COMPARATIVE EXERGETIC PERFORMANCE ANALYSIS FOR CERTAIN THERMAL POWER PLANTS IN SERBIA

by

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Traditional methods of analysis and calculation of complex thermal systems are based on the first law of thermodynamics. These methods use energy balance for a system. In general, energy balances do not provide any information about internal losses. In contrast, the second law of thermodynamics introduces the concept of exergy, which is useful in the analysis of thermal systems. Exergy is a measure for assessing the quality of energy, and allows one to determine the location, cause, and real size of losses incurred as well as residues in a thermal process. The purpose of this study is to comparatively analyze the performance of four thermal power plants from the energetic and exergetic viewpoint. Thermodynamic models of the plants are developed based on the first and second law of thermodynamics. The primary objectives of this paper are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses. Finally, by means of these analyses, the main sources of thermodynamic inefficiencies as well as a reasonable comparison of each plant to others are identified and discussed. As a result, the outcomes of this study can provide a basis for the improvement of plant performance for the considered thermal power plants.

Key words: *power plant, efficiency, exergy, losses, destruction*

Introduction

When two systems with different conditions interact, there is a tendency to create a state of equilibrium and there is a possibility to perform work. If one of the systems is assumed to be surroundings and the other the system being studied, exergy can be defined as the maximum possible work to be obtained when the two systems reach equilibrium, assuming that heat is exchanged only with the surroundings. Every system not in equilibrium with its surroundings can perform work, and a system in the equilibrium state with its surroundings, by definition, can not. Exergy can be treated as the measure or distance between the state of the system and the state of the surroundings. It becomes the attribute for both composition of the system and the surroundings. Exergy can be destroyed but it can not be conserved. In boundary cases, exergy is being completely destroyed, and this occurs when a system spontaneously comes in equilibrium with its surroundings without performing work, *i. e.* the initial ability of the system to perform work gets consumed by the spontaneous process. Since no work is needed for spontaneous equilibrium, it can be concluded that exergy can have a min-

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imum value of zero, and it cannot be negative. One of the main characteristics of exergy is that it can be transferred between the systems.

Exergy analysis is a method that combines energy and mass conservation with the second law of thermodynamics. The exergy analysis method enables the determination of the location, cause and source of losses. This information can be useful in designing a new energy efficient system but also for improving the performance of existing systems. Exergy analysis provides an insight into and a possibility to find causes for thermodynamic inefficiency of the system.

The exergy analysis method has been applied to thermal plants by numerous researchers in order to obtain efficiency, and optimize and improve the operation of such plants. Results show that most of exergy destruction occurs during fuel combustion, more than 80% [1, 2]. Cihan *et al.* [3] and Ameri *et al.* [4] performed the exergy analysis for thermal power plants and showed that it can pinpoint components with high inefficiency. Kanoglu *et al.* [5] stressed out the importance of understanding energy and exergy efficiency in order to improve the operation of power plants. The exergy concept plays an important role in developing energy policy by implementing the analysis for energy conversion or the impact of energy systems on the environment [6, 7].

Description of the system

This paper analyzes two electricity production plants and one cogeneration plant. The plants are the property of the Republic of Serbia, and their lifespan is more than 20 years. The classical thermal power plants use coal as primary fuel, while the combined power plant uses natural gas. In all three cases heating of the main condensate is done in low pressure heaters (LPH). Steam degasation is done in a deaerator (DA3), after which the supply water is heated in high-pressure heaters (HPH). The number of low and high pressure heaters varies from case to case and is given in tab. 1. The cogeneration plant has two additional heaters for the district heating system, which are bypassed when the plant is in the condensing operation mode. A steam boiler (SB) generates superheated steam with defined parameters. Steam from a turbine is used for heating the main condensate and supply water, for district heating and deaerator operation (Ep1, Ep2,...). After the expansion in the turbine, steam is completely condensed in a condenser (CN). The turbine set comprises a high pressure turbine (HTP), a medium pressure turbine (MTP), and a low pressure turbine (LTP). After the expansion in the high pressure turbine, reheating occurs, followed by the expansion in the medium and low pressure turbines. Technical details of the plants are given in tab. 1.

Numerical simulation of operation

For simulation purposes, the software was developed, which uses Microsoft Excel as the platform with built in Visual Basic for Applications, so the macros were written in the VBA editor. Thermodynamic properties were calculated by IAPWS-IF97 (water steam pro) method with VBA.

