Impact of Large-Scale Wind Farms on the German Generation System - Part II

F. Gottelt*, S. Meinke*,
E. Hassel*, J. Nocke*

** Institute of Technical Thermodynamics*

University of Rostock
Albert-Einstein-Str. 2, 18059 Rostock, Germany
Tel: 0049-381-498-7106; e-mail: christian.ziems@uni-rostock.de

Abstract: Today, all over the world the number of installed wind turbines is increasing continuously. This causes more and more problems concerning the integration of the fluctuating wind power production into the existing electrical supply systems. To handle these problems the main aim for the future must be to guarantee the security of supply and the quality of electrical energy by taking the maximum advantage of the available wind potentials.
In this part of the paper, a method based on thermodynamical modelling is described to evaluate the limits and options of dynamic operation mode of coal-fired power plants.
The thermodynamic model is presented briefly followed by a validation of the model. In order to interpret actual loads i.e. of the thick-walled components of the steam generator two strategies of fatigue estimation based on guidelines of the Deutsche Dampfkessel Ausschuss (DDA) and the Forschungskuratium Maschinenbau (FKM) are proposed.

Keywords: Coal-fired power plant, simulation, Modelica, predictive control system, thermodynamic model

1. INTRODUCTION

As stated in the first part of this paper the market's demand of dynamically operating thermal power plants will rise in the next years. This fact raises the question of how the specifications of future plants must be changed and how existing power units can be retrofitted to match these future demands.
There will be a need of a certain number of fast reacting power stations which have to compensate the fluctuations of the wind supply. In the following sections a method is presented for evaluating and optimising the capabilities of thermal power plants to serve the upcoming demands in a new market which will be characterized by fluctuating supply of renewable energy sources.

2. MODEL OF THE POWER PLANT ROSTOCK

In order to judge the expected impacts of a more dynamic power plant operation a detailed model of an existing hard coal power plant based on thermodynamic fundamental equations has been developed. The reference for the model is the hard coal power plant of Rostock due to its long rest life time. Thus its operation mode is highly influenced by future market changes.
The focus of the investigation has been put on the water-/steam side of the plant, as well as its dynamics and the influence of different operation modes on the different devices e.g. thick-walled headers and turbine shafts.

Published on the 18. IFAC Symposium on Power Plants & Power System Control, 5th-8th of July 2009, Tampere/Finnland

2.1 *Thermodynamical Model*

A simplified schematic of the water-steam cycle is shown in figure 9. Indicated are the different turbine stages, the steam generator, feed water tank as well as the condensate- and feed water pumps. Additionally the high pressure (HP) and low pressure (LP) pre-heaters which are heated by the corresponding turbine tapings appear in a reduced manner.
The most elaborate part is the model of the steam generator whereas the implementation of the HP- and LP pre-heaters and their correlated tapping are more challenging from the numerical point of view.

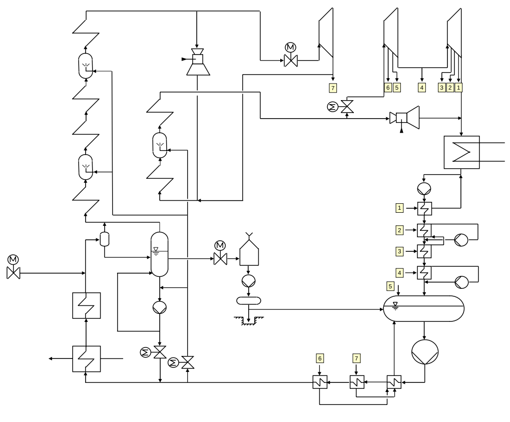

Fig. 9: Simplified schematic of a steam power plant

Describing the model details of each component is far beyond the scope of this paper but as a short overview the governing equations should be discussed for a control volume of discrete size with N in- and outlets in a general way.
The mass balance of an open system can be simplified to a stationary balance for components with a small volume (e.g. turbine stage).

$$\frac{dm}{dt} = \sum_{i=1}^{N} \dot{m}_i$$

Herein $\dot{m}_i$ denotes the i-th mass flow (negative sign for outgoing flows) and m the stored mass within the control volume. For heat exchanger and thick-walled components for which the wall temperature is said to be calculated the energy balance is of importance:

$$\frac{dU}{dt} = \sum_{i=1}^{N} \left(h_i + \frac{c_i^2}{2} + g\rho_i z_i \right) \dot{m}_i + \dot{Q} + \dot{W}_t$$

