Exergy assessment of a cogeneration system with micro-turbine and absorption chiller

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Abstract. *This work consists of the development and application of a methodology of exergy analysis to a cogeneration system with a 30 kWe gas micro-turbine and a 35 kWt absorption chiller. The system was studied as a whole and divided into three subsystems: micro-turbine, heat exchanger and absorption chiller.*

The system was assessed based on data from simulations carried out on the micro-turbine at the NEST's Laboratory at the Federal University of Itajubá – UNIFEI and on the results attained by using the Thermoflex software.

The exergy efficiency of the system is 26.72% and each subsystem comprise of: micro-turbine 45.83%, heat exchanger 73.71% and absorption chiller 11.89%, which indicates the significant irreversibilities caused basically by the combustion process and by the heat exchange due to finite difference temperatures. The results are satisfactory as far as low capacity systems like this one are concerned.

Keywords*: cogeneration, exergy, gas micro-turbine, absorption chiller*

1. Introduction

Cogeneration is the sequential production of electrical or mechanical power and thermal energy, to be used for heating, cooling or both. A cogeneration system using a gas micro-turbine, a heat exchanger and a lithium bromide/water absorption chiller is being assembled in one of the laboratories of the Excellence Group in Thermal and Distributed Generation (NEST) of the Federal University of Itajubá.

The gas micro-turbine generates electric power and through the exhaust gases it supplies thermal energy for water heating at the heat exchanger. The energy obtained by the water is used as a resource to feed the generator of the chiller, where a closed cycle cools the water. A mass flow of cooling water also circulates through the chiller in order to extract the heat corresponding to the condensation of the cooling agent (water) and the heat dissipated during its absorption by the LiBr/water solution. At both the heat exchanger and the chiller there are pumps that make the fluids circulate, consuming a small part of the electric power that was produced.

Some studies on small-scaled cogeneration systems have been published. Experimental analysis of a cogeneration plant with a Capstone micro-turbine and a Yasaki lithium bromide/water absorption chiller was done by (Ho *et al*, 2004). This study presents the results of the system operating at partial load. In addition, it shows the influence of the thermal load and of the operation time on the system performance.

Rücker and Bazzo (2003) carried out an economic and technical analysis of a cogeneration system with gas micro-turbine and absorption chiller. They considered the global efficiency of the system as a function of the generated power at partial load and different ambient temperatures.

Although there are professional softwares that can calculate the energy of the flows and facilitate exergy analyses, it is important to be able to rely on a calculation methodology that, in a systemic and algorithmic way, shows the functional relations between the flows and the subsystems and allows simple exergy assessments to be carried out.

2. Theoretical development

In order to facilitate the study, the system was divided into three subsystems as it is shown in Figure 1:

I) Gas Micro-turbine (high pressure 330 Capstone with 30 kW of electric power);
II) Heat exchanger (Enedis);
III) Lithium bromide/water absorption chiller (Thermax with 35 kW of cooling capacity).

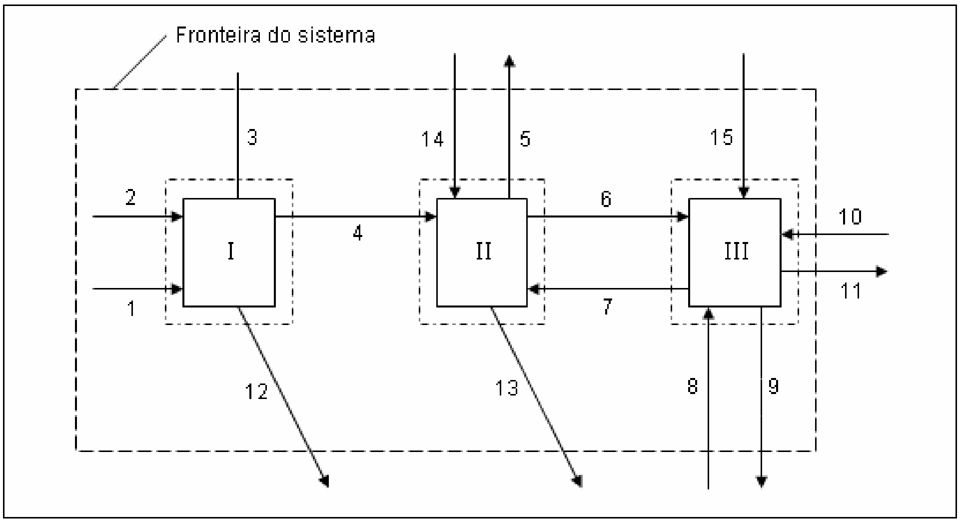

Figure 1. Scheme for the assessment of the proposed cogeneration system

The processes that take place at each subsystem were not analyzed and the analysis does not include other complementing elements, such as the cooling tower, because they are not essential for the exergetic evaluation.

The method developed takes the following considerations into account:

- The selected ambient reference condition is standard (atmospheric pressure of 0.101 MPa and ambient temperature of 25 °C);
- The thermal interactions in tubes, valves and other components of the system were discarded;
- The volume of the gases is determined for the normal temperature and pressure (atmospheric pressure of 0.101 MPa and ambient temperature of 0 °C) and afterwards the corresponding correction to bring them to the standard ambient condition is carried out;
- The kinetic and potential energy of the flow are neglected because they are very small in relation to the other energies involved in the system;
- The gases involved in the system are considered to be ideal.