During simulation, the dependence of turbine efficiency on the design and constructive characteristics was taken into account. At the same time, turbine efficiency varied with operation and was a function of mass flow rates (loads) and other relevant parameters. Characteristics were obtained from measurements or directly from the component manufacturer.

There are methods in literature for calculating turbine internal efficiency for different operating regimes as functions of various parameters in graphic or analytical form. The most complete one is given in the paper by Spencer *et al.* [8]. This paper provides the method

Table 1. Plant operation parameters

	Plant			
	Kostolac	Kolubara	TE-TO Novi Sad	TE-TO Novi Sad
Nominal electric output [MW]	348.5	110	117	115
Nominal heat output [MW]	–	–	0	140
Steam production [th^{-1}]	1000	383.4	432	486
Steam pressure at GV1 [MPa]	18.6	12.7	12.75	12.75
Steam temperatureat GV1 [°C]	540	535	555	555
Steam reheat pressure [MPa]	4.37	3.18	2.31	2.6
Steam reheat temperature [°C]	540	535	349.2	356.8
Pump efficiency, mechanical [%]	0.98	0.98	0.98	0.98
Generator efficiency [%]	0.96	0.96	0.96	0.96
Low/high pressure pre-heater	4/2	5/2	4/3	4/3
Heater number	–	–	2	2
Condensing pressure (MPa)	0.0042	0.01	0.0065	0.0061
Cooling water temperature [°C]	12	27	27	27
Coolng water flow [th^{-1}]	46800	15169	16070	7115
Flue gasses temperature [°C]	150	150	150	150
Fuel	Lignite	Lignite	Natural gas	Natural gas
Heating value [kJkg^{-1}]	6350	6350	50000	50000

for calculating turbine internal efficiency as well as analytical functions for efficiency correction due to a regime change over for different types of turbines. These functions are empirically derived and based on testing turbines manufactured by the General Electric Company. Turbine internal efficiency, for every operating regime, is calculated from correction factors using the equation:

$$\eta_i = \eta_{i0}^* \prod_{j=1}^{n} (1+\varepsilon_j) \tag{1}$$

where η_{i0}^* is the *zero* or normalized internal efficiency without corrections, ε_j – the deviation of efficiency based on the j^{th} influential parameter, n – the total number of corrections. The *zero* or normalized turbine efficiency is obtained from the nominal turbine efficiency:

$$\eta_{i0}^* = \frac{\eta_{i0}}{\prod_{j=1}^{n_1} (1+\varepsilon_j^0)} \tag{2}$$

where η_{i0} is the turbine nominal internal efficiency.

Corrective functions for the deviation of internal efficiency are given in two basic forms [8] depending on whether the correction is a function of one or two parameters. The first form of the corrective function with two influential parameters is given:

$$\varepsilon_i = \frac{1}{a_0} \sum_{j=1}^{n_2} \sum_{k=1}^{n_1} a_{ijk} x^{k-1} y^{j-1} \tag{3}$$

While the other form with one influential parameter is much simpler:

$$\varepsilon_i = \frac{1}{c_0} \sum_{j=1}^{n_3} c_{ij} w^{j-1} \tag{4}$$

In [8] all the values for coefficients a_0, c_0, n_1, n_2, n_3, a_{ijk}, and c_{ij}. are given.

The flow characteristic of a turbine defines its performance in changing input and output parameters. In this simulation model it is defined in the form of the Cone law [9].

Heat exchangers can be modeled using the number of transfer units NTU method. The method is based on knowing the mass flow rate and enthalpies entering and leaving the heat exchanger in the design regime. Pressure drop is also taken into account. Finally, heat transfer coefficients for every part of the heat exchanger should be known in order to determine these coefficients in different operating regimes. Since the NTU method is complicated for implementation, an alternative heat exchanger model is used in this paper which is based on connections between terminal temperature differences (TTD) for different loads. Total heat balance is used for deriving the steam mass flow rate taken from the turbine. Numerical correlation was introduced by Marcuello [10] and it is used for solving TTD in the heat exchanger (entering Δt_2 and leaving Δt_1).

