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EXPERIMENTAL RESULTS AND PERFORMANCE PARAMETERS FOR ASSESSMENT OF A NATURAL GAS SMALL SCALE COGENERATION PLANT

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Abstract. *This work presents experimental results and performance parameters for assessment of a small scale cogeneration plant. The present system consists of a 28 kW_e natural gas microturbine, a heat recovery steam generator, ammonia-water absorption chillers of 5 RT (17.6 kW_t) for refrigeration and 3.8 RT (13.3 kW_t) for freezing purposes, fan coils and a cooling storage system. The absorption chillers were previously powered by natural gas burned direct upon the generator. In the current system, the residual heat of the microturbine exhaust gases is recovered into the HRSG, where it is supplied to the chiller's generator whether by hot water or steam. The performed tests were focused on the 3.8 RT absorption chiller, which can reach temperatures down to -10°C. Four modes of operation were tested. The results obtained for the best mode were 23% and 31% for the microturbine and cogeneration efficiency, respectively, and 0.49 for the COP. The electrical power generated was 25 kW and the cooling power produced was 10 kW, although the minimum temperature reached was +5°C. The cooling power and the COP increased with higher output electrical power and HRSG pressures. The exhaust gases temperature downstream of the HRSG still showed potential for cogeneration. The assessment parameters found were satisfactory, although the cogeneration system needs further improvements in order to enhance the global efficiency.*

Keywords*: Small scale cogeneration plant, natural gas microturbine, absorption chiller, energy saving.*

1. INTRODUCTION

The search for efficient energy conversion systems has increased lately due to the limited availability of fossil fuels and the concern about global warming. In Brazil, the need for expansion of the national electric grid, the constraints in building new hydroelectric power plants by social and ambiental matters and the large offer of natural gas coming from South America countries, adds for the encouragement of researches in new technologies of energy conversion systems.

The cogeneration systems related with combined cooling, heating and electrical power (CCHP) have been used in quite few commercial buildings via absorption chillers, using "waste" heat from microturbines or rejected heat from different industrial processes. The use of waste heat to drive absorption chillers is an alternative way to cut operating costs, whether for air conditioning or freezing purposes.

A small scale cogeneration plant with high performance parameters is essential for its operational and economic feasibility. Besides the particular benefits for the owner of these kinds of systems, the use of this technology reduces the total electrical demand from the national grid and, consequently, the need of expansion of the grid, what brings investment savings for the country. Cogeneration can provide significant increase of energy use, distributed generation and should be considered as an alternative technology for stores that demands electricity, cooling and heating.

Recent results concerning experimental and theoretical investigation have been reported in order to confirm the technical feasibility and to show the potential to achieve high global performance with small scale cogeneration systems using absorption chillers (Medrano *et al.*, 2006, Takeshita *et al.*, 2005, Rucker and Bazzo, 2004). Sun, Z. (2008) addressed his work to commercial buildings, providing electricity and cooling/heating using residual heat. Results showed that the primary energy saving of the considered cogeneration system was 37% more efficient compared to conventional separate systems. More recently, Rossa and Bazzo (2009) reported a thermodynamic modeling and Second Law analysis for a small scale cogeneration system. An increase of about 67% of the energy use was found. The corresponding COP of the absorption chiller was calculated around 0.49, in this case assuming the actual heat removed from the microturbine exhaust gases and cold water temperature set to +5ºC. Experimental results, but using hot water as an intermediate working fluid, have shown a COP around 0.30 (Bazzo *et al*, 2008).

This work reports experimental data of an existing small scale cogeneration plant in operation at LabCET/UFSC. The experimental results are reported concerning the temperatures, the heat removed from the exhaust gases, the cooling power produced, the COP and the efficiencies of the microturbine and the cogeneration plant. Some assessment parameters are presented and discussed for different modes of operation. Results are presented for two distinct ways of supplying heat to a -10°C absorption chiller: through hot water or through saturated steam.

2. DESCRIPTION OF THE SMALL SCALE COGENERATION PLANT

Test facilities consist of one system to store and supply fuel to the plant, one 28 kW_e microturbine working as prime mover, one heat recovery steam generator, one 5 RT (17.6 kW_t) and one 3.8 RT (13.3 kW_t) ammonia-water absorption chillers, for refrigeration and freezing purposes, respectively, two fan coils for air conditioning, one cooling storage system, one data acquisition system and several sensors for temperature, pressure and flow rate measurements. Figure 1 shows the schematic diagram of the cogeneration system.

