THERMOECONOMIC ANALYSIS OF A SMALL SCALE COGENERATION SYSTEM USING A MICROTURBINE AND AN ABSORPTION CHILLER

Claudio Pinto Rebello Rücker and Edson Bazzo
UFSC – Federal University of Santa Catarina
Department of Mechanical Engineering
88.040-900 Florianópolis, SC – Brazil
Phone 55 48 331-9390 Fax 55 48 234-4615
claudio@cet.ufsc.br

Mathias Nils Rognar Jonsson and Svante Johan Karlsson
KTH – Royal Institute of Technology
Department of Energy Technology
Stockholm - Sweden

Abstract. *Brazil presents expressive potential for the implantation of small cogeneration plants, most in the tertiary sector, where there is a large number of small commercial companies or business organizations with the needs relate to electric power, steam, hot water and/or cold water. The objective of this work is to carry out an exergetic and exergoeconomic evaluation of a small scale microturbine-absorption chiller cogeneration system. The system purposes are to generate 28 kW net power and to provide chilled water for refrigeration. The analysis carried out demonstrates the application of the exergetic and exergoeconomic methodology for the proposed cogeneration system. This methodology gives a vast view of the system exergy destruction location and evaluates where are the higher cost rates of exergy destruction and exergy loss. The results showed that great part of the system exergy destruction is located in the microturbine and chiller connection (heat exchanger) and in the evaporator assembly (condenser, absorber and evaporator).*

Keywords. *small-scale cogeneration systems, microturbines, absorption chilling, combined heat and power system, exergoeconomic analysis*

1. Introduction

Distributed electricity generation has gained increased attention from the market during some time. Advantages are, for example, decreased losses in distribution, lower investment costs for grid connection and increased independency of the user. The concept of cogeneration increases the total amount of useful work that can be achieved by a system and thereby making it more feasible.

The use of cogeneration units can help to decentralize and to increase the reliability of the Brazilian electric transmission system, diversifying the energetic power sources and increasing the installed capacity of electricity generation through private capital. Brazil presents expressive potential for the implantation of small cogeneration plants, above the whole tertiary sector, where there is a considerable number of small commercial companies or business organizations with the needs relate to electric power, steam, hot water and/or cold water.

Taking into account the particularities of a system configuration, some research is required in order to examine and improve its efficiency. During the last years several thermoeconomic methodologies have been developed, most of them involving the second law of thermodynamics. The exergoeconomic evaluation consists of a detailed exergy analysis, an economic analysis conducted at the component level, calculation of the cost of each stream using an appropriate exergy-costing method and a final evaluation at the component level conducted with the aid of some relevant exergoeconomic variables.

The objective of this work is to carry out an exergoeconomic evaluation of the proposed cogeneration system, based on the methodology as proposed by Tsatsaronis et al. (1994) and specific assumptions for the absorption chiller (Misra et al., 2003). The purposes of the cogeneration system selected for this work are to generate a net power of 28 kW through a microturbine and to provide cold water for refrigeration system through an absorption chiller.

2. The system description

The cogeneration system principally consists of a microturbine having natural gas as the fuel and with a nominal electrical output of 28 kW and an absorption chiller with a cooling capacity of 29,8 kW. The basic layout of the system can be seen in Fig. (1). The microturbine system consists of an air compressor, an air preheater, a combustor chamber and a gas turbine. Other essential parts of the system are the heat exchanger and the cooling tower. The heat exchanger is used to recover heat from the flue gases of the turbine and to supply the chiller with heat. The cooling tower is used to cool the absorber and condenser of the chiller. The chiller works as a single stage absorption system having hot water as the heat source, and its configuration is shown in Fig (2). The main chiller components are one evaporator, a condenser, a generator, an absorber, an electric pump, a heat exchanger and two expansion valves.