Energy and exergy analysis

Energetic performance analysis is based on the first law of thermodynamics. According to the first law of thermodynamics, the common main performance criteria are power output and thermal efficiency. These parameters are also decisive performance criteria in the economic analysis of power plants. In this analysis, the input and output values of the plant components can be determined using the measured/calculated thermodynamic variables such as enthalpy, pressure, temperature, entropy, mass flow rate, and quality. For every individual component, the three balance equations are applied to find the work output, the heat added, the rate of exergy decrease, the rate of irreversibility, and the energy and exergy efficiencies. The balance equations are then written as follows.

The energy balance equation:

$$-\dot{W} + \dot{Q} = \sum \dot{m}_i (h_{\text{out},i} - h_{\text{in},i}) \tag{5}$$

The thermal efficiency of the CHP plants can be calculated:

$$\eta = \frac{\dot{W} + \dot{Q}_\text{H}}{\dot{m}_\text{f} \text{LHV}} \tag{6}$$

where LHV is the lower heating value of coal, and $\dot{m}_\text{f}$ – the coal flow rate.

Disadvantages of energy efficiency as criteria to evaluate efficiency of CHP plants were analyzed by Huang [11] for a basic CHP plant with a gas turbine. The defined energy efficiency does not take into account the qualitative (thermodynamic, technical, economical) difference between heat and electricity. In order to eliminate these disadvantages, *factorized* efficiency is introduced:

$$\eta_{\mathrm{CHP\_F}} = \frac{\alpha_{\mathrm{H}}\dot{Q}_{\mathrm{H}} + \alpha_{\mathrm{e}}\dot{W}}{\alpha_{\mathrm{f}}\dot{Q}_{\mathrm{f}}} \tag{7}$$

where α is the weighing (ponderation) factor for all three forms of energy (heat, electricity and chemical energy of fuel). This factor is influenced by economical criteria or it can be stated by the government. Feng *et al.* [12] concluded that this approach is adopted in various legislation acts in various countries and most often it is $\alpha_{\mathrm{e}} = \alpha_{\mathrm{f}}$. US PURPA (*Public Utility Regulatory Policy Act*), gives values $\alpha_{\mathrm{e}} = \alpha_{\mathrm{f}} = 1$, and $\alpha_{\mathrm{H}} = 0.5$ [13], so:

$$\eta_{\mathrm{CHP\_F\_PURPA}} = \frac{0.5\dot{Q}_{\mathrm{H}} + \dot{W}_{\mathrm{e}}}{\dot{Q}_{\mathrm{f}}} \tag{8}$$

In some countries, the legislation weighing factor, α_{H}, is not considered constant but rather as a ratio of efficiency for electricity production and efficiency for heat production seen as separate processes:

$$\alpha_{\mathrm{H}} = \frac{\eta_{\mathrm{e\ SEP}}}{\eta_{\mathrm{t\ SEP}}} \tag{9}$$

Generally, *factorized* efficiency represents a simpler form of exergy efficiency leading to a conclusion that the weighing factor selection can lead to inaccurate results.

Exergetic performance analysis

Exergetic performance analysis is based on the second law of thermodynamics. The results obtained from such an analysis can be used as a guide for diminishing the irreversibilities in the power plants and thereby enhancing their performances. In fact, exergy is a thermodynamic indicator that shows the transformation potential and convertible limit of an energy carrier to maximum theoretical work under the conditions imposed by an environment at given pressure and temperature [14, 15]. In the scope of this exergetic performance analysis study, exergy efficiency and exergy destruction rate of both the plant and plant component are determined. In addition, exergy losses per unit power output in the plants are defined and used as a new exergetic performance criterion.

For stationary conditions, exergy balance for the control volume is given:

$$\sum_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - \dot{W}\sum_i \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \sum_i \dot{m}_{\mathrm{out}}e_{\mathrm{out}} - \dot{E}_{\mathrm{D}} = 0 \tag{10}$$

The first term on the left side of the equation represents the exergy transfer related to heat transfer, $\dot{Q}_j$, which takes place in the area of temperature, T_j, the second term represents the exergy transfer related to work, $\dot{W}$, while the third and fourth terms represent exergy entering and leaving the control volume. The fifth term represents the exergy destruction due to process irreversibility. It is calculated according to Gouy-Stodola theorem:

$$\dot{E}_{\mathrm{D}} = T_0\dot{S}_{\mathrm{gen}} \tag{11}$$

The fifth term incorporates exergy loss (exergy of products of combustion, wastewater in condenser, heat rejected to surroundings...). When applying exergy analysis it is very important to have control volume boundaries defined in a proper manner, since heat transfer can be attributed to exergy destruction or exergy losses. The most acceptable way, and the one which enables partial analysis, is to treat every component of the system as a separate control volume.

