EXERGY ANALYSIS OF THE FIGUEIRA THERMAL POWER PLANT OPERATION

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Abstract. *Exergy analysis is a powerful tool to evaluate, design and improve the thermal systems. The method of exergy analysis or availability analysis is well suited for furthering the goal of increasing the efficiency of existing power generation systems, and the capability of more effective energy resource use. Exergy analysis of the FIGUEIRA thermal power plant is presented. Exergy losses occurring in various components are considered and the exergy balance is shown in tabular form. Results clearly reveal that the steam generator is the principal site of thermodynamic losses, while the condenser is relatively unimportant.*

Keywords: *Exergy Analysis, Rankine Cycle, Thermodynamic Optimization*

1. INTRODUCTION

Built up on the Rio Laranjinha (or Fish River) riverside in the Figueira District, the state of Paraná in Brazil, the FIGUEIRA thermal power plant is a power generation system which now develops a maximum net electric power output of 20 MW. At the beginning, in 1963, it operated with two ALSTHON turbo-generators made in France. Since 1974, when a third SIEMENS turbo-generator was installed, until 1986, when a serious accident completely broken-down one of the ALSTHON turbo-generators, the FIGUEIRA power plant fully functioned reaching an one billion kWh production of electric energy in 1978.

Owned firstly by the FIGUEIRA Thermoelectric Company S/A (UTELFA), the Figueira thermal power plant has been incorporated in 1969 by the Paraná Energy State Company (COPEL). Twenty-eight years later, in 1997, based on a renting contract for ten years, the administration, operation and maintenance charges have been passed to the Cambuí Coal-Bearing Company in the Figueira District. As a part of the renting contract, the rent ower completely renovated the steam generator of the ALSTHON turbo-generator. Cambuí Coal-Bearing Company also invested US$ 2 million to install two ash filters, one for each of the two steam-generators. Thus, the dust emission into the atmosphere was reduced by 99.9 % of the total ash. The reform has been completed at the end of 1998.

The two coal-fired steam generators of the FIGUEIRA thermal power plant use the mineral coal mined by the Cambuí Coal-Bearing Company in the Paraná state, in Brazil.

We present in this paper an exergy analysis of the FIGUEIRA thermal power plant functioning based on the fundamental concepts of conservation of mass, conservation of energy and the second law of thermodynamics. Appropriate values of all variables (mass flow rates, pressures and temperatures) have been experimentally determined during a 6.7 MW electric power output functioning regime of the ALSTHON turbo-generator, and an 8.1 MW regime of the SIEMENS turbo-generator.

2. MATHEMATICAL MODEL

2.1 Thermodynamic cycle of the ALSTHON turbo-generator (cycle A)

To complete the exergy analysis of cycle A we consider first the following steps:

- setting up the mass balance

$$\dot{m}_{18} - \dot{m}_1^* - \dot{m}_5^* = 0 \tag{1a}$$

$$\dot{m}_1^* - \dot{m}_2^* - \dot{m}_3^* - \dot{m}_4^* = 0 \tag{1b}$$

$$\dot{m}_9^* - \dot{m}_5^* - \dot{m}_6^* = 0 \tag{1c}$$

$$\dot{m}_{16} - \dot{m}_2^* - \dot{m}_{15}^* = 0 \tag{1d}$$

$$\dot{m}_4^* + \dot{m}_8^* + \dot{m}_{10}^* - \dot{m}_6^* - \dot{m}_{11}^* = 0 \tag{1e}$$

$$\dot{m}_3^* = \dot{m}_7^* = \dot{m}_8^* \tag{1f}$$

$$\dot{m}_9^* = \dot{m}_{10}^* \tag{1g}$$

$$\dot{m}_{11}^* = \dot{m}_{12}^* = \dot{m}_{13}^* = \dot{m}_{14}^* = \dot{m}_{15}^* \tag{1h}$$

$$\dot{m}_{16} = \dot{m}_{17} = \dot{m}_{18} \tag{1i}$$

$$\dot{m}_{w,19}^* = \dot{m}_{w,20}^* = \dot{m}_{w,21}^* \tag{1j}$$

where the subscripts correspond to the principal states of the cycle A in Figs. 1a and 1b.