Where $U, h_i, c_i, \rho_i, z_i, \dot{Q}, \dot{W}_t$ represent the inner Energy of the control volume, specific enthalpy, velocity density and height of flow i, heat flow rate and technical power transferred at the system's borders, respectively. Usually the kinetic and potential energy are small compared to the enthalpies and thus may be neglected
Sonic wave effects in tubes and components are considered of minor importance during normal plant operation and axial change of momentum flow is small compared to the changes in different pressure term. Therefore the following general one dimensional momentum balance (considering the passed through surface A_i, one dimension of the unit vector orthogonal to A_i, n_i, a friction term Δp_f) can be reduced to the terms of pressure, friction and geostatic pressure:

$$\frac{d(\dot{m}c)}{dt} = -\sum_{i=1}^{N} A_i \rho_i c_i (c_i n_i) - \Delta p_f - \sum_{i=1}^{N} p_i A_i n_i - \sum_{i=1}^{N} g \rho_i z_i n_i$$

Each of this equation has to be solved for every component of the plant's model. These three balances are completed by mostly empirical formulas accounting for friction, heat transfer, phase separation and material data. The set of equation compose a large coupled system of differential-algebraic equations (DAE).

For making simulation-based statements about the influence of different power plant operation modes the thermo dynamical model is coupled to a reduced copy of the power plant control system.

The implemented control system uses the currently calculated physical values (i.e. live steam parameter, generated power at a specific coal input) and in a consequence adjusts set values (e.g. life steam pressure) and manipulated variables (e.g. rotational speed of the feed water pump) of the water-steam cycle. Because of this feedback the grade of details as well as the accuracy of the modelled steam cycle and its interfaces to the control system needs to be reasonably high to match the corresponding values of the referenced power plant Rostock.

In detail the power plant control system sets the manipulated values using a map based pilot control. The expected control variable is predicted by a transfer function based model of the process. The difference between this predictive value and the corresponding measurement is adjusted via a corrective control loop (VDI/VDE 3508, 2003).

The parameterisation of the power plant control system and the thermo dynamical model is conducted based on data of the power station Rostock.
Subsequently the modelling and validation process of the boiler is further explained, since the highest thermal stresses are expected here during the in-stationary operation.

The model of the boiler differentiates eight separate heating surfaces: economizer, evaporator, four super heaters and two reheaters. Between the super heaters S1 and S2 as well as S3 and S4 plus in between the reheaters R1 and R2, spray atemperators are located for live steam temperature control. The up- and downstream of the heating surfaces positioned inlet and outlet headers are also modelled and are of particular interest as the spatial and temporal temperature gradient limit significantly the allowed load gradient. Thus, the heat entering and outgoing heat flow is considered. The metal wall of the device is spatially discretised in radial direction, only. Wall temperature gradients in axial direction are neglected, due to unheated and well-insulated headers.

A physical and chemical modelling of the furnace regarding the fuel conversion and the heat transfer via radiation and convection is excluded The heat release at furnace side is simplified represented by a characteristic diagram. The heat surface specific fraction of the released heat is plotted in dependence of the applied fuel mass flow rate.
The modelling is conducted in Modelica (Casella *et al.*, 2003, Casella *et al*, 2005, Fritzson, 2004) using the simulator Dymola®. The modelling with Modelica is characterised by its modular concept.

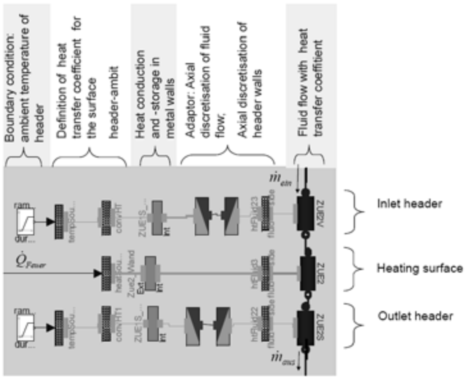

Fig. 10: Dymola view of a generic reheater which distinguishes inlet and outlet header and heating surface

Thus, the superheater is represented by several partial models: The one-dimensional pipe flow and the convective heat transfer occurring at the inner wall are reproduced by a module, the conductive heat transfer inside the header wall by an additional module. The heat flow at the system boundary between these two modules is implemented by a so called connector. This couples the two separate partial models to one device. In order to attach modules with connectors of different dimensionality, adapter modules are available. The module of a reheater is shown in figure 10.

