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Analysis of Coal-Fired Power Unit Operation in Reduced Minimum Safe Load Regime

Piotr Żymełka, Maciej Żyrkowski and Maciej Bujalski

Additional information is available at the end of the chapter

Abstract

Large coal-fired power plants were typically designed as a base load units. Any changes in load level, as well as start-up time, are noticeably slow on that kind of units. However, in order to adapt to changing market conditions with increasing number of renewable energy sources, coal-fired power plants need to improve their flexibility. In the paper, 200 MWe class unit has been taken into consideration. During the test campaign, a minimum safe load of the unit was decreased from 60 to 40%. Paper presents results of a model that was made using Ebsilon®Professional software. The simulation model is comprised of boiler and turbine part of the power unit. Obtained results were validated using measurements collected from the test campaign. Parameters important from the technical and economical point of view were investigated. Results revealed that simulation model can be utilised successfully to scrutinise coal-fired units under off-design operation conditions. As the outcome of the performed analysis, a number of issues related to low load operation of the coal-fired unit are presented and discussed. Paper indicates sensitive areas that need to be addressed when operation in decreased safe load is considered. Finally, overall potential for flexibility improvement for 200 MWe class coal-fired units has been evaluated.

Keywords: flexibility, simulation modelling, power plant, power production, coal

1. Introduction

Large coal-fired power plants used to be designed as a base load unit for electricity market. Power units in such power plants can only change their load level in limited range. What is more, any load changes as well as start-up periods, are considerably slow. That has not been the case for many years, currently though, power units require much more flexibility in operation. The reason for that is a growing number of intermittent energy sources, like wind or

PV units. Due to the currents EU's policy, they are privileged on the electricity market, therefore other units role is to balance power level in the system. Power plants with the highest ability to balance renewable energy sources are those equipped with gas turbines, because of a very high speed of load changes as well as short start-up time they are able to provide. However, many energy markets still rely on large coal-fired plants and therefore they need to increase their flexibility. Two aspects of the flexible operation are taken into consideration. The first is the ability to start and change load faster. The second aspect is related to load level of considered units and possibility of its extension [1]. Paper refers to the latter case and tries to evaluate the potential for flexibility improvement of 200 MWe class coal-fired power unit, including both turbine and boiler part.

Flexibility issues with the steam turbine are mostly related to start-up periods. Material limitations impose a maximum temperature gradient that is allowed when the turbine warms up, which for saturated steam is equal to about 1 K/min and for superheated steam about 5 K/min [2]. Minimum power output that steam turbine can produce depends on the pressure in the condenser, which must be kept close to zero. Much more problems however, are associated with the boiler. In case of the reference unit, the most critical element in terms of thermal stress is a drum, for which maximum allowed temperature gradient for is equal to about 4.5 K/min [2]. Next flexibility limitation is related to combustion process itself, which in case of large-scale boilers cannot be controlled too fast. In fact, any changes in boiler operation must be done gradually. When it comes to the boiler load range, it mainly depends on its water circulation technology. Natural circulation boilers have limited flexibility, due to the fact that certain level of heat from combustion process must be delivered to the evaporator, to maintain water circulation. Boilers with forced circulation can operate with much lower minimum safe load. In the reference unit, that is natural circulation boiler, the minimum safe load was designed to 60%, which is equal roughly to 390 t/h of steam flow and 135 MWe of power output. As a matter of fact, minimum safe load operation of the boiler is crucial in this investigation as it indicates overall minimum load of the entire unit. That is because steam turbine has typically wider range of operation than the boiler. Decreasing minimum safe load of the boiler would subsequently allow to decrease the electric power output of the unit. This, in turn, can be strongly beneficial during periods with low energy consumption, when coal-fired units are often forced to shut down. Avoiding the latter, by ability to decrease the power production, is highly desirable by power plant operators. The reason for that, is because shutting down and starting-up coal fired power plants is a rather costly and time consuming process. In the chapter, authors present outcomes of in situ tests as well as simulation modelling of reference unit operation at 40% load level that is below its safe design minimum.

2. Flexibility of reference coal-fired power unit

Reference coal-fired units were produced between 1970 and 1974 in Poland. It is comprised of 225 MWe condensing steam turbine and pulverised coal boiler with a nominal steam output of 650 t/h. Turbine consists of the high, medium and low-pressure part. Inlet steam pressure equals to about 13.5 MPa. The boiler has natural water circulation and is equipped with the drum as well as front wall burners configuration with additional over fire air nozzles. Fuel mixed with primary air is supplied by six coal mills to low-NOx burners and steam nominal
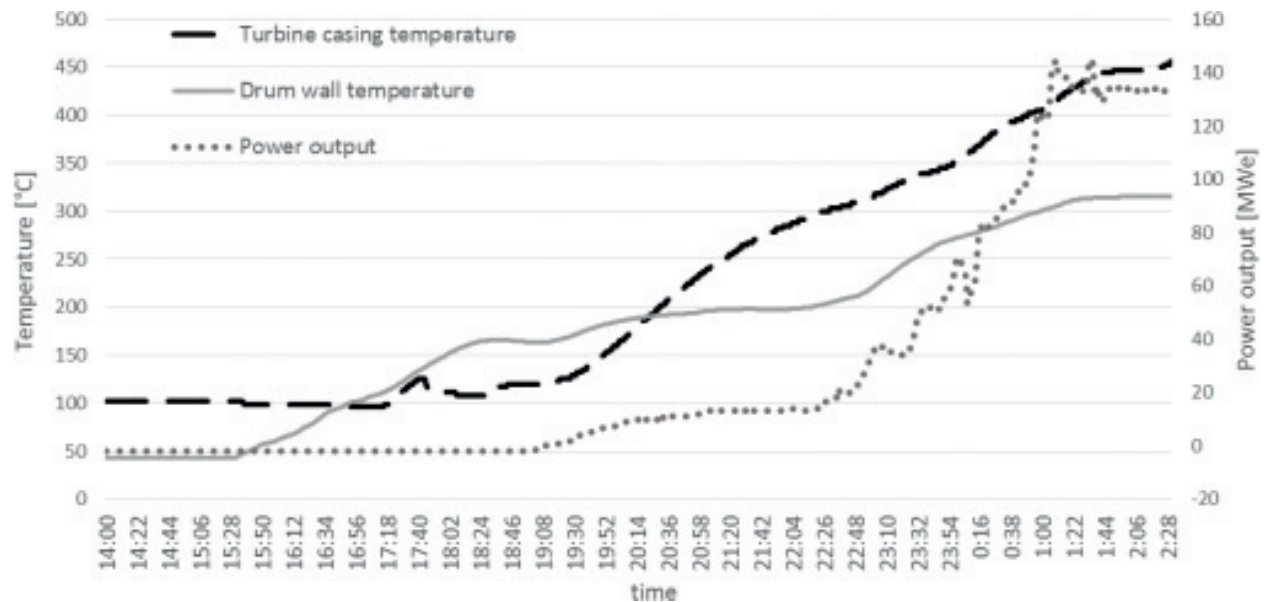
Figure 1. Reference unit cold start-up.

outlet temperature is equal to 540°C. The boiler is comprised of 5 superheaters and 2 reheaters as well as double water heater and three rotary air preheaters. At the boiler's outlet, there is one electrostatic precipitation unit (ESP).