Total exergy of some system, E, can be divided into physical exergy, E^{PH}, kinetic exergy, E^{KN}, potential exergy, E^{PT}, and chemical exergy, E^{CH}, *i. e.*:

$$E = E^{\mathrm{PH}} + E^{\mathrm{KN}} + E^{\mathrm{PT}} + E^{\mathrm{CH}} \tag{12}$$

Using the same approach, total exergy of a system can be expressed in specific values in the form:

$$e = e^{\mathrm{PH}} + e^{\mathrm{KN}} + e^{\mathrm{PT}} + e^{\mathrm{CH}} \tag{13}$$

Physical exergy of a closed system is defined:

$$e^{\mathrm{PH}} = (u - u_0) + p_0(v - v_0) - T_0(s - s_0) \tag{14}$$

i. e.

$$e^{\mathrm{PH}} = (h - h_0) - T_0(s - s_0) \tag{15}$$

Kinetic and potential exergy are calculated:

$$e^{\mathrm{KN}} = \frac{V^2}{2} \tag{16}$$

$$e^{\mathrm{PT}} = \mathrm{g}\ z \tag{17}$$

Usually, in exergy analysis, kinetic and potential exergy are neglected. Chemical exergy is a component associated with the chemical composition of the system and its surroundings. Standardized values of molar chemical exergy for chemical elements and compunds are available in the literature as functions of temperature and pressure of the surroundings (temperature, T_0, and pressure, p_0) [14].

For mixtures containing gases other than those present in the reference tables, chemical exergy can be evaluated using the equation:

$$\dot{E} = \dot{m}\left[(h - h_0) - T_0(s - s_0) + \sum x_k e_k^{\mathrm{CH}} + \mathrm{R}T_0 \sum x_k \ \ln x_k \right] \tag{18}$$

where x_k is the mole fraction of the k^{th} gas in the mixture, and R – the universal gas constant. In exergy analyses, another significant matter which must be noted is the reference conditions [16, 17].

Fuel chemical exergy can be calculated with simple equations containing standard chemical exergies of fuel components. In the case of hydrocarbons C_aH_b, chemical exergy is defined as [18]:

$$\begin{aligned}\bar{e}^{\mathrm{CH}} &= \left[\bar{h}_{\mathrm{F}} + \left(a + \frac{b}{4} \right) \bar{h}_{\mathrm{O_2}} - a\bar{h}_{\mathrm{CO_2}} - \frac{b}{2}\,\bar{h}_{\mathrm{H_2O(g)}} \right](T_0,\ p_0) - \\ &- T_0 \left[\bar{s}_{\mathrm{F}} + \left(a + \frac{b}{4} \right) \bar{s}_{\mathrm{O_2}} - a\bar{s}_{\mathrm{CO_2}} - \frac{b}{2}\,\bar{s}_{\mathrm{H_2O(g)}} \right](T_0,\ p_0) + \\ &+ \mathrm{R}T_0 \ \ln \left[\frac{(x_{0,\mathrm{O_2}})^{a+\frac{b}{4}}}{(x_{0,\mathrm{CO_2}})^{a} (x_{0,\mathrm{H_2O(g)}})^{\frac{b}{2}}} \right]\end{aligned} \tag{19}$$

For solid fuels, a semi-empirical formula depending on fuel composition (carbon, hydrogen, oxygen and sulfur), is used in the form [16]:

$$e^{\mathrm{CH}} = \mathrm{LHV}\left(1.0438 + 0.0013\frac{x_{\mathrm{H}}}{x_{\mathrm{C}}} + 0.1083\frac{x_{\mathrm{O}}}{x_{\mathrm{C}}} + 0.0549\frac{x_{\mathrm{N}}}{x_{\mathrm{C}}}\right) + 6740x_{\mathrm{S}} \tag{20}$$

Often in the literature one can find approximate values of chemical exergy as a function of the lower heating value, given as the ratio between chemical exergy and the lower heating value $\varphi = e^{\mathrm{CH}}/\mathrm{LHV}$. Table 2 gives values of φ for some fuels [19].

