanne and Köykkä (2009). Natural circulation drum boilers typically consist of feed water and combustion air preheaters, evaporator, drum, super- and reheaters. Boiler water is partly vaporized in vertically mounted evaporator tubes. After the evaporator saturated water-steam mixture is separated into saturated water and steam in the drum. The generated steam is replaced with feed water and this boiler water is circulated back to the inlet of the evaporator. Natural circulation is caused by the density difference between the water filled down comer pipes and water-steam mixture filled riser tubes. After the drum superheaters convert saturated steam into superheated steam, Spliethoff (2010).

There are some clear differences between the once through (OTU) and the natural circulation boilers. In the once through boiler there is no separate evaporator loop and no drum, but the water-steam cycle consists of a straight forward pipe system without any internal circulations. A boundary between vaporization and superheating sections is not fixed like in drum boilers but it varies as a function of the heat power - feed water ratio and boiler load, Spliethoff (2010). Moreover, a supercritical OTU boiler can be operated both in subcritical and supercritical pressures. Thus, the boiler tubes can contain
Subsaturated water, saturated water-steam mixture, superheated steam and supercritical fluid water in different points through risers. Different water phases set challenges to the modelling work. This paper describes how different phases and their boundary layers are taken into consideration using water side model blocks. The generalized structure of the model block makes possible the use of the block for the modelling of different types boilers such as the natural circulation and OTU boilers. The model block has been developed in a research project with Metso Power Oy. The modeling work is a part of development work of dynamic CFB boiler models.

2. DYNAMIC PROCESS MODEL

A dynamic process model for a CFB OTU boiler is presented. The model consists of the air-flue gas and the water side models combined together. The model describes the boiler process from the preheaters to the turbine and the reheaters. Figure 1 illustrates the simplified structure diagram of the process model. The modelled boiler is a Benson type boiler, which is the most commonly used once-through boiler type, Franke and Kral (2001).

Fig. 1. The simplified diagram of the process model.

The water is preheated, evaporated and superheated in one pass. The water is preheated almost up to the saturation point in the economisers. The evaporator, which means in this case the risers in furnace walls, vaporizes the water. The superheaters increase steam or supercritical fluid temperature. After the superheaters steam or supercritical fluid flows to the turbines. The reheaters increase steam temperature after the high pressure turbine. The water side of the boiler has been modelled using a generalized model block. The purpose is that all water side components can be modelled using the same generalized model block. The model is based on the first principle laws of mass, energy, and momentum balances and experimental correlations about reaction kinetics and heat transfer. The model was built using Simulink (Version 7.5) and Matlab (Version R2010a) by The MathWorks.

3. GENERALIZED MODEL BLOCK

The generalized water side model block contains pressure drop, heat transfer, energy and mass balance calculations. The properties of water are interpolated from steam and water tables. The input variables of the model block are mass flow of water \( \dot{m}_{\text{in}} \), inlet pressure \( p_{\text{in}} \), inlet enthalpy of water \( h_{\text{in}} \) and heat flow \( Q_g \) from the gas side. In addition, some properties of water \( w_{\text{in}} \) from the previous model block are transmitted to the current block to increase the effectiveness of the calculation. Output variables are water mass flow \( \dot{m}_{\text{out}} \), output pressure \( p_{\text{out}} \), output enthalpy of water \( h_{\text{out}} \) and some other variables \( o \) such as tube wall outer and inner temperatures. Input and output variables are partly equivalent, so that model blocks can be connected together such as in Figure 2.

![Fig. 2. General model blocks of the water side can be connected together.](image)

The model block needs parameters (constant variables) such as the number of tubes (in how many tubes the water mass flow is divided), tube inner diameter, tube thickness, mass, heated length, heat transfer area etc. Parameter values are found from the construction data of the boiler. A general model block can be used to represent different process components such as the parts of an evaporator, superheaters and economizers. The general model structure makes it easier to build the process model, because the general model block is easy to copy and only parameterization is needed. The maintenance of the process model is also simple, because modifications of the parent model block are copied spontaneously to all uniform model blocks.