For coal-fired units, it is important if the start-up is being done from the cold, warm or hot state [2, 3]. **Figures 1** and **2** present parameter like electric power output and critical factors (drum wall and turbine casing temperatures) of reference unit during cold and warm start-ups. According to presented data, the warm start-up time of the unit is equal to about 7 h 20 min, while cold start-up is equal to about 10 h 40 min. Typical load variation of the reference unit is presented in **Figure 3** and it changes from about 140 to 220 MWe. The minimum safe load for the reference unit is equal to 60%.
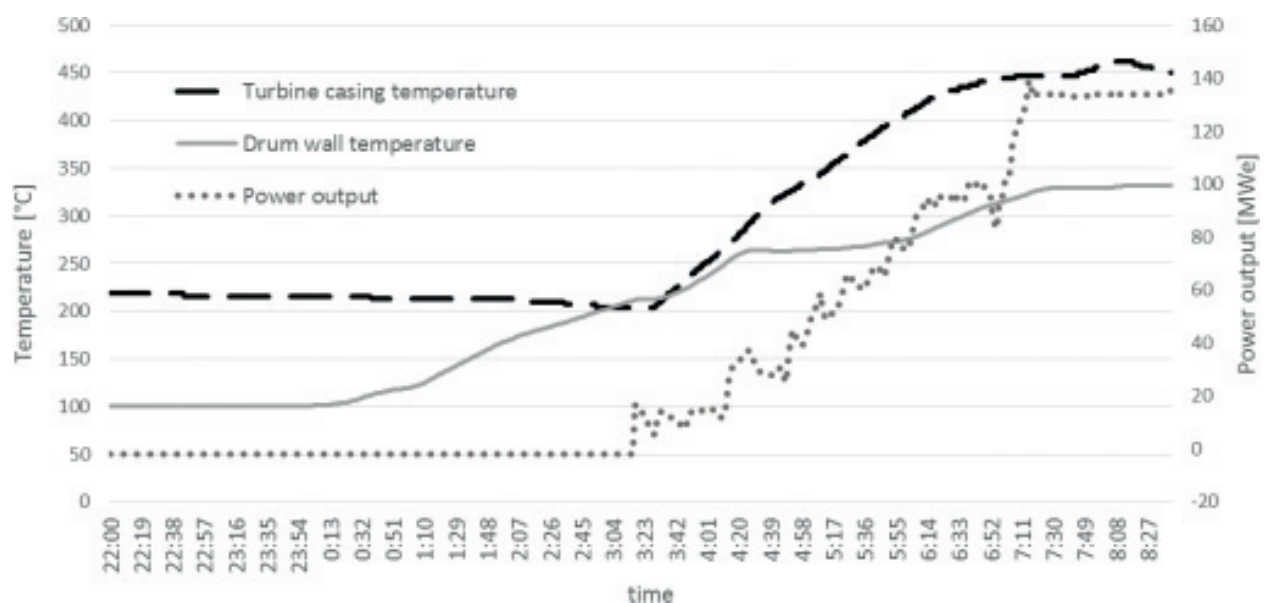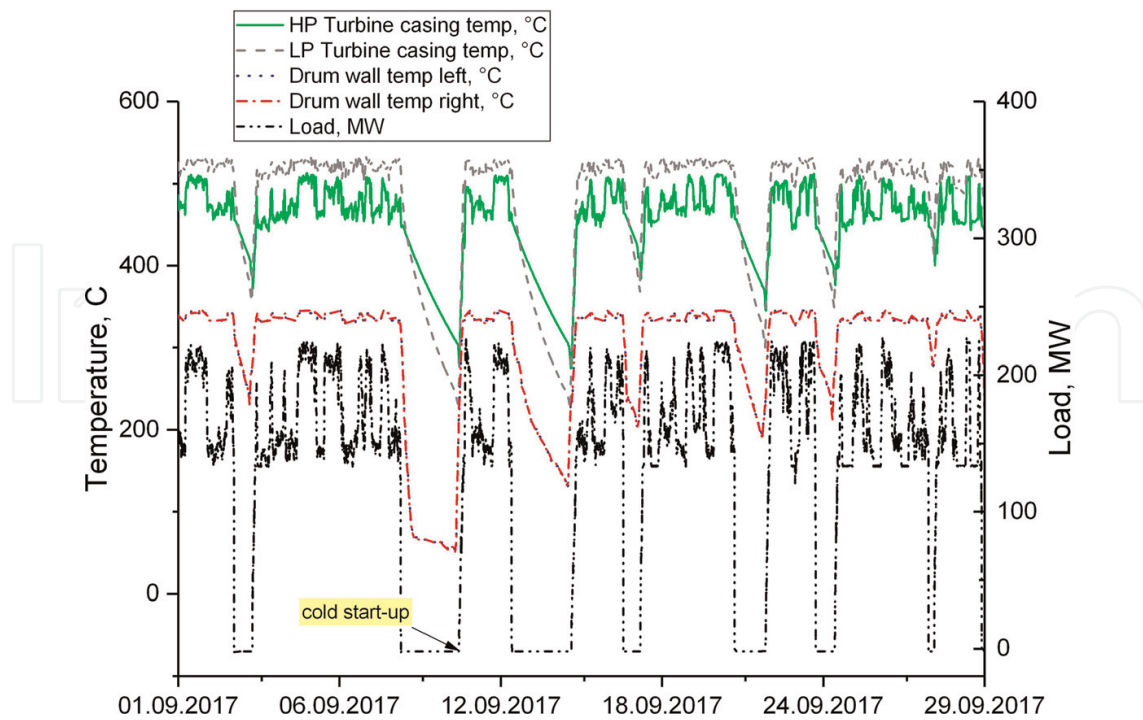

Figure 2. Reference unit warm start-up.

Figure 3. Reference unit critical operation parameters (during September 2017).

The output load of the reference unit has been studied between January and October 2017. Histogram of the electric power is presented in **Figure 4**. During this period, the unit was under flexible operation with frequent load changes over a single day. Most of the time the unit worked with output power below 180 MW which corresponds to the load level below 80%. The unit worked almost 30 h with maximum capacity—225 MW and was shut down several
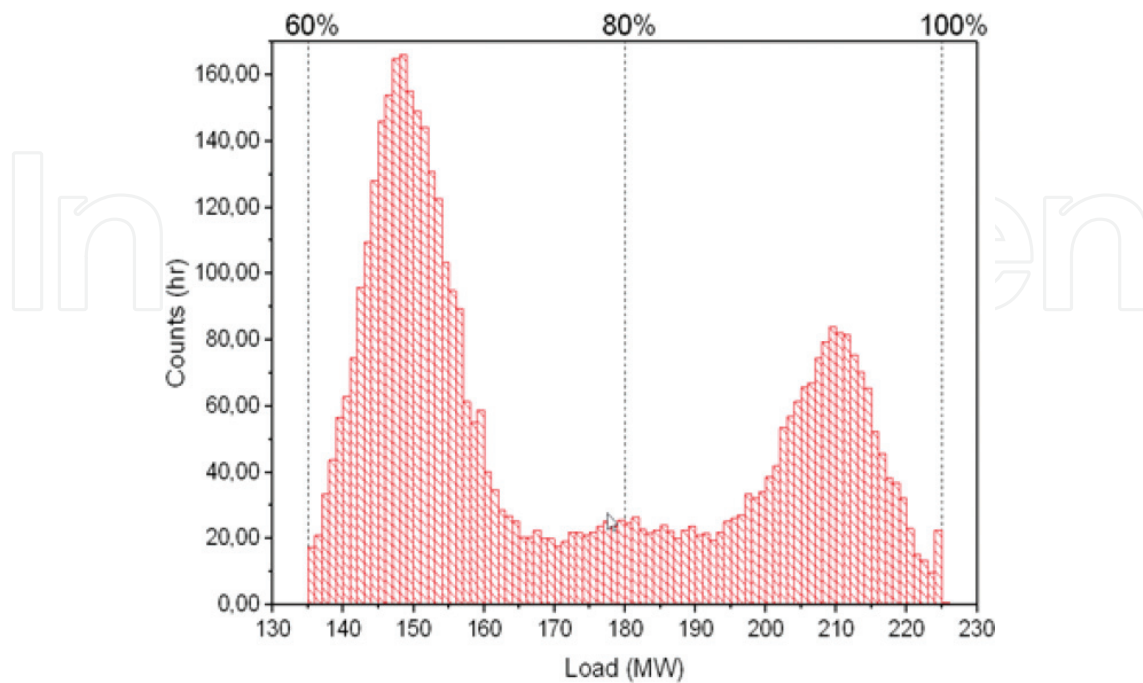
Figure 4. Histogram of the output power (from 1 January to 30 October 2017).

Month	Avg. start-up time, h	No. of cold start-ups, –	No. of warm start-ups, –	Stand-by time, h
January	7.3	1.0	2.0	227.7
February	9.9	1.0	0	520
March	7.0	1.0	2.0	225.7
April	0	0	0	0
May	10.1	1.0	0	571.5
June	9.6	2.0	0	106.0
July	8.1	1.0	1.0	63.0
August	7.9	1.0	2.0	111.3
September	6.0	1.0	6.0	183.2
October	5.1	0	6.0	230.6

Table 1. A number of different start-ups of the reference unit between January and October 2017.

times a month. **Table 1** shows the number of start-ups with the start-up type (from the cold or warm state) during individual months. Aggregated stand-by and average start-up time are also provided. Until the end of October, the unit was launched 28 times. The critical factors (load, drum wall and casing turbine temperatures) of the reference unit during September are presented in **Figure 3**. Over this month, the unit was launched 7 times in total (including one cold start-up).

3. Turbine modelling

3.1. Description of simulation model

Steam turbine of a 225 MW power unit is considered for the simulation analysis. The mathematical model of the coal-fired power unit was developed using the Ebsilon®Professional computer software [4], including detailed models of turbine and boiler part. Boiler model description and simulation results are presented in the next chapter.

The subcritical steam cycle structure of analysed power unit is presented in **Figure 5**. The hatched areas show the main components of analysed thermal system: steam boiler, three casing turbine (consisting of a high-pressure part HP, an intermediate-pressure part IP and a low-pressure part LP), eight regenerative heat exchangers (consisting of three high-pressure HPH and five low-pressure LPH), two-part condenser (CO-1 and CO-2), an electric generator (G), a feed-water tank with deaerator (FW), condensate pumps (CP-1 and CP-2) and feed water pump (FWP).

