EXERGY EFFICIENCY ANALYSIS OF LIGNITE-FIRED STEAM GENERATOR

by

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The operation of steam generators and thermal power plants is commonly evaluated on a basis of energy analysis. However, the real useful energy loss cannot be completely justified only by the First law of thermodynamics, since it does not differentiate between the quality and amount of energy. The present work aims to give a contribution towards identification of the sources and magnitude of thermodynamic inefficiencies in utility steam generators. The work deals with a parallel analysis of the energy and exergy balances of a coal-fired steam generator that belongs to a 315 MWe power generation unit. The steam generator is designed for operation on low grade coal - lignite with net calorific value 6280 to 9211 kJ/kg, in a cycle at 545 °C/177.4 bar, with feed water temperature 251 °C, combustion air preheated to 272 °C and outlet flue gas temperature 160 °C. Since the largest exergy dissipation in the thermal power plant cycle occurs in the steam generator, energy, and exergy balances of the furnace and heat exchanging surfaces are established in order to identify the main sources of inefficiency. On a basis of the analysis, optimization of the combustion and heat transfer processes can be achieved through a set of measures, including retrofitting option of lignite predrying with flue gas and air preheating with dryer exhaust gases.

Key words: *coal-fired power plant, steam generator, exergy, energy efficiency, combustion, heat transfer*

Introduction

The energy needs for electricity generation in the world rely heavily on fossil fuels, since the majority of the power generation is met by coal, natural gas, and oil. Even though renewable energy sources are being rapidly developed, their cost and current technology state have not advanced to a stage where they can significantly reduce the global dependence on fossil fuels. Therefore, it is important that fossil fuel plants, and coal-fired ones in particular, improve the efficiency and reduce their environmental impact, including the reduction of GHG emissions. The energy conversion in the thermal power plants (TPP) is a thermodynamic process, which can be improved by energy analysis that enables identification of energy efficient measures to be addressed. The improvement of the power plant efficiency leads to lower electricity cost and lower emissions, making it an attractive option. The conventional methods of energy analysis based on the First law of thermodynamics focus on conservation of energy. Taking into account the limitations with this approach, achieving higher efficiency warrants additional analysis based on the Second law, which includes analysis of the exergy

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efficiency and irreversibility. The objective of this work is to carry out an exergetic analysis of the efficiency of the solid fuel power plant and particularly of steam boiler as its key component, in order to get a complete picture about the possibilities to improve its energy and environmental performance.

The term *exergy* was coined in 1956 by Zoran Rant [1] by using the Greek words *ex* and *ergon* meaning *from work* [2], although the concept has existed since Gibbs (available energy of the body and medium, 1873) [3]. The work [4] presents a number of exergy-based concepts and methods, *e.g.* efficiency concepts, exergy flow diagrams, exergy utility diagrams, life cycle exergy analysis and exergy economy optimization, that are useful tools in order to describe, analyze and optimize energy conversion systems. The exergy of a system determines its availability to produce work. In that sense, the study [5] deals with comparison of energy and exergy analyses of power plants on fossil fuels, throwing light on the scope for further research and recommendations for improvement in the existing units. The exergy analysis aims to determine maximum work that can be obtained in a work producing process from a given system. An exergy analysis of an existing 300 MW lignite-fired thermal power plant is carried out in the study [6] and, in order to promote a more efficient solution, its performance is compared with three different cogeneration systems, employing Rankine cycle, designed to use the same amount of fuel as the existing plant. In the work [7] a parametric thermodynamic exergy analysis of a 32 MW subcritical coal-fired power plant unit is performed under various operating conditions, including different flow rates, pressures and temperatures. The work [8] deals with an exergy analysis of flue-gas pre-dried lignite-fired power system, based on steam generator with open pulverizing system, aiming to explore the energy saving potential of various options. Habib and Zubair [9] conducted a Second thermodynamic law analysis of regenerative Rankine-cycle power plant with steam reheating. In the work [10], Dincer has performed a thermodynamic analysis of reheat cycle power plants. An exergy analysis of a 210 MW thermal power plant is accomplished by Sengupta *et al.* [11].

Thermodynamic models that combine the concepts of economy and exergy analyses provide a possibility of optimizing complex energy-generating systems to achieve the best balance between the thermodynamic efficiency and economic cost. In the work [12], optimization of a 300 MW coal-fired power plant operation is accomplished based on the structure theory of thermo-economic analysis. Rosen and Dincer [13] in their work have performed an exergo-economic analysis of power plants that operate on various fuels, by investigating the relationship between the capital costs and the thermodynamic losses. Kwak *et al.* [14] presented an exergetic and thermo-economic analysis of power plants. In the study [15], input and output thermodynamic properties of a thermal power plant unit were used to perform energy and exergy analyses. The obtained results were used to evaluate and suggest improvements and exergoeconomic analysis was applied based on the plant exergy losses.

The aim of the work [16] is to present how nature-inspired swarm intelligence (especially particle swarm optimization) can be applied in the field of power plant optimization and how to find solutions for the problems arising and also to apply exergoeconomic optimization technics on TPP. A comparative analysis of four TPP performances from the energetic and exergetic viewpoint has been conducted in [17]. Thermodynamic models of the plants are developed with primary objectives to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses. Erdem *et al.* [18] have conducted a comparison among coal-fired power plants in Turkey based on energy and exergy analysis and discussed thermodynamic inefficiencies. In the study of Aljundi [19], the energy and exergy analysis of a power plant in Jordan is presented with a primary objective to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses. Datta *et al.* [20] have presented an exergy analysis of a coal-based thermal power plant conducted by splitting up the entire plant cycle into three zones. Reddy and Butcher in their work [21] analyzed a waste heat recovery power generation system, based on Second law of thermodynamics. Suresh *et al.* [22] compared the exergetic performance of a coal-based power plant using subcritical, supercritical, and ultra-supercritical steam conditions.