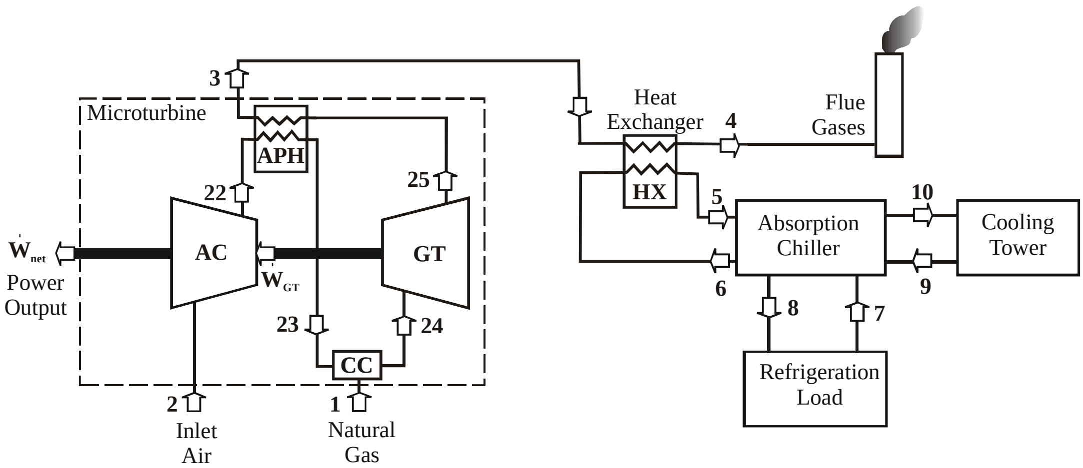

Figure 1. Cogeneration system layout.

3. The thermodynamic model

The calculation procedure consists in solving an equations system based on the First and Second Law of the Thermodynamics and Mass Conservation, taking into account the main equipments of the cogeneration plant.

The environmental conditions are defined as T_O=25°C and P_O=101,3 kPa. To simplify this model, the following assumptions were made: (i) the air and the combustion gases behave as ideal gases with constant specific heats; (ii) all components are adiabatic. According to Valero et al. (1994), the governing equations for the microturbine physical model used in the simulation are presented in the following.

Air compressor (AC):

$$T_{22}^{K} = T_{2}^{K} \cdot \left\{ 1 + \frac{1}{\eta_{AC}} \cdot \left(\left(\frac{P_{22}}{P_{2}} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right) \right\} \tag{1}$$

$$\dot{W}_{AC} = \dot{m}_2 \cdot cp_a \cdot (T_{22} - T_2) \tag{2}$$

The superscript K in the Eq. (1) means that the temperatures are in Kelvin scale. The pressure P_2 is equal to the environmental pressure, cp_a is the air specific heat at constant pressure, γ_a is the air specific heat ratio (γ_a =1,4) and η_{AC} is the air compressor efficiency (η_{AC} =0,85). The temperature of the stream 2 (T_2) is set to be equal to 15°C.

Combustion Chamber (CC):

$$\dot{m}_{24} = \dot{m}_1 + \dot{m}_{23} \tag{3}$$

$$\dot{m}_2 \cdot h_{23} + \dot{m}_1 \cdot LHV = \dot{m}_{24} \cdot h_{24} \tag{4}$$

$$P_{24} = P_{23} \cdot (1 - \Delta P_{CC}) \tag{5}$$

The fuel for the combustion chamber is the natural gas with a lower heating value (LHV) equal to 47680 kJ/kg. The combustion chamber pressure drop (ΔP_{CC}) is equal to 5%. The natural gas mass flow rate ($\dot{m}_1$) is equal to 0,00213 kg/s.

Air Preheater (APH):

$$\dot{m}_2 \cdot cp_a \cdot (T_{23} - T_{22}) = \dot{m}_{24} \cdot cp_g \cdot (T_{25} - T_3) \tag{6}$$

$$P_{23} = P_{22} \cdot (1 - \Delta P_{a,APH}) \tag{7}$$

$$P_3 = P_{25} \cdot (1 - \Delta P_{g,APH}) \tag{8}$$

The combustion gas specific heat at constant pressure (cp_g) is considered to be equal to cp_a at the corresponding stream temperature. The air and gas streams pressure drops ($\Delta P_{a,APH}$) and ($\Delta P_{g,APH}$) are equal to 5% and 3%, respectively. The mass flow rate of the combustion gases stream is equal to 0,297 kg/s. The natural gas and combustion gases mass flow rates are values of a commercial model microturbine available in the American market.

Gas Turbine (GT):

$$T_{25}^{K} = T_{24}^{K} \cdot \left\{ 1 - \eta_{GT} \cdot \left(1 - \left(\frac{P_{24}}{P_{25}} \right)^{\frac{1-\gamma_g}{\gamma_g}} \right) \right\} \tag{9}$$

$$\dot{W}_{GT} = \dot{m}_{24} \cdot cp_g \cdot (T_{24} - T_{25}) \tag{10}$$

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} \tag{11}$$

The gas turbine efficiency (η_{GT}) is equal to 0,91 and the gas specific heat ratio (γ_g) is equal to 1,33.

