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DETERMINATION OF EXERGETIC AND ECONOMIC PARAMETERS OF A NATURAL GAS COMBINED CYCLE THERMOELECTRIC POWER PLANT WITHOUT/WITH SUPPLEMENTARY FUEL BURNING

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Abstract. *The present article a natural gas combined cycle thermoelectric power plant, without and with supplementary fuel burning (SFB), was evaluated through an exergetic and economic analysis. The objective of this research is to determine and compare the exergetic parameters such as the exergetic efficiency and the destruction of exergy associated with the irreversibilities in each equipment of both systems. Finally, the economic analysis determined the expected annual benefit due to the pay-back and the electricity production cost (PCEE), incorporating the effects of the environmental impacts. When developing the present study in the combined cycle without/with SFB it was evidenced a global efficiency of 50.81% vs. 47.39%, respectively. In addition, the system without SFB showed better results of exergetic efficiency and irreversibility. The economic analysis verified the annual benefit due to the pay-back for an annual interest rate of 4%, 8% and 12%, and the PCEE showed an increase between 10% to 12% in the case of the system with SFB.*

Keywords: *combined cycle, economic parameters, exergetic efficiency, natural gas, supplementary fuel burning*

1. INTRODUCTION

The growing commitment to the environment and the increase in the electricity market drive the development of new generation techniques that help reduce greenhouse gas emissions and also that imply lower investment costs (Barros, 2013). One of the causes to be highlighted is industrial development such as energy consumption in the steel industry (Mariños *et al.*, 2020) and the refrigeration industry (Mariños *et al.*, 2019a) lead to an increase in energy demand; therefore, the solution to this problem is to design more efficient and environmentally friendly power plants. To deal with this problem, the combined cycle power generation technology is presented, which is one of the most efficient, with less environmental impact and greater responsiveness.

Many works have been developed to reduce energy consumption in combined cycle thermoelectric plants. The aspects as energy saving, reduction of pollutants and cost reduction were addressed in works that carry out an energetic, exergetic, environmental and economic analysis. Several thermoeconomic optimization methods have been applied to the optimization of energy systems since 1980. Like Valero's exergetic cost theory (Valero, 1994), Frangopoulos' functional method (Frangopoulos, 1996), Spakovsky's engineering functional analysis (Spakovsky, 1994), and Tsatsaronis exergoeconomic method (Tsatsaronis, 1993), among others.

To carry out the objective, the following articles published by the author as Mariños *et al.*, (2019b) and Mariños *et al.*, (2021) were used as reference. In these articles the authors applied the first and second laws of thermodynamics to analyze thermoelectric plants with an installed capacity of 60 MW, operating in combined cycle with gas turbine associated to a HRSG, without/with SFB, and considering a condensation steam turbine. For both analyses, the performance of the thermoelectric power plants was investigated, determining the energetic efficiencies and environmental parameters. To the authors knowledge there is no work in the available literature that compares the combined cycle thermoelectric power plant without/with supplementary fuel burning (SFB) and that shows an analysis of the effect of the ecological efficiency and the environmental impacts on the cost of electricity production.

Finally, the economic analysis focuses on the electricity production cost for the natural gas combined cycles. The methodology presented incorporates the concepts of exergy, economic engineering and ecological efficiency. The model uses functional diagrams in a way that allows the formulation and minimization of the exergetic manufacturing cost, presents the cost of electricity production and incorporates the effects of environmental impacts.

2. METHODOLOGY

2.1 Parameter for selected gas turbines

The parameters of the selected gas turbines must be corrected according to the local conditions of the plant installation, because the ambient temperature, the altitude and the relative humidity influence the performance of the turbine. Thus, the following conditions were adopted (Antunes, 1999): Ambient temperature of 27 °C, altitude of 900 m (local conditions of the plant), and relative humidity 60 %. Estimated data for the gas turbine selection was shown in the work of Mariños *et al.*, (2019b) and Mariños *et al.*, (2021).

2.2 Combined cycle without/with supplementary fuel burning

The scheme of thermoelectric power plant with supplementary fuel burning (with SFB) is shown in Fig. 1. The scheme presents seven components. In atmospheric conditions, air enters the compressor and is compressed to the combustion pressure. Subsequently, it is sent to the combustion chamber, where the fuel is burned continuously under practically constant pressure conditions and the combustion gases expand in the gas turbine producing electricity. Then, the exhaust gases are directed to the HRSG, where superheated steam is produced and sent to the steam turbine, producing more electrical energy (Silveira *et al.*, 2007).

When supplementary burning is used, a large amount of excess air is also used to moderate the flame temperature in the gas turbine combustion. The additional fuel is supplied to the HRSG on supplementary burners and is burned using excess oxygen levels in the exhaust gases to increase steam production rates. This technique is used when there are high peaks in demand with high electricity prices (González *et al.*, 2014). In the case of a thermoelectric power plant without supplementary fuel burning (without SFB), points 13 and 14 of Fig. 1 are not taken into account.