A new combined cycle concept is proposed in the study [23], in an effort to improve the thermal power plant cycle efficiency by reducing the energy loss in the plant components. The proposed combined cycle plant uses water as a working fluid in the topping cycle and an ammonia-water mixture in the bottoming cycle and, according to the study, it is 4% more efficient than the standalone Rankine cycle operating on a condensing mode. The work [24] presents a simplified method for determination of exergy efficiency of a steam-turbine co-generation installation, taking into account the efficiency of separate plant components. A case study of detailed exergy analysis conducted in a coal-fired power plant unit is presented in [25]. The analysis identifies the locations and magnitude of exergy destruction in the system and its components and assesses different efficiency improvement options. In the work [26] the author explains his views that, to better understand and address environmental concerns, we need to focus on the linkages between exergy and the environment, and that much more research is needed if the (potentially immense) benefits are to be fully tapped.

The review exposes the weaknesses of the traditional approach for power plant efficiency analysis that relies solely on the principle of conservation of energy and suggests that an in-depth comprehensive analysis should necessarily include appropriate exergetic analysis of the plant and its components. Current research in this area mainly takes place in two general directions – exergo-economic analysis and detailed analysis of system components. As fossil fuels dominate the world primary energy supply and will do it at least for the next few decades, further improvement of the fossil-fueled power plants is needed due to many reasons, including the environmental impact and plant economic operation. The substantial approach to thermal power plants efficiency analysis must involve an exergy analysis, with emphasize on the sources of exergy dissipation and process irreversibility. The present work aims to give contribution towards the gaining further knowledge and experience in the use of exergy method for performance evaluation of coal-fired power plant and of a steam generator as one of the plant's key components. In this study, an exergy analysis of a 315 MW lignite-fired TPP is carried out. Since lignite is the only significant fossil fuel resource in SEE Europe, it is of a great importance to define possible strategies to maximize the efficiency of the existing power plants. The objective of this work is to perform a steady-state simulation in order to identify the locations, sources and magnitude of thermo-dynamic inefficiencies in the utility steam generator, as one of the key components of a TPP.

Technical description of the analyzed TPP

The present analysis considers the two units of a thermal power plant that have been put into operation in 1983 and 1984, each with nominal power output of 315 MW (339 MW total power) [27]. The steam generators are designed to be fueled on lignite with net calorific value (*NCV*) in range 6280-9211 kJ/kg as received, with a guarantee *NCV* of 7327 kJ/kg [27]. They are aimed for operation in a cycle with feed water temperature 251 °C, steam temperature 545 °C and pressure 177.4 bar. The average ultimate analysis of the fuel is presented in tab. 1, giving *NCV* of 7580 kJ/kg. The combustion air is preheated at 272 °C and the flue gas-

Table 1. Lignite ultimate analysis (as received, in %)

Carbon (C)	Hydrogen (H)	Oxygen (O)	Sulphur (S)	Nitrogen (N)	Ash (A)	Moisture (W)
23.0	2.3	13.0	0.6	0.6	16.0	44.5

es average temperature at the rotating air-heaters outlet is 160 °C. Figure 1 illustrates schematically the basic layout of the considered thermal power plant, consisting of several subsystems: steam generator, turbine, regenerative pre-heating subsystem including water thermal treatment plant (deaerator), and part of the solid fuel treatment facility.

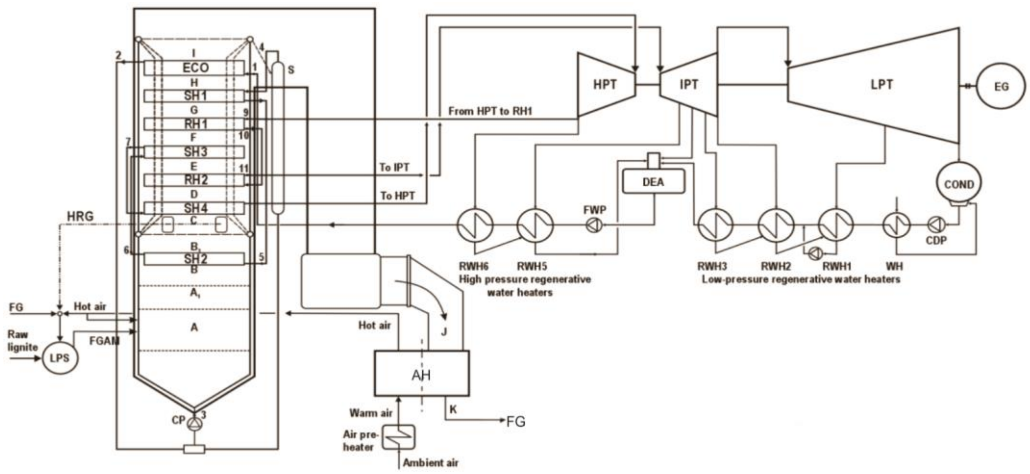

Figure 1. Flow diagram of the power plant: *FG – flue gases, LPS – lignite pulverizing system, FGAM – fuel-gas-air mixture, HRG – hot re-circulation of gases, CP – circulating pump, SH1 to SH4 – steam superheater stages, RH1, RH2 – steam reheaters, ECO – boiler economizer, S – separator, HPT – high pressure turbine, IPT – intermediate pressure turbine, LPT – low pressure turbine, EG – electric generator, COND – condenser, CDP – condensate pump, DEA – deaerator, RWH – regenerative water heaters, FWp – feed-water pump, AH – air heater*

The steam generator is a single gas path tower-type, with forced circulation, a direct tangential combustion system and with a system for slag removal in a solid-state. Some of the basic features of the boiler are a large rectangular cross-sectional area (15.75 × 15.38 mm), hot flue gas re-circulation from the section between the second and fourth stages of the primary superheater and a large number of steam soot blowers (102) [27]. The burner system consists of eight wind box assemblies with staged fuel-air introduction, arranged in a way to direct the coal and air streams at slight angles off of the diagonals and tangent to an imaginary firing circle in the furnace center. Each burner assembly is equipped with separate facility for pulverized coal preparation. The grinding and drying of coal is carried out in eight fan mills with inertial separators. The designed fineness of grinding is such that 45% of the pulverized lignite should be below 90 μm and up to 3% particles can be larger than 1 mm. Gas temperature at the mill entrance before the lignite introduction is 650 °C and temperature at its outlet is 180 °C. The properties of the lignite mineral matter are presented in tab. 2 [27].

