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Tuning of a dynamic boiler model using a nonlinear multivariable optimisation method

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Abstract: Wide and complex process models set challenges to the modelling work. Especially the determination and tuning of the parameters of complex models are often laborious and time-consuming. The strong cross-interconnection of modelled variables also makes tuning work more difficult. Efficient tuning tools can be used to accelerate tuning work. In this paper it is proposed how a nonlinear multivariable optimisation method can be adapted for the tuning of the parameters of a dynamic circulating fluidized bed (CFB) drum boiler model, which based on the first principle laws of mass and energy. In the fine-tuning example, the tuneable variables are the gas side heat transfer coefficients and the water side local resistance coefficients of the heat exchanger model blocks into the flue gas duct.

Keywords: Dynamic models, Modelling, Optimisation, Circulating Fluidized Bed Boiler

1. INTRODUCTION

Software environments for modelling include usually ready model block libraries or the opportunity for the doing of them. Model block libraries can contain individual process components models for example from tanks, valves, pumps and heat exchangers. The building of wide models accelerates by using model block libraries. It is easy to connect the ready submodel blocks into wider process models. However, the values of the fixed parameters of model blocks have to be determined separately to correspond to the process to be modelled. On wide and complex process models the number of the fixed parameters can be very large in which case the definition work and tuning work may become laborious and may delay modelling work significantly. Interconnections between modelled variables are also complicated by the tuning work (Halmevaara *et al.*, 2007, Yli-Fossi, 2013).

The most fixed parameters of boiler models can usually be determined straightforward such as dimensions and natural constants. In this context, the term fixed parameter determines a constant is passed to a model before a simulation run. The typical fixed parameters of a process model are information which is related to the dimensioning of processes and can be for example lengths and diameters of pipes and heat transfer areas of heat exchanger. Usually the properties of the structures are supposed to be constant.

The determination of the initial values of the state variables may be also laborious. The suitable values can be calculated or the model can be run to a desired state. The opportunity for the saving and loading of state values facilitates simulation work.

To improve the modelling accuracy, some of the parameters may have to be trimmed based on the measured process data or other information. To facilitate model parameter tuning work different tuning methods have been applied to dynamic power plant models. A method, called iterative Tuning (IRT), has been used to tune heat transfer coefficients of a reheater model of a nuclear power plant (Halmevaara *et al.*, 2007). A nonlinear optimisation algorithm that minimizes the quadratic cost function is also used in an other nuclear power plant model by Fazekas and Varga, 2008. A circulating fluidized bed boiler model has been analysed and tuned using a particle filtering method by Ikonen *et al*, 2012. The algorithm of the integral controller has also been applied into the fine-tuning of the gas side heat transfer coefficients of a boiler model (Yli-Fossi et al., 2012).

The most significant challenges of the tuning of dynamic power plant models are the strong interconnections between process variables, nonlinearity and time variant characteristics. Power plant models usually describe water and flue gas flows and heat transfer between water and flue gas. The temperature differences and flow rates affect the transition of the heat. The variables depend strongly on each other. The nonlinearities are caused by the behaviour of heat transfer coefficients. The water side heat transfer coefficient depends on the properties of water and the flow rate strongly. In the same way, the properties of flue gas and flow rate affect also heat transfer from flue gas to heat transfer surfaces. Furthermore, the time variant behaviour can be caused by the variation of the properties of the fuel. (Yli-Fossi, 2013)

In this work it is presented how the model can be tuned with an optimisation method which considers to the cross-interactions and nonlinearity of the model.

2. DYNAMIC PROCESS MODEL

A dynamic process model of a circulating fluidized (CFB) drum boiler has been developed. The model represents the real boiler process from the preheaters to the turbine. The presented process 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 and Matlab by The MathWorks. The main goal is that the dynamic model can be used for several purposes such as control design and process development.