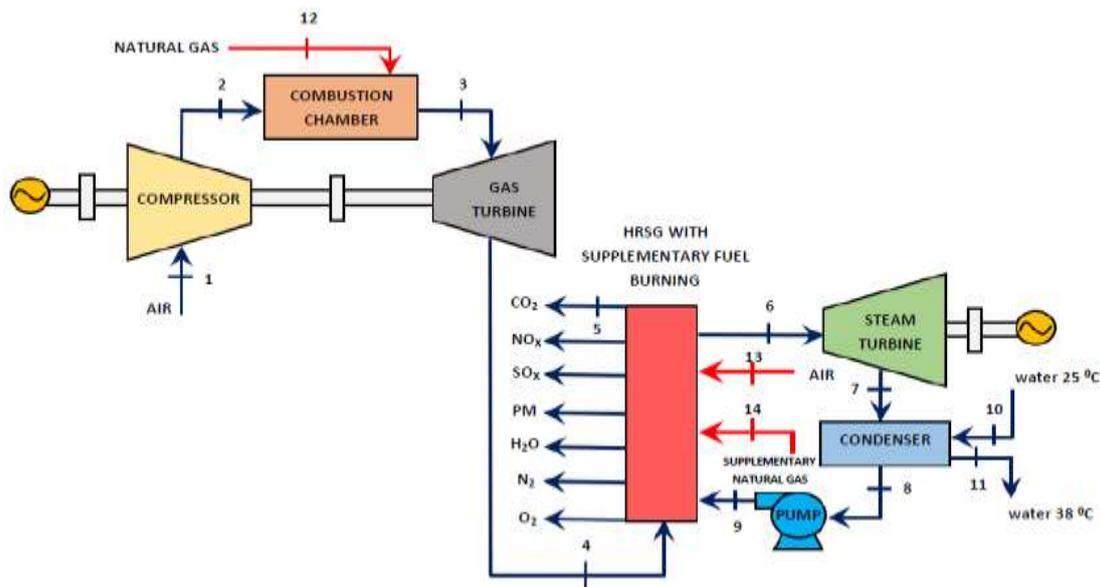


Figure 1. Combined cycle thermoelectric plant with supplementary fuel burning (Mariños *et al.*, 2019b).

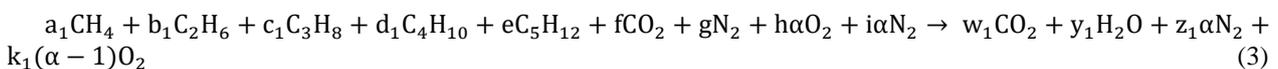
2.3 Second law of thermodynamics applied to the cycle: Exergetic analysis

Exergetic analysis has developed rapidly in recent years and consists of a qualitative assessment of losses through the concept of exergy by applying the second law of thermodynamics. The exergy at each point of the systems is determined with Eq. (1) for steam/water and Eq. (2) for air and combustion gases which are considered perfect gases.

$$Ex_i = (h_i - h_0) - T_0 (S_i - S_0) \quad (i = 6,7,8,9,10,11) \quad (1)$$

$$Ex_i = C_{p,ar,G} \left[(T - h_0) - T_0 \ln \left(\frac{T}{T_0} \right) + R_{ar,G} T_0 \ln \left(\frac{P}{P_0} \right) \right] \quad (i = 1,2,3,4,5) \quad (2)$$

The composition of the combustion gases is obtained using Eq. (3).



Combustion is assumed complete and the NG is composed of 89.3% CH₄, 8% C₂H₆, 0.8% C₃H₈, 0.05% C₄H₁₀, 0.05% C₅H₁₂, 0.5% CO₂ and 1.3% N₂.

Specific heats at constant pressure for air and for the combustion gases, for the system without and with SFB are determined using the mass percentages of each reaction component according to Eq. (4) and the parameters of Table 1.

$$Cp_j = a_0 + a_1T + a_2T^2 + a_3T^3 + a_4T^4 \quad (4)$$

Table 1. Parameters for specific heat of air and gases for the system without (*) and with (**) SFB.

j	a ₀	a ₁	a ₂	a ₃	a ₄
Air	1.040841	-0.000383719	9.45378x10 ⁻⁷	-5.49031x10 ⁻¹⁰	7.92981x10 ⁻¹⁴
NG*	1.030004086	0.534942x10 ⁻⁶	3.38630626x10 ⁻⁷	-1.41686667x10 ⁻¹¹	0
NG**	1.040107893	0.2018666262x10 ⁻⁴	3.304738433x10 ⁻⁷	-1.39016938x10 ⁻¹¹	0

The exergetic efficiency incorporates a notion of the unavailable thermodynamic increase, reflected by the increase in entropy in the system or process. The Eq. (5) and the indices of Table 2 determine the exergetic efficiency of each individual component of the system. In Fig. 2 the balance for each component of the cycle is presented.

$$\zeta_C = \frac{a Ex_w - b Ex_x + c W_{GT} + d W_{ST}}{e Ex_y - f Ex_z + g W_C + h W_P + i \dot{m}_f LHV} \quad (5)$$

Table 2. Indices to determine the exergetic efficiency in each equipment of the system without (*) and with (**) SFB.