The flows that interconnect the subsystems among themselves and the environment, which are displayed in Figure 1, are the following:

1) Fuel (natural gas);
2) Air for the combustion;
3) Electric output power from the micro-turbine electrical generator;
4) Micro-turbine exhaust gases;
5) Heat exchanger outlet gases;
6) Output hot water (chiller inlet);
7) Input hot water (chiller outlet);
8) Chilled water input;
9) Chilled water output;
10) Cooling water input;
11) Cooling water output;
12) Heat losses at subsystem I;
13) Heat losses at subsystem II;
14) Electric power consumed by the heat exchanger pump;
15) Electric power consumed by the cooling and weak solution fluid pumps.

In a sequential way, the used methodology forecast the calculation of the different volumetric flows, enthalpies, exergies and etc that are necessary to carry out the exergy balance.

1. Calculation of the volume of exhaust gases.

For a gaseous fuel, the gases are determined by the following expressions [references 2, 3 and 5] and then modified:

The relation of dry air (R_{dr}) is given by:

$$R_{dr} = 0.0476\left[0.5\,C_{CO} + 0.5\,C_{H_2} + 1.5\,C_{H_2S} + 2C_{ME} + 3.5\,C_{ET} + 5\,C_{PR} + 6.5\,C_{BU} + 8C_{PE} - C_{O_2}\right], \frac{m^3N_{dr}}{m^3N_{NG}} \tag{1}$$

The relation of nitrogen (R_{N2}), oxygen (R_{O2}), carbon dioxide (R_{CO2}), hydrogen sulphides (R_{H2S}), steam (R_S) to the NG are given, respectively, by:

$$R^{eg}_{N_2} = 0.01\,C_{N_2} + 0.79\,\alpha\,R_{as}\ , \frac{m^3N_{N_2}}{m^3N_{NG}} \tag{2}$$

$$R^{eg}_{O_2} = 0.21\,(\alpha - 1)\,R_{as}, \frac{m^3N_{O_2}}{m^3N_{NG}} \tag{3}$$

$$R^{eg}_{CO_2} = 0.01\left[C_{CO} + C_{CO_2} + C_{ME} + 2\,C_{ET} + 3\,C_{PR} + 4\,C_{BU} + 5\,C_{PE}\right], \frac{m^3N_{CO_2}}{m^3N_{NG}} \tag{4}$$

$$R^{eg}_{SO_2} = 0.01\,C_{H_2S}, \frac{m^3N_{SO_2}}{m^3N_{NG}} \tag{5}$$

$$R^{eg}_{S} = 0.01\left[C_{H_2S} + C_{H_2} + 2\,C_{ME} + 3\,C_{ET} + 4\,C_{PR} + 5\,C_{BU} + 6\,C_{PE} + 0.124\,df\right] + 1.6\,da\,\alpha\,R_{as}\ , \frac{m^3N_{S}}{m^3N_{NG}} \tag{6}$$

where: α = air excess coefficient
da = content of humidity in the air supplied for the combustion

The relation of the exhaust gases to the NG (R_{eg}) is:

$$R_{eg} = R^{eg}_{N_2} + R^{eg}_{O_2} + R^{eg}_{CO_2} + R^{eg}_{SO_2} + R^{eg}_{S}, \frac{m^3N_{eg}}{m^3N_{NG}} \tag{7}$$

The molar fraction (χ) of each gas is obtained from equation 8:

$$\chi^{eg}_{N_2} = \frac{R_{N_2}}{R_{eg}}, \chi^{eg}_{O_2} = \frac{R_{O_2}}{R_{eg}}, \chi^{eg}_{SO_2} = \frac{R_{SO_2}}{R_{eg}}, \chi^{eg}_{CO_2} = \frac{R_{CO_2}}{R_{eg}},\ \chi^{eg}_{S} = \frac{R_S}{R_{eg}} \tag{8}$$

2. Calculation of the enthalpy of the exhaust gases, air and gaseous fuel.

a) Exhaust gases enthalpy (h_{eg}) (equation 9):

$$h_{eg} = 0.03734\left(R^{eg}_{N_2}\overline{Cp}_{N_2} + R^{eg}_{O_2}\overline{Cp}_{O_2} + R^{eg}_{SO_2}\overline{Cp}_{SO_2} + R^{eg}_{CO_2}\overline{Cp}_{CO_2} + R^{eg}_{S}\overline{Cp}_{S}\right)(T - 298), \frac{kJ}{m^3{}^{N}_{NG}} \tag{9}$$

where: $\overline{Cp}$ = specific heat at constant pressure
T = absolute temperature

Constant 0.03734 is used for changing the volumes of each gas into kmol of gas/m³ fuel at standard ambient condition.

The values of $\overline{Cp}$ at standard state (0.101 MPa and 298 K) for each gas are provided in [7] in kJ/kmol.K.