2.2 Validation of the Boiler

In order to check the created model on correctness, comparative measurements during normal operation are recorded in the power plant Rostock, which are shown below in contrast to the results of the simulation.
In the investigated period of time the load changes multiple times between 55 and 95 percent. The development of the generator power shown in figure 11 is a typical load profile of the power plant Rostock

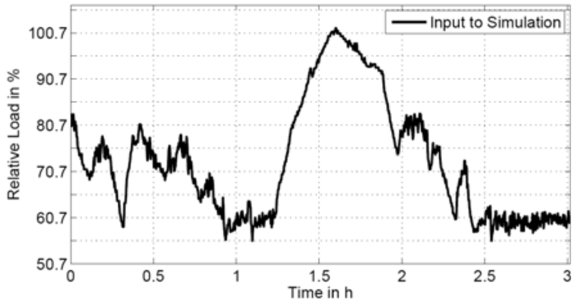

Fig. 11: Definition of validation setup - generator output fluctuates corresponding to the TSO's demands

As boundary conditions for the boiler in the validation setup measurements of live steam pressure, mass flow rate and temperature at boiler inlet are used to simulate the cropped parts of the water steam cycle. In this way it is possible to validate the steam generator's model independently from the rest of the cycle and to check the correctness of the made assumptions. Among these assumptions the representation of the furnace via a characteristic diagram that generates the proportion of the totally released energy transferred to each heating device is probably the most critical and will be examined exactly.

The economiser inlet pressure and the live steam mass flow arise from the current heating by the furnace, the cooling from the working medium due to the feed by the feedwater pump. In figure 12 simulated and measured pressures at inlet of the boiler are compared. From the good conformity (the maximum relative error is less than 3%) of both graphs a good reproduction of the hydraulic characteristic can be stated. Note that the sample rate of measurements is not constant but is triggered by actual gradient of the value in order to reduce the disc storage demand. This leads to the characteristic block-shaped graph of figure 12.

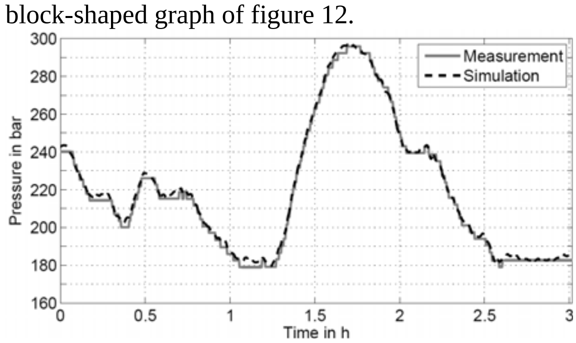

Fig. 12: Feedwater pressure slides during variable pressure operation

As the furnace and its combustions processes are represented via a characteristic diagram it is important to check this approach for suitability.
For this purpose calculated steam temperatures at the outlet header of each heating device are compared to the corresponding measurement. In the following some representative plots are discussed.
In figure 13 the steam temperature in the separator after evaporator is brought out with respect to time.

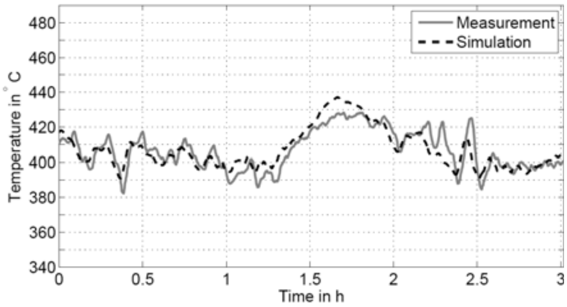

Fig. 13: Comparison of calculated and measured steam temperatures at the separator

The simulation shows a good agreement with the measurement – relative errors reach their maximum at about 2.5%.

The trend of the temperature after super heater S1 shows analogue results, hence the simplified furnace model is capable to emulate the heat flow rates transferred to the different heating areas and their temporal behaviour (see figure 14).