The superheated steam from the steam boiler at 535°C temperature under 12.17 MPa pressure and 180.6 kg/s mass flow feed the high-pressure turbine. The steam after the expansion process in HP part is reduced to the pressure equal to 2.67 MPa. The HP turbine part supplies two regenerative heat exchangers (HPH-3 and HPH-2) and at the outlet of HP section, the mass flow rate is reduced to 158.8 kg/s. After the HP part, the steam is reheated from 324 to 535°C. The nominal pressure drop in the reheated system is equal to 0.36 MPa. The reheated steam at
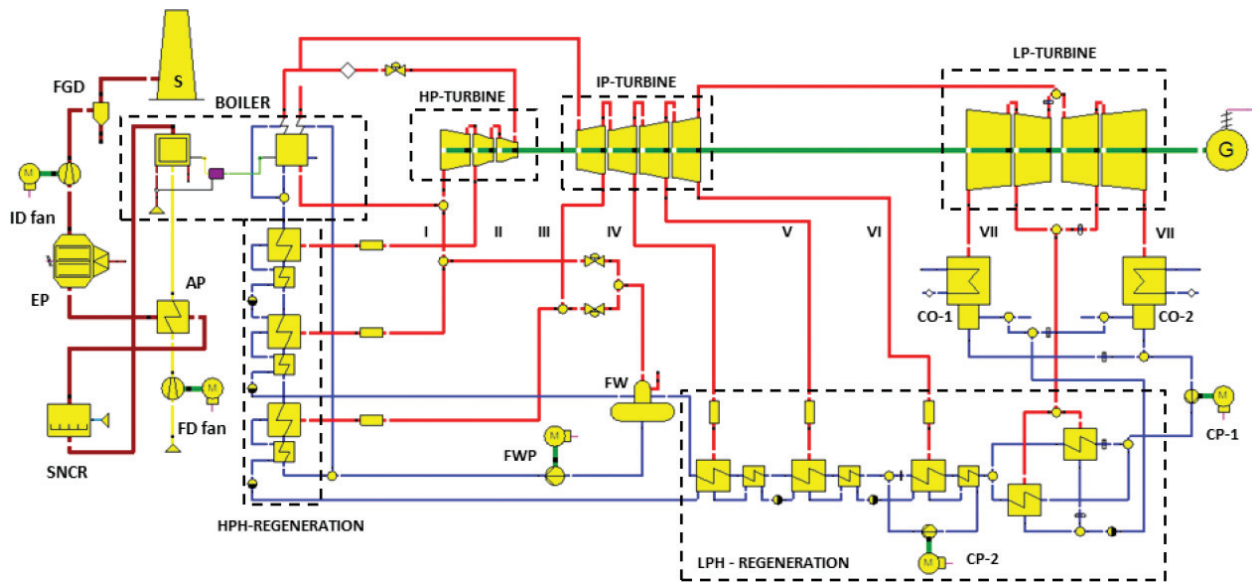
**Figure 5.** A simulation model of the power unit.

535°C and with 2.31 MPa pressure enters the intermediate-pressure (IP) turbine. In the IP turbine part, the steam is expanded to the of temperature 188.0°C and the pressure of 0.13 MPa. The IP section has four extractions to the regeneration system (HPH-1, LPH-5, LPH-4 and LPH-3). The steam mass flow leaving the IP part of the turbine is reduced to 134.9 kg/s. Steam from the IP turbine section feeds the low-pressure LP part. The LP section has one extraction to the low-pressure regeneration heat exchangers (LPH-1/2), so the mass flow rate at the outlet of LP part of the turbine is equal to 129.4 kg/s. The low-pressure steam from the outlet of LP turbine part goes to condensers where is subcooled and then is pumped to the LPH regeneration system. In the LPH section water is reheated to the temperature of 146°C and then enters the deaerator and feed water tank. The water pressure after the deaerator is increased to 16.7 MPa and then it flows to the HPH regeneration system. In the HPH section water is reheated to the 250°C under 15.5 MPa and then feeds the steam boiler. The design steam parameters (pressure, temperature and mass flow rate) at selected turbine extractions are presented in **Table 2**.

The steam expansion line for nominal conditions of the analysed power unit is presented in **Figure 6**. The numerical designation on the expansion curves correspond with the steam parameters in the following locations in turbine system: 1—after shut-off valve, 2—HP-regulating stage, 3—exhaust I (to HPH-3), 4—exhaust II (to HPH-2), 5—before IP stage, 6—exhaust III (to HPH-1), 7—exhaust IV (to LPH-5), 8—exhaust V (to LPH-4), 9—exhaust VI (to LPH-3), 10—exhaust VII (to LPH-1/2), 11—turbine outlet.

The structure of simulation model of 225 MW power unit was developed on the basis of information and data included in the turbine operational manual and performance documentation (power unit energy balance). In the design phase, the average numerical values from thermal measurements included in the in the performance measurements report were used for the unit power load equal to 225 MW. Additionally, values of thermal parameters according to operational data from the measuring system were assumed during the simulation model preparation. The efficiencies of main power plant components (turbine, pumps, generator)

Extraction	Bleed number	Pressure, MPa	Temperature, °C	Mass flow rate, kg/s
I	9	4.03 (3.72)	376.4 (368.0)	8.56 (7.64)
II	12	2.70 (2.48)	324.0 (317.0)	13.17 (11.83)
III	15	1.23 (1.14)	447.0 (447.0)	7.50 (6.72)
IV	19	0.51 (0.47)	334.0 (334.0)	3.72 (3.31)
V	21	0.28 (0.26)	268.0 (268.0)	5.28 (4.78)
VI	23	0.13 (0.12)	188.0 (188.0)	7.39 (6.75)
VII	25	0.026 (0.025)	63.0 (64.6)	5.50 (4.89)

Table 2. Design steam parameters at selected turbine extractions.
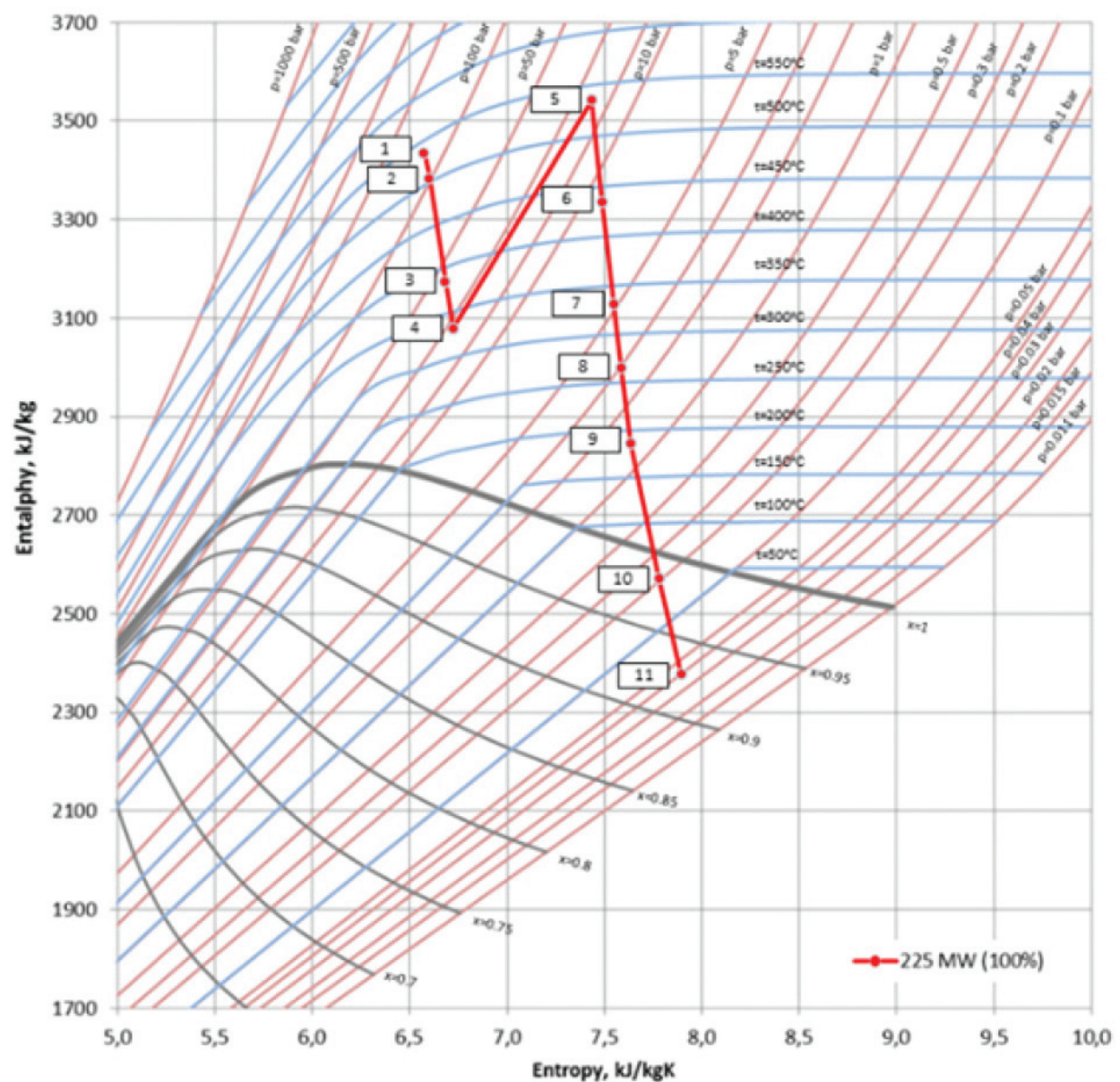
Figure 6. Course of steam expansion line for nominal load of power unit.

were assumed to achieve the nominal operating conditions (in the design mode) and are presented in **Table 3**. The topology constructing process of the analysed power unit thermal cycle was conducted with the following simplifications:

- deaerator is supplied from the third turbine extraction (IP turbine section) for power load above 190 MW, whereas for power load below 190 MW—from the second turbine extraction (HP turbine section),

- steam distribution in the outer glands of the turbine shaft and the turbine sealing system were not included in the simulation model calculations.

