INCREASING FLEXIBILITY OF COAL POWER PLANT BY CONTROL SYSTEM MODIFICATIONS

by

Ante MARUŠIĆ[a*], Dražen LONČAR[a], Jakov BATELIĆ[b], and Valdi FRANKOVIĆ[b]

[a] Department of Energy, Power Engineering and Ecology, Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Zagreb, Croatia
[b] HEP – Croatian Power Utility, Zagreb, Croatia

Original scientific paper
DOI: 10.2298/TSCI160314159M

Expanding implementation of intermittent renewable energy sources has already started to change the role of thermal power plants in energy systems across Europe. Traditionally base load plants are now forced to operate as peaking plants. A familiar transition in upcoming years is expected in Croatia and coal power plant operators are preparing accordingly. To evaluate cycling capabilities and control system operation for flexible operation of selected 210 MW coal plant, series of tests with different load gradients were performed and results were thoroughly analyzed. Two possible "bottlenecks" are identified, thermal stress in superheater header, and achievable ramping rate considering operational limitations of coal feeders, firing system and evaporator dynamics. Several unexpected readings were observed, usually caused by malfunctioning sensors and equipment, resulting in unexpected oscillations of superheated steam temperature. Based on superheater geometry and experimental data, maximal steam temperature gradient during ramping was evaluated. Since thermal stress was well inside the safety margins, the simulation model of the whole boiler was used to evaluate achievable ramping on electric side.

Key words: *coal powered plants, flexible generation, thermal stress, simulation*

Introduction

Increased participation of intermittent sources in the Croatian power system has forced the system operator to finally consider flexible generation from thermal power plants which are traditionally operated in base load. Due to the high share of hydro power, system needs for primary and secondary reserves are covered most of the year. However, low load operation of hydro power plants is not profitable during periods with good hydrological conditions. Therefore some space is left for engagement of thermal power plant fleet to provide required auxiliary services. Since most of gas turbines are part of cogeneration plants and their operation profile is defined with heat demand, achievable control reserve of coal power plants has to be estimated.

Flexible coal plants are subject of interest for over two decades, but it gained significant attention only in the last decade as, thanks to the high share of renewable resources, coal plants started losing their role as baseload plants [1]. Flexibility improvements during this period were based on experience and improvements of the control system. The exponential growth of renewable resources in the last few years demands even faster power plant respons-

* Corresponding author; e-mail: ante.marusic@fsb.hr

es and better damage mitigation which can be achieved only with optimization based on detail dynamic models. Therefore the research interest in coal plants started to grow rapidly recently. A new, complex dynamic model of large coal plants for simulation and further investigation of flexibility is described in [2]. Furthermore, advance model for boiler start-up optimization taking into account thermal stresses is developed recently [3]. Apart from conventional coal plants, flexibility of modern coal plants with carbon capture is also addressed [4]. Further, process simulation model coupled with computational fluid dynamics for tangentially hard coal fired 700 °C boiler is developed [5]. With detailed simulation of heating profiles, calculations based on existing standards (*i. e.* EN-12952-3 [6] and EN-12952-4 [7]) are inadequate and other, more detailed approaches should be used. One such method for estimation of optimum fluid temperature during start-up process, based on finite elements method and taking into account two points at the edge of the collector opening is described in [8].

The maximum ramp rate and number of load cycles in coal plants are generally limited by thermal stresses, creep and fatigue of critical thick wall components. Furthermore due to specific design every plant can have a different "bottleneck". Traditional base load operation did not require steep ramps, hence recommended ramping rates were kept relatively low, deep within the safe operation zone. However, recent experiences have shown that ramp rate improvements of up to 300% are possible with simple sensor and control system retrofits. Thermal stresses are caused by temperature changes inside thick wall components (*e. g.* collector shell), therefore the quality of steam temperature control is of essential value.

Control system modernization offers key operational improvements that help to increase operational flexibility, decrease costs and manage emissions. By reducing oscillations and excursions through tighter plant control, plants can improve unit efficiencies and realize operational savings, while at the same time improving ramp rates 1-2% [9].