Heat Exchanger (HX):

$$\dot{m}_{5,6} \cdot (h_5 - h_6) = (\frac{1 - Loss_{HX}}{100}) \cdot \dot{m}_{24} \cdot cp_g \cdot (T_3 - T_4) \tag{12}$$

The heat loss for the heat exchanger ($Loss_{HX}$) is assumed to be 5%.

The absorption chiller behavior was simulated through mass and energy balances applied to each component of the Fig. (2). The streams 5 and 6 are the heat source (hot water) for the chiller. The streams 9 and 10 cool the condenser and the absorber through the cooling tower shown in Fig. (1). The streams 7 and 8 are the inlet and outlet chilled water for refrigeration, respectively. The streams 12, 13 and 14 are the weak solution and the streams 15,16 and 17 are the strong solution of LiBr of the water/LiBr mixture, respectively.

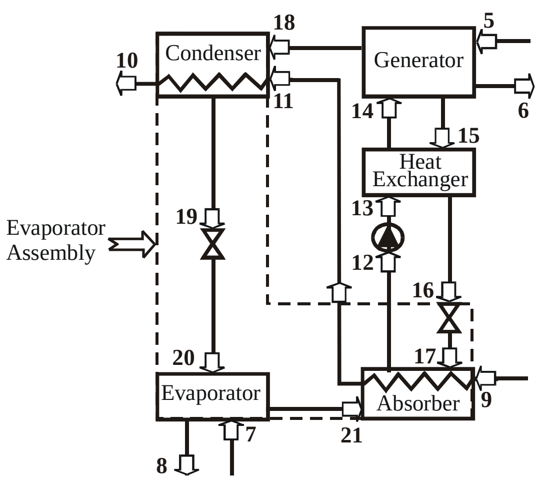

Figure 2. Absorption chiller layout

The solution and the LiBr mass balance are shown in the Eqs. (13) and (14), respectively:

$$\dot{m}_W = \dot{m}_S + \dot{m}_{H2OC} \tag{13}$$

$$X_W \cdot \dot{m}_W = X_S \cdot \dot{m}_S \tag{14}$$

Where, $\dot{m}_W$ and $\dot{m}_S$ are the mass flow rates of the weak and strong LiBr/water solution, respectively and $\dot{m}_{H2OC}$ is the mass flow rate of the water desorbed in the generator X_W and X_S are the weak and strong LiBr concentration, respectively.

Following the energy balances over the absorption chiller components are presented.

Generator (GEN):

$$\dot{Q}_H + \dot{m}_w \cdot h_{14} - \dot{m}_{H2OC} \cdot h_{18} - \dot{m}_S \cdot h_{15} = 0 \tag{15}$$

$$\dot{Q}_H = \frac{(1 - Loss_{GEN})}{100} \cdot \dot{m}_{5,6} \cdot (h_5 - h_6) \tag{16}$$

Condenser (COND):

$$\dot{m}_{9,10} \cdot (h_{10} - h_{11}) = \frac{(1 - LOSS_{COND})}{100} \cdot \dot{m}_{H2O} \cdot (h_{18} - h_{19}) \tag{17}$$

Evaporator (EVAP):

$$\dot{m}_{7,8} \cdot (h_8 - h_7) = \dot{m}_{H2OC} \cdot (h_{21} - h_{20}) \tag{18}$$

Absorber (ABS):

$$\dot{m}_{H2OC} \cdot h_{21} + \dot{m}_S \cdot h_{17} - \dot{m}_W \cdot h_{12} - \dot{Q}_{ABS,\max} = 0 \tag{19}$$

$$\dot{m}_{9,10} \cdot (h_{11} - h_9) = \frac{(1 - Loss_{ABS})}{100} \cdot \dot{Q}_{ABS,\max} \tag{20}$$

Absorbent pump (PUMP):

$$\dot{W}_{PUMP} = \dot{m}_W \cdot (h_{13} - h_{12}) \tag{21}$$

$$\dot{W}_{PUMP} = \dot{m}_W \cdot v_{12} \left(\frac{P_{high} - P_{low}}{1000} \right) \tag{22}$$

P_{high} and P_{low} are the pressures in the generator and in the absorber, respectively, $\dot{W}_{PUMP}$ is the electric work input to the pump and v_{12} is the LiBr/water solution specific volume of the stream 12.

