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Efficiency and power upgrade by an additional high pressure economizer installation at an aged 620 MWe lignite-fired power plant

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ABSTRACT

An additional high pressure economizer was installed at Unit B1 of the 620 MWe lignite-fired Power Plant "Nikola Tesla B" after 30 years of its operation. An innovative connection of the new additional economizer was applied. It is in parallel connection to the first section of the originally built economizer and it is directly fed with the feedwater from the outlet of the feedwater pump. The analysis of Unit B1 operation with such an economizer arrangement is performed and it is supported by measured data. It is shown that more than 30 MWth of the flue gas waste heat is recovered. The Unit gross efficiency is increased by 0.53 percentage points, which leads to 9.4 MWe of the electric power production. The parallel connection of the additional economizer also leads to the partial feedwater bypass of the high pressure heaters, which enables an increase of the plant electric power by up to 24.5 MWe. The accompanying effects are the reduction of the pressure drop in the feedwater line and the economizers, which leads to the decrease of the energy consumption for the main feedwater pump operation. The applied solution is presented together with measured and calculated energy and economic benefits.

1. Introduction

Electricity is indispensable for modern societies. For example, many industrial machines and devices, household appliances and information technologies are powered by electricity. Its consumption is related to the Human Development Index, which is a composite measure of health, education and income or a measure for human well-being [1]. In 2011 the share of electricity in the final energy consumption in the world was 17.7%, while this share in the economically developed part of the world represented by the OECD (the Organisation for Economic Co-operation and Development) countries is even higher 22% [2]. Hence, a lot of attention is applied to the efficiency of electricity final consumption, for instance in residential and industrial sectors [3]. At the same time, a substantial increase of the overall efficiency of energy systems can be achieved by measures on the electricity supply side, especially in countries where electricity is produced in thermal power plants by coal combustion. Coal is still the main energy source for electricity production in the world, according to [2] its share was 41.3% in 2011. Recent commercial developments of the technology of thermal power plants that operate under steam Rankine cycles have risen their efficiency, but the majority of thermal power plants are built decades ago and their efficiency is still below 40% [4]. Therefore, the increase of efficiency of aged thermal power plants leads to the reduction in coal consumption and the increase of overall energy efficiency, especially in countries that are dependent on coal consumption. Examples of other important benefits are the reduction of carbon-dioxide emission and increase of electricity generation economy.

The utilization of waste heat of flue gases at thermal power plants leads to an important increase of efficiency in electricity production. In Ref. [5] it is indicated that a reduction of the exit flue gas temperature from 130 °C to 120 °C at a pulverized coal-fired thermal power plant increases the net plant efficiency by about 0.3 percentage points. Although this is only a fraction of one percent it leads to substantial fuel conservation, especially at the large thermal power plants that operate with the electric power of several hundred megawatts or even in the range of 1 GW throughout the year. The utilization of waste heat and the cooling of

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flue gases to temperatures below 130 °C is a common procedure at thermal power plants equipped with the flue gas desulphurization systems. Besides the waste heat utilization, the cooling of the flue gas provides benefits for the desulphurization process, such as a reduction of energy and water consumption and an increase of desulphurization efficiency. At the 900 MWe lignite-fired thermal power plants Schwarze Pumpe, Lippendorf and Boxberg in Germany, specially designed heat exchangers were installed behind the electrostatic precipitator and in front of the flue gas desulphurization plant with the aim of waste heat utilization from the exhaust flue gas, as shown in Fig. 1 [6]. The heat recovered from the flue gas is transferred by a water circulation system and through intermediate heat exchangers into the condensate line in the steam turbine plant. This solution provides the flue gas cooling from 170 °C to 130 °C and an increase of the power plant efficiency by approximately 0.5%. A similar design solution was analyzed in Ref. [7] for a 600 MWe coal-fired power plant. It was assumed that the heat recovered in a heat exchanger from the flue gas is transferred to water directly taken from the condensate line in the steam turbine plant, without application of the intermediate heat exchangers. The application of finned tubes was considered in the flue gas-water heat exchanger. Cases analyzed in Ref. [7] show the flue gas cooling from 123 °C to temperatures below 100 °C. Comprehensive design solutions for the utilization of waste heat from the flue gas were applied at the 1000 MWe lignite-fired power plant Niederaussem Unit K [8]. The heat recovered by cooling the flue gas in a gas-water heat exchanger, from approximately 160 °C–100 °C or even lower, in front of the desulphurization plant, is transported by a closed water circuit to a water–air heat exchanger that preheats the combustion air (Fig. 2). In addition, high pressure and low pressure economizers are installed in a bypass flue gas channel behind the main economizer bank at the top of the steam generator. The high pressure bypass economizer provides heat directly to the high pressure feedwater in parallel with the high pressure heaters 1 to 3. The low pressure bypass economizer provides heat to a naturally recirculating system incorporating a steam drum that transfers heat to the condensate line through a heat exchanger, which is in parallel connection with the low pressure heater 5. The installation of the low pressure economizer for the flue gas heat recovery was analyzed in Ref. [9] based on the data from an existing 1000 MW typical power generation unit in China. Four possible arrangements for water taking from the condensate line, heating in the economizer and returning to the condensate line were investigated. It was found that a delivery of the recovered heat to a higher temperature section of the condensate line with regenerative heaters provides higher energy-saving effects than a delivery to a colder part. The example of low pressure economizer installation in the Shanghai Waigaoqiao No. 3 power plant is reported in Ref. [9] based on the original paper published in Chinese in Ref. [10]. In this case the condensed water is used from the inlet of the 7th low-pressure regenerative heater to retrieve the waste heat of flue gas. This solution reduces the design temperature of the flue gas

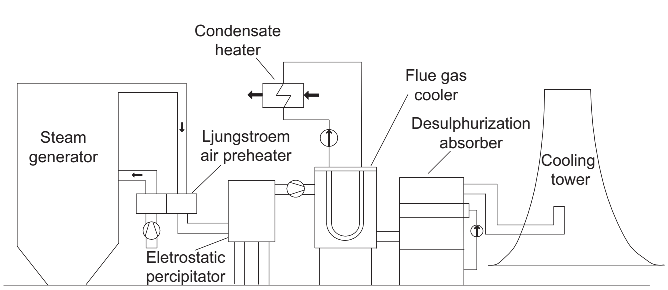

Fig. 1. Flue gas cooling in front of the desulphurization plant [6].

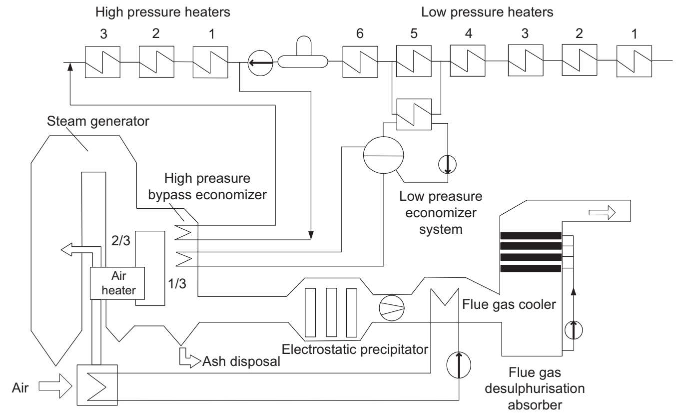

Fig. 2. Multiple solutions for the recovery of waste heat from the flue gas at the power plant Niederaussem Unit K [8].

from 125 °C to 85 °C, which improves boiler efficiency by 2% points and overall unit efficiency by 0.8–0.9% points.

It should be mentioned that there are numerous methods for the utilization of the flue gas waste heat. In Ref. [11] the usage of the heat pumps is analyzed, in Ref. [12] various solution methods applicable to waste-to-energy process plants are presented, and a development of systems based on the Organic Rankine Cycles for the waste heat utilization from low-temperature flue gases is presented in Refs. [13], to mention a few.

In this paper an innovative and applied solution for the waste heat utilization from the steam boiler exit flue gas is presented. It was applied at an aged, 30 year old unit of the lignite-fired Thermal Power Plant “Nikola Tesla B” in Serbia. It consists of an additional economizer that was installed parallel to the first section of the originally built economizer. The additional economizer is fed with the feedwater by a newly installed separate feedwater line that is connected to the main feedwater pump discharge line. In Section 2 the motivation for the retrofit of Unit B1 at the Thermal Power Plant “Nikola Tesla B” is presented together with the solution of the additional economizer installation. The applied approach to the energy analysis of the retrofitted power plant is presented in Section 3. In Section 4 presented and discussed are the results of the energy efficiency and electric power upgrade at the power plant, which are achieved by the additional economizer installation. The economic evaluation of the project is also performed. The achieved energy and economic benefits are outlined in the last section under Conclusions.