- setting up of the energy balance

$$\dot{W}_{el} - \dot{m}_1(h_1 - h_2) - (\dot{m}_1 - \dot{m}_2)(h_2 - h_3) - (\dot{m}_1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4^*) = 0 \tag{2a}$$

$$\dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_9 h_9^* = 0 \tag{2b}$$

$$\dot{m}_4 h_4^* + \dot{m}_3 h_8 + \dot{m}_9 h_{10} - \dot{m}_6 h_6 - \dot{m}_{11} h_{11} - \dot{m}_{w,19}(h_{20} - h_{19}) = 0 \tag{2c}$$

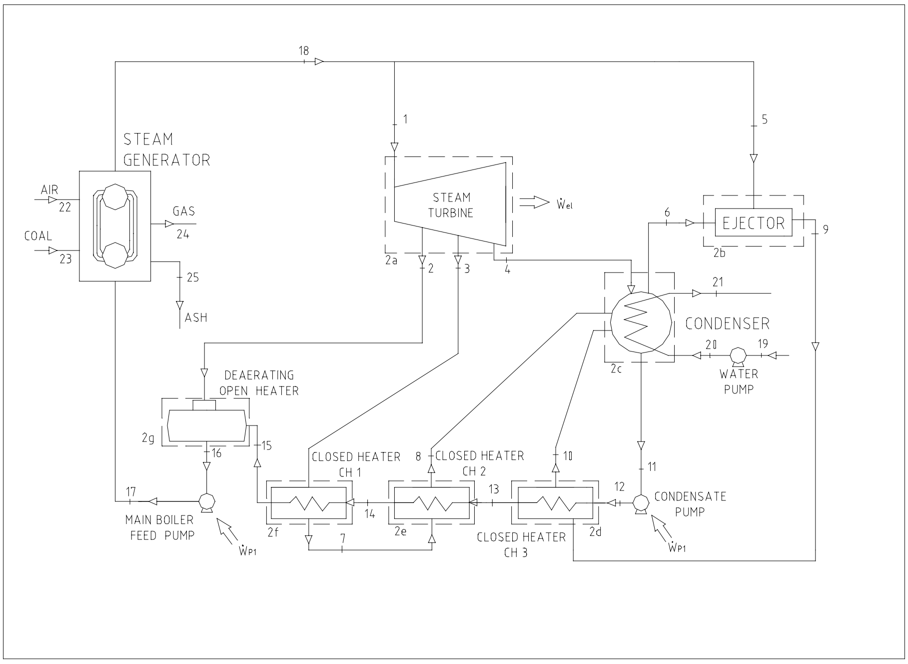
$$\dot{m}_{11}(h_{13}^* - h_{12}) - \dot{m}_9(h_9 - h_{10}) = 0 \tag{2d}$$

$$\dot{m}_{11}(h_{14}^* - h_{13}^*) - \dot{m}_3(h_7^* - h_8) = 0 \tag{2e}$$

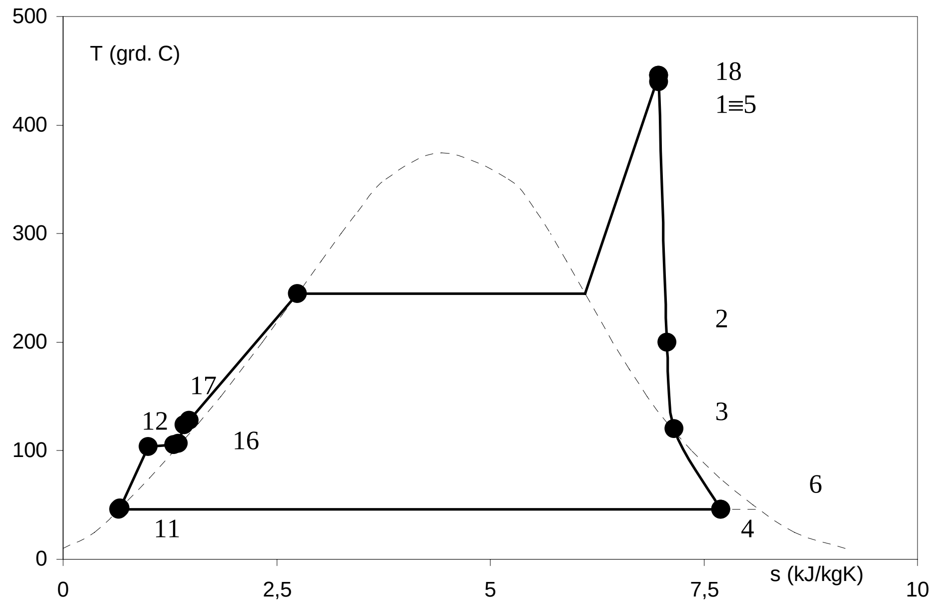
$$\dot{m}_{11}(h_{15}^* - h_{14}^*) - \dot{m}_3(h_3 - h_7^*) = 0 \tag{2f}$$