Equipment	w	x	y	z	a	b	c	d	e	f	g	h	i
Compressor	2	1	-	-	1	1	0	0	0	0	1	0	0
Combustion chamber	3	-	2	12	1	0	0	0	1	-1	0	0	0
Gas turbine	-	-	3	4	0	0	1	0	1	1	0	0	0
HRSG *	6	9	4	-	1	1	0	0	1	0	0	0	0
HRSG **	6	9	4	-	1	1	0	0	1	0	0	0	1
Steam turbine	-	-	6	7	0	0	0	1	1	1	0	0	0
Condenser	11	10	7	8	1	1	0	0	1	1	0	0	0
Pump	9	8	-	-	1	1	0	0	0	0	0	1	0

The exergetic inefficiency of a system consists in the destruction of exergy associated with irreversibilities. According to Valero *et al.* (1994), in a system, irreversibility can be decomposed into internal irreversibility, known by the second law of thermodynamics, as energy destruction and external irreversibility.

With irreversibilities it is possible to quantify whether a process is reversible or not, in other words, it is a way of comparing the work that the process exchanges with the reversible work associated between the same initial and final states. Irreversibility is determined by Eq. (6) and the total irreversibility by Eq. (7).

$$I = \sum Ex_{in} - \sum Ex_{ou} \quad (6)$$

$$I_t = \sum I = I_C + I_{CC} + I_{GT} + I_{RB} + I_{ST} + I_{CO} + I_B \quad (7)$$

In Eqs. (8) - (14) the irreversibilities for each component are presented according to the balance in Fig. 2.

The atmospheric air, captured by the compressor, is compressed to a pressure of about 13 bar, and a temperature of the order of 375 °C, which rises to 1250 °C after the combustion process. The Eq. (8) determines irreversibilities in the compressor.

$$I_C = Ex_1 + W_C - Ex_2 \quad (8)$$

The combustion chamber is the place where NG is mixed with pressurized air, and the combustion process is carried out. The NG is injected into the chamber together with the combustion air that was previously filtered and compressed in the GT's internal compressor.

$$I_{CC} = Ex_2 + E_f - Ex_3 \quad (9)$$

In the gas turbine, the inlet gases can present temperatures up to 1400 °C, while the outlet gases have temperatures of the order of 600 °C. When the energy of the flue gases expands, the main axis of the GT coupled to the generator rotates, transforming mechanical energy into electrical energy.

$$I_{GT} = Ex_3 - Ex_4 - W_{GT} \quad (10)$$

HRSG is the element that harnesses the energy of the GT exhaust gases, using it to generate steam. The post-combustion boiler is the most common type of boiler used in combined cycles. The Eq. (11) determines irreversibilities in the recovery boiler, where, $\alpha = 0$ for the system without SFB and $\alpha = 1$ for the system with SFB.

$$I_{RB} = Ex_4 + Ex_9 + 0.3 \alpha E_f - Ex_5 - Ex_6 \quad (11)$$

Steam turbine expands the high-pressure steam generated in the HRSG to the condensing pressure, transforming the steam energy into kinetic energy of the rotor. This steam expanded in the turbine, coupled to an alternator, constitutes the second stage of electrical generation.

$$I_{ST} = Ex_6 - Ex_7 - W_{ST} \quad (12)$$

Condenser is the element that captures the expanded low-pressure steam from the ST outlet. The energy of the steam that is not usable in the form of mechanical energy in the expansion process is transferred in part to the cold condenser focus, where the water steam is transformed into liquid water.

$$I_{CO} = Ex_7 + Ex_{10} - Ex_8 - Ex_{11} \quad (13)$$

At the condenser outlet, one or two pumping stages are responsible for raising the water pressure from the condensate well to the pressure of the HRSG, where heating, water vaporization and overheating of the steam occurs at constant pressure, thus closing the cycle. The irreversibilities in the pump are obtained by Eq. (14).

$$I_p = Ex_8 + W_p - Ex_9 \quad (14)$$

2.4 Economic analysis of the combined cycle

The thermoeconomic model proposed is based on the concepts of exergy and functional diagrams. This model addresses economic aspects, introducing environmental impacts through the ecological efficiency parameter. The main advantage of this model is that it does not need any complicated computational tool, as it is an algebraic model that is easy to implement and interpret (Silveira *et al.*, 2012).

The model includes emissions in the chimney of the HRSG. It should be noted that, for an adequate monitoring of the development of this thermoeconomic model, each unit will receive an identification letter, that will allow to visualize the transposition between the physical diagram and the corresponding functional diagram. The thermoeconomic diagram is represented in Fig. 3 for the systems with SFB. For the system without SFB, the supplemental fuel and air are not considered.

The notation adopted is $Y_{i,j}$ (j-th input of the i-th component) and $Y_{i,k}$ (k-th output of the i-th component). Note that a comma ($Y_{i,j}$) is used to identify the input streams and a point ($Y_{i,k}$) for the output streams. However, for the system as a whole the notation is opposite, that is, $Y_{O,k}$ represents input and $Y_{O,j}$ represents output, indicating respectively the inputs and products of the thermal plant (0 - system) to the environment. The exergetic functions of each unit are determined by neglecting losses through the pipes, it is emphasized that this does not invalidate the proposed model.

2.5 Thermoeconomic optimization function

The thermoeconomic optimization method is a powerful technique for the design of thermal systems associated with economic factors and details of thermal plants. In this case, the thermoeconomic optimization function aims to minimize the electricity production cost (PC_{EE}) that also incorporates the concept of ecological efficiency. In order to determine it an adequate mathematical model is necessary.