Chiller Heat Exchanger (CHX):

$$\dot{m}_s \cdot (h_{15} - h_{16}) = (\frac{1 - Loss_{CHX}}{100}) \cdot \dot{m}_w \cdot (h_{14} - h_{13}) \tag{23}$$

For simplification, the evaporator and the absorbent pump are considered adiabatic and isentropic, respectively. All the heat losses ($Loss_i$) in the ith chiller component are equal to 5%. The input variables for the chiller are the following: T_5=95°C, T_6=85°C, T_7=12,2°C, T_8=6,7°C, T_9=29,4°C, T_{10}=35,9°C, $\dot{m}_{5,6}$=1,09 kg/s, $\dot{m}_{7,8}$=1,28 kg/s and $\dot{m}_{9,10}$=2,52 kg/s. These are values of a commercial model single stage hot water absorption chiller available in the Indian market. The following assumptions were made in order to simulate the model: (i) the state of the refrigerant leaving the condenser is saturated liquid; (ii) the state of the refrigerant exiting the evaporator is saturated steam; (iii) lithium bromide solutions in the generator and the absorber are assumed to be in equilibrium at their respective temperatures and pressures; (iv) pressure losses in all the heat exchangers and pipelines are neglected.

A simplified model for calculating the exergy of the streams was considered as proposed by Valero et al. (1994). It is assumed that the physical exergy is negligible for the natural gas entering the microturbine. The natural gas (stream 1) exergy rate, $\dot{E}_1$, is calculated as shown:

$$\dot{E}_1 = \dot{m}_1 \cdot \sum x_k \cdot \bar{e}_k \tag{24}$$

For the *jth* combustion gases stream, the exergy rate, $\dot{E}_j$, is calculated as shown:

$$\dot{E}_j = \dot{m}_j \cdot \left(\sum x_k \cdot \bar{e}_k + \bar{R} \cdot T_0^K \cdot \sum x_k \cdot \ln x_k\right) + \dot{m}_j \cdot cp_g \cdot \left(T_j^K - T_0^K - T_0^K \cdot \ln \frac{T_j^K}{T_0^K}\right) \tag{25}$$

Where x_k is the mole fraction, shown in Tab. (1) for the fuel and in Tab. (2) for the combustion products, $\bar{e}_k$ is the standard chemical exergy of the specie *k*, $\bar{R}$ is the gas constant, T_0^K is the reference temperature in Kelvin scale (298,15 K) and T_j^K is the flue gas temperature in Kelvin scale. Complete combustion is assumed in the microturbine combustor.

Table 1. Natural gas composition.

Specie	Mole Fraction (%)
CH_4	89,0
C_2H_6	6,4
C_3H_8	1,6
C_4H_{10}	0,6
C_5H_{12}	0,3
C_6H_{14}	0,1
CO_2	0,8
N_2	1,2

Table 2. Combustion products composition

Specie	Mole Fraction (%)
CO_2	1,4
H_2O	3,5
N_2	77,2
O_2	17,9

For the water streams, the physical exergy rates, $\dot{E}_j^{PH}$, are calculated for the jth stream as shown:

$$\dot{E}_j^{PH} = \dot{m}_j \cdot \left[h_j - h_0 - T_0^K \cdot (s_j - s_0)\right] \tag{26}$$

Here, $\dot{m}_j$, h_j and s_j are, respectively, the mass flow rate, enthalpy and entropy for the jth water stream and h_0 and s_0 are the enthalpy and entropy for the reference state, respectively.

The calculation procedure of the chemical exergy of various substances based on standard chemical exergy values of respective species is widely discussed by Bejan et al. (1996) and Szargut et al. (1988). As there is no chemical reaction in the components of absorption chiller, including the generator and the absorber, the chemical state of the absorbent remains the same throughout the absorber circuit. Therefore, the change in chemical exergy of the water is only considered, while the LiBr chemical exergy is neglected (Misra et al., 2003). In this analysis, for the absorption system, the chemical exergy of the flows for water (refrigerant), $\dot{E}_j^{CH}$, can be calculated using the following relation:

$$\dot{E}_j^{CH} = \dot{m}_j \cdot \left(\frac{1-X}{M_{H_2O}}\right) \cdot \bar{e}_{H_2O} \tag{27}$$

Here, *X* is the LiBr concentration, M_{H_2O} is the water molecular weight and $\bar{e}_{H_2O}$ is the standard chemical exergy of water and its value is taken from Bejan et al. (1996). For calculating the entropy for the LiBr solution streams, it was used the following equation, proposed by Kaita (2001).

$$s_j = \sum_{a=0}^{3} \sum_{b=0}^{3} B_{ab} X_j^b T_j^a \tag{28}$$

Where the constants B_{ab} are taken from the Tab. (3).