The time rate of energy in exhaust gases (H_{eg}) will be:

$$H_{eg} = \dot{V}\,h_{eg}\ , kW \tag{10}$$

where: $\dot{V}$ = volume flow of natural gas at standard condition

b) Air enthalpy

As the supplied air is humid, its specific enthalpy at molar basis ($\overline{h}_{ha}$) is:

$$\overline{h}_{ha} = \left(\overline{C}p_{dr} + 1.61\, da\, \overline{C}p_S\right)\left(T - 298\right), \frac{kJ}{kmol_{dr}} \tag{11}$$

The time rate of energy in the humid air will be:

$$H_{ha} = 0.03734\, \alpha\, R_{as} \dot{V}\, h_{ha} \tag{12}$$

c) Natural gas enthalpy:

$$h_{NG} = 0.03734 \left(\begin{array}{l} R_{N_2} \overline{C}p_{N_2} + R_{O_2} \overline{C}p_{O_2} + R_{CO} \overline{C}p_{CO} + R_{CO_2} \overline{C}p_{CO_2} + R_{H_2S} \overline{C}p_{H_2S} + R_S \overline{C}p_S \\ + R_{H_2} \overline{C}p_{H_2} + R_M \overline{C}p_{ME} + R_{ET} \overline{C}p_{ET} + R_{PR} \overline{C}p_{PR} + R_{BU} \overline{C}p_{BU} + R_{PE} \overline{C}p_{PE} \end{array} \right) (T - 298), \frac{kJ}{m^3{}_{NG}} \tag{13}$$

The time rate of energy in the natural gas will be:

$$H_{NG} = \dot{V}\, h_{NG} \tag{14}$$

3. Calculation of the water enthalpy

It is determined by the water and steam tables [4], from their pressure and temperature values.

4. Calculation of the exergy of the gaseous mixtures

4.1) The calculation of the specific chemical exergy at molar basis ($\overline{e}_{ch}$) of gaseous mixtures can be carried out by using the following equation:

$$\overline{e}_{ch} = \sum_{i=1}^{n} \chi_i \left(\overline{e}_i^0 + \overline{R} T_0 \ln \chi_i\right) \tag{15}$$

where: $\overline{e}_i^0$ = specific chemical exergy at Standard state
$\overline{R}$ = gas universal constant at molar basis
To = temperature at the standard state (= 298 K)

Equation 15 has a different form for each gas involved in the system.

a) Air chemical exergy:

$$\overline{e}^{a}_{ch} = \left[\chi^a_{N_2}\left(\overline{e}^0_{N_2} + RT_0 \ln \chi^a_{N_2}\right)\right] + \left[\chi^a_{O_2}\left(\overline{e}^0_{O_2} + RT_0 \ln \chi^a_{O_2}\right)\right] + \left[\chi^a_S\left(\overline{e}^0_S + RT_0 \ln \chi^a_S\right)\right], \frac{kJ}{kmol_{dr}} \tag{16}$$

where:

$$\chi^a_{N_2} = 0.79, \frac{kmol_{N_2}}{kmol_{dr}}, \chi^a_{O_2} = 0.21, \frac{kmol_{O_2}}{kmol_{dr}}, \chi^a_S = 1.61 da, \frac{kmol_S}{kmol_{dr}}$$

The values of the standard specific chemical exergy are shown in table [7].

b) Fuel chemical exergy

An alternative of equation 15 for the gaseous fuels is:

$$e^{NG}_{ch} = e_{ch_{CO_2}} + e_{ch_{CO}} + e_{ch_{H_2}} + e_{ch_{H_2S}} + e_{ch_{N_2}} + e_{ch_{O_2}} + e_{ch_S} + e_{ch_{ME}} + e_{ch_{ET}} + e_{ch_{PR}} + e_{ch_{BU}} + e_{ch_{PE}} \tag{17}$$

Where:

$$\left.\begin{array}{l}
\overline{e}_{ch\,N_2} = \chi^{NG}_{N_2}\left(\overline{e}^0_{N_2} + RT_0 \ln \chi^{NG}_{N_2}\right), \overline{e}_{ch\,O_2} = \chi^{NG}_{O_2}\left(\overline{e}^0_{O_2} + RT_0 \ln \chi^{NG}_{O_2}\right), \overline{e}_{ch\,H_2} = \chi^{NG}_{H_2}\left(\overline{e}^0_{H_2} + RT_0 \ln \chi^{NG}_{H_2}\right) \\
\overline{e}_{ch\,H_2S} = \chi^{NG}_{H_2S}\left(\overline{e}^0_{H_2S} + RT_0 \ln \chi^{NG}_{H_2S}\right), \overline{e}_{ch\,CO} = \chi^{NG}_{CO}\left(\overline{e}^0_{CO} + RT_0 \ln \chi^{NG}_{CO}\right), \overline{e}_{ch\,CO_2} = \chi^{NG}_{CO_2}\left(\overline{e}^0_{CO_2} + RT_0 \ln \chi^{NG}_{CO_2}\right) \\
\overline{e}_{ch\,S} = \chi^{NG}_{S}\left(\overline{e}^0_{S} + RT_0 \ln \chi^{NG}_{S}\right), \overline{e}_{ch\,ME} = \chi^{NG}_{ME}\left(\overline{e}^0_{ME} + RT_0 \ln \chi^{NG}_{ME}\right), \overline{e}_{ch\,ET} = \chi^{NG}_{ET}\left(\overline{e}^0_{ET} + RT_0 \ln \chi^{NG}_{ET}\right) \\
\overline{e}_{ch\,PR} = \chi^{NG}_{PR}\left(\overline{e}^0_{PR} + RT_0 \ln \chi^{NG}_{PR}\right), \overline{e}_{ch\,BU} = \chi^{NG}_{BU}\left(\overline{e}^0_{BU} + RT_0 \ln \chi^{NG}_{BU}\right), \overline{e}_{ch\,PE} = \chi^{NG}_{PE}\left(\overline{e}^0_{PE} + RT_0 \ln \chi^{NG}_{PE}\right)
\end{array}\right\} \tag{18}$$