$$\dot{m}_2 h_2 + \dot{m}_{11} h_{15}^* - \dot{m}_{16} h_{16} = 0 \tag{2g}$$

The superscript * labels the unknown quantities. All component in the cycle, the steam generator, the steam turbine, the condenser, the three closed heaters CH1, CH2, CH3, the ejector, and the deaerating open heater are analyzed each one as a control volume at steady state, assuming the kinetic and potential energy effects to be negligible. The control volumes are shown in Fig. 1a by dashed lines.



a) - Layout of the ALSTHON turbo-generator



b) - T-s diagram of themodynamic cycle A (not in scale).

Figure 1 - Schematic view of the ALSTHON turbo-generator (a) and the temperature - specific entropy diagram of the regenerative vapor power cycle A (b).

The third step is to carry out the exergy analysis based on the numerical solution of the algebraic system in Eqs. (1a)-(1j) and Eqs. (2a)-(2g).

The standard environmental conditions used to perform the exergy analysis are T_0 = 298.15 K and P_0 = 101.325 kPa. On a unit-of-mass basis the expression of the total exergy transfer associated with a stream is given by (Bejan et al., 1996):

$$e = e^{PH} + ½\, V^2 + gZ + e^{CH} \tag{3}$$

where the physical component, e^{PH}, can be expressed as

$$e^{PH} = (h - h_0) - T_0\,(s - s_0) \tag{4}$$

and the chemical exergy component, e^{CH}, reads

$$e^{CH} = \sum x_k e_k^{CH} + RT_0 \sum x_k\, ln\, x_k \tag{5}$$

where x_k is the mole fraction of component k.

The second-law efficiency or the exergetic efficiency is a useful tool in order to measure the performance of a thermal power plant functioning regime. We focus our attention on the exergetic efficiency of the steam generator, steam turbine and condenser. As suggested by Bejan et al., 1996, in a guidelines for evaluating and improving thermodynamic effectiveness, irreversibilities related to combustion and heat transfer are generally primarily in importance. To improve the use of energy sources, it is also important to minimize the losses generated by thermodynamic irreversibility in the lower temperature stages of steam turbines. Thermodynamic losses associated with the heat transfer have a great contribution to inefficiency, especially at lower temperature levels. Therefore, we also investigate the exergetic losses into the condenser.

2.2 Thermodynamic cycle of the SIEMENS turbo-generator (cycle B)

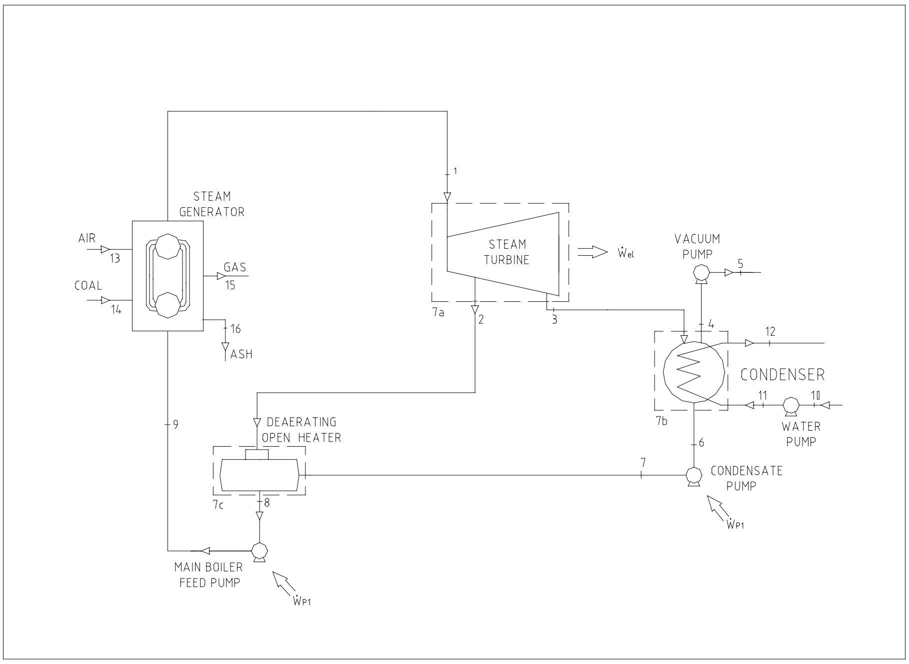


Figure 2 - Schematic view of the SIEMENS turbo-generator.