The exergetic efficiency of the *ith* plant component, ε_i, shows what percentage of fuel exergy, $\dot{E}_{F,i}$, can be found in the exergy of the product, $\dot{E}_{P,i}$, and is defined by:

$$\varepsilon_i = \dot{E}_{P,i} \big/ \dot{E}_{F,i} \tag{29}$$

Table 3. **C**onstants used for entropy calculations

a	B_{a0}	B_{a1}	B_{a2}	B_{a3}
0	5.127558E-01	-1.393954E-02	2.924145E-05	9.035697E-07
1	1.226780E-02	-9.156820E-05	1.820453E-08	-7.991806E-10
2	-1.364895E-05	1.068904E-07	-1.381109E-09	1.529784E-11
3	1.021501E-08	0	0	0

4. The thermoeconomic model

For evaluating the costs of a plant, it was considered the annual cost of fuel and the annual cost associated with owning and operating each plant component. The purchase costs of the components (Z_i) are presented in Tab. (4). The microturbine and absorption chiller total cost were taken from the manufactures Capstone and Thermax, respectively. The microturbine components and absorption chiller components costs were assumed to be an estimated percentage of the total cost. The general equation for the cost rate ($\dot{Z}_i$ in US$/s) associated with the capital investment and the maintenance costs for the *ith* component is:

$$\dot{Z}_i = \frac{Z_i \cdot CRF \cdot \varphi}{N \cdot 3600} \tag{30}$$

where *CRF* is the annual capital recovery factor (*CRF* = 54%), φ is the maintenance factor ($\varphi = 1{,}06$) and *N* represents the number of hours of plant operation per year (*N* = 7200 h). The costs per exergy unit (c_j) of the streams are calculated through cost balances applied to each component, as shown:

$$\sum_{in,i} (\dot{C}_{in}) + \dot{Z}_i = \sum_{out,i} (\dot{C}_{out}) \tag{31}$$

$$\dot{C}_{in,out} = c_{in,out} \cdot \dot{E}_{in,out} \tag{32}$$

The fuel cost per unit of exergy is $c_1 = 6{,}02$ US$/GJ. For solving the microturbine cost balances, the following propositions are considered: (i) all the gas streams have the same costs per unit of exergy; (ii) the air stream cost per unit of exergy is zero.

The thermoeconomic analysis for the absorption chiller is carried out according to the propositions and assumptions developed by Misra et al. (2003). The purpose of the generator is to provide pure water steam (stream18) and weak solution of refrigerant (stream 15). This is achieved by supplying exergy from the hot water (streams 5-6). Hence, streams 15 and 18 are charged with all costs associated with owning and operating the generator. The purpose of the solution heat exchanger is to transfer the exergy of the weak solution of refrigerant to stream 14 (strong solution of refrigerant). Therefore, stream 14 is charged with all costs associated with owning and operating the heat exchanger. The purpose of the pump is to increase the exergy through external electric work. Thus, stream 13 is charged with all costs associated with purchasing and operation the pump. Accordingly, the following equations are developed:

$$\dot{C}_5 + \dot{C}_{14} + \dot{Z}_{GEN} = \dot{C}_{18} + \dot{C}_{15} + \dot{C}_6 \tag{33}$$

$$c_5 = c_6 \tag{34}$$

$$\frac{\dot{C}_{18} \big/ \dot{m}_{H2Oc} - \dot{C}_{14} \big/ \dot{m}_W}{\dot{E}_{18} \big/ \dot{m}_{H2Oc} - \dot{E}_{14} \big/ \dot{m}_W} = \frac{\dot{C}_{15} \big/ \dot{m}_S - \dot{C}_{14} \big/ \dot{m}_W}{\dot{E}_{15} \big/ \dot{m}_S - \dot{E}_{14} \big/ \dot{m}_W} \tag{35}$$

$$\dot{C}_{13} + \dot{C}_{15} + \dot{Z}_{CHX} = \dot{C}_{14} + \dot{C}_{16} \tag{36}$$

$$c_{15} = c_{16} \tag{37}$$

$$\dot{C}_{12} + c_{el} \cdot \dot{W}_{PUMP} + \dot{Z}_{PUMP} = \dot{C}_{13} \tag{38}$$

Where, the electricity cost for the absorbent pump is c_{el} =26 US$/GJ.