The molar fractions are the ones from each element or gaseous fuel compound.

c) Chemical exergy of the exhaust gases

Similarly to the fuel chemical exergy, but taking into account that the exhaust gases have 5 compounds and fundamental elements at most, their exergy is calculated by using the following:

$$\bar{e}_{ch}^{eg} = \bar{e}_{ch\,N_2} + \bar{e}_{ch\,O_2} + \bar{e}_{ch\,CO_2} + \bar{e}_{ch\,SO_2} + \bar{e}_{ch\,S} \tag{19}$$

Where:

$$\left.\begin{array}{l} \bar{e}_{ch\,N_2} = \chi_{N_2}^{eg}\left(\bar{e}_{N_2}^{0} + RT_0 \ln \chi_{N_2}^{eg}\right), \bar{e}_{ch\,O_2} = \chi_{O_2}^{eg}\left(\bar{e}_{O_2}^{0} + RT_0 \ln \chi_{O_2}^{eg}\right), \bar{e}_{ch\,CO_2} = \chi_{CO_2}^{eg}\left(\bar{e}_{CO_2}^{0} + RT_0 \ln \chi_{CO_2}^{eg}\right) \\ \bar{e}_{ch\,SO_2} = \chi_{SO_2}^{eg}\left(\bar{e}_{SO_2}^{0} + RT_0 \ln \chi_{SO_2}^{eg}\right), \bar{e}_{q\,S} = \chi_{S}^{eg}\left(\bar{e}_{S}^{0} + RT_0 \ln \chi_{S}^{eg}\right) \end{array}\right\} \tag{20}$$

The values of the molar fraction of each gas are determined by the set of equations 8.

4.2) Physical exergy

The specific physical exergy at molar basis ($\bar{e}_{ph}$) of the mixture of gases can be calculated by the equation:

$$\bar{e}_{ph} = \bar{C}p\left[T - T_0 - T_0 \ln\left(\frac{T}{T_0}\right)\right] + \bar{R}\, T_0 \ln\left(\frac{p}{p_0}\right), \frac{kJ}{kmol_{eg}} \tag{21}$$

where: p_0 = pressure at the standard state (= 0.101 MPa)

Specific heat at constant pressure ($\bar{C}p$) is obtained through the following expression:

$$\bar{C}p = \sum_{i=1}^{n} \chi_i \bar{C}p_i, \frac{kJ}{kmol\,K} \tag{22}$$

To the dry air, the value of $\bar{C}p$ is obtained directly from [7], with subsystem inlet temperature.

To the exhaust gases:

$$\bar{C}p^{eg} = \chi_{CO_2}^{eg} \bar{C}p_{CO_2} + \chi_{SO_2}^{eg} \bar{C}p_{SO_2} + \chi_{O_2}^{eg} \bar{C}p_{O_2} + \chi_{N_2}^{eg} \bar{C}p_{N_2} + \chi_{S}^{eg} \bar{C}p_{S} \tag{23}$$

To the gaseous fuel (equation 24):

$$\bar{C}p^{eg} = \left(\begin{array}{l} \chi_{N_2}^{eg} \bar{C}p_{N_2} + \chi_{O_2}^{eg} \bar{C}p_{O_2} + \chi_{H_2}^{eg} \bar{C}p_{H_2} + \chi_{CO}^{eg} \bar{C}p_{CO} \\ + \chi_{CO_2}^{eg} \bar{C}p_{CO_2} + \chi_{H_2S}^{eg} \bar{C}p_{H_2S} + \chi_{S}^{eg} \bar{C}p_{S} + \chi_{ME}^{eg} \bar{C}p_{ME} \\ + \chi_{ET}^{eg} \bar{C}p_{ET} + \chi_{PR}^{eg} \bar{C}p_{PR} + \chi_{BU}^{eg} \bar{C}p_{BU} + \chi_{PE}^{eg} \bar{C}p_{PE} \end{array}\right), \frac{kJ}{kmol\,K} \tag{24}$$

The time rate of the total exergy ($\dot{E}$) of each gas mixture will be the sum of the chemical exergy and the physical exergy.

To the air:

$$\dot{E}^{a} = 0.03734 \alpha\, R_{as} \dot{V}\left(\bar{e}_{ch}^{a} + \bar{e}_{ph}^{a}\right), kW \tag{25}$$

To the exhaust gases:

$$\dot{E}^{eg} = 0.03734\, \alpha\, R_{eg}\, \dot{V}\left(\bar{e}_{ch}^{eg} + \bar{e}_{ph}^{eg}\right), kW \tag{26}$$

To the gaseous fuels (equation 27):

$$\dot{E}^{NG} = 0.04076\, \dot{V}\left(\bar{e}_{ch}^{NG} + \bar{e}_{ph}^{NG}\right), kW \tag{27}$$

5. Calculation of the water exergy

As the water does not suffer any sort of chemical transformation in any of the cycles (hot water, chilled water and cooling water cycles) its chemical exergy does not vary, so, only the physical exergy is determined (equation 28).