The steam turbine simulation model was developed as a number of turbine stages. The number of turbine sections was determined in terms of main inlets and outlets (extractions) in steam turbine cycle. The simulation model of steam turbine was divided into the high-pressure part, intermediate-pressure part and low-pressure part in accordance with its actual construction. The steam turbine model is based on the variable isentropic efficiency, what allows to receive accurate results of simulation calculations for the different load operation. This is very important in part load modelling of steam turbine which is working in sliding pressure mode. The performance operation of steam turbine cycle dependents from many operational parameters, e.g. the temperature, pressure and speed of the working medium. This fact also applies to other elements in the thermal cycle of power systems. The characteristics are used in simulation modelling for an accurate thermodynamic description of components behaviour under part load operation. The characteristic lines describing steam turbine performance were developed for each of turbine stage and then implemented in a simulation model of power unit [4, 5].

The local design mode was assumed for high and low-pressure feed water heaters during the model constructing phase in the design mode. The special off-design mode was used for HPH and LPH regeneration sections in the off-design calculations (at different load levels). The regeneration heat exchangers of HP/LP sections and also the condenser system was developed in separate files for design values and then was subsequently transferred to the simulation model of the power unit. The special off-design mode enables maintenance of constant rated values for given components during the development of the whole power unit designing without the possibility of such values overwriting after operation parameters change of the components directly influencing their functioning [5]. The nominal parameters of HP and LP regeneration system developed in the local design mode are presented in **Figures 7** and **8**, respectively.

Efficiency, %	Turbine			Pumps	Generator
	HP	**IP**	**LP**		
Isentropic	0.852	0.837	0.823	0.800	–
Mechanical	0.996	0.996	0.996	0.985	0.996
Electrical	–	–	–	–	0.986

Table 3. Power unit components efficiency.

3.2. Verification of steam turbine simulation model

The simulation model of power unit was verified with available measurements from the DCS system and also with the performance documentation which contains the results of power unit energy balances. Results obtained from simulation calculations in the design mode were compared to results of the guarantee measurements for four different load levels: 100% (225 MW), 80% (180 MW), 60% (135 MW) and 40% (90 MW). In the verification process of simulation model built in the Ebsilon®Professional software, over 50 measurement points (temperature, pressure and mass flow rate) were used at the characteristic points of the thermal structure of analysed power unit. The off-design calculations were performed by entering the required generator active power (power load) and the external (ambient) conditions for the temperature of cooling water inlet to the steam turbine condenser. The characteristic lines were developed for each of steam turbine stages based on the turbine performance data. The real characteristics of a turbine stage group allow to determine the
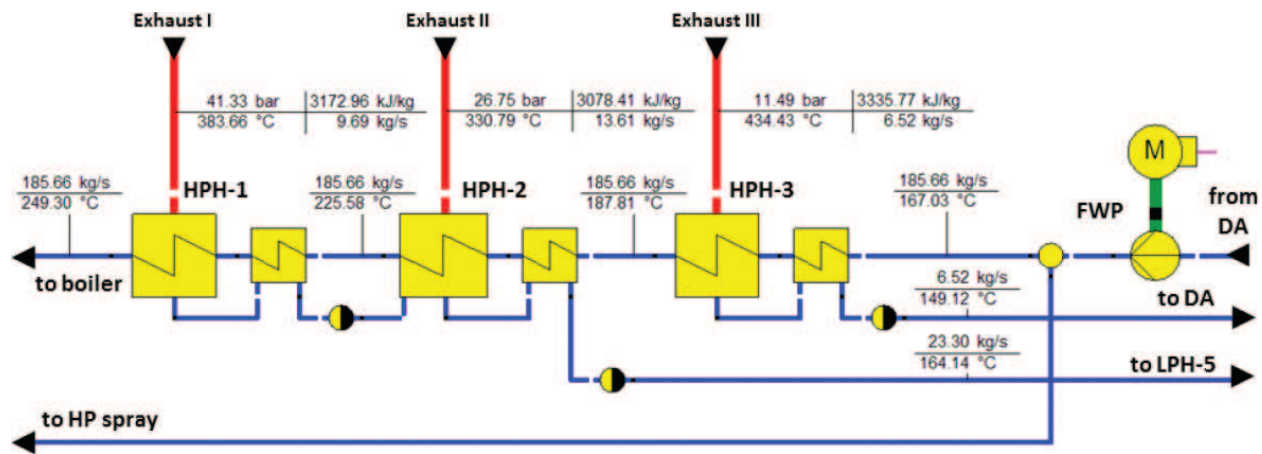

Figure 7. High-pressure regeneration system in the local design mode.
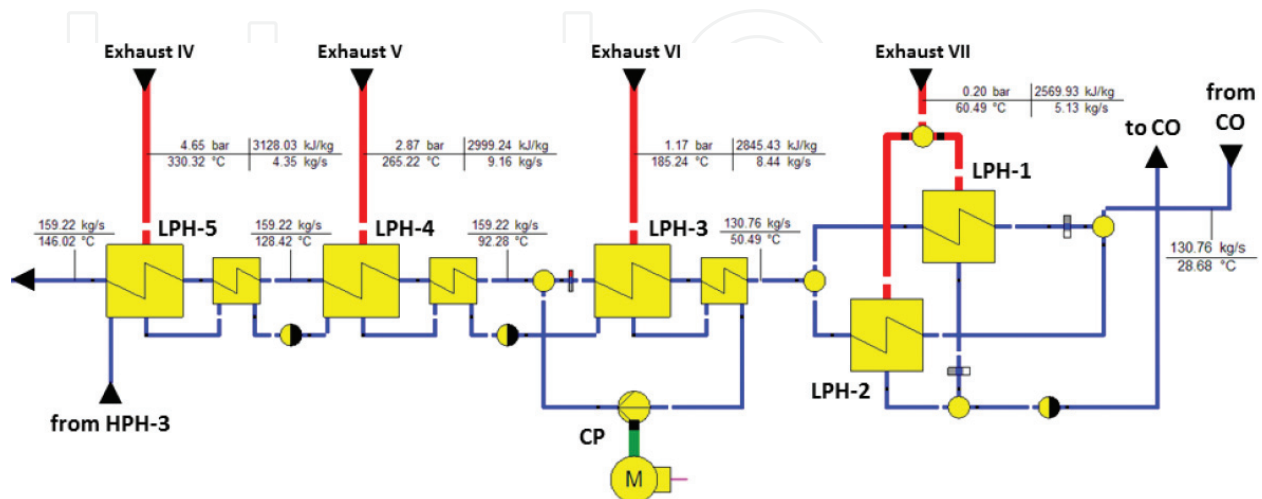

Figure 8. Low-pressure regeneration system in the local design mode.

changes of the isentropic efficiencies in part load calculations and to obtain the reliable results of thermodynamic parameters in steam turbine system.

The relative error for different parameters at the characteristic points in the power unit thermal cycle was used to assess the quality of the developed simulation model. The values of relative error δx_i of selected parameters were calculated according to the following formula:

$$\delta x_i = \frac{|x_{i\ REF} - x_{i\ EBS}|}{x_{i\ REF}} \tag{1}$$

where:

$x_{i\ REF}$ —reference value of the i-th parameter,

$x_{i\ EBS}$ —simulation model value of the i-th parameter.