2. Upgrade of the lignite-fired thermal power plant by the installation of the additional economizer

The Thermal Power Plant “Nikola Tesla B” consists of two identical Units B1 and B2 with designed electric power 620 MWe per each unit. The fuel is pulverized lignite with the lower heating value in the range from approximately 5800 to 8000 kJ/kg. Unit B1 started operation in 1983 and Unit B2 in 1985, and their accumulated operating time is respectively more than 210,000 and 196,000 h. Due to the long operating period, projects of revitalization and modernization have been conducted, together with a project for the power increase. A substantial potential for the increase of the plant’s efficiency and power was observed in the recovery of waste heat from the steam boilers’ exit flue gas. Namely, the temperature of the exit flue gas after the regenerative air preheaters of the Ljungstrom type is in the range from 180 °C to 190 °C, while the design temperature for the clean heat transfer surfaces in the boiler is 151 °C (the minimum temperature that does not lead to the sulfuric acid condensation and low temperature corrosion). The main cause of the high temperature of the exit flue gas is the low cold air flow from the surrounding atmosphere through the Ljungstrom air preheater. This air flow is kept low because of the great inleakage of cold air from the surrounding into the boiler's furnace and gas path through untight joints between various elements and through the opening at the dry-bottom furnace (the boiler operates at a negative pressure formed by induced-draft fans) [14]. The low tightness of the boiler has been the constant problem for the previous long operational period. Therefore, the additional economizer is installed at Unit B1 in 2012 as a solution for the reduction of the exit flue gas temperature and waste heat utilization.

Fig. 3 shows the configuration of the power plant Unit B1 with the installed additional economizer ECO 1A. ECO 1A is installed at the top of the once-through tower type boiler. Regarding the feedwater flow, ECO 1A is in parallel connection to the first section ECO 1 of the originally built economizer. ECO 1A is fed with water by the newly installed additional feedwater pipeline (marked with 1 in Fig. 3), which is connected to the main feedwater pump discharge line. The feedwater to ECO 1 is delivered through the main feedwater pipeline (2 in Fig. 3) with high pressure feedwater heaters (17,18,19). From 25% to 40% of the total feedwater flow to the boiler economizers ECO 1 and ECO 1A is delivered to the additional ECO 1A, while the rest flows through the high pressure heaters to ECO 1. The feedwater from ECO 1 and ECO 1A is mixed and enters the second original economizer section ECO 2. From ECO 2 the feedwater is transported through the downcomer pipes (3) to the bottom of the boiler furnace and fed to the spiral evaporator pipes (4), which form the membrane water walls of the furnace. Above the furnace the steam generation is continued in vertical evaporating tubes, and after that the steam is superheated in four superheaters. The live steam from the boiler enters the high pressure section of the steam turbine (5). After the expansion in the high pressure section the steam is returned to the boiler for reheating (6). The reheated steam is fed to the intermediate pressure section of the steam turbine (7) with two passes and then to the two low pressure sections (8) with two passes per each section. The exhaust steam is condensed in a condenser (9) cooled by the river water. The condensate is pumped through the condensate line and heated in four low pressure heaters (10,11,12,13) by the steam extracted from the intermediate and low pressure turbine sections. The condensate line feeds the feedwater tank (14), and water from the feedwater tank is pumped by the main feedwater pump (15) to the steam boiler. The main feedwater pump is driven by the auxiliary steam turbine (16). The feedwater is heated in three high pressure heaters (17,18,19) with steam extracted from the high pressure and intermediate pressure section of the steam turbine.

The economizer sections are shown in Fig. 4. The additional economizer ECO 1A has a heat transfer surface of 19,200 m^2, the economizer tubes outer diameter and wall thickness are $\phi 33.7 \times 4.5$ mm, the number of tube panels is 199, the number of tubes within one panel is 2, the length of one tube within the panel is approximately 456 m. The heat transfer surface of ECO 1A is 56% of the sum of heat transfer surfaces of ECO 1 (15,550 m^2) and ECO 2 (18,450 m^2).

The presented solution of the installed additional economizer at Unit B1 of the Thermal Power Plant "Nikola Tesla B" is similar to the application of the high pressure bypass economizer at the power plant Niederaussem Unit K [8]. The difference is in the connection of the additional economizer outlet with the main economizer section. At Unit B1 the outlet of the additional economizer ECO 1A is connected to the outlet of the first section of the originally built economizer ECO 1, i.e. ECO 1A and ECO 1 are in parallel connection (Fig. 3). In Unit K of the plant Niederaussem the high pressure bypass economizer is in serial connection to the main economizer section (Fig. 2). The rationale behind the solution with the parallel connection of ECO 1A and ECO 1 at Unit B1 was to reduce the pressure drop from the main feedwater pump to the inlet of the high pressure section of the steam turbine, since pressure drops much higher than the design values were indicated in the aged steam boiler pipelines prior to the upgrade and they were the limitation for the increase of the working fluid flow rate. Other solutions with the installed additional economizers at the steam boilers of power plants, as presented in the Section 1 Introduction, are also different from the solution presented in this paper, since they are based on the delivery of heat recovered from the flue gas to the condensate line, such as in the power plants Schwarze Pumpe, Lippendorf and Boxberg in Germany [6] and in the power plant Shanghai Waigaoqiao No. 3 in China [10]. The same holds for the low pressure economizer connected with the condensate line, thermodynamically analyzed as the possible solution for the coal-fired power plants of 600 MWe [7] and 1000 MWe [9] in China.

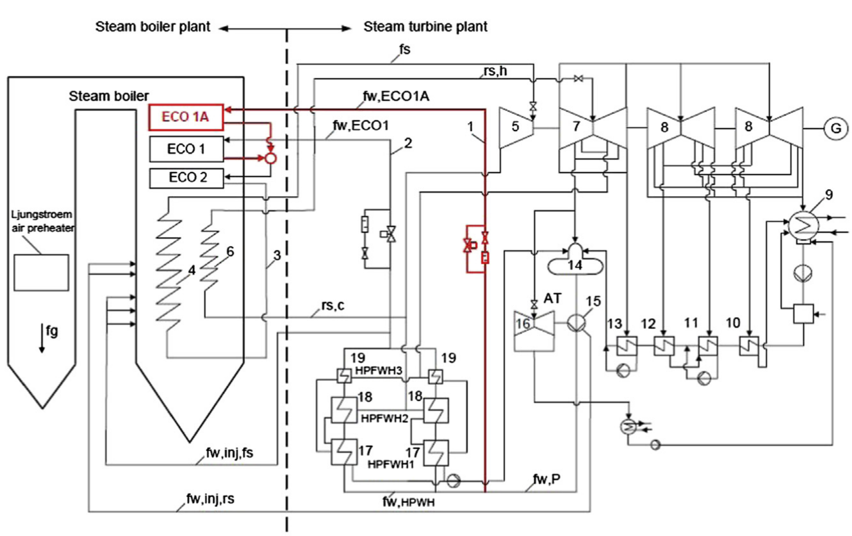

Fig. 3. Additional economizer ECO 1A installation at Unit B1 of the Thermal Power Plant "Nikola Tesla B".

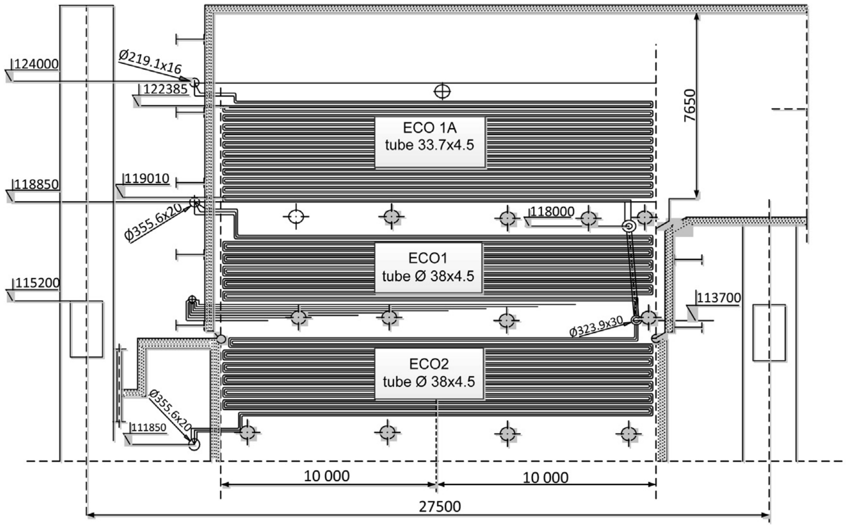

Fig. 4. Layout of the additional economizer ECO 1A.

3. Approach to energy analysis of the retrofitted plant operation

The installation of the additional economizer at Unit B1 of the Thermal Power Plant "Nikola Tesla B" has two dominant effects on the unit's energy efficiency. The heat recovered from the flue gas increases the steam boiler efficiency, but the reduction of the feedwater flow through the high pressure heaters reduces the steam extraction from the high and intermediate pressure sections of the turbine for the regenerative heating and consequently reduces the steam turbine plant efficiency. Nevertheless, the influence of the steam boiler efficiency increase is more dominant and the total efficiency of Unit B1 increases. The influence of the additional economizer installation on the plant's efficiency and electric power upgrade is analyzed as follows.