Different potential modifications for flexibility improvement and low load operation can produce different results depending on plant specifications but also on the energy market regulations, energy system configuration, *etc*. As presented in [10] in a case of Northern Ireland, modifications of one coal power plant can have significant consequences on the whole system, affecting wind curtailment, carbon emissions and financial savings.

Reference coal power plant

The reference plant is a 210 MW plant with pulverized coal-fired once through boiler, reheating and sea-water cooled condenser. The main boiler design parameters are presented in tab. 1. Although the steam generator was designed for domestic coal from a nearby exploitation site, since its commissioning the coal is imported from overseas. Therefore the coal choice depends only on market circumstances and the composition of coal can vary significantly. Operational measurements during burning different coal types can be valuable for plant model validation.

Water-steam tract consists of economizer, evaporator, two superheaters and two reheaters.

After each superheater steam is collected in the collector, because of the collector size and high pressure, a thick wall is necessary to ensure safe operation, on the other hand, thick walls combined with temperature variations can cause significant thermal stress. The temperature in the collector is function of number of variables, most important being fuel rate and feedwater flow. However, none of these variables is directly controlled by steam temperature. While fuel rate is governed by desired overall power, feedwater flow rate is controlled by steam pressure. Therefore, steam temperature is controlled by two pairs of attemporators (desuperheaters). One pair is placed after the evaporator and the second one is between the first and the second superheater. Steam temperature is controlled by the second attemporators pair. The water flow through the first pair of attemporators is controlled to keep steam temperature in control range for the second attemporator pair. The steam temperature control is realized by cascaded control, where simple model of the process is included in the master loop while PI controller is placed in subordinated loop. There is a room for loop improvement by feed-forward signal based on expected load changes.

Table 1. Reference power plant

Power	Fuel	Number of coal feeders	Design steam temperature	Design steam pressure	No. of superheaters	No. of reheaters
210 MW	Hard coal	6	530 °C	150 bar	2	2

Real ramping rates

In order to evaluate current ramping capabilities several tests with different heat rate gradients and different boundary conditions were conducted. Load changes were between 160 MW and 210 MW which equals 76.2% and 100%, respectively. Real plant measurements for +8 MW/min thermal gradient are shown in fig. 1, together with the electrical power output for the same period.

