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PERFORMANCE ANALYSIS OF A SMALL SCALE COGENERATION PLANT BASED ON A THERMAL FLUID HEAT EXCHANGER CONNECTING AN ABSORPTION CHILLER TO A MICROTURBINE

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Abstract.
A thermodynamic modeling is proposed to a small scale cogeneration system, consisting of a 3.5 RT water-ammonia absorption chiller connected to a 28 kWe natural gas microturbine. The proposed configuration changes the absorption chiller, replacing the prior natural gas burning system for thermal fluid as heat source. A computational algorithm is programmed for energetic and exergetic efficiency estimation, taking into account a thermal fluid heat exchanger connecting the absorption chiller to the microturbine. The coefficient of performance (COP) is also evaluated, particularly for different environmental temperatures and different chilled brine requirements. First results have shown the consistency of the proposed thermodynamic model and a good performance of the cogeneration system. The residual energy associated to the microturbine exhausting gases is already enough to supply the absorption chiller load, for electrical power output about 19 kWe, resulting in a highest heat recovery ratio. Otherwise, for electrical output above 19 kWe, a very little influence on the CHP efficiency was observed. The exergy efficiency decreases slightly for increasing environment temperature. In additional, the CHP efficiency and COP decrease strongly for environment temperature above 30 °C.

***Keywords**. Cogeneration, Natural gas microturbine, Ammonia-water absorption chiller, Second law analysis,*

1. Introduction

The natural gas availability and continuous increasing on the energy cost have recommended new technologies or the use of high efficiency equipment in cogeneration systems, for both electrical power and thermal loads generation.

Among this scenario, small scale cogeneration plants based on absorption chillers connected to natural gas microturbines are here proposed for electrical power and chilled brine generation. Absorption refrigeration should become economically attractive when there is a source of inexpensive heat energy at temperatures above 100ºC (Çengel & Boles, 2002). Recent results concerning a combination of a BrLi-water absorption machine 10 RT (35.2 kW_T) with a propane gas microturbine 24 kW_e have been reported by Ho *et. al.* (2003), showing favorable impacts, as an important conclusion, on the performance of the cogeneration system, when longer operating hours is observed. The real performance and behavior of ammonia-water absorption chiller was reported by Lazzarin et al. (1996). A thermodynamic modeling has been also proposed to a small scale cogeneration system, consisting of an absorption chiller 5 RT connected by a thermosyphon heat exchanger to a natural gas microturbine 28 kWe (Rossa & Bazzo, 2006). The proposed configuration changes the heat source of the absorption chiller, replacing the prior natural gas burning system. The exhausting gases from the microturbine are conducted through a proposed thermosyphon heat exchanger. The heat is directly transferred to the ammonia-water solution. No intermediate fluid was considered. The mass flow rate, the energy balance and the exergy destruction for the microturbine, heat exchanger and each component of the chiller were evaluated.

In this work, a thermodynamic modeling is presented for the small scale cogeneration plant to support simultaneous production of power output and chilled brine, this consisting of 45% of water and 55% of ethylene glycol. The small scale cogeneration plant consists of a 3.5 RT (12.3 kWT) ammonia-water absorption chiller connected to a 28 kWe natural gas microturbine. The proposed configuration changes the heat source of the absorption chiller, replacing the direct natural gas burning system by a heat exchanger connected to the microturbine, which warms a thermal fluid as an intermediate fluid. The warmed thermal fluid is continuously pumped to the chiller generator, providing the required ammonia separation. The thermodynamic simulation takes into account environmental changes. A computational implementation was carried out using the software EES, considering the ammonia-water properties calculation. First results show the reliability and a good performance of the proposed configuration. These values are used to analyze the influence of operation parameters on the absorption chiller performance coefficient, on the microturbine and especially on the combined heat power (CHP). The second law analysis was considered to determine the exergetic efficiency.

2. System Descriptions

The cogeneration system, consists of a 3.5 RT (12.2 kW_T) gas fired Robur Chiller connected through a cross flow heat exchanger to a 28 kW_e Capstone turbine model C30 LP. The residual energy of exhausting gases is recovered by a thermal fluid recovery system, for driving the absorption chiller. Thermal fluid consisting of an hydrotreated natural hydrocarbon is used for partially recovering the exhausting gases residual energy to take to the absorption chiller. Figure (1) shows the schematic cogeneration system in four subsystems, including the central natural gas supply, the microturbine, the heat recovery system and the absorption chiller.