Fig. 5 shows the energy flows among the steam boiler plant, the steam turbine plant and the environment. The installation of the additional economizer decreases the rate of heat loss with the exit flue gas $\dot{Q}_{fg}$ and the heat flow rate recovered in the additional economizer is transferred from the steam boiler to the steam turbine plant, which increases the electric generator power P_G. In addition, the reduced flow of the feedwater through the high pressure heaters (since a part of the total feedwater flow towards the economizers is directly transported from the discharge of the main feedwater pump towards the additional economizer and bypasses the high pressure heaters) reduces the steam extraction from the high pressure and intermediate pressure sections of the steam turbine, which leads to the increased steam turbine power and consequently to the increase of the plant electric power P_G.

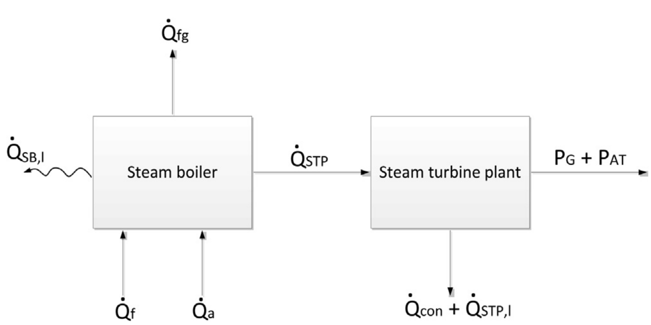

Fig. 5. Heat flow rates between the steam boiler, the steam turbine plant and the environment ($\dot{Q}_f$-the fuel heat power, $\dot{Q}_a$-the heat flow rate of the air input, $\dot{Q}_{SB,l}$-the rate of steam boiler heat loss (without the exit flue gas loss), $\dot{Q}_{fg}$-the heat flow rate with the exit flue gas, $\dot{Q}_{STP}$-the heat power delivered to the steam turbine plant, $\dot{Q}_{con} + \dot{Q}_{STP,l}$-the power of conversion consumption and other losses, P_G-the electric generator power, P_{AT}-the mechanical power of the auxiliary turbine).

A change of the total gross efficiency of the thermal power plant unit is defined as the difference in the plant gross efficiencies after and before the economizer installation

$$\Delta\eta_{TPP} = \eta_{TPP} - \eta_{TPP,0} \tag{1}$$

Since the total gross efficiency of the thermal power plant unit is determined by the product of the efficiencies of the steam boiler and the steam turbine plant, Eq. (1) is rewritten as

$$\Delta\eta_{TPP} = \left(\eta_{SB,0} + \Delta\eta_{SB}\right)\left(\eta_{STP,0} + \Delta\eta_{STP}\right) - \eta_{SB,0}\eta_{STP,0} \tag{2}$$

which results in

$$\Delta\eta_{TPP} = \Delta\eta_{SB}\eta_{STP,0} + \Delta\eta_{STP}\eta_{SB,0} \tag{3}$$

where the product $\Delta\eta_{SB}\,\Delta\eta_{STP}$ is neglected, since it is close to zero.

A change of the steam boiler efficiency is defined as

$$\Delta\eta_{SB} = \eta_{SB} - \eta_{SB,0} = \frac{\dot{Q}_{STP}}{\dot{Q}_f} - \eta_{SB,0} = \frac{\dot{Q}_{STP}}{\dot{Q}_{f,0} + \Delta\dot{Q}_f} - \eta_{SB,0} \tag{4}$$

The change of the fuel heat power is defined from the steam boiler energy flow balance, which reads

$$\dot{Q}_f + \dot{Q}_a = \dot{Q}_{STP} + \dot{Q}_{fg} + \dot{Q}_{SB,l} \tag{5}$$

The explanation of the terms in Eq. (5) is given in the title of Fig. 5. The installation of the additional economizer leads to the change of these energy flows and the change of the fuel heat power is calculated as

$$\Delta\dot{Q}_f = \Delta\dot{Q}_{STP} + \Delta\dot{Q}_{fg} + \Delta\dot{Q}_{SB,l} - \Delta\dot{Q}_a \tag{6}$$

The rate of heat losses from the steam boiler $\dot{Q}_{SB,l}$ takes into account the losses due to the incomplete combustion, the unburned carbon, the cooling through the boiler lining and the slag removal. These heat loss rates are expressed as a part of the fuel heat power input available by the combustion of pulverized lignite

$\dot{Q}_{SB,l} = f\dot{Q}_f$, where the estimated value of fraction f is not greater than 0.03 (3%) [14] (for the pulverized lignite combustion the heat loss by the incomplete combustion is neglected, by the unburned carbon is up to 1%, by cooling through the boiler lining is up to 0.1% and the heat loss with physical heat of slag is up to 2%). It is assumed that the fraction f does not change with the installation of the economizer and $\Delta\dot{Q}_{SB,l} = f\Delta\dot{Q}_f$ The rate of energy input into the boiler by atmospheric air intake is expressed as a fraction of fuel power $\dot{Q}_a = g\dot{Q}_f$, where the value of fraction g is approximately 0.01 (1%). Assuming that g is constant it is obtained $\Delta\dot{Q}_a = g\Delta\dot{Q}_f$. Finally, the change of fuel heat power is derived from Eq. (6)

$$\Delta\dot{Q}_f = \left(\Delta\dot{Q}_{STP} + \Delta\dot{Q}_{fg}\right)\Big/(1 - f + g) \tag{7}$$

It should be mentioned that the change of the exit flue gas heat flow rate $\Delta\dot{Q}_{fg}$ in Eq. (7) is a negative value due to the additional economizer installation.

The energy flow rate from the steam boiler to the steam turbine plant is calculated as

$$\dot{Q}_{STP} = \dot{m}_{fs}\left(h_{fs} - h_{fw,SB}\right) + \dot{m}_{rs,c}\left(h_{rs,h} - h_{rs,c}\right) + \dot{m}_{fw,inj,rs}\left(h_{rs,h} - h_{fw,inj,rs}\right) \tag{8}$$

where the fresh steam mass flow rate from the steam boiler to the high pressure section of the turbine is $\dot{m}_{fs}$ (the fresh steam and other lines are marked in Fig. 3), h_{fs} is the enthalpy of the fresh steam in front of the high pressure section of the turbine, $h_{fw,SB}$ is the mean enthalpy of the feedwater at the steam boiler inlet, $\dot{m}_{rs,c}$ is the cold reheated steam mass flow rate to the steam boiler, $h_{rs,h}$ is the enthalpy of the hot reheated steam at the inlet to the intermediate section of the steam turbine, $h_{rs,c}$ is the enthalpy of the cold reheated steam at the outlet of the intermediate section of the steam turbine, $\dot{m}_{fw,inj,rs}$ is the feedwater injected to the steam boiler for the reheated steam temperature control, $h_{fw,inj,rs}$ is the enthalpy of the feedwater injected to the reheated steam. The mean enthalpy of the feedwater at the steam boiler inlet is calculated as

$$h_{fw,SB} = \left(\dot{m}_{fw,HPFWH}h_{fw,HPFWH,out} + \dot{m}_{fw,ECO1A}h_{fw,HPFWH,in}\right)\Big/m_{fs} \tag{9}$$

where $\dot{m}_{fw,HPFWH}$ is the feedwater flow through both lines of the high pressure feedwater heaters, $h_{fw,HPFWH,out}$ is the feedwater enthalpy at the outlet of the high pressure feedwater heaters, $\dot{m}_{fw,ECO1A}$ is the feedwater flow rate to the additional economizer ECO 1A, and $h_{fw,HPFWH,in}$ is the enthalpy of the feedwater in front of the high pressure feedwater heaters.

The feedwater flow rate towards the additional economizer $\dot{m}_{fw,ECO1A}$ is determined by the control system

$$\dot{m}_{fw,ECO1A} = \varphi\left(\dot{m}_{fw,ECO1} + \dot{m}_{fw,ECO1A}\right) \tag{10}$$

where the feedwater flow through the additional economizer ECO 1A is in the range from one fourth to above one third of the total feedwater flow towards the steam boiler evaporator, i.e. $\varphi = 0.25 \div 0.40$. This amount is justified by the upgrades of the thermal power plant unit efficiency and the electric power, as it is presented in the next section. The feedwater flow rate into the originally built economizer section ECO 1 is calculated as the difference between the feedwater flow rate at the outlet of the high pressure heaters and the feedwater flow rate that is injected into the fresh steam superheating line for the temperature control

$$\dot{m}_{fw,ECO1} = \dot{m}_{fw,HPFWH} - \dot{m}_{fw,inj,fs} \tag{11}$$

The change of the flue gas heat flow rate $\Delta\dot{Q}_{fg}$ at the steam boiler exit (after the air regenerative preheater of the Ljungstrom type) is predicted as

$$\Delta\dot{Q}_{fg} = \dot{m}_{fg}c_{p,fg}T_{fg} - \dot{m}_{fg,0}c_{p,fg,0}T_{fg,0} \tag{12}$$