Since the product of the heat dissipative devices cannot be defined readily, special considerations need to be applied. The methodology described by Bejan et al. (1996) and Misra et al. (2003) explains that all these devices should be considered as a single virtual component together with the component they serve. Following this methodology, the evaporator assembly is a virtual component consisting of evaporator and heat dissipative components (condenser, expansion valve, absorber and throttling valve) and is shown in Fig. (2). The cost rates associated with the capital investment for these components and the cost rates associated with the exergy losses are charged to the chilled water. The cost rates associated with the exergy losses in condenser and absorber are given by:

$$\dot{C}_{18} - \dot{C}_{19} + \dot{Z}_{COND} = \Delta\dot{C}_{COND} \tag{39}$$

$$\dot{C}_{21} + \dot{C}_{17} + \dot{Z}_{ABS} = \Delta\dot{C}_{ABS} + \dot{C}_{12} \tag{40}$$

For the dissipative components and the evaporator, the following cost equations are developed:

$$c_{18} = c_{19} \tag{41}$$

$$c_{19} = c_{20} \tag{42}$$

$$c_{16} = c_{17} \tag{43}$$

$$\frac{\dot{C}_{21} + \dot{C}_{17}}{\dot{E}_{21} + \dot{E}_{17}} = \frac{\dot{C}_{12}}{\dot{E}_{12}} \tag{44}$$

$$c_{20} = c_{21} \tag{45}$$

Since the purpose of the evaporator is to transfer exergy from stream 20 to the product stream 8, therefore all costs associated with owning and operation of the evaporator is charged to stream 8. Thus, the cost rate of the chilled water is given by:

$$\dot{C}_5 + c_{el} \cdot \dot{W}_{PUMP} + \dot{C}_8 = \dot{C}_6 + \dot{C}_7 + \Delta\dot{C}_{ABS} + \Delta\dot{C}_{COND} + \sum_{Chiller} \dot{Z} \tag{46}$$

Table 4. Cogeneration system component costs (Z)

Component	Cost (US$)
Air compressor	16800
Air preheater	4400
Combustion chamber	1600
Gas turbine	17200
Heat exchanger	700
Generator	975
Condenser	5250
Evaporator	525
Pump	300
Absorber	7200
Chiller heat exchanger	750
Total microturbine	40000
Total absorption chiller	15000

For the thermoeconomic evaluation, the cost effectiveness of each component can be evaluated using some exergoeconomic variables (Tsatsaronis et al., 1994). These variables include the average cost per unit of fuel ($c_{F,i}$), the average cost per unit of product ($c_{P,i}$), the exergoeconomic factor (f_i) and the cost flow rate associated with the exergy destruction and exergy loss ($\dot{C}_{D,i}$) and ($\dot{C}_{L,i}$), respectively, considering the fuel exergy rate fixed.

These exergoeconomic variables can be defined by:

$$c_{F,i} = \frac{\dot{C}_{F,i}}{\dot{E}_{F,i}} \tag{47}$$

$$c_{P,i} = \frac{\dot{C}_{P,i}}{\dot{E}_{P,i}} \tag{48}$$

$$\dot{C}_{D,i} = c_{F,i} \cdot \dot{E}_{D,i} \tag{49}$$

$$\dot{C}_{L,i} = c_{F,i} \cdot \dot{E}_{L,i} \tag{50}$$

$$f_i = \frac{\dot{Z}_i}{\dot{Z}_i + \dot{C}_{D,i}} \tag{51}$$

6. Results and discussion

Table (5) summarizes the thermodynamic properties at various plant streams. The results show a high cost per exergy unit of the chilled water (stream 8) and the chiller heat losses (streams 9, 10 and 11), showing a value equal to 2983 US$/GJ and 3234 US$/GJ, respectively. The high value for the chilled water is associated with its low exergy rate. A great part of the heat that supplies the chiller is dissipated through the streams 9, 10 and 11, having the purpose of cooling the absorber and the condenser. This can be a cause of the high cost values of these streams. The net power cost per exergy unit has a value equal to 35,76 US$/GJ.