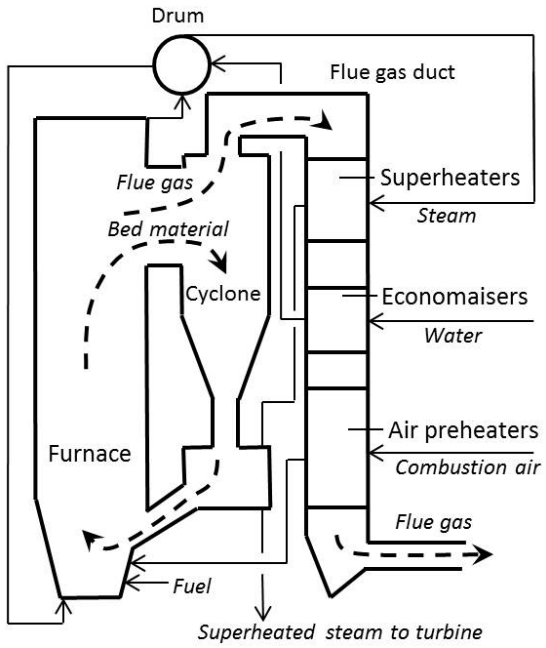

Fig. 1. A simplified diagram of a natural circulation drum boiler process.

The developed model represents a natural circulation drum boiler. Figure 1 presents a simplified diagram of a drum boiler boiler process. 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 of the furnace walls. 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. 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)

The process model includes also the dynamics of the combustion and the fluidized bed material. The different types of fuels can be modelled using the combustion model. In the furnace, the propagation of combustion process and heat transfer depends on the amount of circulating mass and suspension density. Heat flow from the flue gases and solid matter to the furnace walls is transferred by convections and radiation. Solid matter and gases are separated from each other in cyclones. Solid matter flows back into the furnace through the loop seal. In the loop seal heat transfer also takes place to the walls and other heat exchanger elements. From the cyclone flue gases flow to the flue gas duct section of the boiler. (Majanne and Köykkä, 2009)

The flue gas duct includes several heat exchanger elements such as economisers, superheaters, and air preheaters. In the modelled process, the water from the feed water tank to the drum is preheated by the economisers (ECO1, ECO2 and ECO3). The primary (PSH1 and PSH2) and secondary (SSH1 and SSH2) superheaters increase steam temperature before the turbine. Superheated steam temperature is controlled using spray water. Air preheaters preheated combustion air to the furnace. The wall tubes of the flue gas duct vaporized water to the drum as well as the furnace walls. Thus, heat is transferred from flue gas to heat exchangers and the walls of the flue gas duct. The flue gas temperature drops along the duct. Heat flows from the flue gas to water, steam and air are interacting with each other.

The presented power plant model consists of several blocks which have been divided into blocks, water, gas and air. The water side of the process model is modelled using several identical water side model blocks in which the values of parameters can be different. The used water side block able to handle the thermodynamics of water, saturated water-steam mixture, and superheated steam under sub- and supercritical pressures. The developed water side model block includes also the walls or the tubes of heat exchangers. The block can be used represent the water sides of different process components such the parts of an evaporator, superheaters and economisers (Yli-Fossi *et al.*, 2011). The flue gas and air sides were also modelled using model blocks. The separate model blocks have been developed for a furnace, cyclones, a loop seal, and a flue gas duct and air preheaters. (Yli-Fossi et al., 2012).

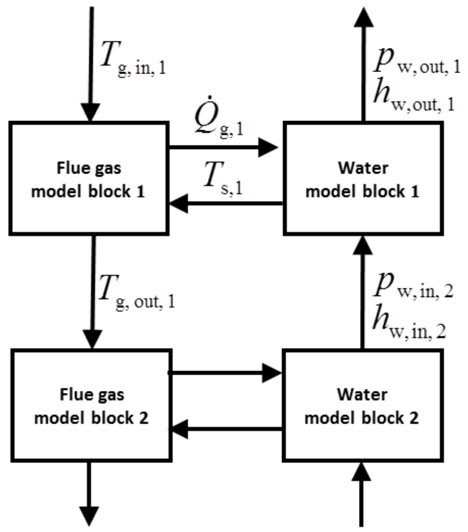

Fig. 2. A simplified example diagram of sub model blocks and some signals of two heat exchanger models.