Figure 3 illustrates the variation of the heat transfer efficiency along the tube. The water is single-phase liquid, when vapour fraction (or steam quality) is zero. When the saturation temperature is reached, bubbles start forming and vapour fraction increases. However, subcooled boiling is also possible, when the temperature of the main body of the liquid is below the saturation temperature. The saturated steam-water mixture flow can have several forms as Figure 3 presents. Steam core and liquid layer on a surface of a tube can be observed in an annular flow.
As the heating continues, the annular liquid layer gets thinner, and eventually disappears with the result that the heat transfer efficiency decreases sharply in the dry-out point, if the heating continues, saturated steam becomes superheated (vapor fraction is 1). Heat transfer capacity can be determined using single-phase convection heat transfer coefficient when liquid water is subsaturated or steam is superheated.

The heat transfer coefficient for single-phase liquid water and superheated steam is calculated using the well-known Dittus and Boelter equation for forced convective heat transfer in turbulent flows and subcritical pressures, Pioro et al. (2004). Several correlations for saturated boiling heat transfer can be found in literature. Our model applies Gungor and Winterton correlation, Webb (2003), for subcooled boiling and saturated boiling. Near supercritical pressure the nuclear boiling heat transfer is calculated using the Thom correlation similarly as Hänninen and Kurki (2008). The heat transfer for the mist flow between the dry-out point and the superheating region is supposed to be the same as the saturated boiling correlation, even if there are several empirical correlations estimating the heat transfer of the mist flow. The model is simplified, because too detailed a model structure would be unnecessary in this purpose.

The general model block must also be able to represent the behaviour of the supercritical water. A supercritical fluid is at pressure and temperature higher than the values of the critical point. For water, the critical point occurs at 374.1 °C and 22.1 MPa. A fluid that is at a pressure above the critical pressure and at a temperature below the critical temperature is called a compressed fluid. The model uses the heat transfer coefficient for forced convection above the critical point by Kitoh et al. and Filenenko et al. (2004a).

The pressure drop in subcritical pressures is determined using correlations by Claxton et al. (1972). The pressure drop of the supercritical water is made according to the equations presented in the survey by Pioro et al. (2004b). In general, total pressure drop consists of frictional, local, accelerational and elevational pressure losses. The model uses equations for the frictional coefficient by Filenenko and the pressure drop due to gravity by Ornatskiy et al. and Razumovskiy, Pioro et al. (2004).

As mentioned earlier, the values of the heat transfer coefficient and the pressure loss depend on the phases of water. This will cause inaccuracy to simulation results, if it is neglected. For example, if the model block represents a long tube, where entering steam-water mixture is totally vaporized and superheated inside the block, the calculated estimate is too rough if it is assumed that water has the same average temperature and pressure throughout the pipe. The model uses average state values, which are determined using inlet and outlet pressures and enthalpies of water. The heat transfer calculation uses also average values of water properties. This means that only one heat transfer coefficient and pressure drop calculation represent the whole pipe. One way to solve the problem is that the long pipe is modelled using multiple blocks, which describes short parts of the long pipe. In this way, specific heat transfer coefficient and pressure drop can be calculated for each block. However, a large number of model blocks decreases the simulation speed. The dependence of the number of the blocks to simulation times has been examined by Yli-Fossi et al. (2011). The simulation speed of the dynamic simulator is, especially from the point of view of the usability, a factor which is significant. The next chapter presents how the accuracy of the model can efficiently be increased when the block is only divided into some sections.

4. SLIDING BOUNDARY

The general model block is divided into two sections able to use different calculations and correlations for each water phase. However, the location of the boundary between the phases effects on the accuracy of the model. Figure 4 illustrates the modelled tube with two heat transfer sections. For instance, the first section represents saturated boiling heat transfer and the second section superheated steam heat transfer.
The specific heat transfer coefficients and pressure drop are calculated for both sections. The location of the boundary between the two sections can be calculated using energy balance

\[ \dot{Q}_{\text{sec1},w} = \dot{Q}_{\text{sec1},\text{out}} - \dot{Q}_{\text{in}} = \dot{m}(h_{\text{sec1},\text{out}} - h_{\text{in}}) \quad (1) \]