The evaporation heat exchanging surfaces – HES (water-walls) are made of membrane panels with vertical tubes stretching from the collectors at the bottom of the furnace, via the super-heaters and re-heaters' zones, up to the upper collectors, above which only the water economizer is located, in an unassembled box. The steam superheaters, reheaters and the economizer are installed in one gas channel along the flue gas path. The superheater second stage (SH2) is built in 'wind-walls' design, covering partially the screen evaporator water-walls in the zone below the suction openings of the hot gas re-circulation. The working fluid parameters of the steam generator HES are given in tab. 3 [27]. The flue gas stream is divided into two parallel lines, with cracks in front of the rotary air heaters. The initial preheating of air is carried out in steam pre-heaters.

Table 2. Lignite ash properties

Ash composition				Ash fusion temperatures	
SiO_2	21.65- 31.76%	MgO	2.96- 9.0%	Initial deformation temperature	1020- 910 °C
Fe_2O_3	5.5- 12.21%	$CaCO_3$	10.35- 14.0%	Softening temperature	1240- 1130 °C
Al_2O_3	5.95- 9.5%	P_2O_5	0.2- 0.5%	Hemispherical temperature	1270- 1170 °C
CaO	28.56- 45.0%			Fluid temperature	1360- 1175 °C

Table 3. Technical data of steam generator HES [27]

Heat exchanging surface	Unit	Entry		Exit	
Economizer (ECO)		1		2	
Pressure (*p*)	bar	210		202	
Temperature (*t*)	°C	251		319	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	1091.5	2.761	1437	3.383
Water flow rate (*m*)	kgs^{-1} (th^{-1})	277.778 (1,000)		277.778	
Evaporator (A, B)		3		4	
Pressure (*p*)	bar	202/196		196	
Temperature (*t*)	°C	319/satur. tempearture 364 °C		364	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	1,437/*h*' = 1,807.5	3.383	*h*" = 2,438	4.9758
Water/steam flow rate (*m*)	kgs^{-1}	277.778		277.778	
Primary superheater (SH1)		4		5	
Pressure (*p*)	bar	196		190	
Temperature (*t*)	°C	364		397	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	2438	4.9758	2831.5	5.593
Steam flow rate (*m*)	kgs^{-1}	1000 t/h = 277.778 kg/s		277.778	
Primary superheater (SH2)		5		6	
Pressure (*p*)	bar	190		185	
Temperature (*t*)	°C	397		420	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	2831.5	5.593	2968	5.802
Steam flow rate (*m*)	kgs^{-1}	277.778		277.778	
Primary superheater (SH3)		6		7	
Pressure (*p*)	bar	185		180	
Temperature (*t*)	°C	420		480	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	2968	5.802	3203	6.137
Steam flow rate (*m*)	kgs^{-1}	277.778		277.778	
Primary superheater (SH4)		7		8	
Pressure (*p*)	bar	180		177.4	
Temperature (*t*)	°C	480		542	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	3203	6.137	3390.7	6.395
Steam flow rate (*m*)	kgs^{-1}	252.589		252.589	
Reheater (RH1)		9		10	
Pressure (*p*)	bar	42		39	
Temperature (*t*)	°C	326,5		480	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	3031.5	6.445	3400	7.040
Water/steam flow rate (*m*)	kgs^{-1}	248.61		248.61	
Reheater (RH2)		10		11	
Pressure (*p*)	bar	39		37.07	
Temperature (*t*)	°C	480		542	
Enthalpy (*h*), entropy (*s*)	$kJkg^{-1}$, $kJkg^{-1}K^{-1}$	3400	7.040	3538.7	7.248
Water/steam flow rate (*m*)	kgs^{-1} (th^{-1})	256.94 (925)		256.94	

Materials and methods

The limitation of the conventional energy analysis approach, based on the First law of thermodynamics, is that it does not take into account properties of the system environment, or degradation of the energy quality through dissipative processes, which means that it does not characterize the irreversibility of the system [28]. Achieving higher efficiency, therefore, warrants a higher order analysis, based on the Second law of thermodynamics, as this enables us to identify the major sources of loss, and shows avenues for performance improvement [28].

Exergy analysis

Exergy analysis characterizes the work potential of a system with reference to the environment conditions, which is the maximum theoretical work that can be obtained from a system when its state is brought to the reference atmospheric conditions. The destruction of exergy in certain process is proportional to the entropy generation in it, which accounts for the inefficiencies due to irreversibility. This means that exergy losses are linked to the entropy generated within a transformation. The exergy analysis of a thermal power plant gives a qualitative picture of exergy dissipation in the energy conversion process and the ability to limit the losses. Its main purpose is to identify the locations, magnitudes, causes and sources of inefficiencies. This information can be useful in designing a new energy efficient system, but also for improving the performance of existing systems.

The First law of thermodynamics efficiency of a system or/and system component is defined as the ratio of energy output to the energy input to the system/component.

The Second law efficiency is defined:

$$\eta_{\mathrm{ex}} = \frac{\text{Actual thermal efficiency}}{\text{Maximum possible (reversible) thermal efficiency}} = \frac{\text{Exergy output}}{\text{Exergy input}} = 1 - \frac{\text{Exergy loss}}{\text{Exergy input}} \tag{1}$$

The overall energy balancing of the steam generator in this work is based on the normative method [29] and the general approach of the exergy balancing is depicted in fig. 2 [30].