Figure 2 illustrates a simplified example diagram of two heat exchangers of water or steam into a flue gas duct. Each heat exchanger, such economaiser or superheater, has been modelled on two model blocks: gas and water model blocks. These blocks have been connected to each other using signals. The heat exchanger 1 has been connected to the heat exchanger 1. Only some of the variables that contain the signal vectors are marked in Figure 2. For example, $\dot{Q}_{g,1}$

means the heat power [J/s] from the gas side to the walls of the heat exchanger 1. The heat power depends several variables such flue gas temperature, wall temperature and gas side heat transfer coefficient, which means heat transfer coefficient between flue gas or suspension matter and the wall of a heat exchanger. The heat power signal in turn has effect back to the surface temperature $T_{s,1}$ [K] and the gas temperature $T_{g,1}$ [K]. The water enthalpy $h_{w,1}$ after the heat exchanger is also dependent on the heat power $\dot{Q}_{g,1}$. The consecutive model blocks are to the series connected. As its consequence, that variations of pressure $p_{w,1}$, enthalpy $h_{w,1}$ and mass flow are conveyed forward in water model blocks.

However, some of the physical parameters may have to be tuned. For example it is difficult to determine gas side heat transfer coefficients and coefficients of local resistance to water and steam flows exactly with physical equations. This reason, these coefficients were chosen the tuneable parameters of the presented model to be tuned in this work. Furthermore, the significance of the coefficients affected a choice.

Gas side heat transfer coefficients are small and flue gas dynamic is complicated (Hubka, 2011). Because of this the coefficients tuned in the model. The coefficient affects the heat transfer heat transfer $\dot{Q}_{g,i}$ from the flue gas to the surface of the tubes of the heat exchanger i in the flue gas duct. The variable i is the index of the heat exchanger. The heat transfer heat transfer can be calculated as (Incropera and DeWitt, 2001)

$$\dot{Q}_{g,i} = \alpha_{g,i} A_{ht,i} \frac{(T_{g,in,i} - T_{s,i}) - (T_{g,out,i} - T_{s,i})}{\ln\left(\dfrac{T_{g,in,i} - T_{s,i}}{T_{g,out,i} - T_{s,i}}\right)} \qquad (1)$$

where $\alpha_{g,i}$ [W/(Km^2)] is the gas side heat transfer coefficient and $A_{ht,i}$ [m^2] is the heat transfer area. $T_{s,i}$ [K] is the temperature of heat transfer surface as the tube walls of the heat exchanger. The $T_{g,in,i}$ [K] is the temperature of the inlet flow of the flue gas to the modelled section. $T_{g,out,i}$ [K] is the temperature of the outlet flow.

The gas side heat transfer coefficient $\alpha_{g,i}$ is can be determined as

$$\alpha_{g,i} = a_{tuning,i}(\alpha_{g,conv,i} + \alpha_{g,rad,i}) \qquad (2)$$

where $\alpha_{g,conv,i}$ [W/(Km^2)] is convective heat transfer coefficient and $\alpha_{g,rad,i}$ [W/(Km^2)] radiant heat transfer coefficient. In the model both coefficients are calculated on equations which are found in the literature. A more exact description of the calculation is found of an earlier publication (Yli-Fossi et al., 2012). $a_{tuning,i}$ [-] is the additional fine-tuning coefficient of the gas side heat transfer coefficient. The default value of the coefficient is 1. The coefficients $a_{tuning,i}$ of each heat exchanger models were specified as parameters to be tuned.

The coefficients of the local resistance of the pipe system pressure drop are also difficultly determined if the system is complex. The total pressure drop $\Delta p_{w,i}$ [Pa] for forced convection water, steam or saturated steam-water mixture flow can be determined as (Pioro *et al.*, 2004)

$$\Delta p_{w,i} = \sum \Delta p_{w,ff,i} + \sum \Delta p_{w,l,i} + \sum \Delta p_{w,ac,i} + \sum \Delta p_{w,g,i} \qquad (3)$$

where $\Delta p_{w,ff,i}$ [Pa] is the pressure drop due to frictional resistance, $\Delta p_{w,l,i}$ [Pa] is the pressure drop due to local flow obstruction, $\Delta p_{w,ac,i}$ [Pa] is the pressure drop due to acceleration of flow and $\Delta p_{w,g,i}$ [Pa] is the pressure drop due to gravity.