To complete the exergy analysis of cycle B we consider first the following steps:

- setting up the mass balance

$$\dot{m}_1 - \dot{m}_2^* - \dot{m}_3^* = 0 \tag{6a}$$

$$\dot{m}_1 = \dot{m}_8 = \dot{m}_9 \quad (6b)$$
$$\dot{m}_3^* = \dot{m}_6^* = \dot{m}_7^* \quad (6c)$$
$$\dot{m}_4^* = \dot{m}_5^* \cong 0 \quad (6d)$$
$$\dot{m}_{10}^* = \dot{m}_{11}^* = \dot{m}_{12}^* \quad (6e)$$

- setting up of the energy balance

$$\dot{W}_{el} - \dot{m}_1(h_1 - h_2^*) - (\dot{m}_1 - \dot{m}_2)(h_2^* - h_3^*) = 0 \quad (7a)$$
$$\dot{m}_3(h_3^* - h_6) - \dot{m}_{10}^*(h_{12} - h_{11}) = 0 \quad (7b)$$
$$\dot{m}_2 h_2^* + \dot{m}_3 h_7 - \dot{m}_1 h_8 = 0 \quad (7c)$$

3. EXERGY ANALYSIS

3.1 Chemical standard exergy of the mineral coal mined from the coal basin owned by the Cambuí Coal-Bearing Company

The procedure for determining the chemical exergy of the mineral coal burnt into the steam generators of the FIGUEIRA thermal power plant is those presented by Bejan et.al., 1996. The chemical composition of the coal is given in Table 1.

Table 1. Chemical composition of the mineral coal.

Constituent	As received	Dry and ash free (DAF)	
	Mass fraction (%)	Mass fraction (%)	kmol kg^{-1} (DAF)
C	53.99	79.82	0.0665
H	3.68	5.44	0.0544
O	5.83	8.62	0.0054
N	1.19	1.76	0.0013
S	2.95	4.36	0.0014
Ash	23.26		
H_2O	9.10		

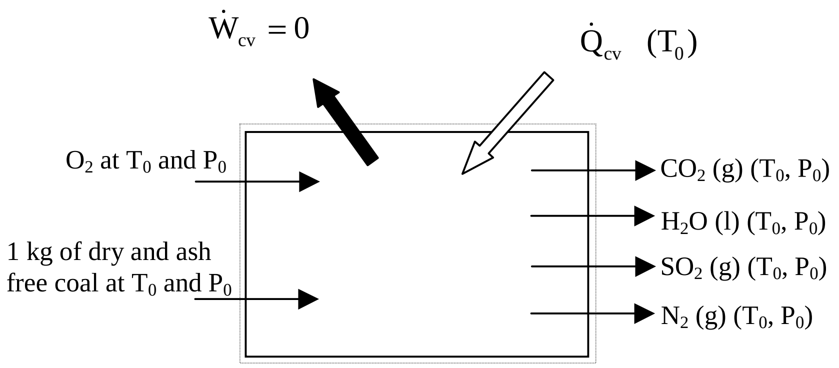


Figure 3 - Schematic view of the control volume used to evaluate the chemical exergy.

It is assumed that 1 kg of dry and ash free coal enters, at T_0 and P_0, the control volume in Fig. 3. During the combustion reaction all combustible elements completely react with the oxygen, that enters the control volume at T_0 and P_0. The combustible elements in the mineral coal react with the oxygen to form carbon dioxide, liquid water, and sulfur dioxide. It is assumed that all substances enter unmixed at T_0 and P_0, and exit the control volume also unmixed at T_0 and P_0. The only heat transfer interaction occurs at the surroundings temperature T_0.