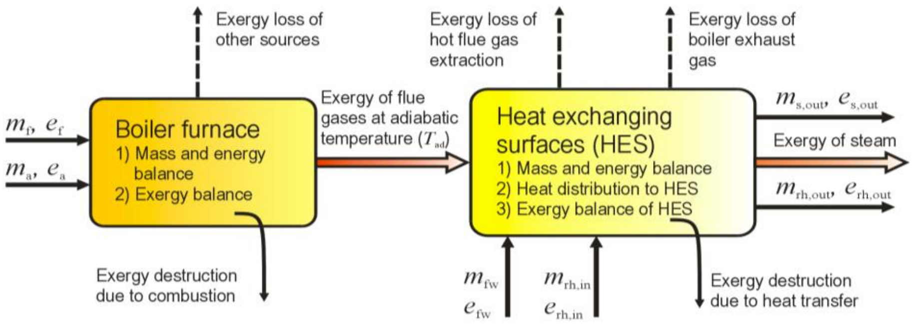

Figure 2. General model concept of the steam generator exergy balance [30]

The calculation of the coal combustion statics, *i. e.* stoichiometric quantities of oxygen and air, as well as flue gas quantities is based on the normative method [29]. The baseline case is set as combustion at 20% excess air, *i. e.* $\lambda = 1.2$, and the results are given in tab. 4.

Energy balance calculations

Adiabatic flame temperature is used as reference temperature for calculation of heat transferred in the furnace. It is calculated from the energy balance of the boiler furnace, under

Table 4. Calculations of combustion statics

Nomen.	Description	Value	Unit
O_{min}	Stoichiometric oxygen quantity	0.673	kgO_2kg^{-1} fuel
$M_{a,min}$	Stoichiometric quantity of air for complete combustion	2.888	$kgairkg^{-1}$ fuel
M_a	Air quantity at 20% excess air, λ = 1.2	3.466	$kgairkg^{-1}$ fuel
M_{CO_2}	CO_2 quantity in flue gases	0.843	$kgCO_2kg^{-1}$ fuel
$M^t_{N_2}$	N_2 quantity in flue gases, λ = 1.0	2.221	kgN_2kg^{-1} fuel
M_{N_2}	N_2 quantity in flue gases, λ = 1.2	2.664	kgN_2kg^{-1} fuel
M_{SO2}	SO_2 quantity in flue gases	0.012	$kgSO_2kg^{-1}$ fuel
$M^t_{H_2O}$	H_2O quantity in flue gases, λ = 1.0	0.681	kgH_2Okg^{-1} fuel
M_{H_2O}	H_2O quantity in flue gases, λ = 1.2	0.687	kgH_2Okg^{-1} fuel
M_{O_2}	O_2 quantity in flue gases	0.129	kgO_2kg^{-1} fuel
$M_{g,t}$	Stoichiometry quantity of flue gases per kg of fuel	3.757	$kgkg^{-1}$
M_g	Total quantity of flue gases per kg of fuel, λ = 1.2	4.335	$kgkg^{-1}$

the assumption that no heat transfer occurs to the surrounding surfaces and there are no energy losses due to combustion inefficiency and to the surroundings:

$$\begin{gathered}\text{Fuel chemical energy + Air physical energy + Physical energy of re-circulating gases =}\\ \text{= Thermal energy in flue gases}\\ (NCV)+M_\mathrm{a}c_\mathrm{p,a}(T_\mathrm{a}-T_0)+x_\mathrm{gr}H_\mathrm{gr}=M_\mathrm{g}c_\mathrm{p,g}(T_\mathrm{ad}-T_0)\end{gathered} \tag{2}$$

As a baseline case, the assumed adiabatic flame temperature at 10% flue gas recirculation (x_{gr} = 0.1), is t_{ad} = 1,550 °C.

The mean specific heat capacity of the combustion products is calculated by the expression:

$$c_{p,\mathrm{g}}=\frac{1}{t_\mathrm{g}-t_0}(c_{p,\mathrm{tg}}t_\mathrm{g}-c_{p,\mathrm{g0}}t_0) \tag{3}$$

The enthalpy of flue gases in certain point is determined with the following expression:

$$H_\mathrm{g}=H_\mathrm{gt}+(\lambda-1)H_\mathrm{at}=M_{\mathrm{CO_2}}h_{\mathrm{CO_2}}+M_{\mathrm{N_2}}{}^\mathrm{t}h_{\mathrm{CO_2}}+M_{\mathrm{SO_2}}h_{\mathrm{SO_2}}+M_{\mathrm{H_2O}}{}^\mathrm{t}h_{\mathrm{H_2O}}+(\lambda-1)M_\mathrm{a,min}c_{p,\mathrm{a}}t_\mathrm{g}\ [\mathrm{kJkg^{-1}}] \tag{4}$$

The obtained flame adiabatic temperature value at the baseline conditions is T_{ad} = 1,826 K.

The temperature at the furnace outlet (in °C) is calculated with the following equation [29]:

$$t_\mathrm{B}=\frac{T_\mathrm{ad}}{M\left[\dfrac{\sigma_\mathrm{o}\psi_\mathrm{m}\varepsilon_\mathrm{w}A_\mathrm{f}T_\mathrm{ad}{}^3}{\eta_\mathrm{fur}B_\mathrm{f}(M_\mathrm{g}c_{p,\mathrm{g}})_\mathrm{m}}\right]^{0.6}+1}-273 \tag{5}$$

The meaning and the value of the variables and coefficients in eq. (5) are given in tab. 5. Since the calculation procedure of t_B is iterative, as a first step, it is assumed that t_B = 1160 °C, which results in mean heat content of flue gases $(M_g c_{p,g})_m$ = 6.469 kJ/kgK, where the mean specific heat capacity of flue gases is calculated by eq. (3), on a temperature range t_A = t_{ad} to t_B. With the values in tab. 5, the obtained temperature of the flue gases at the level below SH2 is t_B = 1161 °C.