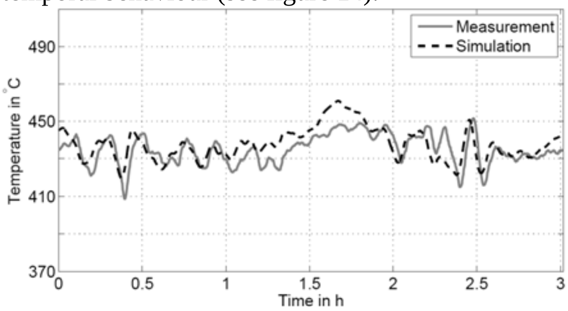

Fig. 14: Comparison of calculated and measured steam temperatures after the first super heater

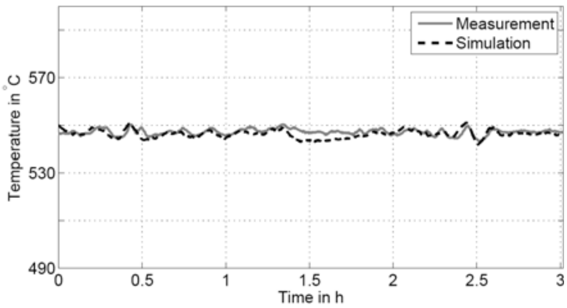

Fig. 15: Comparison of calculated and measured live steam temperature

Figure 15 shows the simulated and measured live steam temperatures. Again a very high correlation between simulation and measurement can be stated. This can only be achieved by copying the cascaded PI-controller that ensures the right tempering of the live steam.

Figure 16 shows the comparability of the implemented temperature control by comparing the simulated and measured injection mass flow rates. Both measured and simulated injection flow rates agree qualitatively very well.

The figure shows that in part load when the temperatures decrease due to sinking fuel mass flows the need for cool injection water drops nearly to zero at this attemperator. The attemperator is operated at its limits which leads to quite extreme changes of valve opening between fully opened and fully closed. This behaviour is uncritical since the task of the first injector is it to keep the second injector within its proper operation limits.

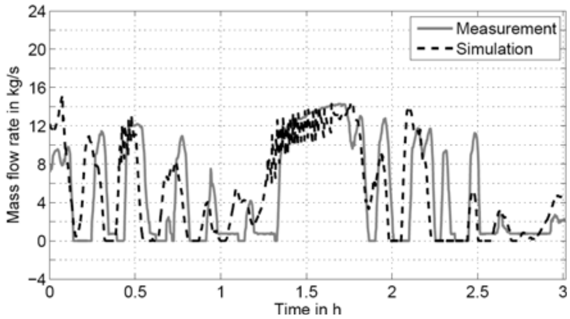

Fig. 16: Comparison of calculated and measured injection mass flow rates at the first HP-injector (after first super heater)

The validation of the model shown above approved the accuracy and the validity of the model assumptions.

2.3 Opportunities and Limits of the Developed Plant Model

With this existing model it is possible to predict temperatures and temperature gradients at points which are inaccessible to measurements like wall temperatures of highly stressed components.

The occurring wall temperatures of the steam separator are presented in figure 17. Obviously the metal temperature at the outside of the wall follows the inner temperature with a certain delay and its amplitudes are considerably smaller. This effect can be explained with time specifics of the heat conduction. The noticeable phase shift of the temperatures leads to relative high temperature differences between the inner and outer phase in case of sharp edged changes in evaporator heating or cooling.

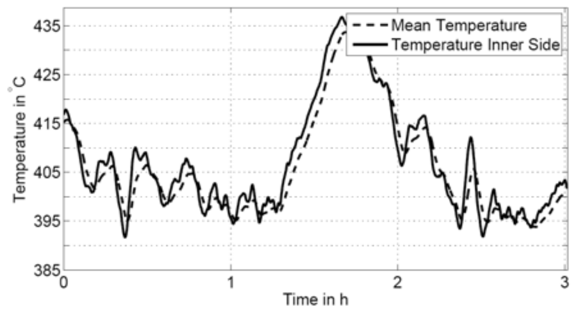

Fig. 17: Metal temperatures in the evaporator outlet header

The evaluation of metal temperatures offers the possibility to benchmark different controller parameter sets with a view to preserving operation at concurrently high load dynamics.