$$e_{ph_w} = h - h_0 - T_0(S - S_0), \frac{kJ}{kg} \tag{28}$$

And the total exergy:

$$\dot{E}_w = \dot{m}_w \, e_{ph_w} \,, kW \tag{29}$$

6. The heat exergy ($\dot{E}_Q$).

The heat exergy, which is lost through convection and radiation from the equipment, is calculated by:

$$\dot{E}_Q = \dot{Q}\left(1 - \frac{T_0}{T_{sa}}\right), kW \tag{30}$$

7. Exergy balance

Exergy balance is carried out on the system and on each subsystem aiming at evaluating their particularities. According to Figure 1:

a) To the whole system (equation 31):

$$\dot{E}_1 + \dot{E}_2 + \dot{E}_8 + \dot{E}_{10} + \dot{E}_{14} + \dot{E}_{15} = \dot{E}_3 + \dot{E}_5 + \dot{E}_9 + \dot{E}_{11} + \dot{E}_{12} + \dot{E}_{13} + \dot{E}_d \tag{31}$$

where: $\dot{E}_d$ = time rate of the exergy destroyed

b) To the gas micro-turbine subsystem (equation 32):

$$\dot{E}_1 + \dot{E}_2 = \dot{E}_3 + \dot{E}_4 + \dot{E}_{12} + \dot{E}_{dI} \tag{32}$$

c) To the heat exchanger subsystem (equation 33):

$$\dot{E}_4 + \dot{E}_7 + \dot{E}_{14} = \dot{E}_5 + \dot{E}_6 + \dot{E}_{13} + \dot{E}_{dII} \tag{33}$$

d) To the absorption chiller subsystem (equation 34):

$$\dot{E}_6 + \dot{E}_8 + \dot{E}_{10} + \dot{E}_{15} = \dot{E}_7 + \dot{E}_9 + \dot{E}_{11} + \dot{E}_{dIII} \tag{34}$$

The system and subsystems exergy efficiencies (η) are calculated according to the general concept of rate of obtained product and the supplied resource.

The goal of the cogeneration system is to obtain electric power from the turbine and a cooling effect of the water in the chiller. Then (equation 35):

$$\eta_{sistema} = \frac{\dot{E}_3 + \left|\dot{E}_Q - \dot{E}_8\right|}{\dot{E}_1 + \dot{E}_2 + \dot{E}_{14} + \dot{E}_{15}} \tag{35}$$

The gas micro-turbine subsystem aims the electric power output and exhausted heat in order to obtain cold. So it is possible to consider that the exergy of the exhaust gases is part of the objective of this subsystem (equation 36):

$$\eta_I = \frac{\dot{E}_3 + \dot{E}_4}{\dot{E}_1 + \dot{E}_2} \tag{36}$$

The goal of the heat exchanger is to heat the water that will supply heat to the chiller so that the chiller can obtain cold water out of the energy of the micro-turbine exhaust gases and out of the power of the hot water pump (equation 37):

$$\eta_{II} = \frac{\dot{E}_6 - \dot{E}_7}{\dot{E}_4 - \dot{E}_5 + \dot{E}_{14}} \tag{37}$$

The objective of the chiller is to obtain cold out of the energy supplied by the hot water and out of the power of the cooling fluid and weak solution pumps (equation 38):

$$\eta_{III} = \frac{\left|\dot{E}_Q - \dot{E}_{10}\right|}{\dot{E}_6 - \dot{E}_7 + \dot{E}_{15}} \tag{38}$$

3. Case study

The calculation is applied to the experimental conditions obtained from [1] and from the results of modeling [12]. The data requests from the cogeneration system are:

- Natural gas (NG) was the fuel used. It presents the following characteristics:
- Composition in % of the volume: CO_2=0.41; N_2=0.43; Methane=90.37; Ethane=6.53; Propane=1.87; Butane=0.35 and Pentane=0.04;
- Lower Heating Value (LHV) = 36,145 kJ/m^3 NG, at 20 ºC;
- NG consumption = 0.00261 m^3/s;
- System input pressure and temperature: p = 0.350 MPa and t = 30 ºC;
- Electric power output = 24.5 kW;
- Air pressure and temperature: p = 0.090 MPa and t = 30 ºC;
- Air excess coefficient = 8.5;
- Micro-turbine exhaust gases pressure and temperature: p = 0.0924 MPa and t = 275 ºC;
- Pressure and temperature of gases at the heat exchanger outlet: p = 0.091 MPa e t = 105 ºC;
- Pressure and temperature of the water at the chiller inlet and outlet (outlet and inlet of the exchanger): p= 0.30/0.25 MPa and t = 90.6/85 ºC;
- Pressure and temperature of the cool water at the chiller inlet and outlet: p = 0.30/0.25 MPa and t = 12.2/6.7 ºC;
- Pressure and temperature of the condensation water at the chiller inlet and outlet: p= 0.30/0.20 MPa and 29.4/36.8 ºC;
- Hot water flow = 1.953 kg/s;
- Chilled water flow = 1.385 kg/s;
- Power of the hot water pump = 0.85 kW;
- Power of the cooling fluid and weak solution pumps of the chiller = 0.60 kW.

The calculations were carried out according to the methodology described at the outset.

The volume and the molar fraction of each gas are shown in Table 1 and the enthalpies of the exhaust gases, air and natural gas in Table 2.