The pressure drop $\Delta p_{w,l,i}$ due to local single-phase flow obstruction of water and steam can be calculated as (Pioro *et al.*, 2004)

$$\Delta p_{w,l,i} = \xi_{w,l,i} \frac{G_{w,i}^{\ 2}}{2\rho_{w,i}} \qquad (4)$$

where $\xi_{w,l,i}$ [-] is the local resistance coefficient $G_{w,i}$ is mass flux [kg/(m^{2}s)] and $\rho_{w,i}$ [kg/m^3] is the density of water or steam.

The pressure drop $\Delta p_{w,l,i}$ due to local two-phase flow obstruction of saturated steam-water mixture can be defined as (Pioro *et al.*, 2004)

$$\Delta p_{l,i} = \xi_{l,i} \frac{G_{w,i}^{\ 2}}{2\left\{\rho_{w,l,i}\left(1 - \varepsilon_{w,r,average,i}\right) + \rho_{w,s,i}\varepsilon_{w,r,average,i}\right\}} \qquad (5)$$

where $\rho_{w,l,,i}$ [kg/m^3] density of (liquid) water and $\rho_{w,s,,i}$ [kg/m^3] density of steam. $\varepsilon_{w,r,average,i}$ [-] is the average void fraction.

The local resistance coefficient $\xi_{w,l,i}$ can be estimated from appropriate correlation for different flow obstructions. The local resistance coefficients affect local pressure drops, which are caused by a change in flow geometry and flow direction. Examples branches, valves and bends drop pressure locally. Reference values and equations for local resistance coefficients are found in the literature. However, the ability to estimate to local pressure drops $\Delta p_{w,l,i}$ of the complex pipe system is very difficult than for other pressure drops $\Delta p_{w,ff,i}$, $\Delta p_{w,ac,i}$ and $\Delta p_{w,g,i}$. Therefore, the coefficients $\xi_{l,i}$ of heat exchangers are considered as parameters which are experimentally tuned in this work. The coefficients are estimated before the tuning. The coefficients are possible to tune using the equation

$$\xi_{l,i} = b_{tuning,i}\xi_{l,i} \qquad (6)$$

where $b_{tuning,i}$ [-] is the additional fine-tuning coefficient of the local resistance coefficient. The default value of the coefficient is 1.

The values of coefficients affect the local pressure losses $\Delta p_{w,l,i}$ and total pressure drops $\Delta p_{w,i}$ of each heat exchangers i.

3. NONLINEAR MULTIVARIABLE OPTIMASATION

The model tuning problem can be usually formulated the optimisation problem. Because the boiler models are usually nonlinear and the models contain multivariable interactions, nonlinear multivariable optimisation is an obvious alternative. In this work, the optimisation problem is solved using the Matlab function *fminsearch*, which find the minimum of the cost function. The function is based on the multidimensional unconstrained nonlinear Nelder-Mead simplex method. This efficient derivative-free direct search method optimizes the cost function merely by comparing function values. The basic algorithm of the method is quite simple to understand and easy to program and use. It is also popular in many fields of science and technology. (Lagarias *et al.*, 1998, Chang, 2012) Mainly for these reasons the Nelder-Mead simplex method was chosen for this work. Also some other optimisation methods were tested with the model.