In exergy analysis, it is very important to correctly assume parameters of the surroundings. These parameters imply reference temperature and pressure, as well as air composition. There are several models in the literature, but the mostly accepted values are $T_0 = 298.15$ K, $p_0 = 1013$ mbar. The reference air composition is given in tab. 3.

Table 2. Ratio between chemical exergy and lower heating value for some fuels

Fuel	Natural gas	Gas	CO	Hydrogen
φ	1.04±5%	1.00±1%	0.973	0.985

Table 3. Molar composition of atmospheric air [20]

Component	N_2	O_2	CO_2	$H_2O_{(g)}$
Molar share [%]	77.48	20.59	0.03	1.9

Exergy efficiency

In the exergetic performance analysis, exergy efficiency gives a measure of the performance of a system or a component. Exergy efficiency of the components in the investigated power plants is defined on the basis the product and fuel approach given in the literature. The fuel represents the net exergy resources spent in this component for generating the product, while the product indicates the desired purpose of including the component into the power plant [14]. Accordingly, exergy destruction and exergy efficiency of the main component in a power plant are given in tab. 4. There are different definitions for exergy efficiency. They all have in common that it is applied for stationary conditions (steady state), where a control volume of asystem is clearly defined and all irreversibilities are taken into account.

Total exergy destruction rate in the plant can be determined as a sum of exergy destruction rates of components.

The simplest form of exergy efficiency is derived from the conventional definition, based on exergy balance of entering and leaving streams. Conventional exergy efficiency is the ratio of leaving exergy stream and total entering exergy streams, where

$$\dot{E}_{\mathrm{in}} = \dot{E}_{\mathrm{out}} + \dot{I} \tag{21}$$

where $\dot{I}$ indicates the irreversibility of the process, while the other forms of exergy are included in entering and leaving exergy the control volume.

Conventional exergy efficiency is given in the form:

$$\eta_{\mathrm{CHPex}}^{\;k} = \frac{\dot{E}_{\mathrm{out}}}{\dot{E}_{\mathrm{in}}} = \frac{\dot{W} + \dot{E}_{\mathrm{Q_H}}}{\dot{E}_{\mathrm{f}}} \tag{22}$$

where $\dot{E}_{\mathrm{Q_H}}$ represents the exergy equivalence of heat, $\dot{E}_{\mathrm{f}}$ – the entering exergy (fuel exergy), while exergy of electricity or mechanical work is equal to energy.

For the whole thermal power plant, the exergy efficiency can be given:

Table 4. Component exergy losses and component exergy efficiency

Component	Exergy loss	Exergy efficiency
Steam boiler	$\dot{E}_{\rm D,B} = \dot{E}_F + \Sigma\dot{E}_{\rm in,B} - \Sigma\dot{E}_{\rm out,B}$	$\eta_{\rm ex,B} = (\dot{E}_{\rm out,B} - \dot{E}_{\rm in,B})/\dot{E}_{\rm F}$
Steam turbine	$\dot{E}_{\rm D,T} = \Sigma\dot{E}_{\rm in,T} - \Sigma\dot{E}_{\rm out,T} - \dot{W}$	$\eta_{\rm ex,T} = \dot{W}_{\rm T}/(\dot{E}_{\rm in,T} - \dot{E}_{\rm out,T})$
Condenser	$\dot{E}_{\rm D,C} = \Sigma\dot{E}_{\rm in,C} - \Sigma\dot{E}_{\rm out,C}$	$\eta_{\rm ex,C} = \dot{E}_{\rm out,C}/\dot{E}_{\rm in,C}$
Pump	$\dot{E}_{\rm D,P} = \Sigma\dot{E}_{\rm in,P} - \Sigma\dot{E}_{\rm out,P} + \dot{W}_{\rm P}$	$\eta_{\rm ex,P} = (\dot{E}_{\rm out,P} - \dot{E}_{\rm in,P})/\dot{W}_{\rm P}$
Heat exchanger	$\dot{E}_{\rm D,H} = \Sigma\dot{E}_{\rm in,H} - \Sigma\dot{E}_{\rm out,H}$	$\eta_{\rm ex,H} = \dot{E}_{\rm out,H}/\dot{E}_{\rm in,H}$
Cycle	$\dot{E}_{\rm cycle} = \Sigma\dot{E}_{\rm all\ components,D}$	$\eta_{\rm ex,cycle} = (\dot{W}_{\rm net\ out} + \dot{E}_{Q_{\rm H}})/\dot{E}_{\rm f}$
Steam exergy	$e_i = \left[(h_i - h_0) - T_0(s_i - s_0)\right]$	
Solid fuel exergy	$\dot{E} = \dot{m}\left[\mathrm{LHV}\left(1.0438 + 0.0013\dfrac{x_{\rm H}}{x_{\rm C}} + 0.1083\dfrac{x_{\rm O}}{x_{\rm C}} + 0.0549\dfrac{x_{\rm N}}{x_{\rm C}}\right) + 6740 x_{\rm S}\right]$ [kJkg^{-1}]	
Gaseous fuel exergy	$\bar{e}^{\rm CH} = \left[\bar{h}_{\rm F} + \left(a + \dfrac{b}{4}\right)\bar{h}_{\rm O_2} - a\bar{h}_{\rm CO_2} - \dfrac{b}{2}\bar{h}_{\rm H_2O(g)}\right](T_0, p_0) - $ $-T_0\left[\bar{s}_{\rm F} + \left(a + \dfrac{b}{4}\right)\bar{s}_{\rm O_2} - a\bar{s}_{\rm CO_2} - \dfrac{b}{2}\bar{s}_{\rm H_2O(g)}\right](T_0, p_0) + \bar{R}T_0 \ln\left[\dfrac{(x_{0,\rm O_2})^{a+\frac{b}{4}}}{(x_{0,\rm CO_2})^a (x_{0,\rm H_2O(g)})^{\frac{b}{2}}}\right]$	
Exergy of air and combustion products	$\dot{E} = \dot{m}\left[(h - h_0) - T_0(s - s_0) + \sum x_k' e_k^{\rm CH} + \bar{R}T_0 \sum x_k' \ln x_k'\right]$	