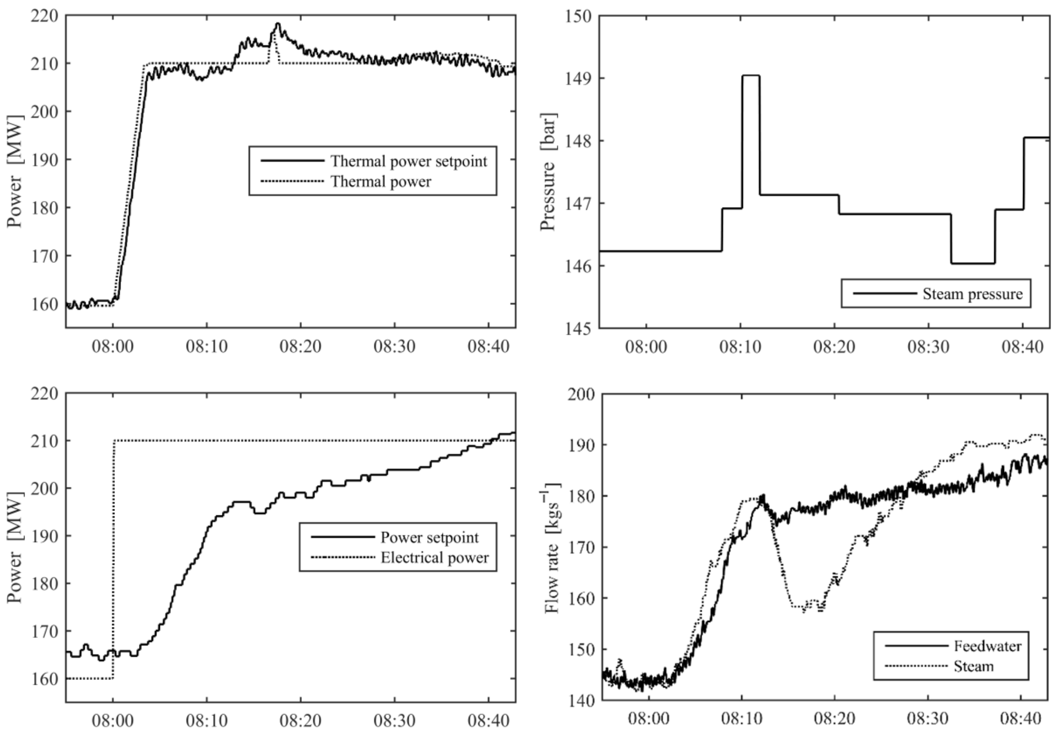

Figure 1. Ramping at 8 MW/min starting at 8:00 a. m.

The steam pressure signal resolution was lowered in the acquisition software by plant operator because of practical reasons during constant pressure operation mode. The electric power gradient for the whole ramping period is around 1.2 MW/min, while the electric power gradient between 8:02 and 8:13 is somewhere around 3 MW/min. This relatively steep ramp is disturbed by sudden steam flow changes caused by the subordinated steam pressure control, as can be seen from lower left diagram in the same figure. Around 8:10, two of four turbine valves were suddenly opened because of the high pressure causing the disturbance in the pressure control system. System responded by closing the valves even more than their prior position. This behavior caused by the pressure control is than continued for about 15 minutes causing the lower steam flow and the pressure increase. Mismatch between the steam and the feedwater flow is further increased by slight thermal power output drop occurring at the same time.

Thermal stress in collector header

The drum is the thickest boiler pressure component, it is therefore defined as the critical component, limiting allowable boiler startup as well as ramping (heating and cooling) rates [8]. Thermal stress is not monitored directly but is calculated from the temperature difference between surface temperature and the temperature in the middle of the collector wall. During results analysis it was observed that one of the temperature sensors is malfunctioning which can be seen from fig. 2, instead of reacting to steam temperatures before middle sensor, it is reacting almost at the same time. The error is probably caused by the displacement of the surface sensor, or its bad installation, since the sensor was not in direct contact with the wall, its time constant increased. Hence, one dimensional mathematical model of heat transfer through a pipe wall was created to acquire more realistic heat profile inside a thick collector wall and ultimately give better approximation of real thermal stress. Temperature distribution through a circular wall can be written as in eq. (1):

$$\frac{\partial T}{\partial t}=\frac{\lambda}{\rho_n c_n}\left(\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}\right) \tag{1}$$

The ratio of inner collector radius and wall thickness is relatively small, therefore it is reasonable to simplify the problem and describe it as a flat wall. Flat wall formulation of the problem in its discretized form can be written as:

$$\frac{\partial T_n}{\partial t}=\frac{\lambda}{\rho_n \Delta r c_n \Delta r}\left[\left(T_{n+1}-T_n\right)-\left(T_n-T_{n-1}\right)\right] \tag{2}$$

and boundary condition on steam side as:

$$\frac{\partial T_1}{\partial t}=\frac{1}{\rho_n c_n \Delta r}\left[\frac{1}{\frac{1}{h}+\frac{\Delta r}{2\lambda}}\left(T_s-T_1\right)-\frac{1}{\frac{\Delta r}{\lambda}}\left(T_2-T_1\right)\right] \tag{3}$$

where λ is thermal conductivity, ρ density, c heat capacity, T_n temperature of discretized segment n, Δr segment width, r_n inner radius of segment n and h heat transfer coefficient. Because of the complex geometry exact evaluation of local h is almost impossible, therefore simple calculation based on fictive velocity in the collector was used. Calculated h is around 1,000 W/(m^{2}K) which is in accordance with calculations from other papers [8, 11]. Because of the thick heat insulation on the outer side of the collector, outer boundary condition was modeled as adiabatic.

Recorded steam temperature was used as simulation input and constant outside temperature was used as outside boundary condition. The results were verified by comparison of simulated and measured middle wall temperature. This way, better assessment of integral wall temperature and consequentially thermal stress was possible.