where \( \dot{Q}_{\text{sec1},w} [\text{J/s}] \) is the convective heat transfer between the tube inner wall and the flowing bulk in the first section. This example, the bulk is saturated steam-water mixture in the first section, and superheated steam in the second section. \( \dot{Q}_{\text{in}} [\text{J/s}] \) is the heat power of the inlet mass flow to the tube. \( \dot{Q}_{\text{sec1, out}} [\text{J/s}] \) is the heat power of the outlet mass flow from the first section to the second section. \( h_{\text{in}} [\text{J/kg}] \) and \( h_{\text{sec1, out}} [\text{J/kg}] \) are the enthalpies of the bulk in to the tube and out from the first section. The mass flow \( \dot{m} [\text{kg/s}] \) is assumed the same in both sections.

The steady rate of the convection heat power can also be expressed as [Cengel, 2002]

\[ \dot{Q}_{\text{sec1},w} = \alpha_{\text{sec1}} A_{\text{sec1}} (T_{\text{sec1},w} - T_{\text{sec1,b}}) \quad (2) \]

where \( \alpha_{\text{sec1}} \) is the convective heat transfer coefficient [W/m\(^2\)•°C] of the first section. \( T_{\text{sec1},w} [\text{°C}] \) and \( T_{\text{sec1,b}} [\text{°C}] \) are the temperatures of the inner wall and the bulk of the first section. In this example, \( T_{\text{sec1,b}} \) can be determined as the saturation temperature. \( A_{\text{sec1}} \) is the convective heat transfer area. In this case, it is the inner surface area of the tube, and it can be written as

\[ A_{\text{sec1}} = 2\pi r L_{\text{sec1},n} \quad (3) \]

where \( r [\text{m}] \) is the inner radius of the tube, \( L_{\text{sec1}} [\text{m}] \) is the length of the first section and \( n [-] \) is the number of tubes. Thus, the location of the boundary between the sections equals \( L_{\text{sec1}} \). When, equations (1), (2) and (3) are connected, \( L_{\text{sec1}} \) can be calculated as

\[ L_{\text{sec1}} = \frac{\dot{m}(h_{\text{sec1, out}} - h_{\text{in}})}{\alpha_{\text{sec1}} 2\pi r m (T_{\text{sec1},w} - T_{\text{sec1,b}})} \quad \text{if } 0 < L_{\text{sec1}} < L_{\text{tot}} \quad (4) \]

Even though \( L_{\text{sec1}} \) can be changed during simulation run, the total heating length \( L_{\text{tot}} \) is the constant parameter of the general model block. Thus, the length of the second section can be calculated as

\[ L_{\text{sec2}} = L_{\text{tot}} - L_{\text{sec1}} \quad (5) \]

If the general model block represents same type heat transfer calculation in the both sections, for example the first and second section using superheated steam heat transfer calculation, then \( L_{\text{sec1}} \) cannot be calculated using (4). In that case, \( L_{\text{sec1}} \) and \( L_{\text{sec2}} \) are determined to be equal. Thus, the general model block includes two sections with their own heat coefficients and other variables such as wall temperatures although water phases are the same in the both sections.

The majority of the parameters of the general model block are constant. However, some parameters, such as mass and heating area, are changed in accordance with length in the sections. Therefore, the length of the first section \( L_{\text{sec1}} \) must be transmitted to the second section. The parameters dependent on the length are calculated during each simulation step if necessary. Figure 5 illustrates how variables are transmitted from the first section to the second section.

![Fig. 5. The general water side model block is divided the two sections.](image)

The location of the boundary between other phases of bulk, such as the superheated steam and supercritical fluid, can also be calculated using (4) and (5). The average temperature \( T_{\text{sec1,b}} \) of the bulk in the first section, can be determined using the average enthalpy [J/kg] of the bulk in the first section

\[ h_{\text{sec1,b}} = \frac{(h_{\text{in}} + h_{\text{sec1, out}})}{2} \quad (6) \]

and the average pressure [Pa] of the bulk in the first section

\[ p_{\text{sec1,b}} = \frac{(p_{\text{in}} + p_{\text{sec1, out}})}{2} \quad (7) \]

where \( p_{\text{in}} [\text{Pa}] \) and \( p_{\text{sec1, out}} [\text{Pa}] \) are the pressures of the inlet and the outlet of the first section. If in question the location of the boundary between superheated steam and supercritical fluid, then \( p_{\text{sec1, out}} \) is the same as the critical pressure