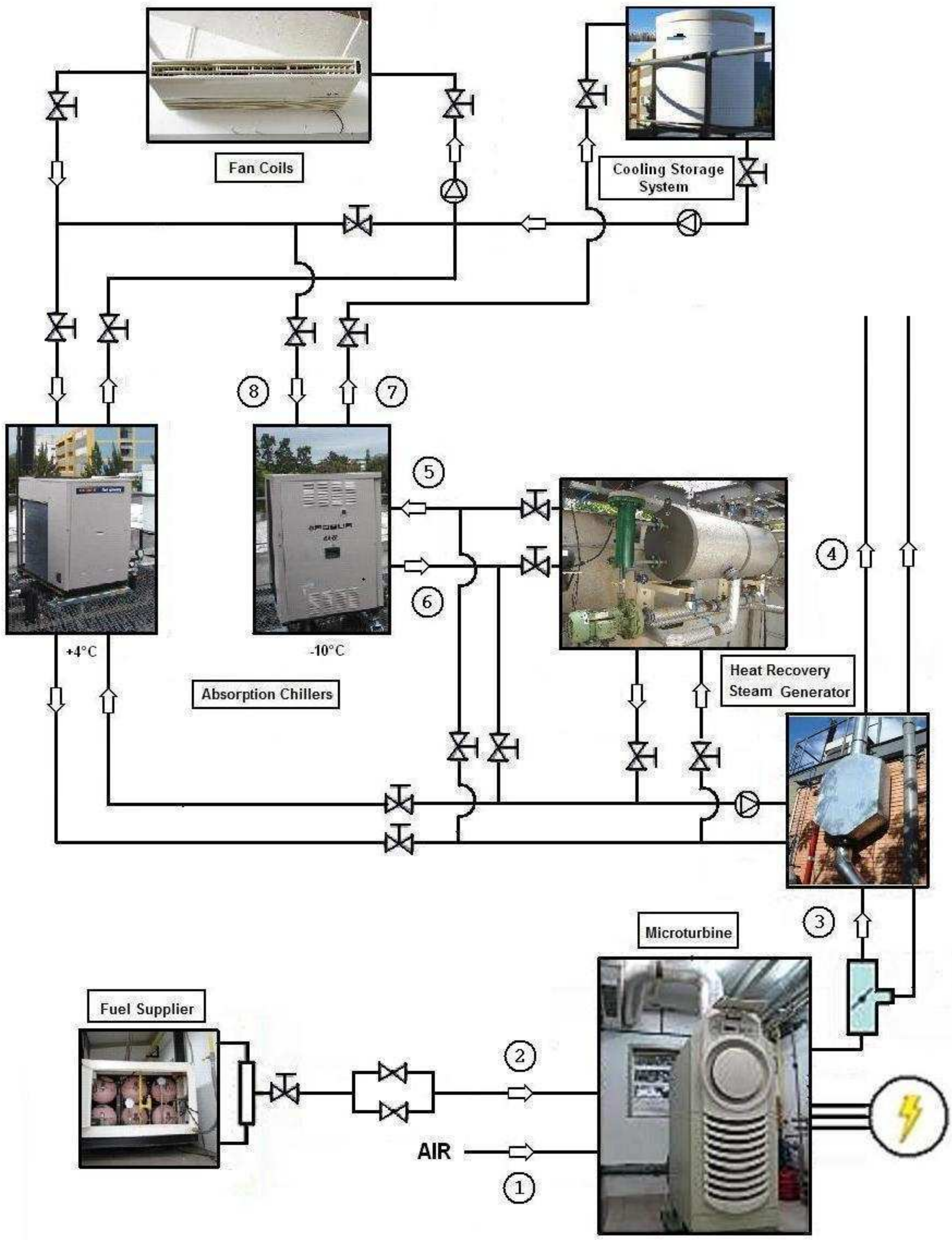

Figure 1. The small scale cogeneration plant scheme.