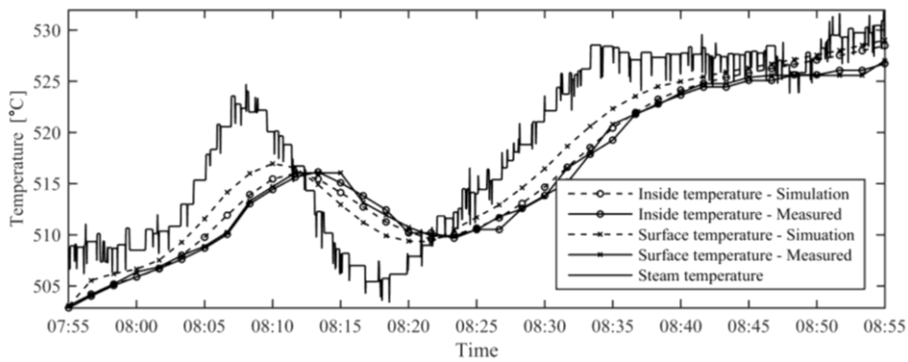

Figure 2. Steam temperature and wall temperatures during ramping at 8 MW/min

Actual stress at the inside corner of the bore of a collector (critical point) was calculated based on expression from EN 12952-3 standard:

$$\sigma = \alpha_m \frac{d_{ms}}{2e_{ms}} p + \alpha_t \frac{\beta_{Lt} E_t}{1-v} \Delta T \tag{4}$$

The first part of the right side expression stands for pressure caused stress and rightmost part for thermal stress. α_m denotes stress concentration factor due to pressure, d_{ms} mean diameter of the collector, e_{ms} mean wall thickness of collector, p pressure in MPa, α_t stress concentration elasticity, v Poisson's ratio and ΔT through-the-wall temperature difference defined as:

$$\Delta T = T_m - T_i \tag{5}$$

where T_m is integral mean wall temperature, and T_i is the temperature at the inside surface of the wall. Actual stress during 8 MW/min thermal load is shown in fig. 3. Constant pressure operation mode excludes any significant pressure changes, stress cycles are practically caused only by temperature changes. The largest stress cycle is approximately 27 N/mm^2, which is significantly smaller than the upper limit of stress range that does not cause fatigue damage (190 N/mm^2 as defined in EN 12952-4 [7]).

Based on the maximal allowed stress an invers equation can be used to calculate maximal ΔT:

$$|\Delta T| = \frac{190 - \alpha_m \dfrac{d_{ms}}{2e_{ms}} \Delta p}{\alpha_t \dfrac{\beta_{Lt} E_t}{1-v}} \tag{6}$$

However, ΔT cannot be directly related to steam temperature deviations since it is not only a function of steam temperature but also of time and heat transfer rate at the surface of the collector. Nevertheless during one cycle ΔT is always smaller than the peak to peak steam temperature, and therefore can be used as limiting value for peak to peak steam temperature variation during load changes.

The calculated allowed ΔT and consequentially the steam temperature variation during operation with 5 bar pressure oscillations (largest one observed during measurements) is around 32.5 °C.

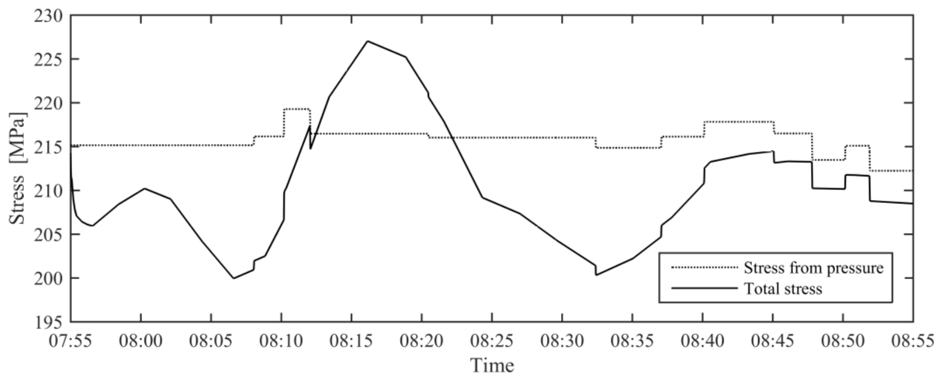

Figure 3. Total stress on critical point

In reality the mentioned temperature difference can only be achieved with high heat transfer coefficients (high local velocities) during steep, uninterrupted loading gradients.