The natural gas is supplied to the microturbine from central natural gas storage, at approximately 150 kPa. Electricity and hot exhausting gases at a temperature about 260°C are produced as energy output. The exhausting gases from the microturbine are directed into the heat exchanger, warming the thermal fluid used to transport energy to the absorption chiller, supplying the required heat load.

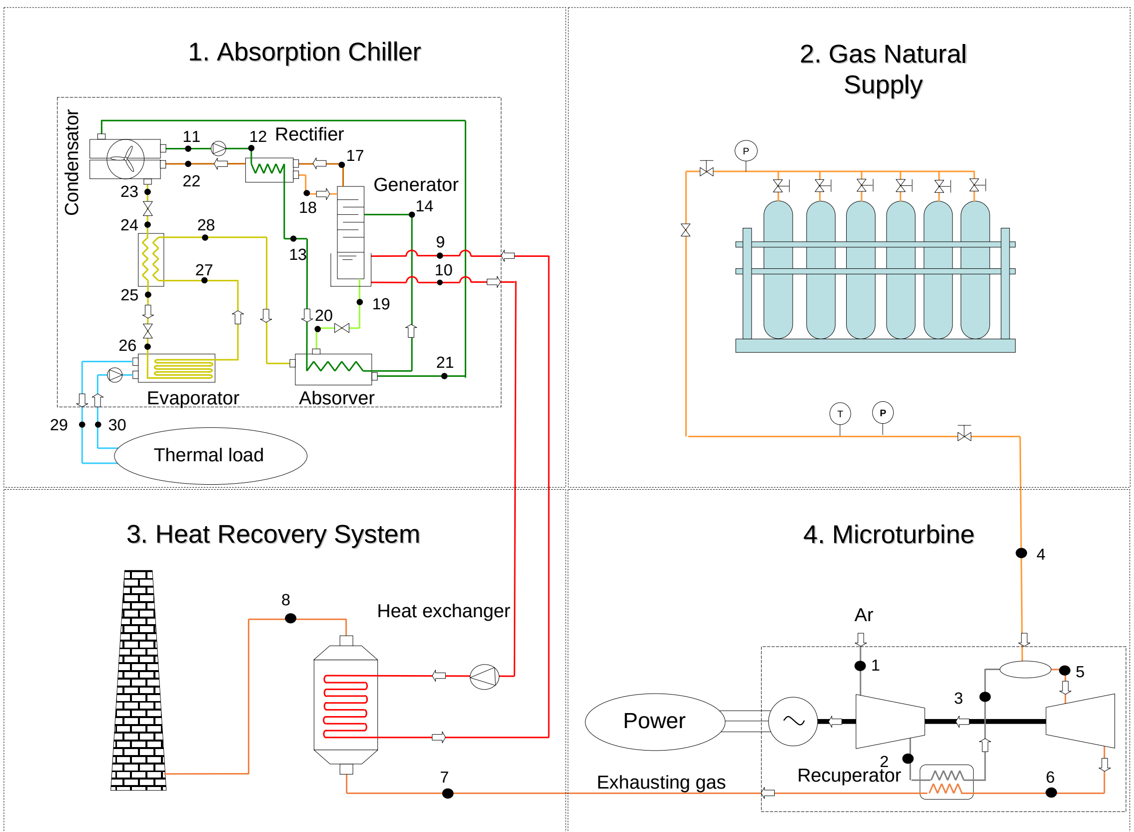

Figure 1. Diagram of the CHP configuration.

The thermal fluid coming from heat exchanger flows throughout a finned wall, transferring heat to the working fluid inside the chiller's generator. In the absorption chiller, a solution of ammonia and water is used as the working fluid, where ammonia is the refrigerant and water is the absorbent fluid. The ammonia-water solution in the generator is warmed, producing both vapor with a strong concentration of ammonia and a liquid solution with a low concentration of ammonia (weak solution).

The ammonia vapor flows into the rectifier, for purification. The hot and pressurized ammonia vapor exiting the rectifier enters the condenser where it is cooled and condensed by rejecting heat to the surroundings.