The chemical reaction into the control volume is described by

$$(cC + hH + oO + sS) + \nu_{O2}\, O_2 \rightarrow \nu_{O2}\, CO_2 + \nu_{H2O}\, H_2O\,(l) + \nu_{SO2}\, SO_2 + \nu_{N2}\, N_2 \qquad (8)$$

where c, h, o, n, s in kmol/kg are indicated in Table 1, and $\nu_{CO2} = c$, $\nu_{H2O} = ½\, h$, $\nu_{SO2} = s$, $\nu_{N2} = ½\, n$, $\nu_{O2} = c + ¼\, h + s - ½\, o$.

Chemical exergy of the mineral coal reads now

$$\begin{aligned} e_{DAF}^{CH} = (HHV)_{DAF} - T_0(s_{DAF} + \nu_{O2}\bar{s}_{O2} - \nu_{CO2}\bar{s}_{CO2} - \nu_{H2O}\bar{s}_{H2O} - \nu_{SO2}\bar{s}_{SO2}) \\ + (\nu_{CO2}\bar{e}_{H2O}^{CH} + \nu_{H2O}\bar{e}_{H2O}^{CH} + \nu_{SO2}\bar{e}_{SO2}^{CH} + \nu_{N2}\bar{e}_{N2}^{CH} - \nu_{O2}\bar{e}_{O2}^{CH}) \end{aligned} \qquad (9)$$

Numerical values of stoichiometric coefficients in Eq. (3) and the specific entropy and standard molar chemical exergy (at T_0 = 298.15 K and P_0 = 101.325 kPa) in Eq. (4) are provided in Table 2 (see Bejan et. al., 1996).

Table 2. Stoichiometric coefficients in Eq. (3) and standard molar chemical exergy at T_0 = 298.15 K and P_0 = 101.325 kPa.

Substance	ν	$\bar{s}_0$	$\bar{e}^{CH}$
	kmol kg^{-1}	kJ kmol^{-1} K^{-1}	kmol kg^{-1}
O_2 (g)	0.0788	205.15	3951
CO_2 (g)	0.0665	213.79	14176
H_2O (l)	0.0272	69.95	45
SO_2 (g)	0.0014	248.09	301939
N_2 (g)	0.0007	191.61	639

Numerical values of the higher heating value $(HHV)_{DAF}$ of a dry and ash free mineral coal of chemical composition as indicated in Table 2, and the absolute entropy s_{DAF} are theoretically calculated as follows (Eiserman et al., 1980):

$$(HHV)_{DAF} = (152{,}19H + 98{,}767)\left[\frac{C}{3} + H - \frac{(O-S)}{8}\right] = 33{,}736\, MJ/kg(DAF) \qquad (10)$$

where H, C, O, S are the mass fractions in the third column in Table 2, and

$$s_{DAF} = c\,[37{,}17 - 31{,}48\, exp(-\frac{0{,}565\, h}{c+n}) + \frac{20{,}1\, o + 54{,}3\, n + 44{,}7\, s}{c+n}] = 1{,}378\, kJ/(kg\, K) \qquad (11)$$

Equation (9) gives the specific chemical exergy of the dry and ash free mineral coal $e_{DAF}^{CH} = 26.27\ \mathrm{MJ/kg(DAF)}$, while the specific chemical exergy of the mineral coal as received is calculated with the formula

$$e^{CH} = 0{,}6764\, e^{CH}_{DAF} + \frac{0{,}091}{18{,}015}\bar{e}^{CH}_{H_2O(l)} = 18{,}00\ MJ/kg\ (DAF) \tag{12}$$

where the higher heating value has been approximated by the measured value $(HHV)_{exp}$ = 25.50 MJ/kg (DAF) and the chemical exergy of the ash has been ignored.

Table 3. Thermodynamic properties of the working fluid (Cycle A).