Based on the physical and functional diagrams, the PC_{EE} can be determined by Eqs. (15) and (16), considering the systems without/with SFB, respectively. These equations follow the method suggested by Silveira *et al.* (2012).

$$PC_{EE} = \frac{f(TI)}{OH(Y_{O,1}+Y_{O,9})\epsilon} + \frac{Y_{O,2}(FC)}{(Y_{O,1}+Y_{O,9})} + OC + \frac{MC_{GT}+MC_{ST}+MC_{RB}}{\epsilon} \quad (15)$$

$$PC_{EE} = \frac{f(TI)}{OH(Y_{O,1}+Y_{O,9})\epsilon} + \frac{Y_{O,2}(Y_{O,2}+Y_{O,5})FC}{(Y_{O,1}+Y_{O,9})} + OC + \frac{MC_{GT}+MC_{ST}+MC_{RB}}{\epsilon} \quad (16)$$

Where, the PC_{EE} is associated with exergetic functions ($ge_1 Y_{O,2} Y_{O,5}$ and $Y_{O,9}$), ecological efficiency (ϵ), maintenance cost (MC), number of hours of operation per year (OH) and fuel cost (FC). The annuity factor (f) is represented by Eqs. (17) and (18). The operating personnel cost (OC) will be evaluated by Eq. (19).

$$f = \frac{q^{k \cdot (q-1)}}{q^{k-1}} \quad (17)$$

$$q = 1 + \frac{r}{100} \quad (18)$$

$$OC = \frac{156000}{7200 E_{p_{total}}} \quad (19)$$

$$TI = 1.4 (\ln_C + \ln_{CC} + \ln_{GT} + \ln_{RB} + \ln_{ST} + \ln_{CO} + \ln_P) \quad (20)$$

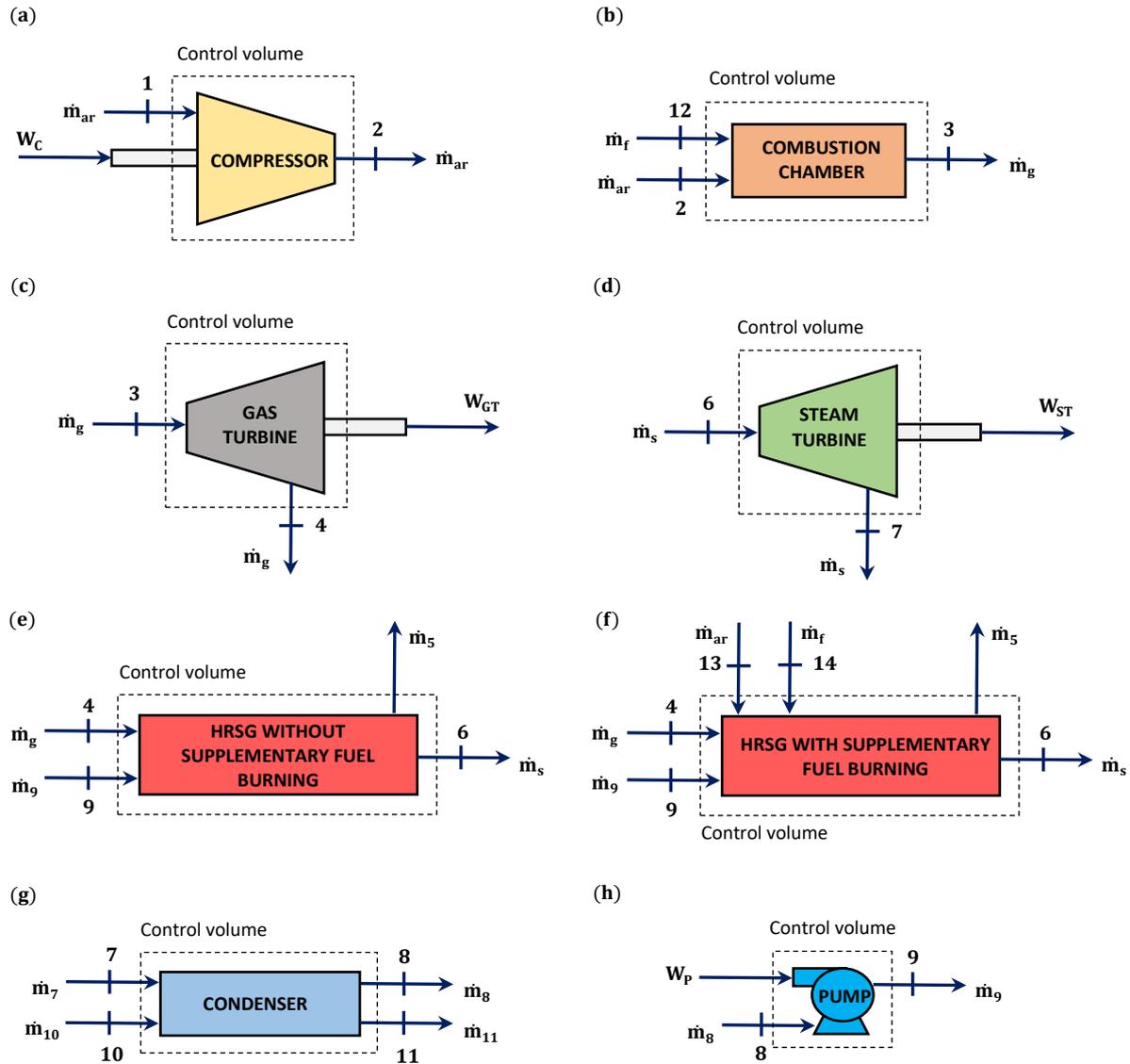


Figure 2. Components of the combined cycle without/with supplementary fuel burning.