The values of relative errors calculated using Eq. (1) for different groups of the parameter at part load conditions are presented in **Table 3**. The values of relative errors obtained from the verification process of steam turbine cycle confirm the correctness and accuracy of the simulation model. The analysis shows that the difference between simulation model results and measured data varying from 1.94 to 6.17%. Based on the obtained results it can be concluded that the simulation model of steam turbine cycle may be used for off-design investigation of the analysed 225 MWe power unit. Results in **Table 4** show that with the decrease of power unit load the average relative error increases. This means that the accuracy of the simulation model is the highest for nominal conditions and in the steam turbine part load operation the precision decreases. The main cause of this is the various simulation model assumptions and the accuracy of the developed characteristics of turbine stages. However, it must be also taken into account the uncertainties of measured data used in the verification process.

3.3. Thermodynamic analysis of power unit in off-design conditions

Through simulation analysis, it is possible to determine the specific parameters of thermal power systems under conditions which are different from the nominal. In this chapter, the thermodynamic analysis was carried out to demonstrate the impact of changes in power unit performance under part load operation. One-dimensional simulation modelling is sufficient to obtain the detailed results of power systems behaviour in steady-state conditions. However, it does not give any information about the effects of dynamic changes in power system operation. This type of thermodynamic modelling is a perfect engineering tool to give a global view of the thermal cycle performance [5–7].

	100% load	80% load	60% load	40% load
Pressure	2.40%	3.78%	4.47%	5.08%
Temperature	2.89%	4.97%	5.38%	6.17%
Mass flow	1.94%	4.32%	5.24%	5.23%
Average	2.42%	4.24%	4.72%	5.48%

Table 4. Relative errors in the verification process of power unit under different load.

To determine the technical-operational parameters of the power unit in off-design conditions the following indices were determined based on the results of simulation calculations: gross power of steam turbine generator, isentropic efficiency of steam turbine, isentropic efficiency of HP, IP and LP part of steam turbine, gross thermal efficiency of power unit, heat supplied to the steam cycle and specific consumption of heat. The mathematical formulas concerning quantities and indices specified above are presented below [8].

The gross power of steam turbine generator is defined as follows:

$$N_{el,G} = (N_{i,HP} + N_{i,IP} + N_{i,LP}) \cdot \eta_{mS,T} \cdot \eta_G \qquad (2)$$

where:

$N_{i,HP}$ —internal power of the HP section of the steam turbine, MW

$N_{i,IP}$ —internal power of the IP section of the steam turbine, MW

$N_{i,LP}$ —internal power of the LP section of the steam turbine, MW

$\eta_{mS,T}$ —mechanical efficiency of the steam turbine,

η_G —generator efficiency.

The isentropic efficiency of the steam turbine is calculated from the following equation:

$$\eta_i = \frac{i_1 - i_2}{i_1 - i_{2s}} \qquad (3)$$

where:

i_1 —enthalpy at the inlet to the turbine, kJ/kg

i_2 —enthalpy at the outlet from the turbine, kJ/kg

i_{2s} —enthalpy at the outlet from the turbine after isentropic expansion, kJ/kg

The values of the gross thermal efficiency of power unit result from the following equation:

$$\eta_{el,G} = \frac{N_{el,G}}{\dot{Q}_{ch}} \qquad (4)$$

where:

$\dot{Q}_{ch}$ —the flux of chemical energy of the fuel, MW

The values of the flux of chemical energy of fuel were taken from the simulation calculations of boiler performance in off-design conditions. The amount of heat supplied to the steam cycle in coal-fired boiler result from the equation:

$$\dot{Q}_{sc} = \dot{m}_{LS} \cdot (i_{LS} - i_{FW}) + \dot{m}_{RH} \cdot (i_{HRH} - i_{CRH}) \qquad (5)$$

where:

$\dot{m}_{LS}$ —live steam mass flow rate, kg/s

i_{LS}—live steam enthalpy at the outlet from the boiler, kJ/kg

i_{FW}—feed water enthalpy at the inlet to the boiler, kJ/kg

$\dot{m}_{RH}$—reheat steam mass flow rate, kg/s

i_{HRH}—hot reheat steam enthalpy at the outlet from the intermediate superheater, kJ/kg

i_{CRH}—cold reheat steam enthalpy at the inlet to the intermediate superheater, kJ/kg

The specific consumption of heat in a steam turbine is calculated from the following equation:

$$q_{h,ST} = 3600 \cdot \frac{\dot{Q}_{sc}}{N_{el,G}} \tag{6}$$

The values of selected characteristic indices obtained from the simulation modelling of steam turbine system are presented in **Table 4**. The part load calculations were performed for four different loads of power unit: 100%, 80%, 60% and 40%. For each of off-design conditions, the simulation results were compared with the data obtained from the performance documentation of the steam turbine. The *EBS* and *REF* abbreviations used in **Table 5** mean the calculation results from the Ebsilon®Professional simulation model and the values of selected parameters from the reference documentation, respectively. The obtained results show that with the decrease in power load, the gross thermal efficiency decreases from 40.0% (for 100% load) to 36.8% (40% load). This means that for 40% part load of the power unit, the gross thermal efficiency is 3.2% less than for the nominal conditions. The steam turbine heat rate of 100% load is equal to 8119 kJ/kWh. The steam turbine heat rate increase with power load drop and for 40% load is 9021 kJ/kWh. On this basis, it can be concluded that for 40% load of power unit it is necessary to supply to the steam cycle about 902 kJ of an additional amount of energy for each of kWh produced in the generator. The isentropic efficiency of a steam turbine for nominal conditions is equal to 85.1% and it decreases to 79.1% in 40% load. The isentropic efficiency of HP and LP section of steam turbine also decreases with the power load from the 81.5 to 65.2% and from 87.2 to 84.1%, respectively. The only exception is an

Indices	100% load		80% load		60% load		40% load	
	EBS	REF	EBS	REF	EBS	REF	EBS	REF
Gross power, MW	222.8		179.6		134.9		90.7	
Gross thermal efficiency, %	40.0	39.9	39.6	39.7	38.4	38.6	36.8	36.7
Steam turbine heat rate, kJ/kWh	8119	8100	8164	8175	8358	8380	9021	8999
Isentropic efficiency of turbine, %	85.1	85.2	83.6	83.7	82.1	82.0	79.1	79.0
Isentropic efficiency of HP, %	81.5	81.5	78.1	78.0	74.0	74.3	65.2	65.3
Isentropic efficiency of IP, %	86.6	86.7	86.3	86.5	87.0	87.1	87.6	87.6
Isentropic efficiency of LP, %	87.2	87.0	86.2	86.4	85.2	85.1	84.1	84.0

Table 5. Selected technical-operational parameters of power unit under different load.

isentropic efficiency of IP turbine for which this indices at nominal conditions is equal to 86.6% while in 40% load increases to 87.6%. The thermodynamic analysis of selected power unit performance factors shows that the difference between simulation results and reference data varying from 0.11 to 1.21%. Based on the obtained results it can be concluded that the simulation model of steam turbine cycle accurately confirms the results of the reference performance measurements.

4. Boiler modelling

Boiler modelling was performed using Ebsilon®Professional software with EbsBoiler package. The simulation was made for 100%, 80%, 60% and 40% load of the boiler and results were validated using data from in-situ test campaign. Ebsilon®Professional software allows creating a boiler model using graphical interface embedded in the software. **Figure 9** presents schematic of the reference boiler, with superheaters (SH), reheaters (RH) and water heater (WH) depicted. A number of particular heat exchanger represents the degree of overheating of the steam—SH 5, for instance, is a final superheater with steam outlet temperature equal to about 535°C. **Figure 10** depicts part of the model, that represents combustion chamber and evaporator. Furthermore, simulated processes are controlled in the same way it is done on the reference unit—for instance amount of air supplied to the combustion chamber in the model is controlled by the level of oxygen at the boiler outlet. Values depicted in **Figure 10** present different parameters of the combustion process in full load operation. Fuel flow is equal to 98.697 t/h while total air to combustion is equal to 618012.633 m^3/h. Primary and secondary air flows are modelled as a single stream. However, if more precision in calculations of combustion chamber itself is required, air staging would have to be modelled. Level of oxygen at the outlet of combustion zone is equal to 4.499% while air-fuel ratio is equal to 1.302. The latter value does not represent real stoichiometry of combustion in the reference boiler, due to simplification that was made in case of primary and secondary air streams. **Figure 11** depicts part of the model that represents areas of the boiler (D, D′, D″) from where heat is taken to particular heat exchangers—for instance, heat flows from area D′ to the superheater 4 and 1 subsequently. There is also a water spray injection model, with water flow equal to 3.722 t/h. In steady-state modelling of the boiler, water spray injection is difficult to simulate. In reality, this value varies heavily during operation, therefore it can only be simulated as an average value. In general, this way of modelling requires deep knowledge about the simulated object and its operating conditions.