$$\eta_{\rm ex,cycle} = \frac{\dot{W}_{\rm net\ out} + \dot{E}_{Q_{\rm H}}}{\dot{E}_{\rm f}} \tag{23}$$

Exergetic efficiency acts as a basis for comparisons among cogeneration systems (of the same type) with different capacities of electricity and heat production, due to the fact that exergy quantifies the difference in the quality of the heat produced in different temperatures.

The other important exergetic performance criterion defined in this study is the amount of exergy loss rate per unit power and thermal output and it can be written [21]:

$$\xi = \frac{\dot{E}_{\rm D,\ cycle\ total}}{\dot{W}_{\rm net} + \dot{E}_{Q_{\rm H}}} \tag{24}$$

where $\dot{E}_{Q_{\rm H}}$ represents the exergy equivalence of heat, expressible [22]:

$$\dot{E}_{Q_{\rm H}} = \int \delta\dot{Q}_{\rm H}\left(1 - \frac{T_0}{T}\right) \tag{25}$$

where T is the temperature at which heat is transferred. This relation is of little practical value unless the functional relationship between the rate of heat transfer, $\dot{Q}$, and temperature T is known. In many cases, heat is utilized by transferring it from the working fluid exiting the heat producing device (*e. g.*, turbine, internal combustion engine) to a secondary fluid, in a heat exchanger. One can express the exergy rate of heating as the exergy increase in the cold fluid in the heater [22]:

$$\dot{E}_Q = \Delta\dot{E}_{x\ \text{cold}} = \dot{m}_{\text{cold}}(\Delta h - T_0 \Delta s)_{\text{cold}} \tag{26}$$

where Δh and Δs are the enthalpy and entropy changes of the cold fluid, respectively.