Table 1. Volume of gases, m^{3}N/m^{3}N GN

	CO_2	Theoretical air	N_2	$H_2O_{(v)}$	O_2	Total gases
Volume	1.11	10.26	68.90	4.33	16.16	90.50
Molar fraction	0.01226	–	76.13	0.0478	0.1785	–

Table 2. Enthalpy of the gases, kW

Natural gas		Air	Exhaust gases	
Enthalpy	LHV		Micro-turbine	Heat exchanger
0.052	92.59	1.30	66.74	20.76

As it can be seen, the enthalpy of the natural gas due to its pressure and temperature can be discarded in further calculations because it presents a very small value.

Once the values are determined, it is possible to carry out an energy balance on the gas micro-turbine subsystem in order to find out the losses to the environment. Based on Figure 1, it can seen that:

$$H_1 = H_{NG}\,; H_2 = H_{ha}\,; H_3 = H_{el}\,; H_4 = H_{eg}\,; H_{12} = H_Q$$

$$\therefore H_{NG} + H_1 = H_{el} + H_{eg} + \dot{Q}$$

where: H = time rate of energy
$\dot{Q}$ = heat loss to the environment

The used LHV at 25 °C (with is the calculation basis) is 35,475.24 kJ/m³GN.

$$H_1 = 35{,}475.24x0.00261 = 92.59kW$$

The generated electric power is 24.5 kW. According to the data:

$$92.59 + 1.30 = 24.5 + 66.74 + \dot{Q}$$

$$\dot{Q} = 2.62\ \mathrm{kW}$$

This value is appropriate for this kind of equipment, for it lies lower than 3% of the input energy.
The values of the specific enthalpy and entropy of the water in the Table [4] are:

$$h_0 = 104.92\,; h_6 = 379.74\,; h_7 = 356.16\ \mathrm{em}\ \frac{kJ}{kg}$$

$$s_0 = 0.3672\,; s_6 = 1.1927\,; s_7 = 1.1344\ \mathrm{em}\ \frac{kJ}{kg\,K}$$

For chilled water:

$$h_8 = 51.6367\,; h_9 = 28.42\ \mathrm{em}\ \frac{kJ}{kg}$$

$$s_8 = 0.1835\,; s_9 = 0.1018\ \mathrm{em}\ \frac{kJ}{kg\,K}$$

It is necessary to carry out an energy balance in the chiller to determine the cool water flow:

$$\dot{m}_6\, h_6 + \dot{m}_8\, h_8 + \dot{m}_{10}\, h_{10} + H_{15} = \dot{m}_6\, h_7 + \dot{m}_8\, h_9 + \dot{m}_{10}\, h_{11}$$

$$h_{10} = 123.49\,; h_{11} = 154.33\ \mathrm{em}\ \frac{kJ}{kg}$$

$$s_{10} = 0.4283\,; s_{11} = 0.5293\ \mathrm{em}\ \frac{kJ}{kg\,K}$$

where: $\dot{m}$ = mass flow

Replacing the values:

$$\dot{m}_{10} = 2.53\ \frac{kg}{s}$$

The exergy for each gaseous flow is shown in Table 3.

Table 3. Exergy of the gases

Exergy	Natural gas	Exhaust gases	
		Micro-turbine	Heat exchanger
Chemical (kJ/kmol)	904,959.71	248.50	248.50
Physical (kJ/kmol)	3,080.64	1,851.47	10.26
Total (kW)	96.60	18.52	2.40

The water exergies at the chiller inlet and outlet are displayed in table 4.

Table 4. Water enthalpy and exergy in the chiller, in kW

	Hot water		Cold water		Condensation water	
	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
Enthalpy	741.63	695.58	71.50	39.36	312.43	390.45
Exergy	56.67	44.16	2.01	3.57	0.91	2.79

The energy balance in the heat exchanger allows the calculation of the losses to the environment regarding this equipment:

$$H_4 + H_7 + H_{14} = H_5 + H_6 + H_{13}$$

$$\mathrm{H}_4 = 66.74\,\mathrm{kW}; \mathrm{H}_5 = \mathrm{H}_9 = 20.76\,\mathrm{kW}; \mathrm{H}_6 = 741.63\,\mathrm{kW}$$

$$\mathrm{H}_7 = 695.58\,\mathrm{kW}; \mathrm{H}_{14} = 0.85\,\mathrm{kW}$$

$$\mathrm{H}_{13} = 66.74 + 695.58 + 0.85 - 20.76 - 741.63$$

$$\therefore \mathrm{H}_{13} = 0.78\,\mathrm{kW}$$

The exergy transferred with the heat in the micro-turbine (considering T_m = 333 K) and in the heat exchanger (considering T_m = 323 K), respectively, will be:

$$\dot{\mathrm{E}}_{\mathrm{ch}\ 12} = \dot{\mathrm{Q}}_{12}\left(1 - \frac{\mathrm{T}_0}{\mathrm{T}_\mathrm{m}}\right) = 2.62\left(1 - \frac{298}{333}\right) = 0.27\,\mathrm{kW}$$

$$\dot{\mathrm{E}}_{\mathrm{ch}\ 13} = \dot{\mathrm{Q}}_{13}\left(1 - \frac{\mathrm{T}_0}{\mathrm{T}_\mathrm{m}}\right) = 0.78\left(1 - \frac{298}{323}\right) = 0.06\,\mathrm{kW}$$

The exergy balances of the system and subsystems allow the determination of the destroyed exergy (E_d) in each one of them. Table 5 shows the values of the destroyed exergy and of the exergy efficiency (η_{ex}) of the system and each subsystem.