Table 5. State properties and exergoeconomic costs for the plant streams.

Stream (j)	T_j (°C)	P_j. (kPa)	$\dot{m}_j$ (kg/s)	Chemical Compos.	LiBr Concent. (%)	$\dot{E}_j^{PH}$ (kW)	$\dot{E}_j^{CH}$ (kW)	$\dot{E}_j$ (kW)	c_j (US$/GJ)
1	15	300	0,00213	Natural gas	-	0	107,8	107,8	6,02
2	15	101,3	0,295	Air	-	0,051	0	0,051	0
3	260	101,3	0,297	Flue gas	-	18,97	1,039	20,01	26,54
4	100,6	101,3	0,297	Flue gas	-	2,47	1,039	3,51	26,54
5	95	101,3	1,097	Water	-	32,73	2,74	35,5	55,32
6	85	101,3	1,097	Water	-	24,46	2,74	27,2	55,32
7	12,2	101,3	1,279	Water	-	1,39	3,20	4,6	2983
8	6,7	101,3	1,279	Water	-	3,01	3,20	6,2	2983
9	29,4	101,3	2,519	Water	-	0,115	6,29	6,4	3234
10	35,9	101,3	2,519	Water	-	1,852	6,29	8,1	3234
11	33,1	101,3	2,519	Water	-	0,931	6,29	7,2	3234
12	34,1	0,8372	0,2365	Water/LiBr	55,58	5,691	0,262	6,0	76,41
13	34,1	7,761	0,2365	Water/LiBr	55,58	5,691	0,262	6,0	77,52
14	60,3	7,761	0,2365	Water/LiBr	55,58	6,8	0,262	7,1	87,9
15	82,3	7,761	0,2239	Water/LiBr	58,71	11,93	0,231	12,2	80,19
16	51,9	7,761	0,2239	Water/LiBr	58,71	10,15	0,231	10,4	80,19
17	41	0,8372	0,2239	Water/LiBr	58,71	14,51	0,231	14,7	80,19
18	75,6	7,761	0,0126	Steam	-	1,642	0,032	1,70	74,63
19	40,9	7,761	0,0126	Water	-	0,0205	0,032	0,1	74,63
20	4,4	0,837	0,0126	Sat. water	-	-31,76*	0,032	-31,7*	74,63
21	4,4	0,837	0,0126	Steam	-	-2,291*	0,032	-2,3*	74,63
22	110,7	242	0,2949	Air	-	25,05	0	25,05	55,39
23	545,8	229,9	0,2949	Air	-	92,07	0	92,07	38,48
24	814,4	218,4	0,297	Flue gas	-	158,2	1,039	159,3	26,54
25	647,8	104,5	0,297	Flue gas	-	96,52	1,039	97,56	26,54
$\dot{W}_{NET}$	-	-	-	-	-	-	-	28	35,76

* The values are negative because both the pressure and temperature are below reference environment state.

The results of exergy analysis are summarized in Tab. (6). The evaporator assembly presents a high exergy destruction ratio value equal to 43,1% and, consequently, a low exergetic efficiency value equal to 26,5%. The heat exchanger that transfers heat from the microturbine combustion gases to the absorption chiller generator has the highest exergy destruction ratio among all the plant components, having a value equal to 45,3%. The air compressor, air preheater and gas turbine present high values of exergetic efficiency, around 90%. Analyzing these results it can be concluded that great part of the cogeneration system exergy destruction rate is located in the microturbine and chiller connection (heat exchanger) and in the evaporator assembly (condenser, absorber and evaporator). These results suggest the need to find means for decreasing the exergy destruction rate of these components. It can be accomplished through an exergoeconomic optimization.

The exergoeconomic analysis results are shown in Tab. (7). The heat exchanger present the lowest exergoeconomic factor among all components, having a value equal to 7,2%. This result means exergy destruction and exergy loss and the same is confirmed by the low value of exergetic efficiency shown in Tab. (6). The combustor chamber has the second highest cost rate of exergy destruction. The highest cost rate of exergy destruction is found in the air preheater, having a value equal to 1,007 US$/h. Taking into account the low values of capital cost rate of the air preheater and the combustion chamber, investments could be done in order to decrease their cost rate of exergy destruction.