Achivable ramping rates

Although the comprehensive control system modification is a general practice for coal power plants ramp improvement, one has to wonder about the achievable ramping rate without significant modifications. As mentioned before, much higher (32.5 °C) temperature variations could be allowed without compromising the boiler life expectancy. Nevertheless during more than 30 ramping tests, the temperature variations of this order were not possible because of the control system constrains. A significant ramp of around 5 MW/min was observed only once, caused incidentally by the engagement of additional coal feeder. Records of this event are shown on lower right diagram in fig. 4 starting around 05:25. Sudden start-up of the additional coal feeder (shown in the upper left diagram in fig. 4) causes thermal load overshoot which is not compensated by the decrease in other five feeders. As can be seen from the upper right diagram in fig. 4, the temperature overshoot caused by this disturbance was not larger than 5 °C. These results lead to conclusion that even without complicated control system modifications and with just simple adjustments of the control system constraints (allowing thermal load overshot during fast load changes), significant ramping rate improvements can be accomplished. To confirm this thesis and asses maximal achievable ramping rates detailed one dimensional boiler model was used.

The boiler model was based on the in-house model which is in detail presented in [12], with small modifications. The model domain (boiler) is discretized on 150 segments on the water-steam side (economizer, evaporator, superheaters and reheaters) and 25 segments on the flue gas side. The boiler scheme used for discretization is shown in fig. 5. The heated water-steam pipes are printed in red and the connecting pipes outside of the firebox are printed in red. Outside contours of the firebox are printed in green. Both convection and radiation were taken into account for calculation of the heat transfer on the gas side, and a simple quasi-steady model of combustion was used. Fuel flow and ambient air temperature were input variables, where load is indirectly defined with fuel flow rate. The simulation was set as open-loop system, that is subordinate local control loops were not simulated.