Quantification of the effects of thermal stress on the different components of a plant is a challenging task as the processes of fatigue are complex and highly

statistical. For this reason the results of a fatigue prediction in this context can only be of qualitative nature and should be understood as a trend indicator that is capable of identifying the most stressed components and predict possible side effects of innovative control strategies on this complex system. For a detailed investigation of certain components a FEM-analysis considering the installation situation (and with it possible pretensions in the component) and the exact geometry should be taken into account.
However, for a first estimation of the effects of future more dynamic plant operation two different approaches are used and should be discussed in the following:
The guidelines of the *Deutsche Dampfkesselausschuss (2000)* TRD 301 and 508 give directives for the estimation of fatigue of thick-walled boiler components under smouldering pressure and temperature due to start-up processes.
For this purpose an effective stress range is evaluated with a Wöhler-diagram for crack initiation. The following equation gives the law for calculating the stress range ($\Delta\sigma$).

$$\Delta\sigma_t = \left(\alpha_m \frac{d_m}{2s_b}\right)\Delta p + \left(\alpha_\vartheta \frac{\beta_{L\vartheta} E_\vartheta}{1-\nu}\right)\Delta\vartheta$$

Herein α_m, α_ϑ, d_m, s_m, $\beta_{L\vartheta}$, E_ϑ, ν, Δp, and $\Delta\vartheta$ denote for mechanical and thermal correction factor for stress superelevation at branches, mean diameter, mean wall thickness, linear expansion coefficient, Young's modulus, Poisson's ratio and the range of pressure and temperature difference during load change, respectively. Figure 18 shows qualitatively the evaluation of the working stress during load change. The maximum number of load changes comparable to the actual one is generated from the Wöhler-curve. The percentile fatigue of the actual load change is then:

$$e = \frac{1}{N} 100$$

This estimation leads to conservative results in order to handle the numerous uncertainties in calculation of working stresses at complex components and material properties.

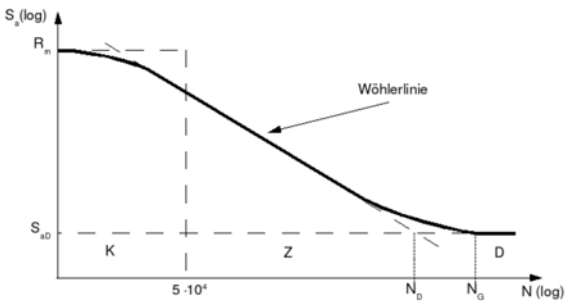

Fig. 18 Principle of evaluation of component stress for cyclic loading (Levin et. al, 1990)

The estimation according to the *Deutsche Dampfkesselausschuss* references mainly start-up and shut-down processes of a boiler which is well-founded in the former static operation of power plants. Considering the flaw growth of damaged component gives far more sensitive view on the operation mode.
The *Forschungskuratorium Maschinenbau* (FKM, 2001) gives guidelines for the calculation of crack progress. Figure 19 gives a general overview on crack propagation rate as function of the range of stress intensity factor ΔK.

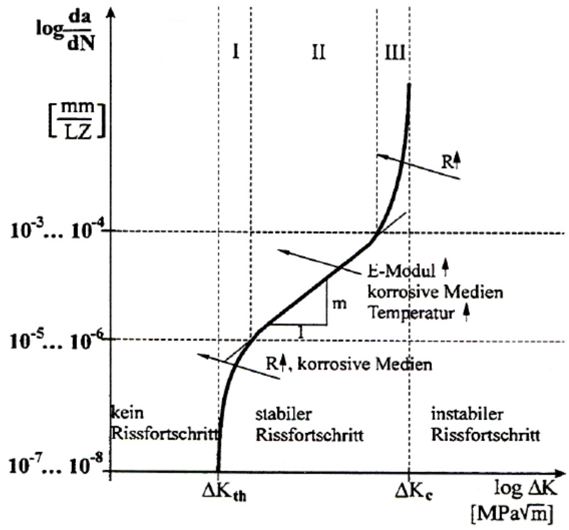

Fig. 19 Overview on crack propagation under cyclic load (FKM, 2001)

There is a certain load that does not leads to crack propagation ($\Delta K \leq K_{th}$). In region I to III there is a stable propagation to be expected ($\Delta K_{th} \leq \Delta K \leq \Delta K_c$) which can be conservatively estimated by the law of Paris and Erdogan:

$$\frac{da}{dN} = C\Delta K^m$$

Where a, N, C, m denotes for crack length, number of cycles, a case-specific factor and a load specific exponent, respectively.
The stress intensity factor has to be calculated depending on the flaw's geometry and size and its position within the component. With this tool it is possible to detect the most strained components by comparing the crack growth over a certain reference time period which is shown in figure 20 for the investigated validation setup.