The liquid ammonia is then throttled to a lower pressure section by means of a first expansion valve and further it is previously cooled in a precooler. In the following, the liquid ammonia pressure is again reduced from 436 kPa to 262 kPa at temperatures about -12 °C by a second expansion valve, flowing to the evaporator, picking up heat from the cold refrigerated space, where a solution of water-ethylene glycol (brine) is chilled for attending the thermal power demand. The low pressure ammonia vapor then leaves the evaporator and enters the absorber, after cooling the liquid ammonia coming from the condenser (precooler).

The ammonia vapor enters the absorber, where it dissolves and reacts with water, in direct contact with the weak solution coming from the generator and through a restrictor. The absorption of ammonia vapor is an exothermic process. Heat is released during this process. The solution flows from the absorber to a portion of the condenser for cooling and complete absorption. The lower is the temperature, the higher the concentration.

At this point, now the liquid solution with high concentration of ammonia (strong solution) is pumped through a coil inside the rectifier and also another coil inside the absorber (GAX system), where it is preheated before entering in the generator. A hydraulically driven diaphragm pump is used to displace the strong solution to the high pressure level inside the generator.

3. Thermodinamic analysis

The mass conservation principle, the first and the second law of thermodynamics were applied to each component of the system. All components of the CHP were considered as a control volume with inlet and outlet streams heat transfer and work interactions. Concerning the mass conservation and the first law of thermodynamics, the governing equations are:

$$\sum \dot{m}_i - \sum \dot{m}_o = 0 \tag{1}$$

$$\sum \dot{m}_i \, z_i - \sum \dot{m}_o \, z_o = 0 \tag{2}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_o \, h_o - \sum \dot{m}_i \, h_i \tag{3}$$

where $\dot{m}$ is the mass flow rate [kg/s], z is the mass fraction of ammonia in the solution, h is the specific enthalpy [kJ/kg], $\dot{Q}$ is the heat transfer [kW] and $\dot{W}$ is the power output [kW], subjected to the following assumptions:

Dead state of T_0 = 25°C and P_0= 101.025 kPa;
Steady state operation;
Equilibrium thermodynamic at all points;
Equal pressure in heat exchange and generator;
Neglected pressure drop inside the tubes and components;
Refrigerant vapor concentration leaving the rectifier fixed to be 0.98 (point 22);
Difference between inlet and outlet cool water fixed to be 5°C.

The COP of the absorption system is defined as the desired output in the evaporator ($\dot{Q}_{cool}$) per required input in the generator ($\dot{Q}_{HR}$) and it can be calculated by:

$$COP = \frac{\dot{Q}_{cool}}{\dot{Q}_{HR}} \tag{4}$$

The heat recovery system ratio (R_{HR}) is obtained from the heat energy recovered and the thermal load ($\dot{Q}_{fuel}$) from the microturbine. Therefore, it is expressed as:

$$R_{HR} = \frac{\dot{Q}_{HR}}{\dot{Q}_{fuel}} \tag{5}$$

Defining the CHP efficiency [η_{CHP}] as the ratio of the cooling load plus electrical load to the input fuel energy to the CHP. Therefore, it is expressed as:

$$\eta_{CHP} = \frac{\dot{W}_e + \dot{Q}_{cool}}{\dot{Q}_{fuel}} \tag{6}$$

The second law was used to calculate the CHP efficiency based on exergy. Ignoring magnetic, electrical, nuclear, and surface tension effect the total exergy of system becomes the summation of physical (ψ^{ph}) and chemical (ψ^{ch}) exergies:

$$\psi = \psi^{ch} + \psi^{ph} \tag{7}$$

where the physical exergy is defined as:

$$\psi^{ph} = \left(h - h_0\right) - T_0\left(s - s_0\right) \tag{8}$$

where T_0 is temperature [°C], s is the specific entropy [kJ/kg K] and the subscript 0 means that the properties are evaluated at the dead state condition.

For the water-ammonia solution considered here, the chemical exergy is roughly calculated using the same equation presented by Misra et al. (2006):

$$\psi^{ch} = \frac{z}{M_{NH_3}} e^0_{ch,NH_3} + \frac{(1-z)}{M_{H_2O}} e^0_{ch,H_2O} \tag{9}$$

where M_{NH_3} and M_{H_2O} are the molar mass of ammonia and water, respectively, and e^0_{ch,NH_3} and e^0_{ch,H_2O} are the standard chemical exergy of ammonia and water, respectively, as given by Ahrendts (1980). The calculation procedure of the chemical exergy of various substances based on standard chemical exergy values of respective species has been discussed by Bejan *et al.* (1996), Szargut *et al.* (1988) and Ahrendts (1980).