The Eq. (20) determines the total investment in the plant (TI), for that it is necessary to consider the associated costs for each plant component (\ln_j). These costs are presented by Eqs. (21) - (27), but some coefficients were adapted, taking into account the installations, electrical equipment, system control, piping and installation location. It was considered that 40% of the investment corresponds to equipment (Mancebo *et al.*, 2016).

$$\ln_C = \left[\frac{75 \dot{m}_{ar}}{0.9 - \eta_C} \right] \left[\frac{P_2}{P_1} \right] \ln \left(\frac{P_2}{P_1} \right) \quad (21)$$

$$\ln_{CC} = \left[\frac{48.64 \dot{m}_{ar}}{0.995 \frac{P_4}{P_3}} \right] \left[1 + e^{(0.018 T_4 - 26.4)} \right] \quad (22)$$

$$\ln_{GT} = \left[\frac{1536 \dot{m}_g}{0.92 - \eta_{GT}} \right] \ln \left(\frac{P_3}{P_4} \right) \left[1 + e^{(0.036 T_4 - 54.4)} \right] \quad (23)$$

$$\ln_{RB} = 4775 \left[\frac{h_s}{\log(T_4 - T_6)} \right]^{0.8} + 11820 \dot{m}_s + 658 \dot{m}_g \quad (24)$$

$$\ln_{ST} = 6000 [E_{p_{ST}}]^{0.7} \quad (25)$$

$$\ln_{CO} = 1773 \dot{m}_s \quad (26)$$

$$\ln_p = 3540 [W_p]^{0.71} \quad (27)$$

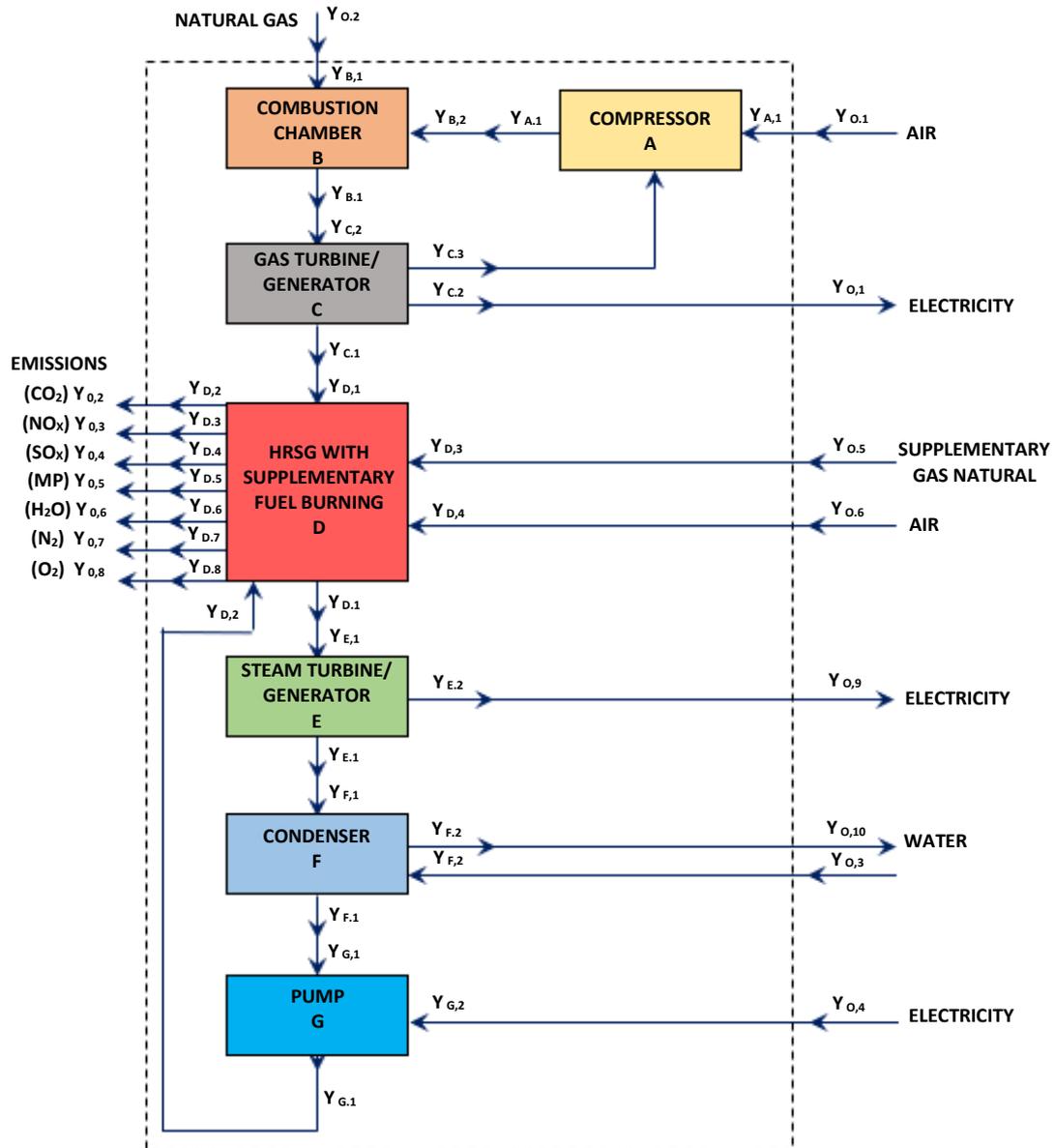


Figure 3. Thermoeconomic functional diagram of the gas turbine system associated with the HRSG and steam turbine with SFB.