The prime mover is a Capstone Microturbine Model 330 which includes a compressor, a heat recuperator, a natural gas combustor, a turbine and a generator. The rotating components are mounted on a single shaft supported by air bearings and spin at up to 96000 RPM. The generator is cooled by the airflow into the microturbine. The output of the generator is variable voltage and variable frequency AC power. Natural gas is the fuel used on the tests, which is supplied by manifold cylinders. Operating at ISO condition, the microturbine produces 28 kW of electrical power and 85 kW usable heat from the exhaust gases with temperatures around 270 °C (Capstone, 2001).

Absorption chillers are single-block brine-chilling units equipped with an air-cooled condenser. The absorption cooling cycle is based upon a solution of water and ammonia. Both chillers that compose the plant had their power system modified from a direct firing system to water jackets enclosing the generators. The jackets were properly designed and installed in place of the chiller's combustion chamber to provide hot water circulation around the external generator fins. One absorption chiller is a 5 RT Robur ACF 60-00 that can reach temperatures down to +4°C (Robur, 2008), for air conditioning purposes, and is powered by hot water. The other chiller is a 5 RT Robur ACF 60-00LB, that can reach temperatures down to -10°C (Robur, 2008), for freezing purposes, and it is powered whether by hot water or steam. The cold water pipeline uses a brine solution of monoethylene glycol 25% and water 75%, in mass, providing freezing temperatures below -10°C.

The HRSG system consists of one diverter valve, one cross-flow heat exchanger, one steam generator and one pump. The residual heat is recovered into the HRSG. For the chillers powered by hot water, the water pipeline is heated in the heat exchanger and it flows direct into the generators and no automatic device is used to control the rate of residual heat recovered.

For the -10°C chiller powered by steam, the hot water pipeline is linked to a steam generator that supplies heat to the generator while the steam condenses. In this case, the heat recovered rate is controlled by the pressure in the HRSG. The heat increases the pressure in the HRSG until a maximum value set. At this moment the diverter valve shifts the exhaust gases stream, releasing it to the atmosphere without taking any advantage of it (Fig. 1). The heat consumed by the chiller's generator decreases the pressure. When the gauge pressure reaches a minimum value, the diverter valve shifts the exhaust gases stream, directing them back to the heat exchanger. This cycle repeats continuously. This system was developed in order to avoid overpressure in the HRSG and to supply the required heat to provide separation of ammonia from the ammonia water solution, inside the chiller's generator.

An air conditioning system, consisting of two 2.5 RT Aquastylus Chilled Water Fan Coil Trane, was installed in the laboratory to use the cooling power produced. A cooling storage system was also installed for the same purposes.

Measurements of temperatures, pressures and flow rate in strategic points are used to obtain thermodynamics properties for further evaluation.

3. THERMODYNAMICS EQUATIONS

3.1. Thermodynamic Model

The model adopted in the present analysis is composed of three subsystems: the microturbine (MT), the heat recovery steam generator (HRSG) and the absorption chiller (CH). This work focuses on the assessment of the subsystems and the whole cogeneration system. Figure 2 presents the model adopted.

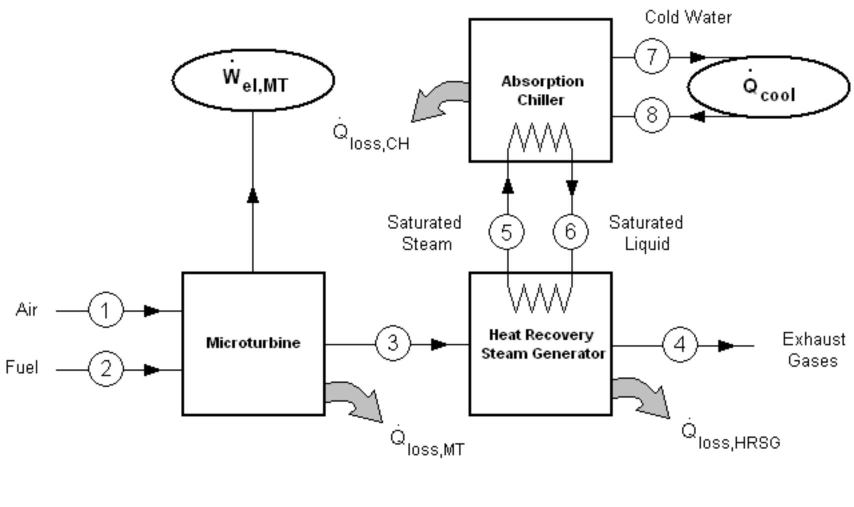

Figure 2. Thermodynamic model of the small scale cogeneration plant

All the thermodynamic properties were obtained from Engineering Equation Solver (Klein and Alvarado, 2008).