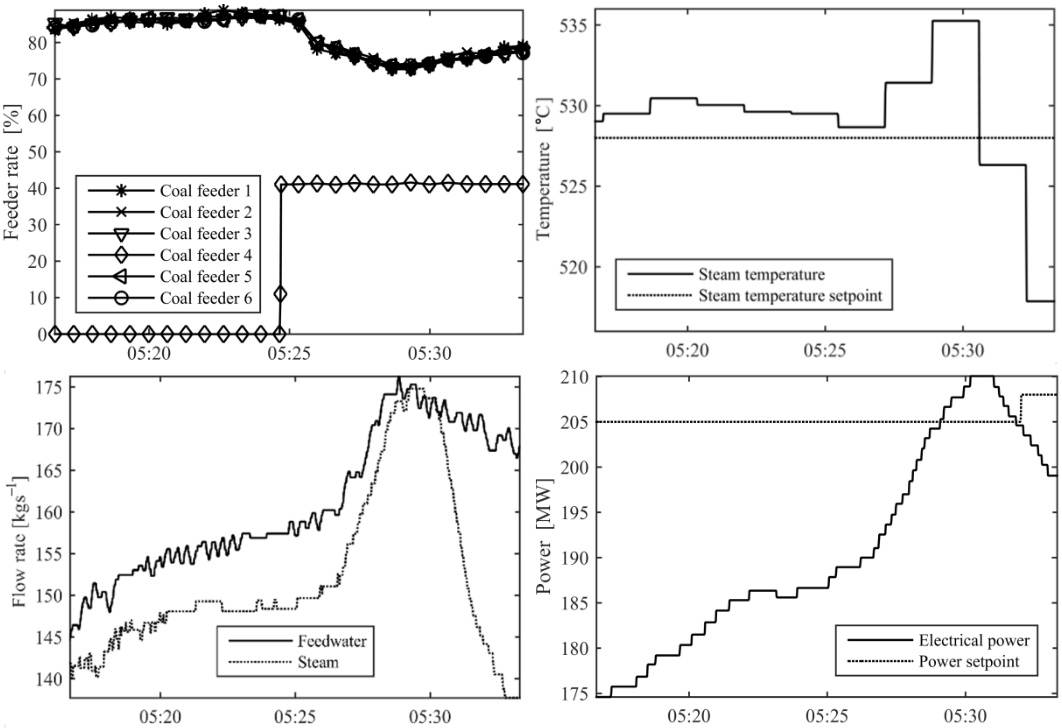

Figure 4. Maximal achieved ramp rate

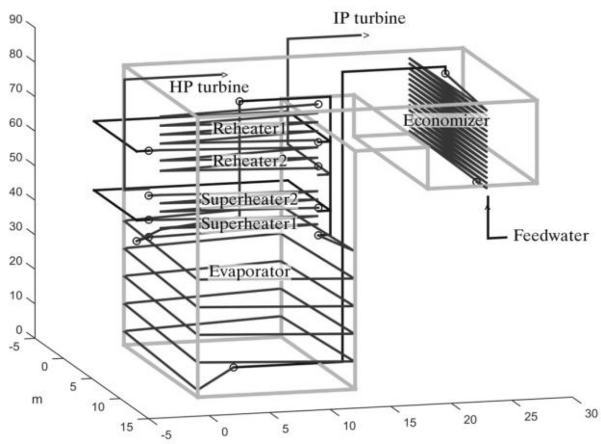

Figure 5. Boiler components scheme used for discretization

Initial conditions used in simulation were steady state conditions during stable operation with 80% load, resulting in the power output of 168 MW. Because of the numerical constraints only 6% load change was simulated, resulting in the power output increase of around 9.5 MW.

Simulation results are shown in fig. 6, where the x axis represents time in minutes. Two different firing methods were simulated, the normal ramp, which is similar as actual firing method at the moment, and the fuel flow overshoot (clearly visible in upper left diagram), which is traditionally used to produce steeper power gradients. As can be seen from the lower right diagram, effective power ramps of around 4 MW/min were observed, where the overshot has produced better results during the ramping and also caused significantly shorter transient period (time necessary for reaching set point load).