Table 5. Destroyed Exergy ($\dot{\mathrm{E}}_\mathrm{d}$) and exergy efficiency (η_{ex})

	System	Micro-turbine	Heat exchanger	Chiller
$\dot{\mathrm{E}}_\mathrm{d}$ (kW)	66.83	52.76	4.4	9.67
η_{ex} (%)	26.72	45.83	73.71	11.84

Figures 2 and 3 (attached) show Sankey's and Grassman's diagrams of the micro-turbine, respectively.

4. Result analysis

The chemical exergy and the physical exergy of the air are very small if compared to the fuel exergy. So, the air exergy was not considered in this study.

The natural gas exergy is, preponderantly, a result of the chemical component. Because of the low temperature and pressure presented by the natural gas, the physical component only represents 0.3 % of the total. Generally, this should be the behavior, given that the Capstone micro-turbine inlet pressure is not higher than 4 bar and the temperature must not be elevated so that the specific volume does not rise to values that will affect the micro-turbine efficiency significantly.

The exergy of the exhaust gases presents a low value, both in the micro-turbine and in the heat exchanger. At the heat exchanger outlet it is convenient to have low exergy values, for this contributes to losses reduction for water heating in the heat exchanger. On the other hand, at the micro-turbine outlet it is necessary for the gas exergy to be higher, once this is the source

The exergy destruction is mainly concentrated in the gas micro-turbine subsystem (78.94% of the total destruction), fundamentally because of the combustion process that causes significant irreversibilities and that implies a low efficiency of this subsystem (45.83%).

Even though, the exergy destruction in the chiller is considerably high owing to heat transferring processes to an ending difference in temperatures. This can also be ratified by the value of the exergy efficiency (11.89% of this subsystem).

The system as a whole presents very low efficiency (26.72%). However it is logical for this sort of system due to the nature of the processes that are developed in it.

Attachments I and II show Sankey's and Grassman's diagrams of the analyzed system, respectively. When the diagrams are compared, observe that the scale of Grassman's diagram is half of Snakey's.

5. Conclusions

The micro-turbine can function with a wide range of fuels, grid connected or isolated locations and producing accepted levels of noise and emissions [1]. In this case, the cogeneration may be an interesting alternative regarding electricity, heat and cool supply in small buildings such as hospitals, schools, hotels, etc [15].

Today, these systems do not reach as high efficiencies as large capacity systems. As it can be seen, the micro-turbine is the main responsible for the irreversibilities. Soon, the advances in the development of new machines will significantly increase the global efficiency of the plant.

However, there are some projects of ceramic micro-turbines being developed. Simulations and tests indicate that the efficiency values may reach 40% [14].

6. References

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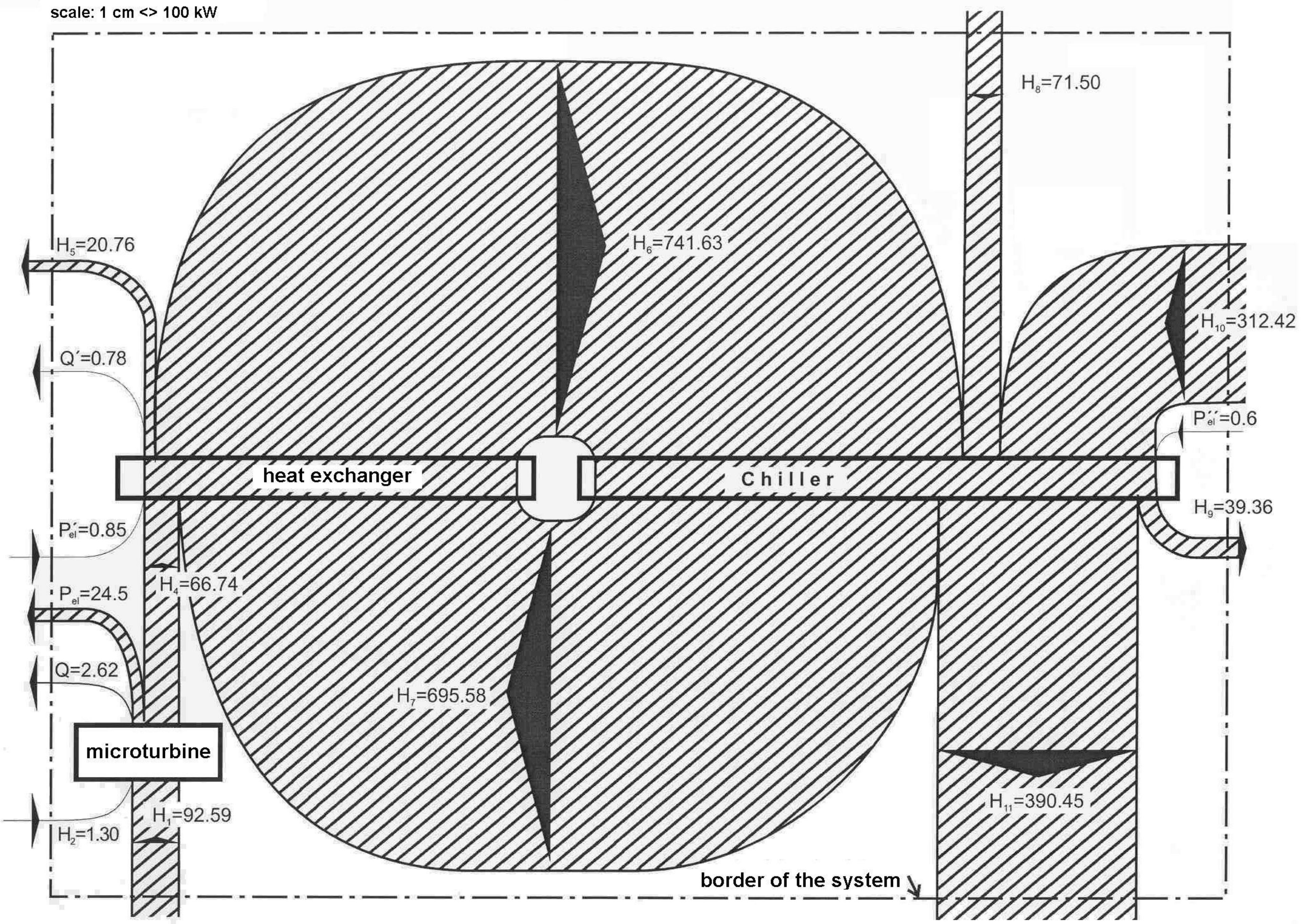