State	$\dot{m}$	P	T	H	s	e^{PH}
	$kg \cdot s^{-1}$	MPa	^0C	$kJ \cdot kg^{-1}$	$kJ \cdot kg^{-1} \cdot K^{-1}$	$kJ \cdot kg^{-1}$
0	-	0.10	25.0	105.0	0.367	0.0
1	9.127	3.53	440.0	3314.0	6.969	1241.7
2	0.280	0.49	200.0	2856.0	7.069	753.9
3	0.040	0.19	120.0	2707.0	7.152	580.2
4	8.807	0.01	46.0	2571.0	8.106	159.5
5	0.040	3.53	440.0	3314.0	6.969	1241.7
6	0.880	0.01	46.0	2585.0	8.150	160.8
7	0.040	0.15	111.0	1587.0	4.346	296.4
8	0.040	0.12	105.0	467.1	1.434	44.2
9	0.920	0.50	152.0	2617.0	6.511	680.9
10	0.920	0.20	76.0	317.6	1.026	16.5
11	8.887	0.01	46.0	191.8	0.649	2.8
12	8.887	0.78	47.0	199.2	0.670	4.1
13	8.887	0.68	104.0	437.2	1.354	38.3
14	8.887	0.55	105.5	442.3	1.368	39.3
15	8.887	0.52	107.0	447.3	1.381	40.3
16	9.167	0.49	124.0	520.9	1.570	57.4
17	9.167	3.63	128.0	540.1	1.610	64.8
18	9.167	3.63	446.0	3326.0	6.974	1252.2
19	477.600	0.10	22.0	92.4	0.325	0.05
20	477.600	0.15	22.5	94.5	0.332	0.07
21	477.600	0.15	32.0	134.3	0.464	0.40

3.2 Exergy analysis of cycle A

Based on the experimental data in Table 3, measured during a 6.7 MW electric power output regime, we calculate the in- and outgoing exergy rates for all components of the ALSTHON turbo-generator. Numerical results for the steam generator, steam turbine and the condenser are shown in Tables 4, 5 and 6. Table 7 presents the overall results of the exergetic analysis of cycle A.

Exergetic efficiencies in Table 7 are calculated as follows:

$$\varepsilon_{steam\ generator} = \frac{\dot{E}_{18} - \dot{E}_{17}}{(\dot{E}_{22} + \dot{E}_{23}) - (\dot{E}_{24} + \dot{E}_{25})} \tag{13}$$

$$\varepsilon_{turbo-generator} = \frac{\dot{W}_{el}}{\dot{E}_1 - \dot{E}_2 - \dot{E}_3 - \dot{E}_4} \tag{14}$$

Table 4. The in- and outgoing flows of the steam generator of the ALSTHON turbo-generator.

Components	P	T	Flow*	Exergy		
	bar	^{0}C	kg s^{-1}	$\dot{E}^{PH}$ (kW)	$\dot{E}^{CH}$ (kW)	$\dot{E}$ (kW)
Air in	1.2	26	14.44	225.32	30.17	255.49
Water in (l)	36.3	128	9.17	553.23	22.93	576.16
Steam out	35.3	446	-9.17	-11454.05	-4399.56	-15853.61
Coal	1.0	25	1.94	0.00	34920.00	34920.00
Combustion products						
CO_2	0.164	183	-3.85	-272.73	-203.43	-476.16
H_2O (g)	0.086	183	-0.82	-161.64	-33.83	-195.47
SO_2	0.003	183	-0.11	-5.12	-2.08	-7.20
N_2	0.744	183	-11.10	-70.86	-188.47	-259.33
O_2	0.003	183	-0.05	-21.04	-0.01	-21.05
Total						18938.83

* The minus sign means that the stream is leaving the control volume.

Table 5. The in- and outgoing flows of the steam turbine of the ALSTHON turbo-generator.

Components	P	T	Flow	Exergy		
	bar	^{0}C	kg s^{-1}	$\dot{E}^{PH}$ (kW)	$\dot{E}^{CH}$ (kW)	$\dot{E}$ (kW)
Steam in	35.3	440	9.13	11323.08	4378.93	15702.01
Steam out						
State 2	4.9	200	-0.28	-210.78	-134.34	-345.12
State 3	1.9	120	-0.04	-23.16	-19.19	-42.35
State 4	0.1	46	-8.81	-1396.94	-4225.40	-5622.34
Net power output of the ALSTHON turbo-generator						-6700.00
Total						2992.19

Table 6. The in- and outgoing flows of the condenser of the ALSTHON turbo-generator.

Components	P	T	Flow	Exergy		
	bar	^{0}C	kg s^{-1}	$\dot{E}^{PH}$ (kW)	$\dot{E}^{CH}$ (kW)	$\dot{E}$ (kW)
Steam in						
State 4	0.1	46	8.807	1396.97	4225.42	5622.39
State 8	1.2	105	0.040	1.76	0.10	1.86
State 10	2.0	76	0.920	14.83	2.30	17.13
Water in state 20	1.5	22.5	477.600	33.43	1194.00	1227.43
Steam out state 6	0.1	46	-0.880	-140.36	-422.21	-562.57
Water out						
State 11	0.1	46	-8.887	-24.19	-22.22	-46.41
State 21	1.5	32	-477.600	-181.23	-1194.00	-1375.23
Total						4884.60

Difference between 100 % and the actual total value in Table 7 is the percentage of the mineral coal exergy lost in the other components of the power plant and the exergy wasted in the system.