The temperature profile along the boiler furnace was determined using the zone method [29]. According to that approach, the mean temperature of flue gases exiting the main combustion zone was calculated by iterative procedure with the expression:

$$t_{\mathrm{g}}" = \frac{\dfrac{\beta_{\mathrm{c}}(NCV)}{\eta_{\mathrm{c}}} + q_{\mathrm{a}} + q_{\mathrm{f,ph}} + q_{\mathrm{r}} - q_6}{(M_{\mathrm{g}} c_{p,\mathrm{g}})"} - \frac{\sigma_{\mathrm{o}} \varepsilon_{\mathrm{w}} (T_{\mathrm{g}}")^4}{B_{\mathrm{f}} (M_{\mathrm{g}} c_{p,\mathrm{g}})"} \psi A \tag{6}$$

Table 5. Coefficients and variables in the equation for t_B

Nomen.	Description	Value
M	Coefficient of temperature field in the furnace $M = A - Bx_F = 0.59 - 0.5 \times 0.2$	0.49
x_F	Location of the zone where maximum flame temperature occurs $x_F = m/n = h_{burners}/H_F$	0.2
σ_o	Stefan-Boltzman constant	$56.7 \cdot 10^{-12}$ kW/m^{2}K^4
ψ_m	Furnace waterwalls thermal efficiency coefficient [29, 31]	0.539
ε_w	Emissivity of the furnace walls (average)	0.55
A_f	Total surface area of furnace walls	2,800 m^2
η_{fur}	Furnace efficiency	0.98
B_f	Fuel consumption	120 kg/s
$(M_g c_{p,g})_m$	Average heat content of flue gases	6.469 kJ/kgK

Temperature of the flue gases at the exit of other zones was also calculated iteratively [29]:

$$t_{\mathrm{g}}" = \frac{\Delta\beta_{\mathrm{c}}(NCV) + (M_{\mathrm{g}} c_{p,\mathrm{g}})'t'}{(M_{\mathrm{g}} c_{p,\mathrm{g}})"} - \left[1 + \left(\frac{T"}{T'}\right)^4\right] \frac{\sigma_{\mathrm{o}} \varepsilon_{\mathrm{w}} (T')^4 \left[A_{\mathrm{m}}(\psi" - \psi') + \psi A_{\mathrm{z}}\right]}{2B_{\mathrm{f}} (M_{\mathrm{g}} c_{p,\mathrm{g}})"} \tag{7}$$

Temperature at the exit of zone with radiate-convective superheater was calculated as [29]:

$$t_{\mathrm{g}}" = \frac{2\Delta\beta_{\mathrm{c}} B_{\mathrm{f}}(NCV)}{2B_{\mathrm{f}}(Vc)" + \alpha_{\mathrm{c}} A_{\mathrm{c}}} + t'\left[\frac{2B_{\mathrm{f}}(M_{\mathrm{g}} c_{p,\mathrm{g}})' - \alpha_{\mathrm{c}} A_{\mathrm{c}}}{2B_{\mathrm{f}}(M_{\mathrm{g}} c_{p,\mathrm{g}})" + \alpha_{\mathrm{c}} A_{\mathrm{c}}}\right] + t_{\mathrm{z}} \frac{2\alpha_{\mathrm{c}} A_{\mathrm{c}}}{2B_{\mathrm{f}}(M_{\mathrm{g}} c_{p,\mathrm{g}})" + \alpha_{\mathrm{c}} A_{\mathrm{c}}} - \left[1 + \left(\frac{T"}{T'}\right)^4\right] \frac{\sigma_{\mathrm{o}} \varepsilon_{\mathrm{l}} (T')^4}{2B_{\mathrm{f}}(M_{\mathrm{g}} c_{p,\mathrm{g}})" + \alpha_{\mathrm{c}} A_{\mathrm{c}}} \psi A_{\mathrm{z}} \tag{8}$$

Heat received by working fluid water/steam in each heat exchanging surface is defined as:

$$Q_{\mathrm{i}} = m_{\mathrm{i}}(h_{\mathrm{out}} - h_{\mathrm{in}}) \tag{9}$$

which is equal to the heat transferred from the combustion products to the HES:

$$Q_{\mathrm{i}} = (H_{\mathrm{g,in}} - H_{\mathrm{g,out}}) B_{\mathrm{f}} \eta_{\mathrm{c}} \eta_{\mathrm{ins}} \tag{10}$$

The previous equations were applied for energy balancing of all the HES in the steam generator, determining that way the energy distribution to the heat exchanging surfaces and the flue gas temperature change along the boiler height, as well. Further step towards an improvement of the overall procedure will be application of CFD analysis of the steam generator, with approach similar to the one implemented in [32, 33].

Exergy balance calculations

For solid fuels the chemical exergy is calculated with the following semi-empirical relation [34]:

$$e_{\mathrm{f,ch}} = (NCV)(1.0438 + 0.0013\frac{h}{c} + 0.1083\frac{o}{c} + 0.0549\frac{n}{c}) + 6{,}740\mathrm{s} \tag{11}$$

The total fuel chemical exergy [kW] at plant nominal operating mode is $E_{\mathrm{f,ch}} = B_{\mathrm{f}}e_{\mathrm{f,ch}}$.

The specific physical exergy of the working fluid water/steam is generally defined:

$$e_{\mathrm{w/s}} = (h - h_0) - T_0(s - s_0) \tag{12}$$

The water/steam side exergy flow rate is calculated as $E_{\mathrm{w/s}} = m_{\mathrm{w/s}}e_{\mathrm{w/s}}$ [kJs^{-1}]. The atmospheric pressure and reference state temperature are taken respectively: p_0 = 101325 Pa and T_0 = 293 K.