For the cost of maintaining of the gas turbine (MC_{GT}) and steam turbine (MC_{ST}) systems, Antunes (1999) and Silveira *et al.* (2012) use the value of 0.015 US\$/kWh, and the value of 0.017 US\$/kWh. For this case, the value of 0.014 US\$/kWh and 0.015 US\$/kWh is adopted for the MC_{GT} and MC_{ST} , respectively.

For the cost of maintenance of the HRSG (MC_{RB}), Antunes (1999) and Silveira *et al.* (2012) use the value of 0.008 US\$/kWh. For this case, the values given by Castro (2007) of 0.005 US\$/kWh and 0.011 US\$/kWh is adopted for the system without SFB and for the system with SFB, respectively.

3. RESULTS AND DISCUSSIONS

The results of Fig. 4 were obtained from the energetic analysis and environmental analysis developed by Mariños *et al.*, (2019b) and Mariños *et al.*, (2021), respectively.

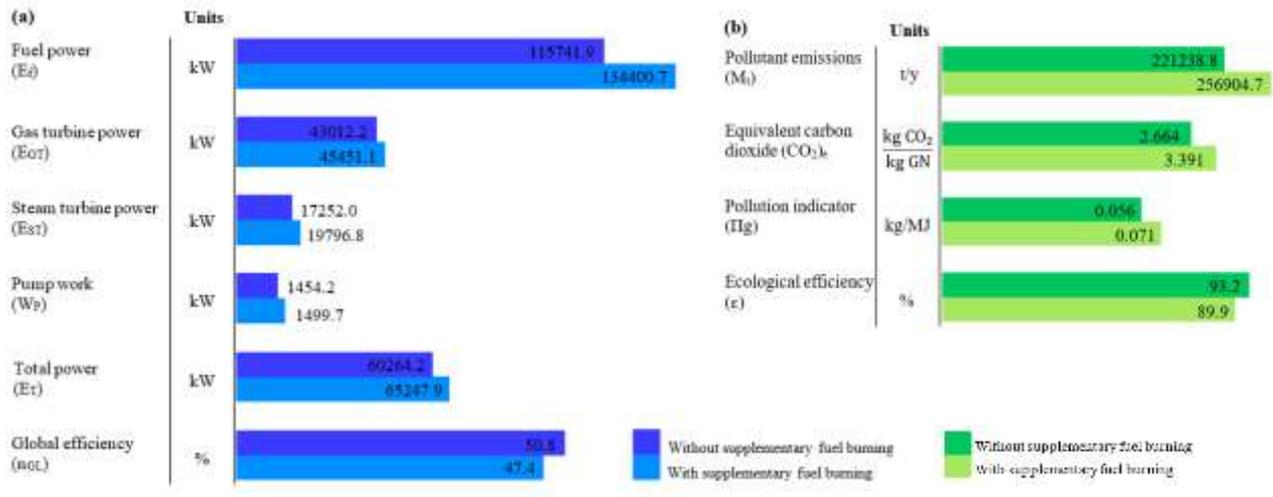


Figure 4. Results of the: (a) energetic analysis (Mariños *et al.*, 2019b) and (b) environmental analysis (Mariños *et al.*, 2021) of the combined cycle thermoelectric power plant without/with SFB.

3.1 Results of exergetic analysis

Figure 5 compares the results obtained from the exergetic and irreversibility analysis of each component of the system without/with SFB. The equations developed in section 2.3 were used.