Results of the simulation indicate boiler behaviour in different load levels, including load 40% which is below minimum safe design load of the reference boiler. **Figure 12** presents result of flue gas temperature distribution inside the boiler, for live steam flow equal to 650 t/h (100% load), 520 t/h (80% load), 390 t/h (60% load) and 260 t/h (40% load). Temperature values depicted in **Figure 12** should be treated as average from given boiler areas (perpendicular to the flue gas flow direction) and not as exact temperature points. What can be seen, is that the difference in temperatures is more significant between 60 and 40% load than between 80 and 60% load.
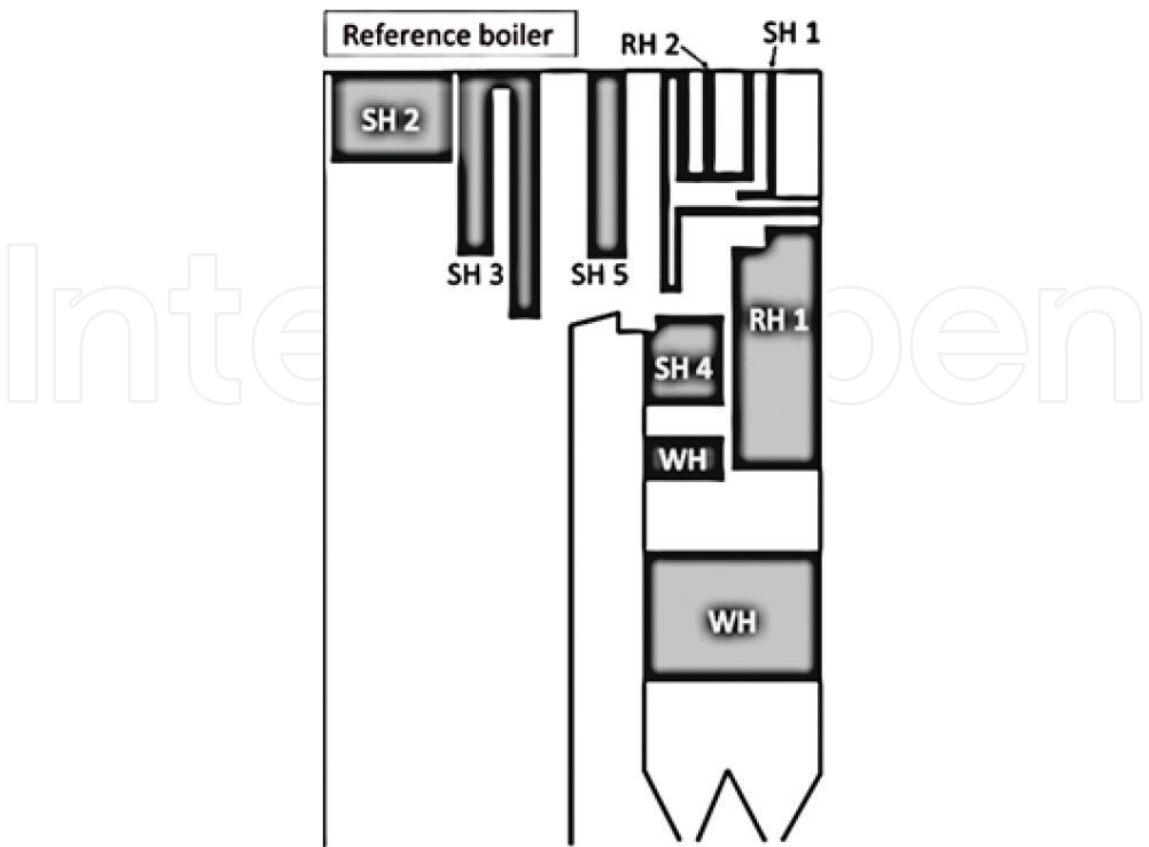
Figure 9. Reference boiler description (SH—superheater, RH—reheater, WH—water heater).
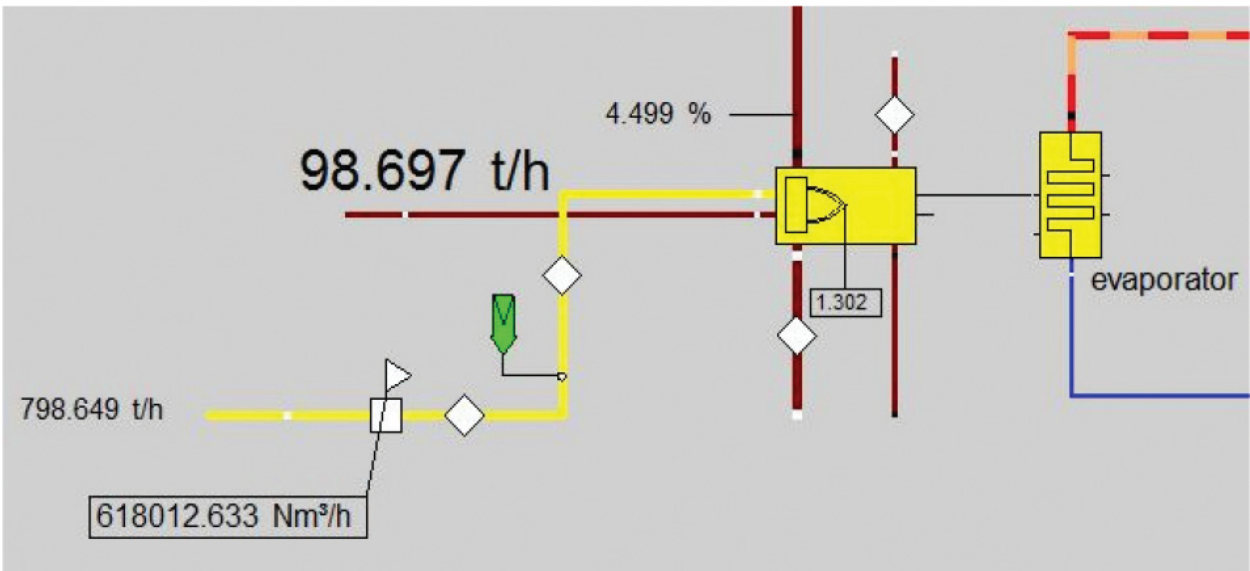

Figure 10. Combustion chamber and evaporator depicted in the model.