A change of the steam turbine plant efficiency $\Delta\eta_{STP}$, due to the installation of the additional economizer ECO 1A, is predicted on the basis of measured process parameters. The data applied in the calculation of the turbine plant efficiency are the arithmetic means of the values sampled in intervals of 2 s during 5 min period. The temperatures were measured with the K-type thermocouples with the absolute error lower than 0.5 K. The pressures were measured with the industrial type transmitters with ceramic measuring cell and electronic module with the relative error 0.25%. The mass flow rate measurements were based on the pressure difference measurements at the orifices. The relative error of mass flow rate measurements for steam was below 2% and for feedwater below 1%. The measured data were collected by the acquisition system that is a part of the steam unit's digital control system. The steam turbine plant efficiency is determined as

$$\eta_{STP} = \frac{P_G + P_{AT}}{\dot{Q}_{STP}} \tag{13}$$

where the heat power used in the steam turbine plant $\dot{Q}_{STP}$ is predicted with Eq. (8) and it is calculated for sets of measured data under operational regimes with different feedwater flows through the additional economizer ECO 1A. The pressure, temperature and mass flow rates of the steam and feedwater streams at the boundaries between the steam boiler and the steam power plant are measured. These are the parameters of the fresh steam in front of the high pressure section of the steam turbine, the steam extracted from the high pressure section of the steam turbine and taken to the steam boiler for reheating, the hot reheated steam in front of the intermediate pressure section of the steam turbine, the feedwater flowing into the economizers ECO 1 and ECO 1A and the feedwater injections into the superheated and reheated steam lines. The enthalpies in Eq. (8) are calculated with equations of state on the basis of measured pressure and temperature data, i.e. $h = h(p,T)$. In addition, measured is the electric generator power P_G. The auxiliary turbine power P_{AT}, which drives the main feedwater pump, is calculated on the basis of the measured feedwater flow rate, the suction and discharge feedwater pressures and information on the main feedwater pump and auxiliary turbine efficiencies. A dependence of the steam power plant efficiency on the mean feedwater temperature is presented in Fig. 6. The measurements were performed with different feedwater flow distributions between originally built economizer ECO 1 and additional economizer ECO 1A, which led to different mean temperatures of the feedwater. Four groups of data are presented, for mean feedwater temperatures equal approximately to 212 °C, 214 °C, 238 °C and 240 °C. Standard deviations of these efficiencies within each four sets of results are not higher than 0.0019. The presented results clearly show that the steam power plant efficiency increases with the increase of the mean feedwater temperature. The highest feedwater temperature close to 260 °C takes place when all feedwater inflows the economizer ECO 1 and the additional economizer ECO 1A is out of operation. In this case the steam extraction from the turbine is the greatest, as well as the heat regeneration within the steam Rankine cycle of the thermal power plant. The mean feedwater temperature decreases with the increase of the feedwater flow towards the additional economizer ECO 1A, which is the result of

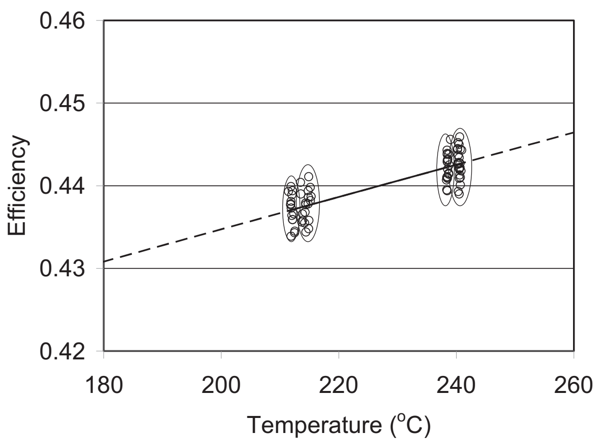

Fig. 6. Steam turbine plant efficiency in dependence of the mean feedwater temperature (mean temperatures calculated from the mean enthalpies (Eq. (9)) of the feedwater flows into the originally built economizer ECO 1 and the additional economizer ECO 1A).

the reduced heating of the feedwater in the high pressure feedwater heaters and corresponding reduced steam extraction from the steam turbine and reduced heat regeneration within the thermodynamic cycle. In case when all the feedwater bypasses the high pressure feedwater heaters, the feedwater temperature is approximately 180 °C. For the change of the feedwater flow through the high pressure heaters from zero to the total flow rate, the steam power plant efficiency increases by 0.0156 percentage points.

The change of power production of the thermal power plant unit is determined as.

$$\Delta P_G + \Delta P_{AT} = \eta_{STP}\dot{Q}_{STP} - \eta_{STP,0}\dot{Q}_{STP,0} \tag{14}$$

In Eq. (14) the change of the auxiliary turbine power is negative, since the distribution of the feedwater flow towards two parallel feedwater pipelines and economizers' sections ECO 1 and ECO 1A leads to the decrease of the feedwater flow pressure drop from the discharge of the main feedwater pump to the inlet of the second section of the economizer ECO 2. This pressure drop decrease leads to the decrease of the main feedwater pump power and consequently to the decrease of the auxiliary turbine power. The additional economizer and its feedwater pipeline are designed to provide a lower pressure drop than the pressure drop along the main feedwater pipeline. In such a way the distribution of the feedwater between the originally built economizer ECO 1 and additional economizer is performed by the throttling of the control valve on the additional feedwater line, while the control valve on the main feedwater line is opened. Hence, the change of the pressure drop from the discharge of the main feedwater pump towards the inlet of the second section economizer ECO 2 is determined as

$$\Delta\left(\Delta p_{fw}\right) = \sum_{i=1,2} K_i\left(\frac{\dot{m}_{fw,i}^2}{2\rho_{fw,i}}\right) - \sum_{i=1,2} K_i\left(\frac{\dot{m}_{fw,i}^2}{2\rho_{fw,i}}\right)_0 \tag{15}$$

where the second term on the r.h.s determines the pressure drop due to friction losses and local resistances in the case without the additional economizer usage and the first term determines the pressure drop in the case with feedwater delivery to the additional economizer. The index i is 1 for the feedwater flow from the main feedwater pump discharge line towards the junction of the main feedwater line with the pipeline for the feedwater injection into the superheated steam line, while $i = 2$ indicates the feedwater flow from this junction towards the exit of economizer ECO 1. The change of the auxiliary steam turbine power is determined as

$$\Delta P_{AT} = \frac{\Delta\left(\Delta p_{fw,P}\right)\dot{V}_{fw,P}}{\eta_P} < 0 \tag{16}$$

The electric power gain due to the increase of the efficiency of the thermal power plant unit, which is achieved by the additional economizer installation and which means the increase of the power without the increase of the fuel consumption, is determined as

$$\Delta P_G^+ = \Delta\eta_{TPP}\dot{Q}_f - \Delta P_{AT} \tag{17}$$

The analysis of energy effects achieved by the installation of the additional economizer is performed for operational conditions with the constant fresh steam mass flow rate, the constant pressure and temperature of the fresh steam in front of the high pressure section of the steam turbine and the constant temperature of the cooling water of the main steam condenser. Process parameters required for the calculation of the change of the steam boiler efficiency $\Delta\eta_{SB}$ with Eq. (4), the change of the steam turbine plant efficiency $\Delta\eta_{STP}$ with Eq. (13), the change of the electric power ΔP_G from Eq. (14), and the electric power ΔP_G^+ achieved by the Unit efficiency increase $\Delta\eta_{TPP}$ (Eq. (17)) and by the auxiliary turbine power decrease are predicted by thermal and hydraulic balances of the steam boiler and steam power plant components. For instance, the enthalpy of the feedwater at the outlet of the high pressure feedwater heaters $h_{fw,HPFWH,out}$ is calculated from energy balance equations written for the high pressure feedwater heaters of the shell-and-tube type. The feedwater from the main feedwater pump is first heated in the high pressure feedwater heater number 1 (HPFWH1, marked with 17 in Fig. 3), where the heat is transferred from the mixture formed by the superheated steam from the high pressure feedwater heater number 3 (HPFWH3, marked with 19 in Fig. 3) and the steam-condensate two-phase stream from the high pressure feedwater heater number 2 (HPFWH2, marked with 18 in Fig. 3). The heat rate taken from the two-phase mixture is

$$\dot{Q}_{HPFWH1} = \dot{m}_{TP,HPFWH2}h'\left(p_{TP,HPFWH2}\right) + \dot{m}_{s,HPFWH3}h_{s,HPFWH3,out} - \left(\dot{m}_{TP,HPFWH2} + \dot{m}_{s,HPFWH3}\right)h'\left(p_{TP,HPFWH1}\right) \tag{18}$$

and the same heat rate is delivered to the feedwater stream

$$\dot{Q}_{HPFWH1} = \dot{m}_{fw,HPFWH}\left(h_{fw,HPFWH1,out} - h_{fw,HPFWH1,in}\right) \tag{19}$$

The heat transfer rate in the high pressure feedwater heater 1 is also predicted as

$$\dot{Q}_{HPFWH1} = (kA)_{HPFWH1}\frac{T_{fw,HPFWH1,out} - T_{fw,HPFWH1,in}}{\ln\dfrac{T_{TP,HPFWH1} - T_{fw,HPFWH1,out}}{T_{TP,HPFWH1} - T_{fw,HPFWH1,in}}} \tag{20}$$

where $T_{TP,HPFWH1}$ is the temperature of the shell side two-phase mixture of steam and condensate in the high temperature feedwater heater number 1, which is predicted as the saturation temperature of the condensing two-phase mixture $T_{TP,HPFWH1} = T_{sat}(p_{TP,HPFWH1})$.