The second law efficiency of the CHP (ε_{CHP}) is defined as the ratio of the useful exergy to exergy supplied to the CHP. Therefore, the exergetic efficiency of the CHP is the ratio of sum the electricity energy ($\dot{W}_e$) generated by microturbine and the required chilled brine exergy at the evaporator to the exergy of the heat source.

$$\varepsilon_{CHP} = \frac{\dot{W}_e + \dot{m}_{29}\left(\psi_{29} - \psi_{30}\right)}{\dot{m}_{fuel}\,\psi_{fuel}} \tag{10}$$

where the $\dot{m}_{fuel}$ and ψ_{fuel} are the mass flow rate and the exergy of the fuel, respectively.

4. Validation of computational code

The input data include the environmental conditions, electrical power, natural gas lower heating value, heat exchanger effectiveness, chilled brine flow rate, inlet and outlet temperatures. The values of pressure, temperature, enthalpy, entropy, mass concentration, mass flow rate, and exergy of the solution are calculated. The COP, exergetic efficiency of the chiller and CHP efficiency were also estimated.

The heat exchangers were computed on the basis of a given an effectiveness or setting a temperature difference, taking into account the effects of flow rates and propriety variations of brine and ammonia-water solutions. Table (1) reports the characterization of the heat transfer devices in the CHP.

Table 1 - Characterization of the heat transfer devices in the CHP.

Device	Type	Effectiveness [%]	ΔT [°C]
Microturbine			
Recuperator	air/air	0.70	
Absorption chiller			
Evaporator	brine/vapor	--	T_{30}-T_{29} = 5
Absorber	solution-solution	0.85	
Generator	thermal fluid /solution	--	T_{10}-T_{19} = 10
Condenser	air cooled	0.50	
Solution cooled absorber	air cooled	0.50	
Rectifier	vapor/solution	0.15	
Precooler	vapor/liquid	--	T_{24}-T_{27} = 32
Thermal recovery system			
Heat exchanger	thermal fluid/exhaust gas	0.75	

The exhaust gas flow rate were supplied by manufacturer and the turbine and compressor isentropic efficiency value were set to achieve the same electric power and efficiency values supplied by manufacturer Capstone – System Manual. Table (2) compares data predicted by program and experimental values supplied by the manufacturer as function of temperature environment. The errors obtained show the program is well set to predict the real microturbine operation.

Table 2 - Comparison of the manufacturer and the predicted internal conditions of microturbine (environment temperature at 35°C).

Parameters	Manufacturer value	Predicted data	Error [%]
Power load [kW_e]	24	24.4	1.67
Exhausting gas flow rate [kgs^{-1}]	0.267	0.274	2.62
Exhausting gas temperature [°C]	250	244	2.40
Microturbine efficiency [%]	24	23.3	2.92

In the absorption chiller, the mass flow rate and temperature were set for brine. Tentative values were assumed for the temperatures at the evaporator, the condenser, the generator and the absorber as well as for the concentration of the developed ammonia vapor. Condensation and evaporation temperatures with vapor concentration give high and low pressures. The strong and weak concentrations were evaluated with once the generator and absorber temperatures given. The code validation was done comparing the heat exchanged on each component predicted by program and the experimental values showed by Lazzarin *et al.* (1996). Table (3) shows experimental and predicted cooling capacity as a function of cooled brine outlet temperature at -10 °C.

Table 3 - Comparison of the experimental and the predicted internal parameters of the chiller for brine outlet temperature equal of -10 °C (Environment temperature at 35°C).

Parameters	Experimental results	Predicted data	Error [%]
Brine (water-ethylene glycol) [% / %]	45/55	45/55	-
Strong solution concentration [%]	29.8	35.3	18.46
Weak solution concentration [%]	5.7	6.6	15.79
Condensation pressure [kPa]	1517	1521	0.26
Evaporation pressure [kPa]	208	262	25.96
Strong solution flow rate [$kg\ s^{-1}$]	0.03	0.032	6.60
Weak solution flow rate [$kg\ s^{-1}$]	0.022	0.023	6.15
Vapor mass flow rate [$kg\ s^{-1}$]	0.008	0.0109	2.37
Generator heat flow rate [kW]	20.8	20.8	0.00
Absorber heat flow rate [kW]	19.9	19.6	- 1.51
Evaporator heat flow rate [kW]	9.3	9.3	0.00
Condenser heat flow rate [kW]	10.2	10.5	2.94
COP [-]	0.44	0.45	2.22

5. Result and discussion

In this section, performance data for the system will be presented for steady state conditions. The first two analyzes were done setting environment temperature at 25°C and brine temperature at -6°C, being analyzed the influence of microturbine electrical output in the CHP parameter. The third analyze shows the comportment of the CHP and the absorption chiller with brine temperature change. In the last one, the electrical output and brine temperature were set at 26.5 kW_e and -6 °C respectively, being showed the influence of environment temperature in the CHP and the absorption chiller parameters.