The specific physical exergy of flue gases is calculated as exergy of ideal gases mixture:

$$e_{\mathrm{g}} = (h_{\mathrm{g}} - h_0)\left(1 - \frac{T_0}{T_{\mathrm{g}} - T_0}\ln\frac{T_{\mathrm{g}}}{T_0}\right) + R_{\mathrm{g}}T_0\ln\frac{p_{\mathrm{g}}}{p_0} = c_{p,\mathrm{g}}\left[(T_{\mathrm{g}} - T_0) - T_0\ln\frac{T_{\mathrm{g}}}{T_0}\right] + R_{\mathrm{g}}T_0\ln\frac{p_{\mathrm{g}}}{p_0} \tag{13}$$

With $p_{\mathrm{g}} \approx p_0$ and $p \approx$ const, the last term in the previous equations becomes ≈ 0. In these equations, h_{g} and $c_{\mathrm{p,g}}$ are calculated as for a gas mixture, $h_{\mathrm{g}} = \Sigma x_k h_k$ and $c_{p,\mathrm{g}} = \Sigma x_k c_{p,\mathrm{k}}$, where x_{k} is mass share of each gas component. The flue gas exergy flow rate [kJs^{-1}] is calculated as $E_{\mathrm{g}} = M_{\mathrm{g}}B_{\mathrm{f}}e_{\mathrm{g}}$. Once the exergy flows of all input and output streams for certain component were determined, (*i. e.* heat exchanging surface), the exergy destruction was calculated as difference between the exergy changes on gas and on water/steam side. The specific exergy of the hot gas re-circulation stream at level C (fig. 1), at $t_{\mathrm{gr}} \approx 1050$ °C, is calculated on a basis of eq. (13). The exergy loss with the flue gases at the boiler exit is calculated in an analogous way.

Exergy dissipation due to combustion irreversibility d_{comb} is calculated from the balance of the boiler furnace:

$$e_{\mathrm{f,ch}} + e_{\mathrm{a}} = e_{\mathrm{ad}} + e_{\mathrm{l}} + d_{\mathrm{comb}} \tag{14}$$

The exergy of the preheated air is:

$$e_{\mathrm{a}} = M_{\mathrm{a}}\left[(c_{p,\mathrm{a}}t_{\mathrm{a}} - c_{p,0}t_0)\left(1 - \frac{T_0}{T_{\mathrm{a}} - T_0}\ln\frac{T_{\mathrm{a}}}{T_0}\right)\right] \tag{15}$$

The exergy of the flue gas mixture at temperature T_{ad} is given:

$$e_{\mathrm{ad}} = M_g\left[(c_{p,\mathrm{g,ad}}t_{\mathrm{ad}} - c_{p,0}t_0)\left(1 - \frac{T_0}{T_{\mathrm{ad}} - T_0}\ln\frac{T_{\mathrm{ad}}}{T_0}\right)\right] \tag{16}$$

Apart of d_{comb}, the exergy losses related to the boiler furnace include losses due to incomplete combustion – of chemical (q_3) and mechanical reasons (q_4), heat loss due to radiation and convection (q_5), and loss due to heat contained in the ash and slag falling through the furnace funnel (q_6):

$$e_{\mathrm{l}} = q_3 + q_4 + q_5\left(1 - \frac{T_0}{T_{\mathrm{b,env}}}\right) + q_6\left(1 - \frac{T_0}{T_{\mathrm{ash}}}\right) \tag{17}$$

The values of q_3, q_4, q_5, and q_6 (kJkg^{-1} fuel) are set on a basis of recommendations given in [29]. In this case, it was assumed that q_3 is 1% of the energy entering the furnace with the fuel q = (NCV), q_4 is 2% of q, q_5 and q_6 are 0.5% of q, each and T_{ash} = 873 K (600 °C). Thus, it is obtained e_{l} = 256 kJ/kg.

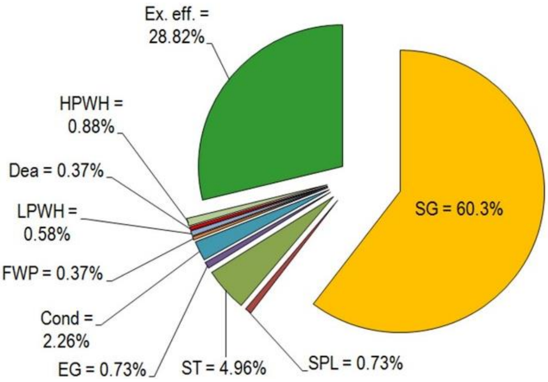

Figure 3. Exergy efficiency of the TPP (green color) and exergy destruction ratios of the plant components (nominal mode)
Ex. eff. – exergy efficiency of the TPP unit, SG – steam generator, SPL – steam pipelines, ST – steam turbine, EG – electric generator, Cond – condenser, FWP – feed-water pump system, LPWH – low-pressure regenerative water heaters, Dea – deaerator, HPWH – high-pressure water heaters (for color image see journal web site)

Results and discussion

The main purpose of the exergy balance analysis is to identify the components of the system where large exergy losses occur. Based on the approach described here and in [5, 8, 25], exergy flow in different streams of the power plant steam cycle was calculated at the points before and after the process components. Several operating modes of the power plant were subject of analysis, with regard to the plant design parameters and measured values from the monitoring system [30]. The overall exergy balance diagram of the power plant operation in nominal mode is given in fig. 3, based on [30]. The analysis shows that the heat transfer processes in the cycle, particularly in the steam generator, are in a relatively good range that suggests a proper design of the power cycle equipment. In different modes, the shares of exergy loss for the plant components change, while the boiler covers yet the major part of loss with about 82-84% of the total exergy destruction. This is a reason for devoting more attention to the processes taking place in the steam generator plant in order to improve its efficiency. In addition to the steam generator, regardless of the operating mode, the turbine system and the condenser are components with significant exergy destruction, but with small potential for optimization. The destruction of exergy in the steam turbine is major contributor to the total plant exergy destruction, due to the irreversibility in the steam expansion process. According to the First law analysis, the plant condenser has the highest heat loss in the cycle, however, due to the low quality of the lost energy, the saving potential is not very significant.