4. MODEL FINE-TUNING

In the fine-tuning example, the gas side heat transfer coefficient $\alpha_{g,i}$ and the local resistance coefficient $\xi_{l,i}$ of the heat exchangers ECO1, ECO2, ECO3, PSH1, PSH2, SSH1 and SSH2 of the presented drum boiler model are tuned using transfer fine-tuning coefficients $a_{\text{tuning},i}$ and $b_{\text{tuning},i}$. The heat exchanger model blocks are indexed to the same order as above: the index i of ECO1 is 1, the index i ECO2 is 2, etc. The last heat exchanger SSH2 is the index i is 7. The tuneable parameters are the manipulated input variables of the optimisation problem. Temperatures $T_{\text{g,out},i}$ or enthalpies $h_{\text{w},i}$ or heat powers $\dot{Q}_{\text{g},i}$ and pressures $p_{\text{w},i}$ and their combinations can be chosen as the controlled variables. The choice is affected by the fact whether the values of the controlled variables to be reached for are known as in the ones calculated or to be measured from the process to be modelled. The selected target values of the controlled variables are individual averages from a certain period of the time in the steady state situation. In this tuning case, the values of the main part of temperatures $T_{\text{g,out},i}$ and $T_{\text{w,out},i}$ have been measured in a certain steady state situation of the boiler process. $T_{\text{w,out},i}$ [K] means the water or steam outlet temperature from the heat exchanger i. $T_{\text{w,out},i}$ can be calculated from pressures $p_{\text{w,out},i}$ and enthalpies $h_{\text{w,out},i}$ in the model. The values of the pressures $p_{\text{w},i}$ are less measured. Target values or setpoints are not been measured direct are based on the calculated values using a design calculation. The calculated values can also be determined indirectly using measured values. Thus, the target values of the optimisation case are measured values and calculated design values. When generally speaking choosing the target values, one must ensure to the reliability of the reference values.

The cost function J to be minimised is defined as

$$J = e_{\text{T}} + e_{\Delta\text{p}} \tag{7}$$

where e_{T} is a squared error of the water or steam outlet temperatures of the heat exchanger model blocks. The error e_{T} can be defined as

$$e_{\text{T}} = \sum_{i=1}^{7} k_{\text{T},i}\left(\hat{T}_{\text{w,out},i} - T_{\text{w,out},i}\right)^2 \tag{8}$$

where $k_{\text{T},i}$ is the vector of the weight coefficients and $\hat{T}_{\text{w,out},i}$ is the vector of the target values of the water or steam outlet temperatures. e_{p} is a squared error of the outlet pressure of the heat exchanger model blocks. The error e_{p} is written as

$$e_{\text{p}} = \sum_{i=1}^{7} k_{\text{p},i}\left(\hat{p}_{\text{w},i} - p_{\text{w},i}\right)^2 \tag{9}$$

where $k_{\text{p},i}$ is the vector of the weight coefficients and $\hat{p}_{\text{w},i}$ is the vector of the target values of the total water or steam pressure drop of the heat exchanger model blocks.

The process model is tuned to correspond to a certain steady state situation. The model and the optimisation function interact with each other during the tuning simulation, so the minimisation of the cost function is more difficult in the strong transient situation than the steady state situation. The subject of the presented tuning work is not the dynamic behaviour of the model. If necessary, the model can be tested in different operation points if the comparison data is available. The dynamic validation must be made separately. To suppose also this is data available from transient situations. The optimisation function is performed in parallel with the process model during the tuning procedure. The optimisation function changes the manipulated variables, $a_{\text{tuning},i}$ and $\xi_{l,i}$, with the intervals of 180 simulated seconds. The changes are made at long intervals because the model is dynamic. The optimisation function must perceive the directions of the controlled variables caused by the changes of the manipulated variables.

Before the tuning simulation, the process model is simulated against to the steady state situation like values that have been measured from the process. Some controlles and variables set manual and constant during the tuning simulation run. Otherwise for example the steam temperature controllers disturb optimisation. The value of each weight coefficients $k_{\text{p},i}$ is 1 and the value of each weight coefficients $k_{\text{p},i}$ is 0.3. Thus, the values of the temperature were set as the most important tuning targets. The units that have been used in the cost function are [ºC] and [bar].

Figure 3 shows the value of the cost function during the tuning simulation. The tuning simulation begins at time 0. The value of the function becomes smaller considerably during the simulation.