Comparing the results shows in Fig. (2), can be observed that both exhaust gas and thermal fluid temperature rise until microturbine electrical output achieve 19 kW_e, resulting in an increase in the heat transferred to thermal fluid and consequently to generator. As of 19 kW_e the higher absorption chiller cooling capacity is achieved, and heat flux transferred to thermal fluid become constant. As result, exhaust gas temperature increase strongly while thermal fluid temperature is established.

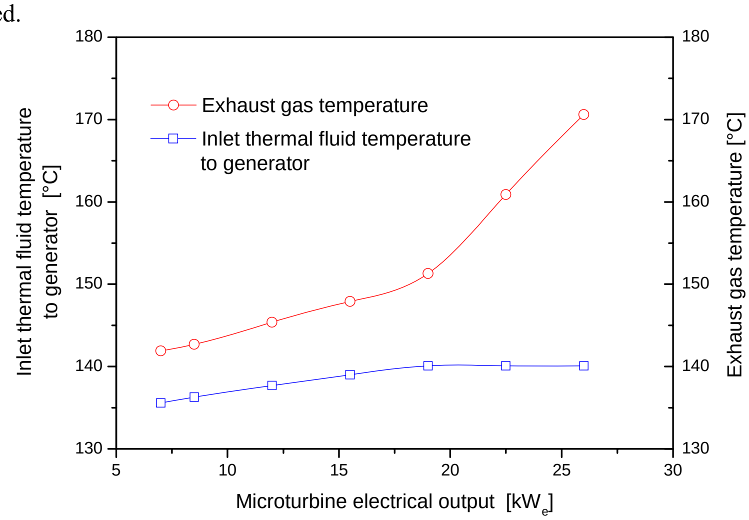

Figure 2 - Changing on the exhaust gas and thermal fluid temperature for microturbine electrical output variation.

The corresponding impact on the heat recovery system ratio and the CHP and exergetic efficiency can be seen from Fig. (3). The highest heat recovery system ratio occurs at 19 kW_e electrical output when the nominal thermal load of generator is established, after this point the heat flux recovery keeps constant and the heat recovery system ratio begins decrease. The CHP efficiency increase strongly until 19 kW_e, where heat recovered is the maximum and as of this value has a slight increase as result of the rise in the electrical output. A raise in the exergetic system efficiency was seen as the microturbine electric output increases.

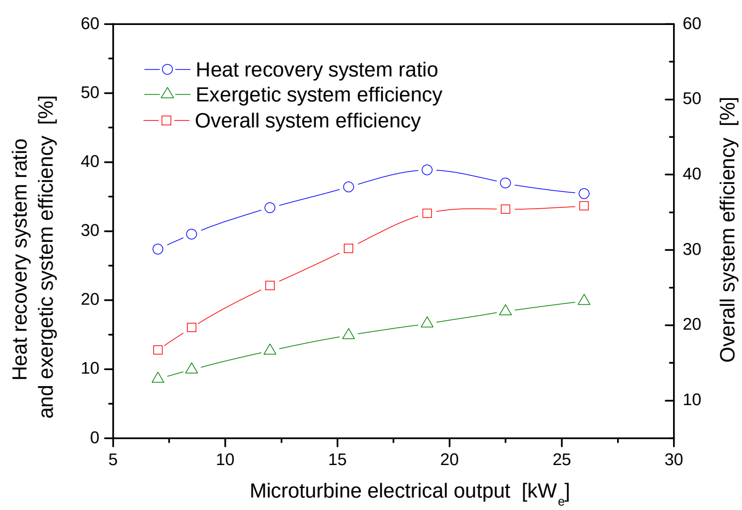

Figure 3 – Changing on the heat recovery system ratio, CHP and exergetic efficiency for microturbine electrical output variation.