The flue gases temperature change along the steam generator gas path, determined with the zone method (according to [29]), is presented in fig. 4 for several operating modes.

The chart in fig. 5 shows the calculated energy distribution along the steam generator sections and HES. It presents how the energy contained in the gases is distributed along separate sections of the boiler (Total-gs), the amount of flue-gas energy that is delivered to evaporator segments (Evap-gs) and to other HES (HExs-gs), the amount of energy that is absorbed by the HES for water heating, steam superheating and air heating (HExs-wsas) and the energy that is absorbed in separate segments of the evaporator (Evap-wss).

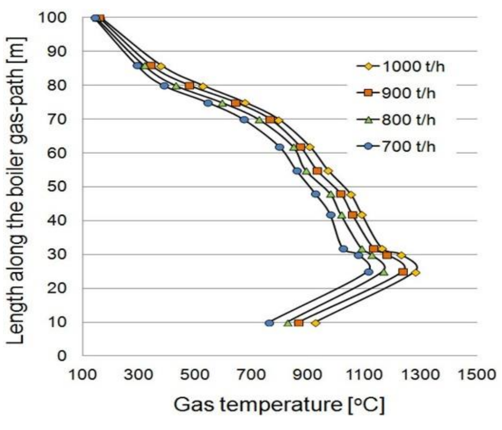

Figure 4. Temperature profile along the boiler gas-path obtained at different operating regimes

Calculated exergy changes of the streams on the sides of the combustion products and water/steam, as well as the exergy dissipation due to combustion and heat transfer in the steam generator in the nominal operating mode are presented in tab. 6. The exergy flow in the steam generator in nominal operating mode is presented in fig. 6. The largest exergy destruction occurs due to the combustion process irreversibility and the heat transfer from the combustion products to the working medium (water, steam and air), which determines the boiler's internal efficiency.

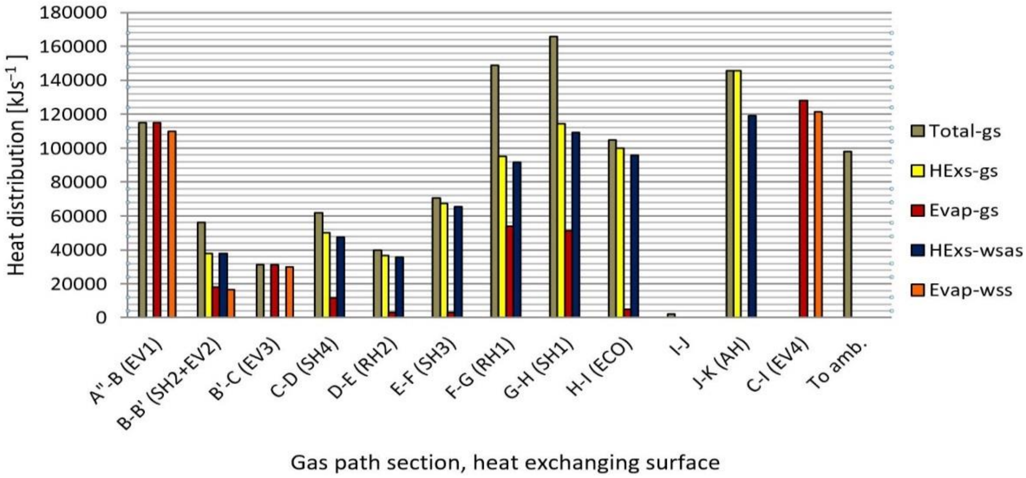

Figure 5. Heat distribution in the steam generator on gas, water-steam and air sides
Total-gs – distribution of available flue gas energy, Evap-gs – energy delivered to evaporator, HExs-gs – energy delivered to other HES, HExs-wsas – energy absorbed in HES for water heating, steam superheating and air heating, Evap-wss – energy absorbed in separate parts of the evaporator

Table 6. Exergy change, exergy dissipation/destruction due to combustion and heat transfer

Location (description)	Exergy change – gas side [kW]	Exergy change – water/steam/air side [kW]	Exergy destruction/loss [kW]
A-B (furnace)	1095640 to 764000	–	309000 (combustion)
A-B (evaporator)	261097	148421	99926
B-B' (SH2)	44122	20768	21806
B'-C (evap.)	21344	–	23394
C-D (SH4)	46842	28317	17551
D-E (RH2)	33425	19979	12444
E-F (SH3)	56137	38013	16575
F-G (RH1)	67853	48271	17131
G-H (SH1)	83053	59072	20361
H-I (ECO)	65960	45348	17079
I-J	1171	0	1081
J-K (AH)	65550	34060	31220
To ambient	17446	0	16104

Much smaller, but still significant exergy destruction occurs due to inefficiencies in the coal treatment system, hot flue gas re-circulation, air-fuel mixing, flue gases releasing to atmosphere (due to temperature difference), boiler blowdown and other reasons. With regard to the steam generator, on the basis of the previous analysis, optimization of the combustion and heat transfer processes can be achieved through a set of measures. Thus, sufficient preheating of combustion air, controlling the excess air at optimal level, lignite pre-drying by use of re-circulating flue gases heat, creating conditions for suitable fuel and air mixing, reducing temperature difference between combustion products and water-steam in every stage of heat transfer process, utilization of the energy contained in the exit gases and other measures can be effective in controlling the exergy losses.