The heat transfer rate in the high pressure feedwater heater number 2 (HPFWH2) is calculated as

$$\dot{Q}_{HPFWH2} = (kA)_{HPFWH2}\frac{T_{fw,HPFWH2,out} - T_{fw,HPFWH2,in}}{\ln\dfrac{T_{TP,HPFWH2} - T_{fw,HPFWH2,out}}{T_{TP,HPFWH2} - T_{fw,HPFWH2,in}}} \tag{21}$$

This heat rate is taken from the steam extracted from the high pressure section of the steam turbine and which condenses in the heat exchanger HPFWH2

$$\dot{Q}_{\mathrm{HPFWH2}} = \dot{m}_{\mathrm{TP,HPFWH2}}\left(h_{\mathrm{s,HPFWH2,in}} - h'\left(p_{\mathrm{TP,HPFWH2}}\right)\right) \tag{22}$$

The same heat rate is delivered to the feedwater

$$\dot{Q}_{\mathrm{HPFWH2}} = \dot{m}_{\mathrm{fw,HPFWH}}\left(h_{\mathrm{fw,HPFWH2,out}} - h_{\mathrm{fw,HPFWH2,in}}\right) \tag{23}$$

The heat transfer rate in the high pressure feedwater heater number 3 (HPFWH3) is calculated as

$$\dot{Q}_{\mathrm{HPFWH3}} = (\mathrm{kA})_{\mathrm{HPFWH3}}\frac{\left(T_{\mathrm{s,HPFWH3,in}} - T_{\mathrm{fw,HPFWH3,out}}\right) - \left(T_{\mathrm{s,HPFWH3,out}} - T_{\mathrm{fw,HPFWH3,in}}\right)}{\ln\dfrac{T_{\mathrm{s,HPFWH3,in}} - T_{\mathrm{fw,HPFWH3,out}}}{T_{\mathrm{s,HPFWH3,out}} - T_{\mathrm{fw,HPFWH3,in}}}} \tag{24}$$

This heat rate is taken from the steam extracted from the intermediate pressure section of the steam turbine

$$\dot{Q}_{\mathrm{HPFWH3}} = \dot{m}_{\mathrm{s,HPFWH3}}\left(h_{\mathrm{s,HPFWH3,in}} - h_{\mathrm{s,HPFWH3,out}}\right) \tag{25}$$

The same heat rate is delivered to the feedwater

$$\dot{Q}_{\mathrm{HPFWH3}} = \dot{m}_{\mathrm{fw,HPFWH}}\left(h_{\mathrm{fw,HPFWH3,out}} - h_{\mathrm{fw,HPFWH3,in}}\right) \tag{26}$$

The equalities between the upstream heat exchanger outlet feedwater enthalpy and the downstream heat exchanger inlet enthalpy are added to the above system of equations $h_{\mathrm{fw,HPFWH2,in}} = h_{\mathrm{fw,HPFWH1,out}}$ and $h_{\mathrm{fw,HPFWH3,in}} = h_{\mathrm{fw,HPFWH2,out}}$. Feedwater temperatures at the inlets and outlets of the high pressure heat exchangers, which figure in the balances of heat transfer rates (Eqs. (20), (21) and (24)), are calculated with the equations of state for water $T = T(h,p)$, where h and p are heat exchanger inlet and outlet enthalpy and pressure. The feedwater, steam and steam-water pressures at the inlet and outlet of the high pressure heat exchangers are calculated from the momentum balances written for connecting pipelines between feedwater heaters, steam turbine extraction pipelines and heat exchangers' flow channels

$$\Delta p_i = -\varsigma_i\frac{\dot{m}_i^2}{2\rho_i A_i^2} - \rho_i g \Delta H_i \tag{27}$$

where ς_i is the equivalent coefficient of the resistance to flow, which takes into account the local resistances and the friction pressure drop along the flow path, ρ_i is the fluid density, A_i is the flow channel cross section area, the second term on the r.h.s. represents the hydrostatic pressure change with the difference between the flow channel outlet and inlet level, and index i denotes the flow channel section.

The steam pressures at the inlet of the high pressure heat exchanger number 2 ($p_{\mathrm{s,HPFWH2,in}}$) and number 3 ($p_{\mathrm{s,HPFWH3,in}}$) are predicted starting form the pressure of steam extracted from the high and intermediate pressure sections of the steam turbine by applying Eq. (27). The steam enthalpies in front of the feedwater heaters HPFWH2 and HPFWH3 is equal to the enthalpies of corresponding extracted steam, i.e. the isenthalpic flow is assumed along the extraction steam pipelines.

The pressure of steam extracted from the high and intermediate pressure section of the steam turbine is calculated with the empirical relation [15]

$$\frac{\dot{m}}{\dot{m}_0} = \sqrt{\frac{p_{\mathrm{in}}^2 - p_{\mathrm{in},0}^2}{p_{\mathrm{out}}^2 - p_{\mathrm{out},0}^2}\frac{T_{\mathrm{in}}}{T_{\mathrm{in},0}}} \tag{28}$$

where index 0 denotes a referent operating condition with known input (index in) and output (index out) steam pressure p, temperature T and mass flow rate $\dot{m}$. For the known steam inlet pressure p_{in}, temperature T_{in} and mass flow rate $\dot{m}$ at a new operating state, the outlet pressure p_{out} is predicted from Eq. (28). The extracted steam enthalpy is predicted with the equation for the steam turbine section efficiency

$$\eta_i = \frac{h_{\mathrm{in}} - h_{\mathrm{out}}}{h_{\mathrm{in}} - h_{\mathrm{out,s}}} \tag{29}$$

where h_{in} is the steam enthalpy at the inlet of the turbine section, h_{out} is the enthalpy of outlet (extracted) steam and $h_{\mathrm{out,s}}$ is the outlet enthalpy for the isentropic expansion to the steam outlet pressure.

The flue gas temperature at the steam boiler exit after the Ljungstrom air preheater, which figures in Eq. (12), is predicted with the thermal balances of heat transfers between water, steam and flue gas streams in heat exchanger components in the steam boiler. The thermal and hydraulic calculation of the steam boiler is performed according to normative method [16] (this method is presented in Ref. [17] in English). Necessary relations among water and steam thermo–physical parameters (such as water and steam enthalpy and density dependence on pressure and temperature) are calculated with corresponding polynomials derived from the steam tables [18]. The set of stated or described thermal and hydraulic balance equations is solved iteratively for prescribed fresh steam parameters and the flow rate in front of the high pressure section of the steam turbine. After the steam boiler and steam turbine plant thermal and hydraulic calculation, the energy effects of the additional economizer installation are calculated according to the following algorithm:

1. The heat power delivered to the steam turbine plant $\dot{Q}_{\mathrm{STP}}$ is calculated with Eq. (8) for different feedwater flow rates through ECO 1A.
2. The change $\Delta\dot{Q}_{\mathrm{STP}}$ is calculated for the operation with and without the additional economizer by using the results of the previous step1.
3. The change of the exit flue gas heat flow rate $\Delta\dot{Q}_{\mathrm{fg}}$ is calculated with Eq. (12).
4. The change of the fuel heat power $\Delta\dot{Q}_{\mathrm{f}}$ is calculated with Eq. (7).
5. The change of the steam boiler efficiency $\Delta\eta_{\mathrm{SB}}$ is calculated with Eq. (4).
6. The change of the steam turbine plant efficiency $\Delta\eta_{STP}$ is predicted by the relation based on the least squares of data presented in Fig. 6.
7. The change of the gross efficiency of the thermal power plant unit $\Delta\eta_{\mathrm{TPP}}$ is calculated with Eq. (3) for prescribed steam boiler and steam turbine plant efficiencies for the operation without ECO 1A.
8. The decrease of the auxiliary turbine power ΔP_{AT} is predicted with Eq. (16).
9. The increase of the electric power ΔP_{G} is predicted from Eq. (14).
10. The increase of the electric power based on the unit gross efficiency increase $\Delta P_{\mathrm{G}}^{+}$ is predicted with Eq. (17).

4. Results and discussion

The analysis of energy effects of the additional economizer installation is performed for operating conditions that are defined with plant measured parameters presented in Table 1. The presented data are the arithmetic mean of the values sampled in intervals of 2 s during 5 min period. Table 1 also presents the maximum relative deviation of all measured parameters from the arithmetic mean value, calculated as $\hat{\varepsilon}_r = \max(|x_i - \overline{x}|/\overline{x})$. The higher relative deviations $\hat{\varepsilon}_r$ are observed only for the feedwater injections into the superheating and reheating steam lines, but it is the common characteristic of operation of the systems for steam temperature control that cannot be avoided in the real plant operation. The data were measured during the operation with 28.5% of the feedwater mass flow rate through the additional economizer, while the analyzed conditions with other distributions of the feedwater flow through originally built economizer ECO 1 and additional economizer ECO 1A have the same fresh steam parameters and flow rate in front of the high pressure section of the steam turbine, the same steam turbine condenser pressure and the lignite characteristics. The measured data were also utilized for the validation of thermal and hydraulic modeling characteristics of plant components.