Table 5. Exergy destruction and exergy efficiency of components and systems

			Plant			
			Kostolac	Kolubara	TE-TO Novi Sad $\dot{Q}$ = 0 MW	TE-TO Novi Sad $\dot{Q}$ = 140 MW
SB	Exergy destruction	[MW]	521.8	196.2	204.7	227.3
	Exergy destruction	[%]	88.2	82.9	85.1	88.2
	Exergy efficiency	[%]	46.4	45.8	45.6	45.7
HTP	Exergy destruction	[MW]	10.3	7.7	8.13	7.8
	Exergy destruction	[%]	1.7	3.3	3.4	3.0
	Exergy efficiency	[%]	89.7	77.2	84.3	85.9
MTP	Exergy destruction	[MW]	11.5	8.7	13	13.5
	Exergy destruction	[%]	1.9	3.7	5.4	5.3
	Exergy efficiency	[%]	91.6	86.4	79.6	80.5
LTP	Exergy destruction	[MW]	34.9	12.1	9.5	3.8
	Exergy destruction	[%]	5.9	5.1	3.9	1.5
	Exergy efficiency	[%]	79.3	70.1	70.9	75.7
Condenser	Exergy destruction	[MW]	3.1	6.6	1.5	0.4
	Exergy destruction	[%]	0.5	2.8	0.61	0.2
	Exergy efficiency	[%]	57.8	51.7	82.3	89.1
Reheaters	Exergy destruction	[MW]	8.9	4.1	3.7	5
	Exergy destruction	[%]	1.5	1.7	1.6	1.9
	Exergy efficiency	[%]	–	–	–	–
Plant	Exergy destruction	[MW]	591.8	236.6	240.5	257.8
	Exergy destruction	[%]	100	100	100	100
	Exergy efficiency	[%]	35.8	30.3	31.1	36.3
	Energy efficiency	[%]	39.0	31.8	32.3	63.3
	Factorized efficiency	[%]	39.0	31.8	32.3	45.9
	Exergy loss rate per unit power and thermal output	–	1.7	2.2	2.0	1.6

Discussion

Operation parameters of the analyzed plants and some of their components are given in tab. 1 and are used for both energy and exergy analysis of plant operation. By using the mathematical model and software developed in VBA, thermodynamic states and exergies in different points of the plants were calculated, mainly at the component inputs and outputs, and then energy and exergy losses of the components were calculated. The equations given in tab. 2 were used to calculate the exergy efficiency of each component and the whole plant, tab. 5. Based on energy analysis, the obtained energy efficiency is in the range from 31.8% to 63.3%. It can be seen that the CHP plant possesses the highest value of energy efficiency. Energy analysis does not provide enough information needed for improvements in efficiency both for the whole plant and its components. This is why exergy analysis is performed based on the second law of thermodynamics, since it provides an insight into the reasons for thermodynamic inefficiency of components and systems.

From tab. 5, by comparing components and plant exergy losses, it is clear that the maximum exergy destruction occurs in the boiler and it is in the range from 82.9% to 88.2% out of total losses. Losses occur due to combustion irreversibility. Participation of other components in total exergy destruction is significantly lower, and in the case of the condenser which is the most critical component in energy analysis, it is in the range from 0.2% to 2.8%. Component exergy efficiency shows the places where further improvements can be made, which would lead to improving the efficiency of the whole plant.

Plant efficiency in both cases (energy and exergy analysis) shows that the plants with the possibility to transfer heat have better values compared to the plants used only for electricity production. At the same time, exergy loss rate per unit power and thermal output for this type of plant has the lowest value.

Conclusion

This paper presents the exergy analysis for the components of four different plants assuming the operation parameters defined in tab. 1. The biggest exergy loss in every plant was determined in the boiler, up to 88.2% of total exergy loss. A small part of exergy loss occurs in the condenser. The obtained results show that the most critical component is the boiler unlike the results from energy analysis where the condenser is considered to be most critical [23]. Exergy efficiency for the whole plant ranges from 30.3% to 36.3%. Compared with the results from similar studies, where exergy efficiency is in the range 36-37% [24, 25], it is evident that there is good agreement in the obtained results.

Nomenclature

E – exergy, [kJ]
$\dot{E}_D$ – exergy destruction, [kW]
e – specific exergy, [kJkg^{-1}]
g – gravity of the Earth, [ms^{-2}]
h – enthalpy, [kJkg^{-1}]
LHV – lower heatig value of fuel, [kJkg^{-1}]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
p – pressure, [bar]
$\dot{Q}$ – heat transfer rate, [kW]
R – universal gas constant, [Jmol^{-1}K^{-1}]
s – enthropy, [kJkg^{-1}K^{-1}]
T – temperature, [K]
u – internal energy, [kJkg^{-1}]
V – velocity, [ms^{-1}]
v – specific volume, [m^{3}kg^{-1}]
W – power, [kW]
x – part of component in mixture, [%]
z – elevation height difference, [m]

Superscripts

CH – chemical exergy
KN – kinetic exergy
PH – physical exergy
PT – potential exergy

Subscripts

e – electricity
ex – exergetic
f – fuel
H – heating
out – outlet
SEP – separate process
0 – reference state
in – inlet

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