5. SIMULATION EXAMPLES

The short part of the large membrane wall furnace has been modelled for a simulation example. The heating length of the modelled part is 10 m, and it consists of several parallel tubes, where water flow is upwards. For the sake of comparison, modelling has been implemented in three different ways. The first case is that the furnace wall is modelled using the twenty sections of general water side blocks. The modelled length of each block is 0.5 m by default, but the lengths of the blocks can be changed according to (4) and (5), if the boundary between the saturated steam-water mixture and the superheated steam is detected during the simulation run. In the second case, the model consists of four sections with the constant length, 2.5 m. The third case was that the model included also the two sections as the previous way, but the length of the blocks was calculated in the same way as in the first model. The most accurate way to model heat transfer coefficients and water and wall temperatures through the part of the furnace wall was the first way, because the first model was divided into more sections than the second and third model. Thus, the first model can be used as a reference of this case.
Figures 6, 7, 8 and 9 illustrate simulation results of the three models in the steady-state situation. Input parameters such as the enthalpy of inlet mass flow, input pressure, mass flow and heat flux form the gas side were constants during the simulation run. Figure 6 presents the average value of vapour fraction in each section and the output value of vapour fraction from each section. The values were calculated using the average and output enthalpies and pressures from each section. In the last part of the wall saturated steam-water mixture heats up to superheated steam between the two last sections. Figure 6 illustrates the results from the three models. The second model divided the vapour fractions to four sections with equal lengths. The locations of the boundaries between the two last sections of the first and third model are different from those of the second model. Thus, the first and second models applied (4) and (5).

Figure 7 shows the heat transfer coefficient through the parts of the furnace wall. The heat transfer coefficient drops strongly between the boundary between the saturated steam-water mixture and the superheated steam. The locations of the boundaries are nearly the same the first (reference) model and the third model, even the third model using only four sections. Figure 8 presents the modelled average and output temperatures of the saturated steam-water mixture and the superheated steam from each section. It seems that the differences of the average temperatures between the first and the third models are smaller than between the first and the second model. However, the output temperatures are almost the same for all three models.

Figure 9 presents the wall inner surface temperatures. The only average temperature in direction of the heating length can be calculated through each sections. The third model gives more accurate results than the second model in comparison with the first model.

The calculation of the boundary location is static, but temperatures and enthalpies have been modelled using differential equations. The step response test for the calculation of the boundary is shown in Figure 10. The same
first and third models were used in this test as in the previous example. Figure 10 presents the models responses for the step changes in heat flux and mass flow to the position of the phase boundary. The dynamic behaviour can be seen as the location of the boundary, which is caused by the dynamic changes of temperatures and enthalpies. Figure 10 indicates that the simpler model (the third) estimates the location of the boundary almost the same place as the reference model (the first) during the simulation run. However, the dynamic behaviour of the model depends on the number of the blocks. More model variations were tested by Yli-Fossi et al. (2010).

The length of the first section decreases, if heat flux increases. Thus, superheated steam volume increases in the tubes, then the length of the steam-water mixture section decreases. The length of the first section increases when the mass flow through the wall tubes increases. The reason is that the water temperature decreases. Thus, the same heat power transferred from the gas side cannot keep the water temperature in the original level because of the increased mass flow.

The simulation test demonstrates that the location of the boundary between different water phases can be determined using (4) and (5), even when the modelled process is divided into a few model sections. The presented sliding boundary calculation is also accurately than if the boundaries between sections would be kept constant.

6. CONCLUSIONS

The dynamic model of CFB-OTU boiler was modelled. The large and complex process model is challenging from the point of view of maintenance and simulation speed. The water side was built using the presented general water side model block with sliding boundary calculation. The general model type keeps the maintenance easy. The presented sliding boundary calculation has been tested with simulation runs. It seems that the model block is workable and efficient. However, the development work is still continues.

REFERENCES