Fig. 7 shows the mass flow rate of feedwater through both lines of the high pressure heaters, as well as the calculated mass flow rate of the extracted steam towards the high pressure feedwater heaters in dependence on the ratio of the feedwater flow through the additional economizer ECO 1A to the total feedwater flow from the economizers towards the evaporator section of the steam boiler. The total feedwater flow to the evaporator equals the sum of the feedwater flows through the originally built economizer section ECO 1 and the additional economizer ECO 1A, which are in parallel connection, as presented in Fig. 3. The increase of the feedwater flow through the additional economizer leads to the decrease of the feedwater flow through the high pressure heaters and consequently to the reduction of the extracted steam condensation in the high pressure heaters. In Fig. 7 presented is the change of the feedwater mass flow rate through the additional economizer from 0% to 40% of the total feedwater flow towards the evaporator. For this span of the feedwater mass flow rate through ECO 1A the mass rate of extracted steam condensation in the high pressure feedwater heaters decreases from 80.3 kg/s to 53.2 kg/s (the reduction of approximately 34%). This decrease of the steam condensation rate is caused by the decrease of the feedwater mass flow rate through the high pressure heaters for approximately 37% (the reduction from 529 kg/s to 334 kg/s). The change of the mass flow rate of steam extracted from the turbine stages is practically linear in Fig. 7. It should be mentioned that the steam extraction from the turbine stages towards the feedwater heaters is not controlled by valves or other flow control devices; it is governed only by the steam condensation and the heat transfer from the condensing steam towards the colder feedwater. The high pressure heaters are shell-and-tube heat exchangers, where steam condenses on the shell side, while feedwater flows inside tubes.

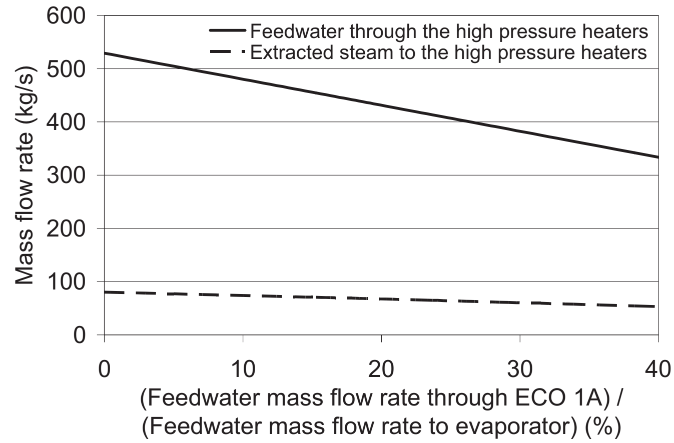

Fig. 7. Feedwater mass flow rate through the high pressure heaters and the corresponding mass rate of extracted steam condensation.

Calculated and measured feedwater temperatures at several locations are presented in Fig. 8. The increase of the feedwater delivery to the additional economizer ECO 1A and corresponding reduction of the feedwater flow through the high pressure heaters lead to the slight increase of the feedwater temperature at the outlet of the high pressure heaters from 257 °C to 261 °C. The mean feedwater temperature at the inlet of the steam boiler, calculated from the enthalpy obtained by the energy balance Eq. (9), considerably drops with the reduction of the feedwater flow through the high pressure heaters from 257 °C to 231 °C. Both feedwater temperature at the outlet of the high pressure heaters and the mean feedwater temperature at the steam boiler inlet change linearly with the feedwater flow through the additional economizer ECO 1A. At low feedwater flow rates through the additional economizer, the feedwater temperature at the additional economizer exit is close to the flue gas temperature of 305 °C. The decrease of the water temperature at the exit of ECO 1A is slow with the feedwater mass flow rate increase approximately up to 15% of the total feedwater flow towards the evaporator, while further flow rate increase leads to a steeper feedwater temperature drop at the ECO 1A exit. The feedwater temperature at the exit of the second economizer section ECO 2 first slightly increases with the start of the feedwater flow through the additional economizer, reaches maximum for approximately 15% of the feedwater flow through ECO 1A and after that slightly decreases. Measured values of the feedwater temperatures at the exit of the high pressure heaters, at the exit of the additional economizer and at the exit of the second economizer section ECO 2 are also depicted in Fig. 8 for the

Table 1
Measured parameters of the analyzed operating conditions (the reheated steam data and the main feedwater pump discharge parameters correspond to 28.5% of the feedwater flow through the additional economizer, other data are valid for all analyzed conditions).

- The fresh steam flow rate is 529 kg/s ($\hat{\varepsilon}_r = 0.5\%$).
- The fresh steam pressure and temperature in front of the high pressure section of the steam turbine are 17.3 MPa ($\hat{\varepsilon}_r = 0.8\%$) and 535 °C ($\hat{\varepsilon}_r = 0.3\%$).
- The steam pressure and temperature at the exit of the high pressure section of the steam turbine are 43.0 bar ($\hat{\varepsilon}_r = 0.3\%$) and 335.7 °C ($\hat{\varepsilon}_r = 0.2\%$).
- The reheated steam pressure and temperature at the inlet of the intermediate pressure section of the steam turbine 39.7 bar ($\hat{\varepsilon}_r = 0.3\%$) and 535.7 °C ($\hat{\varepsilon}_r = 0.2\%$).
- The feedwater pressure and temperature at the discharge of the main feedwater pump are 248 bar ($\hat{\varepsilon}_r = 0.8\%$) and 180 °C ($\hat{\varepsilon}_r = 0.03\%$).
- The mass flow rate of the feedwater injected into the superheating fresh steam line for its temperature control is 40.4 kg/s ($\hat{\varepsilon}_r = 9.5\%$).
- The mass flow rate of the feedwater injected into the reheating steam line for its temperature control is 5.4 kg/s ($\hat{\varepsilon}_r = 6.5\%$).
- The pressure in the steam turbine condenser is 0.052 bar ($\hat{\varepsilon}_r = 0.6\%$).
- The lignite lower heat capacity 7554 kJ/kg, the lignite moisture mass fraction 49.9% and the ash fraction 15.6% [19].
- Prior to the upgrade the steam boiler efficiency is 0.865 and the steam turbine plant efficiency is 0.446.

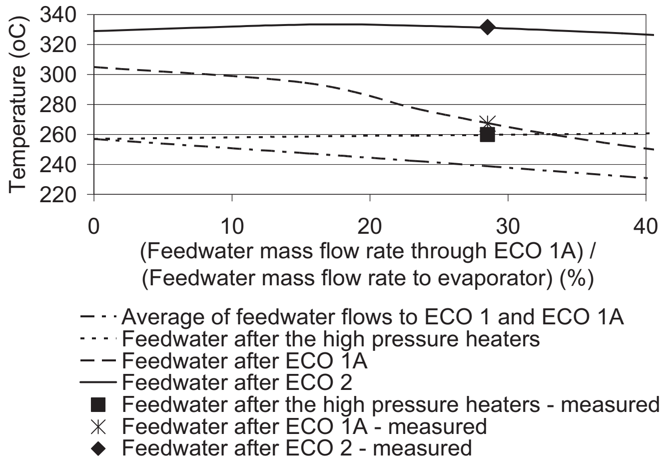

Fig. 8. Calculated and measured feedwater temperatures at various locations.

operation with 28.5% of the feedwater flow through the additional economizer. Good agreement between calculated and measured data is shown. The maximum feedwater temperature within the economizers' sections is at the exit of the second section ECO 2 with a value of nearly 335 °C. The boiling pressure for this temperature is 137 bar, while the feedwater pressure in the economizers' sections is approximately 230 bar or higher. Hence, due to the fact that the boiling pressure is much lower than the actual water pressure in the economizers' sections, boiling does not occur.

Fig. 9 shows the flue gas temperature at the exit of the steam boiler, after the Ljugstrom air preheater, and in a flue gas duct, after the additional economizer, in dependence on the ratio of the feedwater flow through the additional economizer ECO 1A to the total feedwater flow towards the evaporator. The exit flue gas temperature after the Ljungstrom air preheater drops from 188 °C to below 160 °C with the increase of the feedwater mass flow rate through the additional economizer from zero to 40%. Hence, the installation of the additional economizer ECO 1A reduces the exit flue gas temperature by nearly 28 °C. The rate of temperature drop of the exit flue gas is faster with the increase of the feedwater flow through the additional economizer up to approximately 15%, while it is slower for the further increase of the feedwater flow through ECO 1A. As it is shown in Fig. 9, the exit flue gas temperature after the Ljungstrom air preheater is 188 °C in operation without ECO 1A, which is a high value that indicates a substantial heat power loss. Therefore, it should be mentioned that prior to the additional economizer installation, Unit B1 at Thermal Power Plant "Nikola Tesla" had been working only with one line of the high pressure feedwater heaters, while the other line had been bypassed in order to reduce the feedwater temperature at the inlet of the economizer ECO 1 and consequently to reduce the exit flue gas temperature, which leads to an increase of the steam boiler efficiency, as well as the efficiency of the whole plant unit. The other Unit B2, which has not been retrofitted yet, and operates without the additional economizer, still uses only one line of the high pressure feedwater heaters.