Figure 2. Sankey's diagram

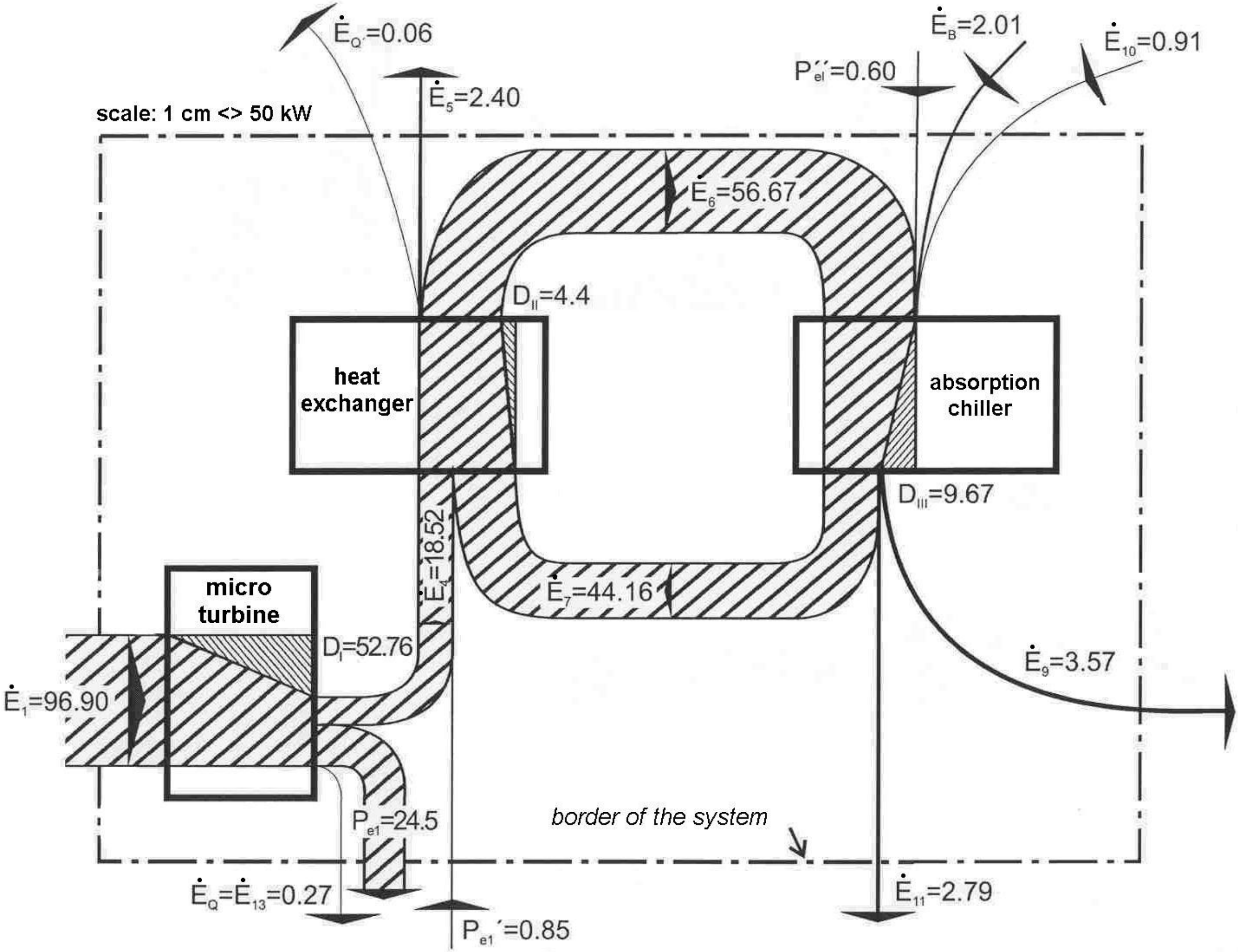

Figure 3. Grassman's diagram