Table 6. Results of exergetic analysis for the plant components

Components	Fuel exergy, $\dot{E}_{F,i}$ (kW)	Prod. exergy, $\dot{E}_{P,i}$ (kW)	Ex. dest., $\dot{E}_{D,i}$ (kW)	Ex. Loss, $\dot{E}_{L,i}$ (kW)	Ex. dest. ratio (%)	Ex. loss ratio (%)	Ex. effic., ε_i (%)
Air compressor	28,43	25,00	3,42	0	12,0	0	87,9
Air preheater	77,55	67,02	10,53	0	13,6	0	86,4
Comb. chamber	107,80	67,20	40,61	0	37,7	0	62,3
Gas turbine	61,71	56,43	5,28	0	8,6	0	91,4
Heat exchanger	16,50	8,20	7,48	0,826	45,3	5,0	49,7
Generator	8,28	6,77	1,14	0,359	13,8	4,3	81,8
Evap. assembly	6,10	1,62	2,63	0,118	43,1	1,9	26,5
Ch. heat exch.	1,78	1,11	0,61	0,061	34,0	3,4	62,3

Table 7. Results of exergoeconomic analysis for the plant components.

Components	Cost per exergy unit		Cost rate of		Capital cost $\dot{Z}_i$ (US$/h)	Exergoeconomic factor, f_i (%)
	Fuel, $c_{F,i}$ (US$/GJ)	Product, $c_{P,i}$ (US$/GJ)	Ex. dest., $\dot{C}_{D,i}$ (US$/h)	Ex. loss, $\dot{C}_{L,i}$ (US$/h)		
Air compressor	35,76	55,50	0,441	0	1,336	75,2
Air preheater	26,54	32,17	1,007	0	0,350	25,8
Comb. chamber	6,02	10,18	0,880	0	0,127	12,6
Gas turbine	26,54	35,76	0,504	0	1,367	73,1
Heat exchanger	26,54	55,32	0,715	0,079	0,056	7,2
Generator	55,32	70,78	0,228	0,072	0,078	25,4
Evap. assembly	82,36	2983	0,779	0,035	1,031	57,0
Ch. heat exch.	80,19	143,6	0,176	0,018	0,060	25,3

7. Conclusion

The analysis carried out in this paper demonstrates the application of the exergetic and exergoeconomic methodology for the proposed microturbine-absorption chiller cogeneration system. This methodology gives a vast view of the system exergy destruction location and evaluates where are the higher cost rates of exergy destruction and exergy loss.

The system product costs per exergy unit net power and chilled water showed values equal to 35,76 US$/GJ and 2983 US$/GJ, respectively. In order to get better results in the thermoeconomic model used in this work it could be paid attention in some points: (i) the model proposed by Misra et al. (2003) for the absorption chiller exergoeconomic analysis could be improved by opening the evaporator assembly and so making possible to analyze the evaporator, absorber and condenser, individually; (ii) the chilled water exergy unit cost presents a very high value because of its low exergy rate.

Analyzing the results it can be concluded that great part of the cogeneration system exergy destruction rate is located in the microturbine and chiller connection (heat exchanger) and in the evaporator assembly (condenser, absorber and evaporator). The found results suggest the need to look for means for decreasing the exergy destruction rate of these components, using an exergoeconomic optimization tool.

8. Acknowledgement

The authors acknowledge the financial support from Petrobras, TBG, SCGás and from ANP – National Agency for Petroleum.

9. References

Bejan, A., Tsatsaronis, G., Moran, M., 1996, “Thermal Design and Optimization”, New York: John Wiley & Sons, Inc.

Kaita, Y., 2001, “Thermodynamic Properties of Lithium Bromide-Water Solutions at High Temperatures”, International Journal of Refrigeration, No. 24, pp. 374-390.

Misra, R.D., et al., 2003, “Thermoeconomic Optimization of a Single Effect Water/LiBr Vapour Absorption Refrigeration System”, International Journal of Refrigeration, No. 26, pp. 158-169.

Szargut, J., Morris, D.R. and Steward, F.R., 1988, “Exergy Analysis of Thermal, Chemical and Metallurgical Processes”, New York: Hemisphere Publishing Corporation.

Tsatsaronis, G. and Pisa, J., 1994, “Exergoeconomic Evaluation and Optimization of Energy Systems – Application to the CGAM Problem”, Energy, Vol. 19, No. 3, pp. 287-321.

Valero, A. et al. , 1994, “CGAM Problem: Definition and Conventional Solution”, Energy, Vol. 19, No. 3, pp. 279-286.