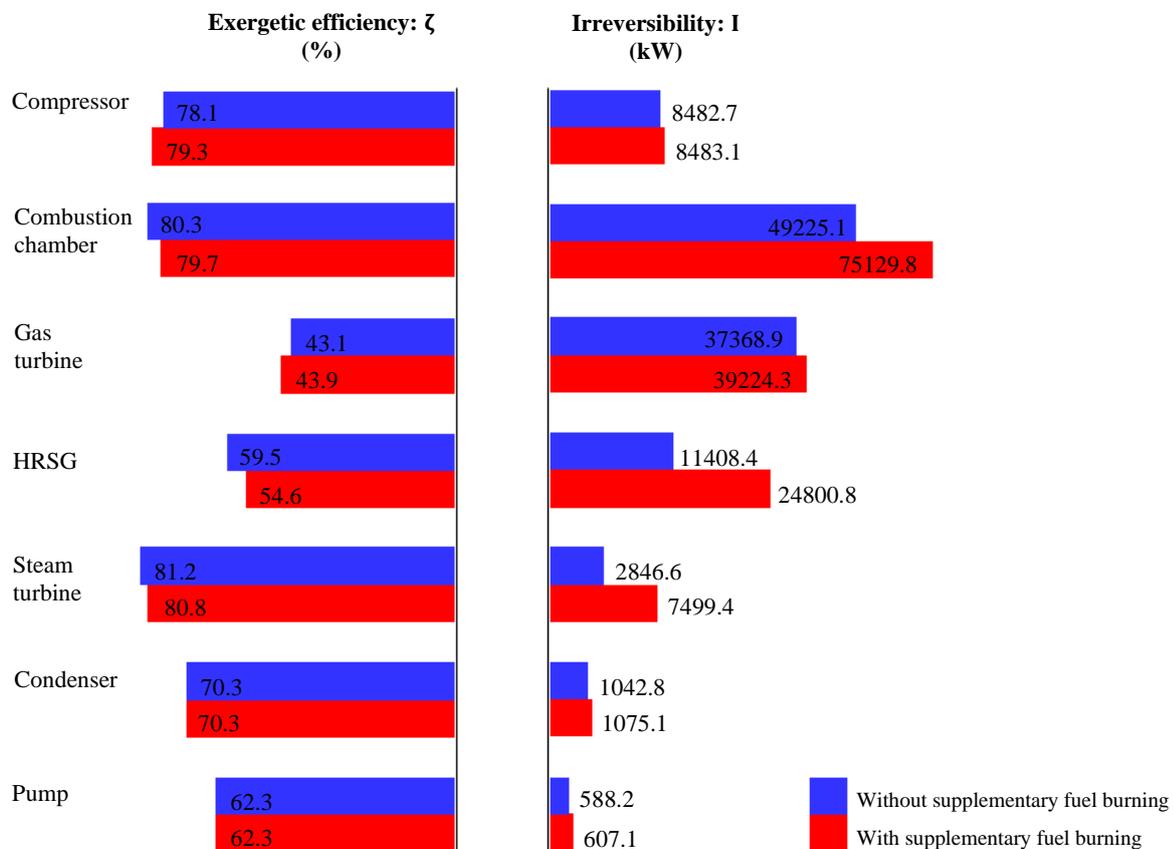


Figure 5. Results of the exergetic analysis: exergetic efficiency (ζ) and irreversibility (I) in the components of the combined cycle thermoelectric power plant without/with SFB.

The results of Fig. 5 are validated with the exergetic analysis performed by Castro (2007), which is shown in Table 3. Castro performed an exergetic analysis of a combined cycle thermal power plant that operates with natural gas and diesel. Table 3 shows the results of this comparison, and it is observed that the exergetic efficiency in each component of the system does not present significant variation with respect to the fuel used. In addition, it is observed that the exergetic efficiency of the HRSG of the cycle without SFB presents an increase of 4.9% compared to the cycle with SFB. This occurs because the input exergy of the HRSG with SFB is higher due to the additional fuel, and that being related to the useful exergy (exergy produced) of the equipment, a decrease in its efficiency is obtained. The results presented in Fig. 5 and Table 3 show that the equipment of the system without SFB presents a better performance as shown by the exergy efficiency.

Table 3. Comparison of exergetic efficiency in the system without (*) and with (**) SFB.

Equipments (%)	NG*	NG**	NG* (Castro, 2007)	NG** (Castro, 2007)	Diesel* (Castro, 2007)	Diesel** (Castro, 2007)	NG*/NG**
Compressor	78.1	79.3	82.3	83.6	85.8	85.8	-1.2
Combustion Chamber	80.3	79.7	85.9	85.2	86.8	86.8	0.6
Gas turbine	43.1	43.9	40.6	40.9	40.7	40.7	-0.8
HRSG	59.5	54.6	55.7	49.8	55.7	49.6	4.9
Steam turbine	81.2	80.8	85.8	85.8	85.8	85.8	0.4
Condenser	70.3	70.3	72.4	72.4	72.4	72.4	0
Pump	62.3	62.3	59.5	59.5	59.5	59.5	0

3.2 Results of economic analysis

3.2.1 Electricity production cost considering ecological efficiency

Table 4 shows the parameters involved in the thermoeconomic analysis and the results for a 5-year amortization period shown in Table 5, considering a variation in the costs of the natural gas used.

When considering the system with SFB for natural gas, there is an increase in the electricity production cost that incorporates the concept of ecological efficiency of 10% to 12% in relation to the combined cycle without SFB system.

Table 4. Parameters of thermoeconomic analysis (Silveira *et al.*, 2012).

Parameters	Specification	Units
Annual interest rate (r)	12	%
Pay-back (k)	5	year
Operating hours (OH)	7200	h/y
Cost of natural gas (FC)	0.011 a 0.017	US\$/kWh

Table 5. Results of the thermoeconomic analysis for the system without/with SFB.

FC [US\$/kWh]	PC _{EE} [US\$/kWh]		
	Without SFB	With SFB	Increase of the system with SFB (%)
0.011	0.0679	0.0768	11.58
0.012	0.0699	0.0788	11.29
0.013	0.0718	0.0807	11.02
0.014	0.0738	0.0827	10.80
0.015	0.0757	0.0847	10.55
0.016	0.0777	0.0866	10.35
0.017	0.0796	0.0886	10.16

3.2.2 Expected annual benefit due to pay-back

In this section, the annual benefit to recover the investment is determined. That is, in how long will the company reach equilibrium for the result to exceed the value of the initial investment. For this development the data shown in Table 6 are used.