The thermal calculation of the heat transfer in the economizer sections and comparisons with measured data show that the achieved overall heat transfer coefficient from the flue gas to the feedwater in ECO 1A is approximately 47 W/(m^{2}K), while its value in economizer sections ECO 1 and ECO 2 is 65 W/(m^{2}K). The lower value in the additional economizer ECO 1A takes place due to its position at the top of the boiler, at the location where the vertical square 20 m × 20 m channel sharply bends to the horizontal section (Fig. 4). This turn of the flue gas flow direction causes the non-uniformity of the gas velocity profile in the cross section of the economizer tube bundle with decrease of gas velocities in the corner sections, which leads to the reduction of the heat transfer coefficient. The installation of the additional economizer in the downward flue gas channel above the Lungstrom air preheater was not acceptable because of the circular cross section of this channel and because its metal structure cannot bear the heavy load of the additional economizer (the building of additional supports for the additional economizer in the downward exit flue gas channel was considered as the more expensive solution as opposed to the present solution with the additional economizer location at the top of the steam boiler).

The power of the waste heat recovered from the exit flue gas and transferred to the steam turbine plant by heating of the feedwater in the additional economizer is shown in Fig. 10 (the upper curve in the diagram). The recovered heat power increases up to 33 MWth with the feedwater mass flow rate increase through the additional economizer up to 40% of the total feedwater mass flow rate to the evaporator. The linear increase of the electric power at the generator is shown in Fig. 10 in the range of the feedwater flow increase through ECO 1A from 0% to 40%. For the maximum feedwater flow of 40% through ECO 1A the electric power increase reaches 24.5 MWe. Since the fresh steam parameters and the flow rate at

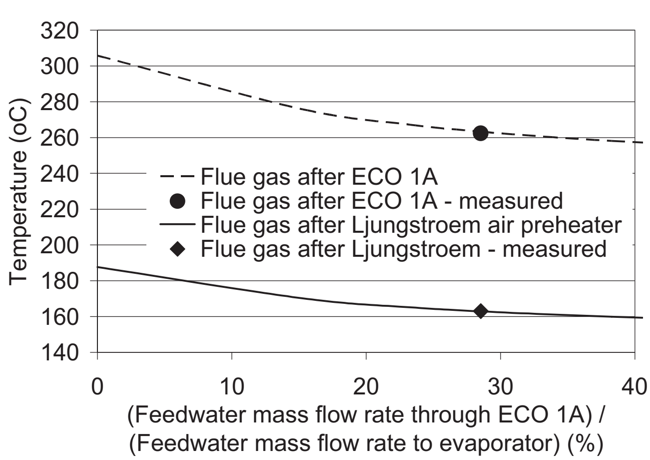

Fig. 9. Calculated and measured flue gas temperatures after additional economizer and the Ljungstrom air preheater.

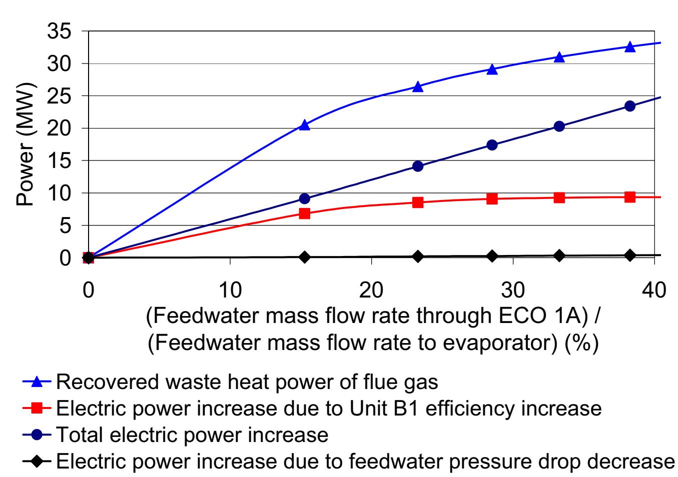

Fig. 10. Recovered waste heat power and corresponding electric power upgrade by the installation of the additional economizer.

the inlet of the high pressure section of the steam turbine are constant, this electric power increase is due to the increased steam flow through the intermediate and low pressure turbine sections due to the reduced steam extraction and condensation in the high pressure feedwater heaters. A part of this electric power increase is produced on account of the Unit B1 gross efficiency upgrade due to the flue gas waste heat utilization, as well as due to the reduction of the feedwater pressure drop from the main feedwater pump discharge to the exit of the economizer section. As shown in Fig. 10 the electric power produced due to the Unit B1 efficiency upgrade rapidly increases with the increase of the feedwater flow through ECO 1A from zero to 15%. In the range of feedwater flow through ECO 1A from 15% to 38% the electric power increases from 6.8 MWe to 9.4 MWe due to efficiency upgrade, while with further increase of the ECO 1A feedwater flow this electric power slightly decreases. The electric power increase due to the reduction of the auxiliary turbine power is presented with the bottom line in Fig. 10 (this electric power increase is achieved by the reduction of the pressure drop in the feedwater flow, and consequently to the reduction of the main feedwater pump power, due to the parallel connection of the additional economizer ECO 1A to the originally built economizer ECO 1). This power increase is from 0.2 MWe to 0.4 MWe for the feedwater flow through ECO 1A increase from 15% to 40%. Fig. 10 shows that the maximum overall electric power increase is achieved with the maximum feedwater flow through the additional economizer ECO 1A. This is explained by the steam turbine power increase with the reduction of the steam extraction to the high pressure feedwater heaters, due to the reduction of the feedwater flow through these heaters with the increase of feedwater flow through the additional economizer ECO 1A. In addition, the maximum of the electric power production by the heat recovery from the flue gas is achieved for the feedwater mass flow rate through ECO 1A in the range from 33% to 38% of the feedwater flow to the evapororator. The maximum of the electric power production based on the recovery of the flue gas heat is determined by the change of the power plant efficiency, which is dependant on the change of the steam boiler and steam turbine plant efficiencies. The explanation follows by description of Fig. 11.

Fig. 11 shows the upgrade of the total gross efficiency of Unit B1, the upgrade of the steam boiler efficiency and the reduction of the steam turbine plant efficiency. The steam power plant efficiency decreases by 0.52 percentage points with the increase of the feedwater flow through the additional economizer from zero to 40% of the total feedwater flow to evaporator. On the other side, the steam boiler efficiency substantially increases by 2.19 percentage points in

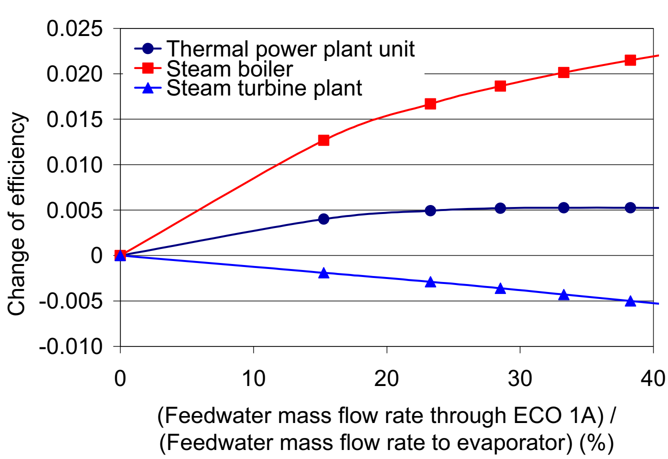

Fig. 11. Efficiency upgrade by the installation of the additional economizer.

the same range of the feedwater flow increase through the additional economizer. As a consequence, the Unit B1 gross efficiency rapidly increases by 0.4 percentage points with the increase of the feedwater mass flow rate through ECO 1A from zero to 15% of the feedwater mass flow rate to the evaporator. The maximum value of the Unit efficiency is 0.53 points and it is achieved by the ECO 1A feedwater mass flow rate in the range from 33% to 38% of the feedwater mass flow rate to the evaporator. In this range of the ECO 1A feedwater mass flow rate the Unit efficiency is practically constant.