Figure (4) shows the COP, CHP and exergetic efficiency variations for different chilled brine outlet temperatures. The COP and CHP increase slightly for chilled brine outlet temperature increasing. On your side, as expected, the exergetic efficiency is almost constant.

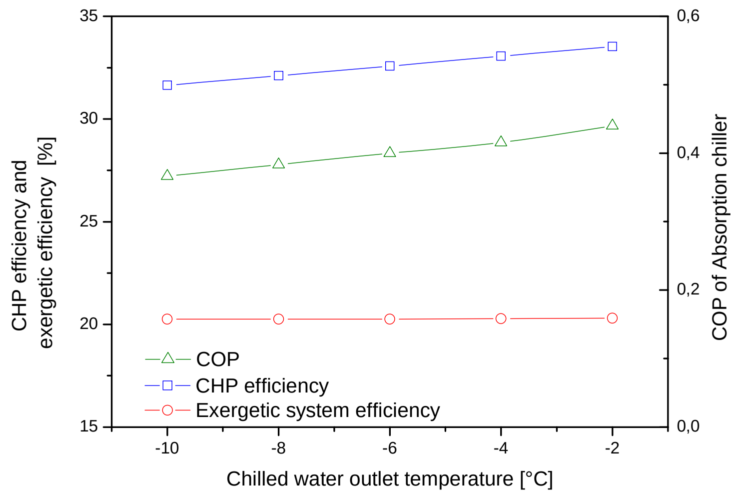

Figure 4 - Variation on the CHP and exergetic efficiency for chilled water outlet temperature changing.

Figure (5) shows the influence of the environment temperature on the CHP, microturbine and exergetic efficiencies, as well as on the COP of the absorption chiller. The higher is the environment temperature, the lower the values for all efficiencies and COP of the absorption chiller. Above 30 °C, it is clear a strongly reduction on the COP and CHP efficiency.

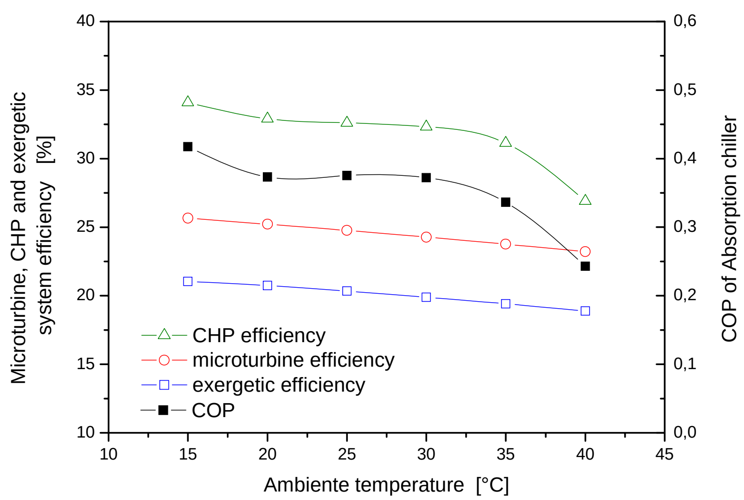

Figure 5 - Variation on the COP, CHP, microturbine and exergetic efficiency for environment temperature.

6. Conclusions

An absorption chiller connected by heat recovery system to a microturbine was analyzed as a promises alternative for cooling water and power generation. The predicted values of microturbine and absorption chiller codes shown low errors as compared with microturbine manufacture (Capstone – System Manual) and experimental results reported by Lazzarin *et al.* (1996). First results have shown the consistency and a potential tool of the proposed thermodynamic model for thermal system analysis, as well as a good performance of the proposed cogeneration system. It was clear that when microturbine electrical output achieves 19 kW_e, the residual energy in the microturbine exhausting is already enough to supply the absorption chiller load, and for this power the heat recovery system ratio is the highest. Concerning the changing in the electrical load until 19 kW_e, a high influence on the CHP efficiency was observed, while for electrical output up to 19 kW_e, the influence on the CHP efficiency was very little. A slightly increase in the COP and the CHP efficiency as the brine outlet temperature raise was observed, when the exergetic efficiency become keep constant. In the analyze influence of environment temperature in the CHP, was observed that to environment temperatures up at 30 °C the COP and CHP efficiency is strongly prejudiced. Also, for changing on the environment temperature, a same trend for both the COP of the absorption chiller and the CHP efficiency was observed.

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