Figure 13 presents results of live and reheated steam temperature at the boiler outlet for different load levels. What can be observed, is that both temperatures undergo a steep descent between 60 and 40% load. In the latter case, the live steam temperature is below 520°C and reheated steam temperature falls down below 490°C. Both values are considerably lower than designed
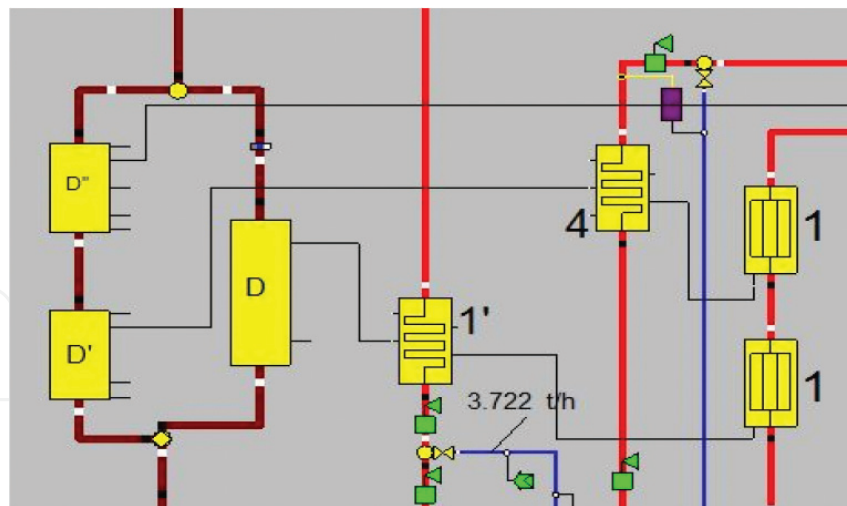
Figure 11. Flue gas path, superheaters and water spray injection depicted in the model.
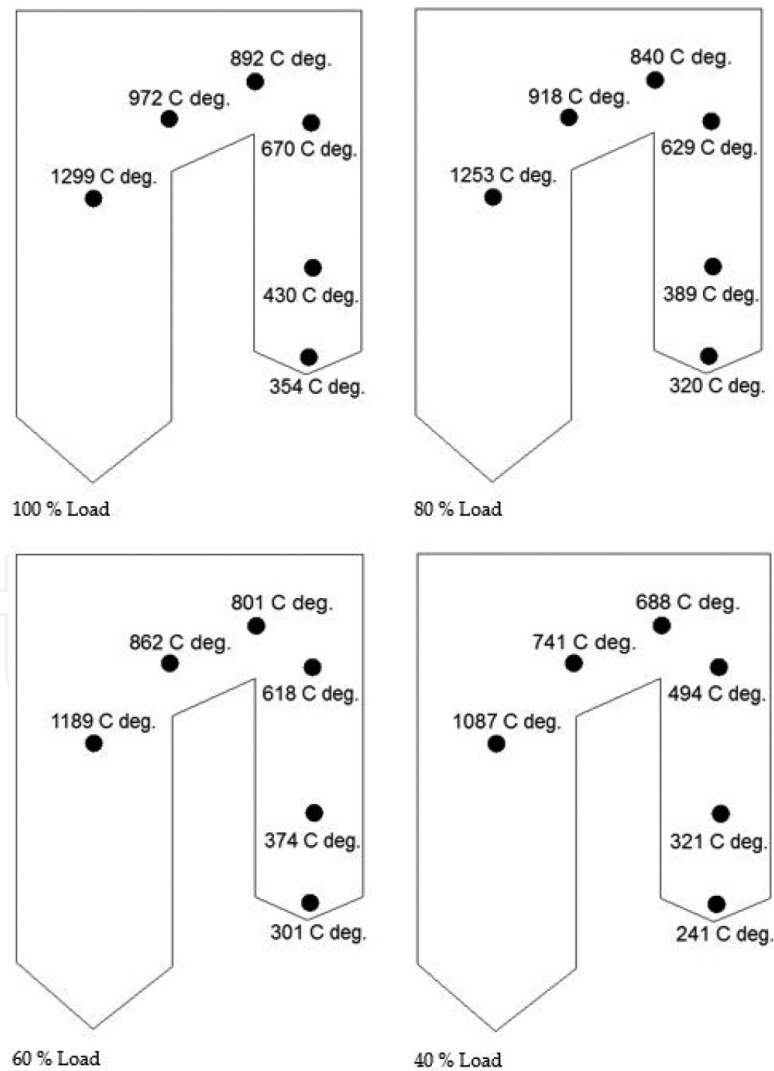
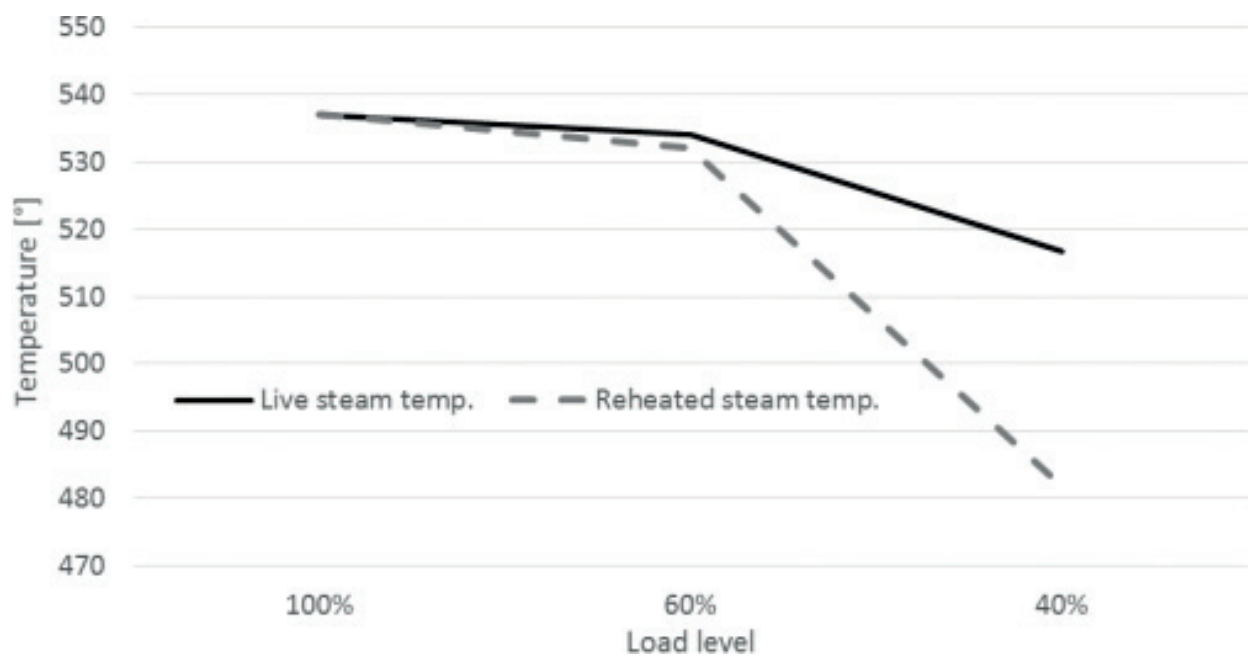**Figure 12.** Flue gas temperature distribution inside the boiler for different load levels.

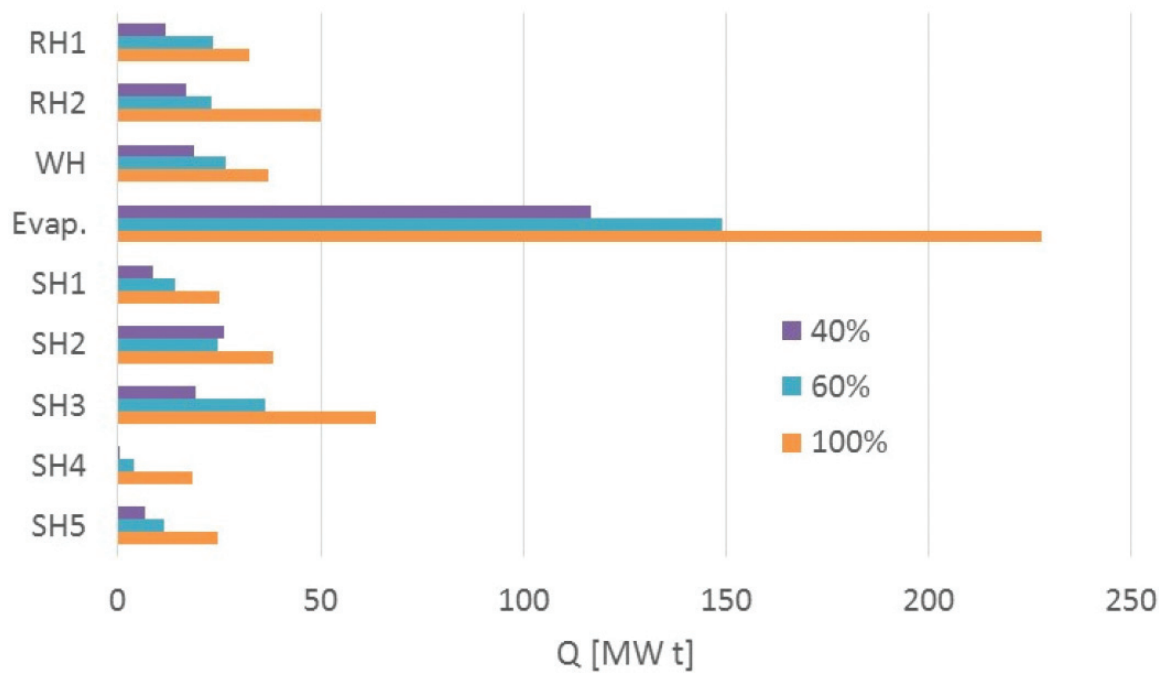**Figure 13.** Steam temperatures in different load levels.

Figure 14. Heating power delivered to each heat exchanger for different load levels (100%, 60%, 40%).

temperature equal to 535°C. **Figure 14** presents results from heating power Q taken by each heat exchanger in the boiler. What is interesting, is that second superheater (SH 2) has got higher power in 40% than in 60% load. Also, power taken by fourth superheater (SH 4) in 40% load is almost negligible. **Figure 15** depicts descend of flue gas temperature at the outlet of air preheater.
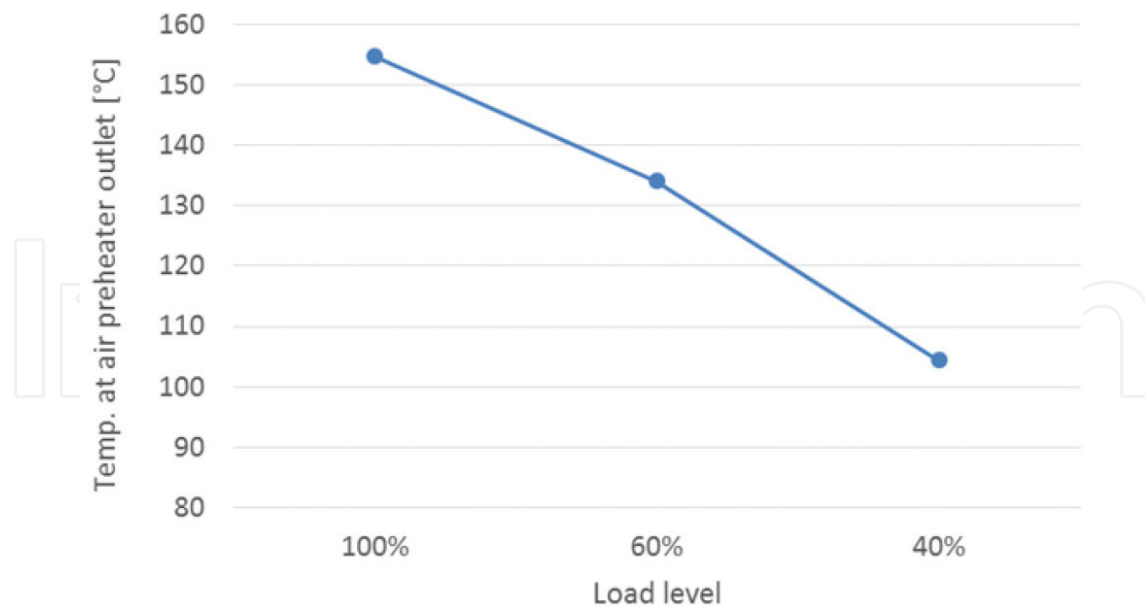
Figure 15. Flue gas temperature at the outlet of air preheater.