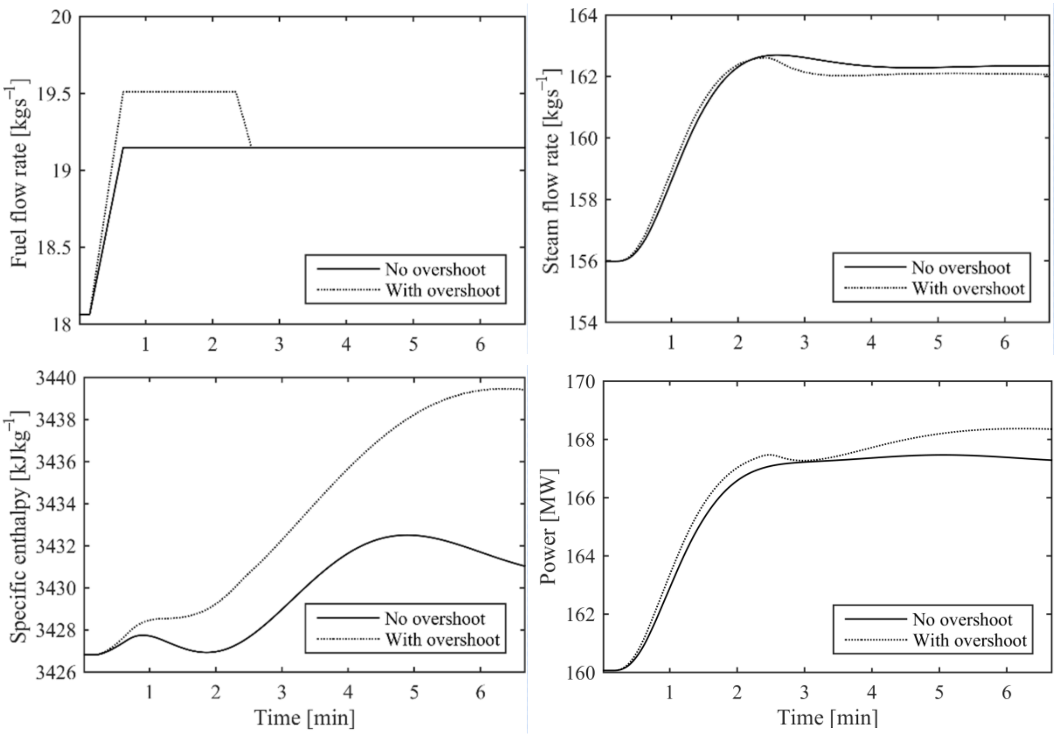

Figure 6. Simulated boiler responses

Conclusions

Two most important operating conditions concerning flexibility are startup and ramping. Both of these states are characterized with sudden temperature and/or pressure changes, hence causing large stress changes in thick wall components. Although older thermal power plants were not designed for cycling operation, most of them are capable of it with none or small hardware modifications. Nevertheless extensive control system modifications are sometimes necessary to ensure high heating rate gradients and safe operation.

To assess thermal stress during ramping rates, a simple one dimensional mathematical model was created and a significant influence of heat transfer rate in the collector was addressed. However, thermal stress caused by ramping is well under specified limit that causes fatigue even during steepest heat gradients and despite malfunctioning of the steam temperature control system.

Considering small calculated thermal stress during operation, the simulation of maximal load gradients was carried out. It is shown that even during maximal firing rate gradients thermal stress is well inside the safe zone leading to the conclusion that, contrary to some cases found in literature, the load gradient in the observed power plant is only limited by its firing mechanism and thermal inertia. Most simple modifications such as fuel load overshoot control can produce significant improvements and often be best solution for the thermal plant flexibilization.

References

[1] ***, Flexible Generation: Backing Up Renewables, Renewables Action Plan (RESAP), EURELECTRIC, Brussels, 2011
[2] Starkloff, R., *et al*., Development and Validation of a Dynamic Simulation Model for a Large Coal-fired Power Plant, *Appl. Ther. Eng.*, *91* (2015), Dec., pp. 496-506
[3] Taler, J., *et al*., Optimization of the Boiler Start-up Taking Into Account Thermal Stresses, *Energy*, *92* (2015), 1, pp. 160-170
[4] Lucquiaud, M., *et al*., Enhanced Operating Flexibility and Optimised Off-design Operation of Coal Plants with Post-combustion Capture, *Energy Procedia*, *63* (2014), Dec., pp. 7494-7507
[5] Schuhbauer, C., *et al*., Coupled Simulation of a Tangentially Hard Coal Fired 700 °C Boiler, *Fuel*, *122* (2014), Apr., pp. 149-163
[6] ***, Brussels: European Committee for Standardization, BS EN 12952-3:2001 – Water-Tube Boilers and Auxiliary Installations, Design and Calculation of Pressure Parts, 2001
[7] ***, Brussels: European Committee for Standardization, BS EN 12952-4:2000 – Water-Tube Boilers and Auxiliary Installations, In-Service Boiler Life Expectancy, 2000
[8] Dzierwa, P., Taler, J., Optimum Heating of Pressure Vessels With Holes, *J. Press. Vess. Technol.*, *137* (2014), 1, pp. 011202
[9] Shankar, R., Feasibility of Wear and Tear Sensors for Flexible Plant Operations, EPRI Tech. Update, 2003
[10] Kubik, M. L., *et al*., Increasing Thermal Plant Flexibility in a High Renewables Power System, *Appl. Energy*, *154* (2015), Sep., pp. 102-111
[11] Kruger, K., *et al*., Optimization of Boiler Start-up Using a Nonlinear Boiler Model and Hard Constraints, *Energy*, *29* (2004), 12-15, pp. 2239-2251
[12] Lončar, D., Application of Fuzzy Logic in Thermal Power Plant Control Systems, Ph. D. thesis, University of Zagreb, Zagreb, 2001

Paper submitted: March 14, 2016
Paper revised: May 23, 2016
Paper accepted: May 24, 2016