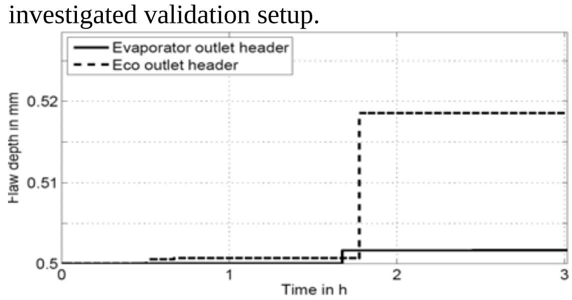

Fig. 20 Comparison of crack growth of two thick-walled components

It becomes apparent that the economizer outlet header is under higher stress than the evaporator outlet header which is due to different geometries.
With this estimation of fatigue a method is given to evaluate the impact of a specific operation mode. In this way, future demands on power plants which

might become necessary in order to realize wind integration successfully at controllable costs can be benchmarked.
This aspect of power plant operation management will probably become more important due to highly increasing wind power production and its fluctuating characteristic.
The detailed manner of the plant model does not allow long term simulation over years or even weeks due to high computing time at this actual state of development. Therefore the fatigue has to be extrapolated in a first step assuming a constant operation mode for a long time period. Future work could cover a model reduction to increase its efficiency.

4. CONCLUSION AND FUTURE WORK

A model, based on thermodynamic balance equations, of a hard-coal-fired power plant was validated and it was shown that this model and the described stress criteria for evaluating of different operation modes with fatigue estimation and flaw growth are a reasonable tool for benchmarking the capabilities of existing plants to compensate future wind fluctuations.
Present simulation results highlight a high potential of power plant control system optimisation. Especially a fine tuned ratio of feed water mass flow and fuel mass flow is a key for a smooth operation with high dynamics and low lifetime consumption.
Future work will expand the analysis of this aspect and will define the quantitative effects of different control strategies as well as the potential of model based controllers.

REFERENCES

Casella,F./ Leva, A. (2005). Object-Oriented Modelling and Simulation of Power Plants with Modelica. In: *proceedings of 44th IEEE Conference on Decision and Control, and the European Control Conference*, Sevilla.

Casella,C./ Leva,A. (2003). Open Library for Power Plant Simulation: Design and Experimental Validation. In*: proceedings of 3rd. International Modelica Conference*, Linköping.

Deutscher Dampfkesselausschuss (2000). Technische Regeln für Dampfkessel (TRD) 301 Berechnung auf Wechselbeanpruchung durch schwellenden Innendruck bzw. durch kombinierte Innendruck- und Temperaturänderungen. Carl Heymanns Verlag KG

Deutscher Dampfkesselausschuss (2000). Technische Regeln für Dampfkessel (TRD) 508 Zusätzliche Prüfungen an Bauteilen berechnet mit zeitabhängigen Festigkeitswerten Carl Heymanns Verlag KG

Forschungskuratorium Maschinenbau (2001). Bruchmechanischer Festigkeitsnachweis für Maschinenbauteile, VDMA-Verlag

Fritzson,F. (2004). Principles of Object-Oriented Modeling and Simulation with Modelica. Wiley-Interscience, Piscataway.

VDI/VDE 3508 (2003). Blockregelung von Wärmekraftwerken. Verein deutscher Ingenieure und Verband der Elektrotechnik Elektronik Informationstechnik, Düsseldorf

Klefenz, G. (1991). Die Regelung von Dampfkraftwerken. Wissenschaftsverlag Mannheim/Wien/Zürich

Levin, G./Lässig,G.(1990). Apparate und Behälter Grundlagen der Festigkeitsberechnung.Verlag Technik Berlin