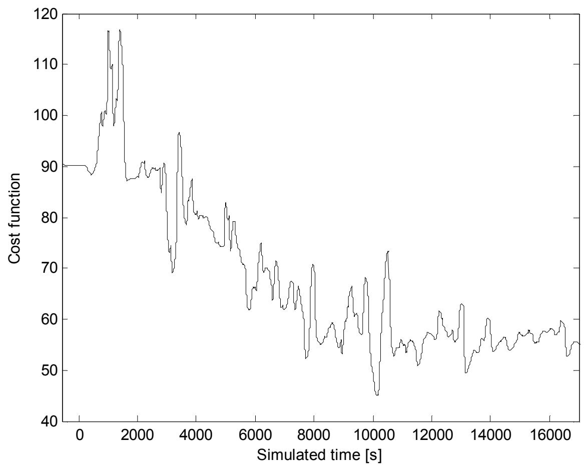

Fig. 3. The cost function during the tuning simulation.

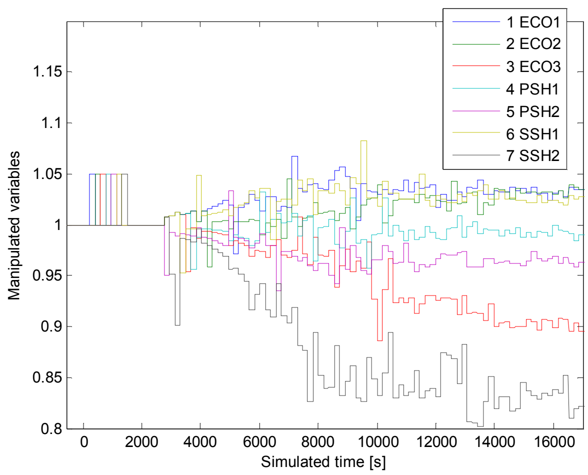

Fig. 4. The manipulated variables (the gas side heat transfer fine tuning coefficients) during the tuning simulation.

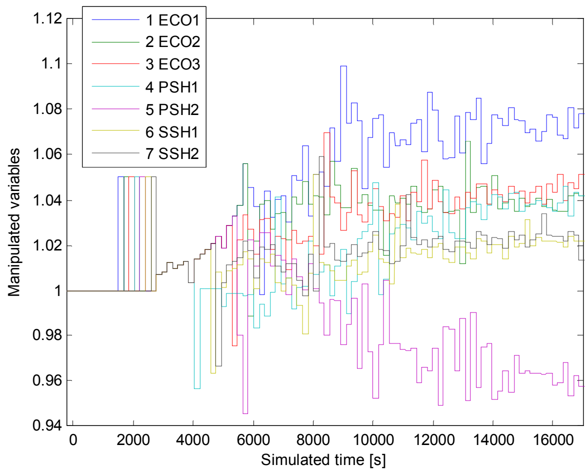

Fig. 5. The manipulated variables (the local resistance fine tuning coefficients) during the tuning simulation.

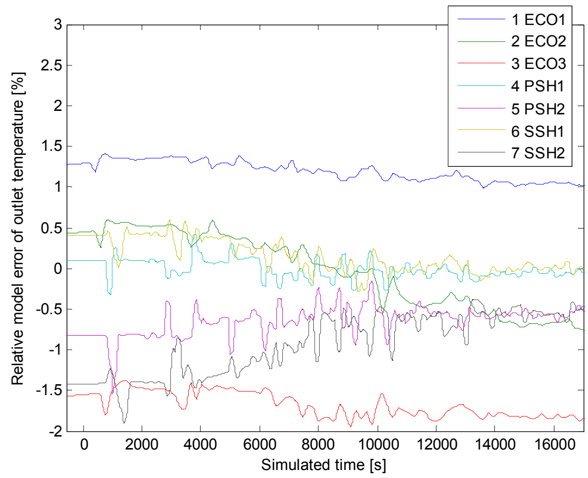

Fig. 6. Relative model errors of the water or steam outlet temperature of the heat exchanger model blocks during the tuning simulation.