When the feedwater mass flow rate through the additional economizer ECO 1A is higher than 33% of the feedwater mass flow rate to the evaporator, it is seen in Figs. 10 and 11 that the Unit B1 gross efficiency is practically constant, while the total Unit electric power still considerably increases with the same trend. Hence, regarding the Unit gross efficiency the acceptable range of the feedwater mass flow rate through the additional economizer is from 33% to 38% of the total feedwater mass flow rate to the evaporator. Regarding the electric power the choice of 33% of the feedwater flow through ECO 1A provides the electric power increase of 20.3 MWe. A variation of the feedwater flow through ECO 1A to 40% for a certain time period leads to the electric power increase by 4.2 MWe (the total power increase is 24.5 MWe), while a reduction of the feedwater flow through ECO 1A to 15% leads to the electric power decrease by 11.1 MWe (from 20.3 MWe to 9.2 MWe, Fig. 10). Hence, the variation of the feedwater flow through the additional economizer is a method for the Unit electric power control, whereas the pressure, temperature and mass flow rate of the fresh steam in front of the high pressure section of the steam turbine are kept constant, which is a considerable advantage for the process control.

For the evaluation of the economic benefits of the efficiency increase by the installation of the additional economizer at Unit B1 of the Thermal Power Plant "Nikola Tesla", it is assumed that Unit B1 operates 7500 h per year on the upgraded power. With the feedwater mass flow rate through the additional economizer of 38% of the total feedwater mass flow rate to the evaporator, the electric power production on account of the waste heat recovered from the boiler exit flue gas and reduced consumption of the main feedwater pump is 9.4 MWe (Fig. 10). Hence, the net annual electricity production is $E_e = 70.5$ GWh. The value of this electricity production is in the range from 3.525 million euros to 7.05 million euros for the electricity price in the range from 50 euros/MWh to 100 euros/MWh respectively. The investment in the production and installation of the additional economizer with separate feedwater line and valves is $I = 6.5$ million euros. The ratio of this investment to the clean electric power of 9.4 MWe results in the specific capital investment of 691 euros/kWe, which is approximately one half of the specific capital investment in the subcritical thermal power plant with the pulverized coal combustion (which is approximately 1400 euros/kWe according to Fig. 4 in Ref. [20]). By taking into account the total power increase of 24.5 MWe, as presented in Fig. 10, the specific capital investment is 265 euros/kWe, which is approximately only 19% of the specific capital investment in the power plant with the pulverized coal combustion [20].

The economic evaluation is performed with the calculation of the internal rate of return and the simple pay back period. The internal rate of return i_{IRR} is calculated by equating the present value of electricity production during the lifetime to the present value of costs [21]

$$E_e c_e \sum_{j=1}^{n} \left(\frac{1+e}{1+i_{IRR}}\right)^j = I + mI \sum_{j=1}^{n} \left(\frac{1+g}{1+i_{IRR}}\right)^j \quad (30)$$

where the plant operation for $n = 20$ years is adopted. The electricity price c_e is varied in the range from 50 euros/MWh to

100 euros/MWh, while the rate of annual increase of electricity price is $e = 0.03$. The annual operation and maintenance costs of the installed equipment are calculated in Eq. (30) as the fraction of the investment mI, where the estimated value of m is 0.001. The annual increase of operation and maintenance costs is $g = 0.03$.

The SPBP (simple pay back period) is calculated as the ratio of the total investment costs to the value of annual electricity production

$$\mathrm{SPBP} = \frac{I}{E_e c_e} \tag{31}$$

The results of the economic analysis are presented in Fig. 12. It is shown that the internal rate of return is in the range from 0.58 (58%) to 1.15 (115%) for the specified electricity price change from 50 euros/MWh to 100 euros/MWh, while the simple pay back period varies from 1.84 years ($\approx$22 months) to 0.92 years ($\approx$11 months). According to these results the project is economically extremely attractive.

The annual reduction in the lignite consumption, achieved by the utilization of the flue gas waste heat and electricity production of 70.5 GWh is approximately 87×10^6 kg. This avoided lignite consumption leads to the reduction of CO_2 emission in the amount of 70.7×10^6 kg. The estimation is done with lignite lower heat value of 7500 kJ/kg, the CO_2 emission factor 108.8 $kgCO_2/GJ$ [22] and the Unit B1 gross efficiency 0.3911, which corresponds to the feedwater mass flow rate through the additional economizer in the range from 335 to 38% of the feedwater flow to the evaporator.

5. Conclusions

The upgrade of efficiency and electric power at the 30 year old Unit B1 of the Thermal Power Plant “Nikola Tesla B” was conducted with the installation of the additional high pressure economizer in parallel with the existing first section of the originally built economizer. The additional economizer is fed with the feedwater by the new separate feedwater line connected to the discharge line of the main feedwater pump. Although the heat recovery from the flue gas to increase the steam boiler efficiency is applied in many power stations, the presented solution is innovative due to the special connection of the additional economizer with the feedwater line and main economizer sections. The presented solution leads to multiple benefits. The additional economizer installation leads to the reduction of the steam extraction from the steam turbine for the feedwater heating in the high pressure heaters, which enables the total electric power increase of the Unit by up to 24.5 MWe for the analyzed operational conditions supported by the measured data. The reduction of the steam extraction also leads to the reduction of the heat regeneration within the Rankine cycle and the decrease of the steam turbine plant efficiency, but this effect has a smaller influence on the Unit gross efficiency than the increase of the steam boiler efficiency due to the recovery of the waste heat in the additional economizer from the exit flue gas. In total, the important increase of the Unit gross efficiency is achieved. These changes of efficiencies depend on the feedwater flow through the additional economizer. The maximum increase of the Unit gross efficiency by 0.53 percentage points is achieved for the feedwater mass flow rate through the additional economizer in the range from 33% to 38% of the total feedwater mass flow rate to the evaporator section of the steam boiler. In the same range of the feedwater flow through the additional economizer, up to 9.4 MWe of the Unit electric power is produced on account of the exit flue gas utilization, where 0.4 MWe is obtained due to the reduction of the pressure drop along the parallel feedwater pipelines and the economizers sections and consequent reduction of the main feedwater pump power. The exit flue gas temperature after the Ljungstrom air preheater is reduced by 28 °C, but this temperature is still higher than the temperature of the sulfuric acid dew point and there is no risk for the low temperature corrosion in the electrostatic precipitator. The additional feature of the performed solution is the possibility to control the steam turbine power by the change of the feedwater flow through the additional economizer, while the parameters and the mass flow rate of the fresh steam are being kept constant in front of the high pressure section of the steam turbine. The economic analysis shows that the project is extremely attractive. For the electricity price in the range from 50 euros/MWe to 100 euros/MWe the internal rate of return varies from 58% to 115%, while the simple pay back period is in the range from 22 months to 11 months respectively.

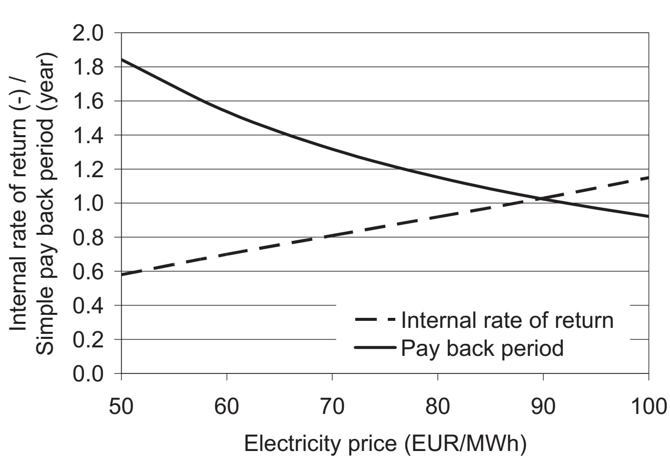

Fig. 12. Internal rate of return and simple pay back period dependence on the electricity price.

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Nomenclature

A	area, m^2
c	price, euros/MWh
E	energy, kWh
e	rate of annual increase of electricity price,
h	specific enthalpy, J/kg
f	steam boiler losses (except losses with the flue gas) as a fraction of the fuel heat power,
g	heat power input to the steam boiler by air inflow as a fraction of the fuel heat power, gravity, m/s^2 rate of annual increase of operation and maintenance costs,
H	height, m
i	discount rate,
K	equivalent pressure loss coefficient in Eq. (15)
k	heat transfer coefficient, $W/(m^2K)$
$\dot{m}$	mass flow rate, kg/s
P_G	electric power, W
ΔP_G^+	electric power production due to the Unit efficiency upgrade
p	pressure, Pa
$\dot{Q}$	heat power, W
SPBP	simple pay back period

T temperature, K, °C
$\dot{V}$ volume flow rate, m^3/s
x process parameter

Greek letters
ε_r relative deviation of the measured parameter from the arithmetic mean value
η efficiency
ρ density, kg/m^3
ς equivalent pressure loss coefficient in Eq. (27)

Subscripts
AT auxiliary turbine
a air
c cold
e electricity
ECO 1 the first section of the originally built economizer
ECO1A additional economizer
ECO 2 the second section of the originally built economizer
f fuel
fg flue gas
fs fresh steam
fw feedwater
G electric generator
HPFWH high pressure feedwater heater
h hot
in inlet parameter
inj injection
iRR internal rate of return
l loss
out outlet parameter
P pump
rs reheated steam
SB steam boiler
STP steam turbine plant
S isentropic process
TP two-phase mixture
TPP thermal power plant

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