Table 7. Results of the exergetic analysis of cycle A.

Unit	Exergy losses		Exergetic efficiency, ε
	kW	%*	%
Steam generator	18938.83	54.2	44.6
Steam turbine	2992.19	8.5	69.1
Condenser	4884.60	14.0	
Total		76.7	
Net power output	6700.00	19.2	
Total		95.9	

* Values expressed as a percentage of the exergy carried into the plant by the mineral coal.

Exergetic overall efficiency of cycle A is given by

$$\varepsilon_{cycle\ A} = \frac{\dot{W}_{el}}{(\dot{E}_{20} + \dot{E}_{22} + \dot{E}_{23}) - (\dot{E}_{21} + \dot{E}_{24} + \dot{E}_{25})} = 19.7\ \% \tag{15}$$

3.3 Exergy analysis of cycle B

Exergetic efficiencies in Table 9 are calculated as follows

$$\varepsilon_{steam\ generator} = \frac{\dot{E}_1 - \dot{E}_9}{(\dot{E}_{13} + \dot{E}_{14}) - (\dot{E}_{15} + \dot{E}_{16})} \tag{16}$$

$$\varepsilon_{turbo-generator} = \frac{\dot{W}_{el}}{\dot{E}_1 - \dot{E}_2 - \dot{E}_3} \tag{17}$$

Table 8. Thermodynamic properties of the working fluid (Cycle B).

State	ṁ	P	T	h	s	e^{PH}
	$kg \cdot s^{-1}$	MPa	^0C	$kJ \cdot kg^{-1}$	$kJ \cdot kg^{-1} \cdot K^{-1}$	$kJ \cdot kg^{-1}$
0	-	0.10	25.0	105	0.367	0.00
1	10.00	3.63	426.0	3280	6.909	1224.50
2	1.07	0.17	138.4	2747.3	7.302	574.63
3	8.93	0.01	46.0	2436.8	7.687	149.34
6	8.93	0.01	46.0	191.81	0.649	2.67
7	8.93	0.17	47.0	196.91	0.665	3.15
8	10.00	0.17	112.0	469.8	1.440	44.77
9	10.00	3.63	133.0	561.3	1.663	69.99
10	503.70	0.10	21.0	88.2	0.311	-0.02
11	503.70	0.15	21.5	90.4	0.318	0.00
12	503.70	0.15	36.0	151.0	0.519	0.74

Exergetic overall efficiency of cycle B is given by

$$\varepsilon_{cycle\,B} = \frac{\dot{W}_{el}}{(\dot{E}_{11} + \dot{E}_{13} + \dot{E}_{14}) - (\dot{E}_{12} + \dot{E}_{15} + \dot{E}_{16})} = 22.7\,\% \tag{18}$$

Table 9. Results of the exergetic analysis of cycle B.

Unit	Exergy losses		Exergetic efficiency, ε
	kW	%	%
Steam generator	19461.47	53.3	45.6
Steam turbine	2194.93	6.0	78.7
Condenser	5405.66	14.8	
Total		74.1	
Net power output	8100.00	22.2	
Total		96.3	

4. CONCLUSIONS

The mathematical model presented in this paper shows that a useful picture of the FIGUEIRA thermal power plant performance can be obtained based on the fundamental concepts of conservation of mass, conservation of energy and the second law of thermodynamics.

Irreversibilities within the steam generators are responsible for the destruction of the largest part of the exergy entering the power plant with the coal. Numerical results in Tables 7 and 9 show that more than 50 % of the fuel exergy is destroyed within the steam generators of the ALSTHON and SIEMENS turbo-generator groups.

An accurate calculation of the exergy exiting the stem generators with the unburned fuel is suggested by the large difference between numerical values of the calculated higher heating value $(HHV)_{DAF}$ = 33.736 MJ/kg(DAF) of the mineral coal, and the measured value of the higher heating $(HHV)_{exp}$ = 25.50 MJ/kg (DAF).

Results of the exergetic analysis of the FIGUEIRA thermal power plant functioning presented in here also suggest a careful investigation of feedwater heaters operation for the two turbo-generators groups.

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