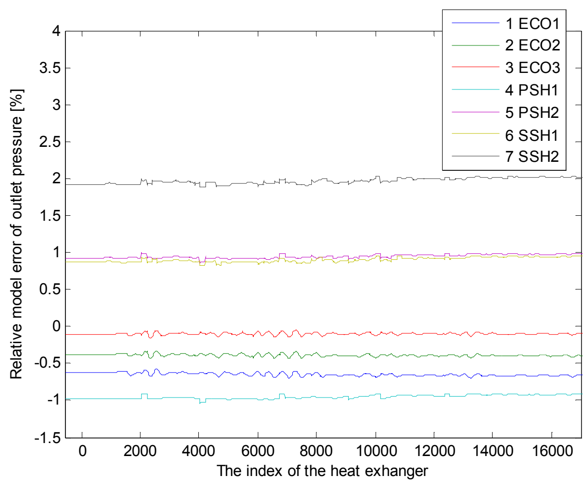

Fig. 7. Relative model errors of the water or steam outlet pressure of the heat exchanger model blocks during the tuning simulation.

Figure 4 presents how the gas side heat transfer fine tuning coefficients changed during the tuning simulation. At the beginning of the simulation, the optimisation function will test all the manipulated variables one by one with small changes. After the testing more variables are changed the same time. Correspondingly Figure 5 illustrates the changes in the local resistance fine tuning coefficients.

Figure 6 shows relative model errors of the model blocks of water or steam outer temperature of each heat exchanger during the tuning simulation. The error becomes smaller in almost all the model blocks. However, the errors of the heat exchanger model blocks ECO1 and ECO3 were not reduced Figure 7 presents relative model errors of the water or steam outer pressure during the tuning simulation. The pressure changes were minor during the simulation. Because of this, the pressure coefficients were tuned once more separately based on Equation 9. The value of each weight coefficients $k_{p,i}$ is 1. The gas side heat transfer fine tuning coefficients remain constant during the second simulation.

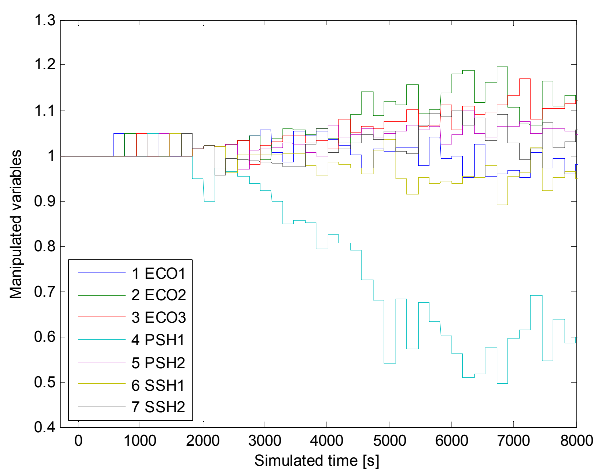

Fig. 8. The manipulated variables (the local resistance fine tuning coefficients) during the tuning simulation.

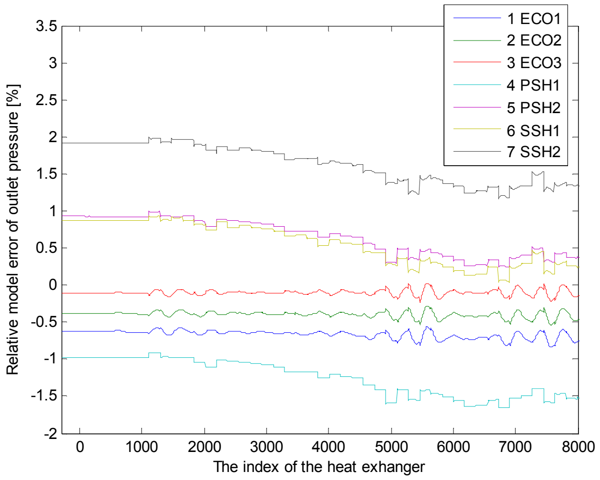

Fig. 9. Relative model errors of the water or steam outlet pressure of the heat exchanger model blocks during the tuning simulation.

Figure 8 presents the local resistance fine tuning coefficients during the second tuning simulation. The variation is stronger than in Figure 5. Figure 9 shows that some of the blocks of the relative model error is reduced by more than in Figure 7.