5. Discussion

Outcomes from simulation modelling revealed that reference power unit can experience a variety of issues when operating with the decreased minimum safe load. Steam temperatures presented in **Figure 8** indicate, that operation in 40% load is much less efficient than in 60% load. However, there is an economic sense of such configuration when shut down and start-up costs are taken into consideration. Flue gas temperatures presented in **Figure 12** indicate other potential issues. The vastly important threat is related to sulphuric acid (H_2SO_4) condensation. Formation of H_2SO_4 occurs in a temperature range between 200 and 400°C while its content depends on the amount of sulphur trioxide SO_3 and water vapour in the flue gas. The temperature of condensation decreases proportionally to the content of sulphuric acid or water vapour in flue gas and it is in the range of 95–160°C [9–11]. However, roughly 90% of sulphuric acid condenses in a temperature range between 115 and 138°C [12]. According to data presented in **Figures 12** and **15**, in 40% load temperature at the outlet of air preheater is close to 100°C. That increases the probability of sulphuric acid condensation inside the air preheater, which can eventually lead to corrosion problems. Corrosion can also be an issue when it comes to electrostatic precipitator (ESP) operation, however low flue gas temperature is in general favourable here because it decreases dust resistivity [13].

Another concern with low load boiler operation is related to de-NOx installations. Temperature value at the outlet of combustion zone depicted in **Figure 12** decreases from about 1190 to 1090°C. Considering low load operation on boilers equipped with SNCR technology, it must be confirmed that reagent injection system is able to adjust to the new conditions—optimal injection temperature window for NOx removal is roughly between 950 and 1025°C [14]. Regarding

SCR operation, most of the current commercial installations are based on V_2O_5/TiO_2 catalyst, which has considerably high working temperature, roughly between 300 and 400°C. Furthermore, the activity of such catalysts decreases with temperature [15, 16]. SCR installation is normally designed for full load operation temperatures, but it should also work well in minimum load conditions. In case of reference boiler, that would be 60% load. Simulation outcomes reveal, that in 40% load flue gas temperature in SCR relevant region drops below 300°C.

Operation of the steam turbine in part load conditions as well as below current technical minimum involves many technical and economic aspects. The simulation results for a selected different load of power unit shows that analysed power unit can operate stably in the range between 90 and 225 MW. The thermodynamic analysis demonstrates, that the gross thermal efficiency varies between 40.0 and 36.8%. The steam turbine heat rate was calculated to determine the quality of steam turbine performance in off-design conditions. The heat rate of steam turbine cycle increases from 8119 kJ/kWh (for 100% load) to 9021 kJ/kWh (40% load). **Figure 16** shows the course of gross thermal efficiency and steam turbine heat rate. What can be seen, is that the heat rate and efficiency have non–linear characteristic. Presented indices rapidly change in the range between 90 and 135 MW. The thermal efficiency decrease from 38.6 to 36.8% and the steam turbine heat rate increases from 8358 to 9021 kJ/kWh in load range below the current technical minimum.

The simulation calculations reveal that the 225 MW power unit can operate between 40 and 100% load with reasonable efficiency. Power unit operation with reduced minimum load allows decreasing start-up and shut-down operation costs. Reducing the minimum load of
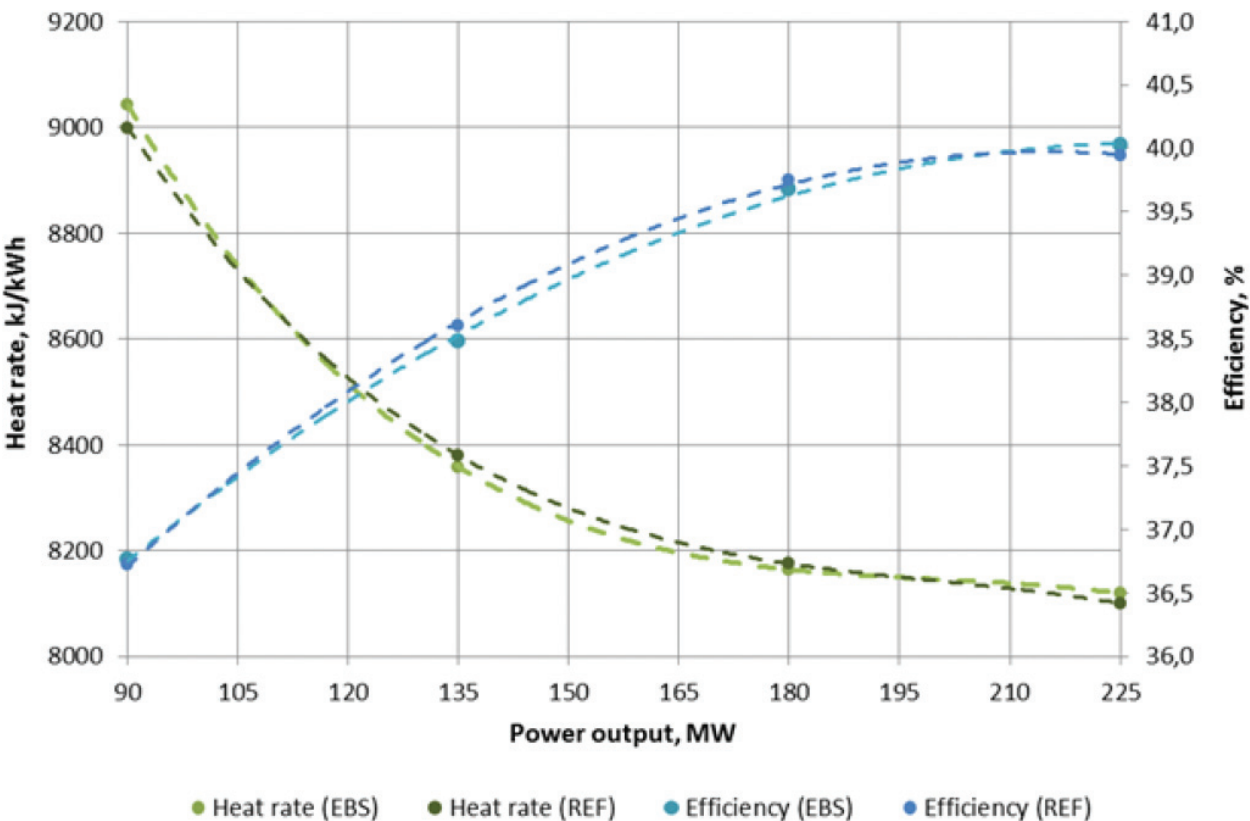

Figure 16. Power unit heat rate and thermal efficiency for part load conditions.

conventional coal-fired power plants strongly influences operation of the steam turbine. The turbine ventilation is one of the main limiting factors in the load range of steam turbine. Insufficient steam flow causes temperature increase and thermal stresses in turbine stages. It is important to revise all of the control loops and measuring devices in case the of power unit operation at low loads to avoid exploitation and performance problems [7, 17]. The steady-state simulation analysis of the power unit model shows a good accordance to reference operating data throughout the power unit operating range. Based on the results of verification process it can be stated that simulation model of steam turbine cycle developed in Ebsilon®Professional can be used as an engineering tool for investigation of power unit performance in off-design conditions.

6. Conclusions

Simulation modelling described in the paper can be recommended as an efficient and accurate method to evaluate power unit operation in different conditions. Authors investigated the operation of '200 MWe' class coal-fired power unit in reduced minimum load regime. Performed tests revealed, that this is a technically feasible way of operation that can improve the flexibility of that sort of units. However, reducing the unit load from designed minimum safe load, can bring about variety of problems. Vastly decreased efficiency, inaccurate measurements and operation problems derived from low flue gas temperature are the most important ones. On the other hand, ability to work in extended load range may decrease number of shut down and start-ups of the unit, which is very desirable from power plant operator point of view. To conclude, it must be said that reference 200 MWe class coal-fired unit, as well as other similar units, have potential of flexibility improvement that is worth considering.

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