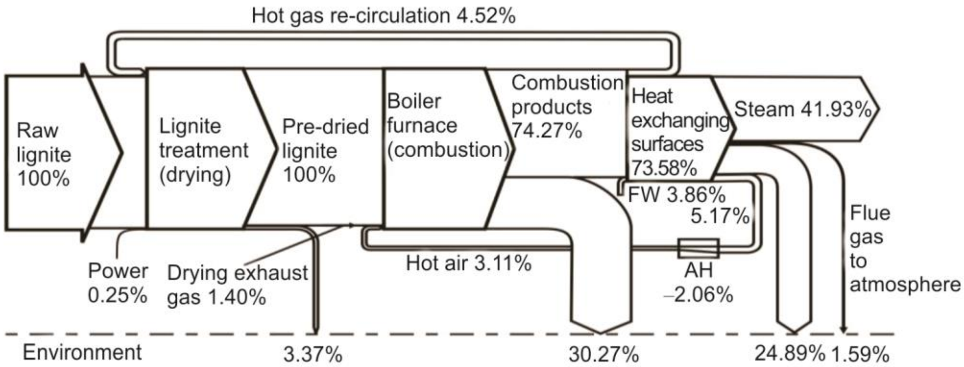

Figure 6. Exergy flow diagram in the steam generator at nominal working regime (315 MW)

Since changes in the steam generator internal with the aim of heat transfer optimization are hard to be undertaken at this point, the retrofit measures related with fuel treatment and combustion process optimization seem to be achievable. In that sense, fuel pre-drying by use of combination of flue gases hot- and cold re-circulation and fresh air preheating with dryer exhaust gases is an attractive option, with significant energy saving potential. In the present case, the raw coal is pulverized and dried in a direct closed pulverizing system, which means that the fuel moisture is not prevented from entering the boiler but evaporated by the supplied energy of the recirculating flue gas. Since the moisture content in the fired lignite is high, a significant amount of flue gas needs to be re-circulated. By implementation of so-called open pulverizing system, the moisture released from lignite in the milling system would not enter the furnace, that will result with boiler efficiency improvement [35]. In that case, the adiabatic combustion temperature would increase and the exergy destruction in the combustion process will be reduced. Air preheating by use of dryer exhaust gases can be integrated within this measure and would further increase the overall boiler efficiency.

Further steam generator efficiency improvement can be achieved by maintaining optimum level of excess air in the furnace and by minimizing disorganized intake of air along the gas path. In addition, an improved combustion system, including staged air and fuel introduction and better mixing, will result in reduced formation of NO_x. This measure is in direct correlation with the opportunity to reduce exergy dissipation due to the irreversibility of the heat transfer, by reducing the temperature difference between the combustion products and working fluid (water, steam and air) in every stage of the HES. Further comprehensive in-depth research, that includes both energy and exergy efficiency analyses, should yield more complete insight into the performance and efficiency enhancement opportunities of the TPP and its components in different operating regimes, including combined heat and power generation modes [35].

Conclusion

The efficiency of the thermal processes occurring in a 315 MW utility lignite-fired power plant is analyzed from the energy and exergy viewpoint. An optimization model is established, based on an energy balance, zonal calculation method and exergy method, in order to determine the magnitude of the exergy dissipation in the plant components. The largest exergy destruction in the power plant occurs in the steam generator, because of the high internal exergy losses due to the irreversibility of the combustion process and the heat transfer between the combustion products and water-steam stream. Depending on the working regime, the steam generator is responsible for about 83% of the total exergy destruction in the power plant. Despite the fact that the considered steam generator is already in use longer than its designed technical life-time, its energy efficiency is still relatively high, since the internal irreversible phenomena do not affect much the energy balance. However, its exergy efficiency is considerably lower, with values slightly above 40% in different operating modes. Based on the conducted analysis, several options are considered to be analyzed in more detailed manner for enhancement of the power plant and boiler efficiencies. In that direction, optimization of combustion through sufficient combustion air preheating by use of dryer exhaust gases, controlling the excess air at optimal level, delivery of pre-dried lignite by use of hot- and cold flue gas recirculation, creating conditions for suitable fuel and air mixing can be effective in order to reduce the exergy dissipation. The opportunities for limiting the exergy dissipation in the heat transfer process are mostly based on reduction of the temperature difference between combustion products and working fluid in HES and, therefore, less feasible.

Nomenclature

A_c – heat transfer surface, [m^2]
A_f – surface area of furnace walls, [m^2]
A, A_z – surface of zone walls, [m^2]
B, B_f – fuel consumption, [kgs^{-1}]
c_p – specific heat capacity, [kJkg^{-1}K^{-1}]
E – exergy flow rate, [kJs^{-1}]
H – enthalpy per kg fuel, [kJkg^{-1}]
h – specific enthalpy, [kJkg^{-1}]
M – mass per kg fuel, [kgkg^{-1}]
m – mass-flow, [kgs^{-1}]
NCV – net calorific value, [kJkg^{-1}]
p – pressure, [Pa]
Q – thermal energy, heat, [kJs^{-1}]
q – heat per kg fuel, [kgkg^{-1}]
q_3-q_6 – energy losses in the boiler, [kJkg^{-1}]
R_g – gas constant, [kJkg^{-1}K^{-1}]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [K]
t – temperature, [°C]
x_{gr} – re-circulating ratio, [–]
x_k – share of gas component in mixture, [–]

Greek symbols

α_c – convective heat transfer coefficient
β_c – combustion efficiency
$\Delta\beta_c$ – fraction of fuel undergone combustion in a considered zone
λ – excess air coefficient
ψ – thermal efficiency coefficient
ψ' – thermal efficiency of a zone inlet
ψ'' – thermal efficiency of a zone outlet
η – efficiency, conversion degree
‹'› – value of variable at the zone entrance
‹"› – value of variable at the zone exit

Subscripts

a – air
ad – adiabatic conditions
ash – ash
b,env – boiler envelope
c – combustion, conversion
ch – chemical
f – fuel
fur – furnace
g – gas, flue gas
in – inlet
ins – insulation
l – loss
m – mean, average
min – minimal, stoichiometric
out – outlet
r – re-circulation
w – wall, water
w/s – water/steam
0 – reference conditions

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