The result can be possibly be improved by changing the cost function, the weighting coefficients and the optimisation time step interval. The results are satisfactory, especially for the outlet temperatures. In this work, it was intended to minimise a certain process wholeness and not individual variables.

5. CONCLUSIONS

A dynamic process model of a circulating fluidized (CFB) drum boiler was presented. The gas side heat transfer coefficients and the local resistance coefficients of water side and of the heat exchanger model blocks of the flue gas duct are tuned using nonlinear multivariable optimisation method. The presented model contains multivariable interactions and nonlinearity. In the fine-tuning example demonstrated that the chosen method was reduced model error in this case. The method can be adapted to different models. However, the use of the presented fine tuning method requires that the tunable parameters of the target is possible to change during the tuning simulation run.

REFERENCES

Chang, K-H. (2012). Stochastic Nelder–Mead simplex method – A new globally convergent direct search method for simulation optimization, European Journal of Operational Research, Volume 220, Issue 3, 1 August 2012, 684–694.

Fazekas, C. and Varga, I. (2008). Modeling and Parameter Estimation of a Nuclear Power Plant , *9th International PhD Workshop on Systems and Control: Young Generation Viewpoint*, 1. - 3. October 2008, Izola, Slovenia.

Halmevaara, K. and Hyötyniemi, H. (2007). Dynaamisten simulointimallien parametrien virittäminen datapohjaisilla tilastollisilla menetelmillä, *Proceedings of Automaatio 07*, Finnish Society of Automation, Helsinki.

Hubka, L. (2011). Temperature Dynamic of Once-through Boiler Based on Flue Gases Heat Transports, *Preprints of the 18th IFAC World Congress*, 28.8-2.9.2011, Milano, Italy.

Ikonen E., Kovacs, J., Aaltonen, H., Ritvanen, J., Selek, I. and Kettunen A. (2012). Analysis and tuning of a CFB model using particle filtering, *Proceedings of the 8th IFAC Power Plant and Power System Control Symposium*, Toulouse, France, 2-5 September 2012.

Incropera, F.P. and DeWitt, D.P. (2001). *Fundamentals of Heat and Mass Transfer*. 5.p. USA, John Wiley & Sons.

Lagarias, J.C., Reeds, J. A., Wright, M. H. and Wright, P. E. (1998). Convergence Properties of the Nelder-Mead Simplex Method in Low Dimensions, *SIAM Journal of Optimization*, Vol. 9 Number 1, 112-147.

Majanne, Y. and Köykkä, P. (2009). Dynamic model of the circulating fluidized bed boiler, *Preprints of IFAC Symposium on Power Plants*, Tampere, Finland.

Pioro, I.L., Duffey, R.B. and Dumouchel, T.J. (2004). Hydraulic resistance of fluids flowing in channels at supercritical pressures (survey), *Nuclear Engineer and Design*, Vol. 231, 187-197.

Spliethoff, H. (2010). *Power Generation from Solid Fuels*,Springer-Verlag, Berlin Heidelberg.

Ylätalo, J. (2009), *Calculation models for two-phase flow in steam boiler tubes*, Bachelor of Science Thesis, Lappeenranta University of Technology, Finland.

Yli-Fossi, T., Köykkä, P. and Majanne, Y. (2011). A Generalized Dynamic Water Side Model for a Once-Through Benson Boiler, *Preprints of the 18th IFAC World Congress*, 28.8-2.9.2011, Milano, Italy.

Yli-Fossi, T., Köykkä P. and Majanne, Y. (2012). A Tuning Tool for Gas Side Heat Transfer Coefficients of a Boiler Model, *Proceedings of the 8th IFAC Power Plant and Power System Control Symposium*, Toulouse, France, 2-5 September

Yli-Fossi, T. (2013). Dynaamisten voimalaitosmallien parametrien määrittäminen ja virittäminen, *Proceedings of AutomaatioXX Seminar*, Finnish Society of Automation, 22 May 2013, Helsinki, Finland.