Table 6. Parameters for the expected annual benefit.

Parameters	Specification	Unit
Annual interest rate	4, 8 e 12	%
Pay-back	1 - 12	year
Operating hours	7200	h/year
Cost of natural gas (Silveira <i>et al.</i> , 2007)	0.014	US\$/kWh
ST power (Mariños <i>et al.</i> , 2019b; Mariños <i>et al.</i> , 2021)	20000	kW
GT power (Mariños <i>et al.</i> , 2019b; Mariños <i>et al.</i> , 2021)	51685	kW
ST maintenance cost (Silveira <i>et al.</i> , 2007)	0.015	US\$/kWh
GT maintenance cost (Silveira <i>et al.</i> , 2007)	0.014	US\$/kWh
RB maintenance cost without SFB (Antunes, 1999)	0.005	US\$/kWh
RB maintenance cost with SFB (Antunes, 1999)	0.011	US\$/kWh

The total cost of the investment and the annual benefit without and with SFB due to the pay-back for an annual interest rate of 4%, 8% and 12% was developed. For a rate of 4%, the investment is recovered in 3 years and 3 months, for a rate of 8% in 3 years and 6 months, and for a rate of 12% in 3 years and 9 months.

3.2.3 Sensitivity analysis

The fuel cost, the annual interest rate, the amortization period for invested capital, among others, are the parameters that affect economic performance. The sensitivity analysis is presented according to the following variables: Fuel prices varying between $\pm 50\%$, capital amortization period varied between 3 to 12 years and annual interest rate of 4% and 12%.

Figure 6 shows the PC_{EE} as a function of the pay-back, with an annual interest rate of 4% and 12%, considering natural gas as a fuel.

In Fig. 6, it can be seen that the electricity production cost is lower for the combined cycle without SFB when compared to the combined cycle with SFB. The costs increase by 10 - 12%, with an annual interest rate of 4% and 12%.

The economic analysis carried out by Silveira *et al.* (2007) to a NG and diesel thermoelectric power plant showed that the electricity production cost for the NG power plant is 0.053 US\$/kWh for an annual interest rate of 7%, while for diesel it was 0.150 US\$/kWh, being the ratio between both costs of 1/3.

The concept of economic engineering and ecological efficiency is incorporated into the model, which makes it possible to determine the electricity production cost in thermoelectric power plants. According to the results obtained, it appears that it is possible to apply this method to determine the electricity production cost considering the level of environmental impact of the generation. The method considers the type of fuel used, the levels of thermodynamic efficiency and ecological efficiency of the generation plant.

It is observed that if NG is used as fuel, there is an increase in the electricity production cost in the system with SFB in relation to the system without SFB, and the ecological electricity production cost (PC_{EE}) is much better (lower) considering the NG fueled combined cycle without SFB when compared to the combined cycle with SFB. NG as a fuel for combined cycle technology (CC) is a great option, due to the lower levels of electricity production cost presented, for all parameters considered. Therefore, it is possible to affirm that CC offer important competitive advantages over conventional ones. As it is a relatively recent technology, there are open opportunities for improvements, either in thermal efficiency or in the performance of the equipment involved, making the global economy of this type of enterprise even more attractive.

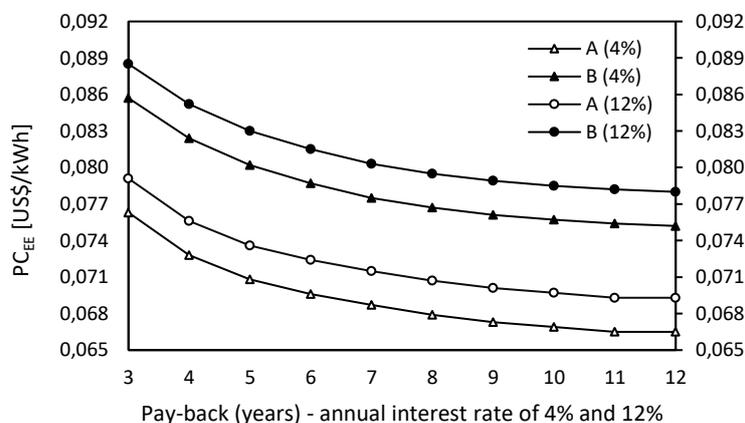


Figure 6. Ecological electricity production cost based on the pay-back without/with SFB with an annual interest rate of 4% and 12%.

4. CONCLUSIONS

Through the exergetic analysis was demonstrated that the components of the system without SFB present a better performance, a result corroborated by the exergetic efficiency. The cycle with SFB has an increase of the irreversibility in the combustion chamber, HRSG and steam turbine of 52.6%, 117.4% and 163.5%, respectively. In this way, it is shown that the components of the system with SFB suffer greater alterations and present an irreversible process caused by friction, mixing of fluids, unrestrained expansion, heat transfer across a finite temperature difference, inelastic deformation of materials, and chemical reactions.

The economic analysis showed that the annual benefit without/with SFB due to the pay-back for an annual interest rate of 4%, 8% and 12% are 3.3 years, 3.5 years and 3.9 years respectively, and there is an increase in the electricity production cost between 10% and 12% for the system with SFB in relation to the system without SFB.

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