3.2. Performance Parameters

The cogeneration system is evaluated accordingly to some performance parameters in order to identify room for improvements. Based on the First Law of Thermodynamics, the following parameters are chosen for evaluation of the subsystems microturbine (MT), heat recovery steam generator (HRSG) and absorption chiller (CH), as well as the cogeneration system (cog):

$$\eta_{MT} = \frac{\dot{W}_{MT}}{\dot{m}_2 LHV_{NG}} \tag{1}$$

$$\eta_{HRSG} = \frac{\dot{Q}_{gen}}{\dot{Q}_{HRSG}} = \frac{\dot{m}_5 \cdot cp_{hw}\left(T_5 - T_6\right)}{\dot{m}_3 \cdot cp_{eg}\left(T_3 - T_4\right)} \tag{2}$$

$$\eta_{cog} = \frac{\dot{W}_{MT} + \dot{Q}_{cool} - \sum \dot{W}_i}{\dot{m}_2 LHV_{NG}} \tag{3}$$

$$COP = \frac{\dot{Q}_{cool}}{\dot{Q}_{HRSG}} = \frac{\dot{m}_7 \cdot cp_b\left(T_8 - T_7\right)}{\dot{m}_3 \cdot cp_{eg}\left(T_3 - T_4\right)} \tag{4}$$

where η is the First Law efficiency, $\dot{W}_{MT}$ is the output power of the microturbine, $\dot{m}_k$ is the mass flow rate for the k_{th} point, LHV_{NG} is the lower heating value of natural gas, $\dot{Q}_{gen}$ is the heating power provided to the chiller's generator, $\dot{Q}_{HRSG}$ is the heating power provided to the HRSG, $\dot{Q}_{cool}$ is the cooling power produced, *cp* is the specific heat, and the subscripts *hw*, *eg* and *b* mean hot water, exhaust gases and brine, respectively, T_k is the temperature for the *k*th point, $\dot{W}_i$ is the electrical power related to the brine and ammonia-water pumps and *COP* is the coefficient of performance of the absorption chiller. The lower heating value was calculated based on the composition of natural gas from SCGÁS Co. and found 47308 kJ/kg. The electrical power consumed by the pumps ($\dot{W}_i$) is 0.93 kW. The pumping power referred to the hot water pipeline was not considered in this work.

COP's evaluation comprises the efficiency for cooling power production of all the processes of heat transfer from the exhaust gases to the chiller's generator. HRSG's efficiency is just applicable when operating with hot water, as the temperatures T[5] and T[6] are the same for steam generation.

4. EXPERIMENTAL RESULTS AND ANALYSIS

The tests were focused on the Robur ACF 60-00LB chiller (-10°C). Different modes of operation were tested concerning to chiller's heat source, electrical output power and the devices that consume the cooling power produced. Four modes of operation are reported:

a) Mode 1 – Steam, 25 kW_{el} and fancoils;
b) Mode 2 – Steam, 25 kW_{el} and cooling storage system;
c) Mode 3 – Steam, 20 kW_{el} and cooling storage system;
d) Mode 4 – Hot water, 10 kW_{el} and cooling storage system.

4.1. Mode 1 – Steam, 25 kW_{el} and fancoils

In Mode 1, the absorption chiller was powered by steam. The test was carried out for maximum and minimum steam gauge pressure set to 6 bar and 2.7 bar respectively. The maximum pressure was set high enough to allow cogeneration plant to reach the steady state condition, aiming no use of the heat recovery rate control device. The fancoils were designated to consume the cooling power.

Figure 3 shows the temperature profiles along time for the thermocouples T[3], T[4], T[5], T[7], T[8] placed at the points indicated in Fig. 1 and Fig. 2:

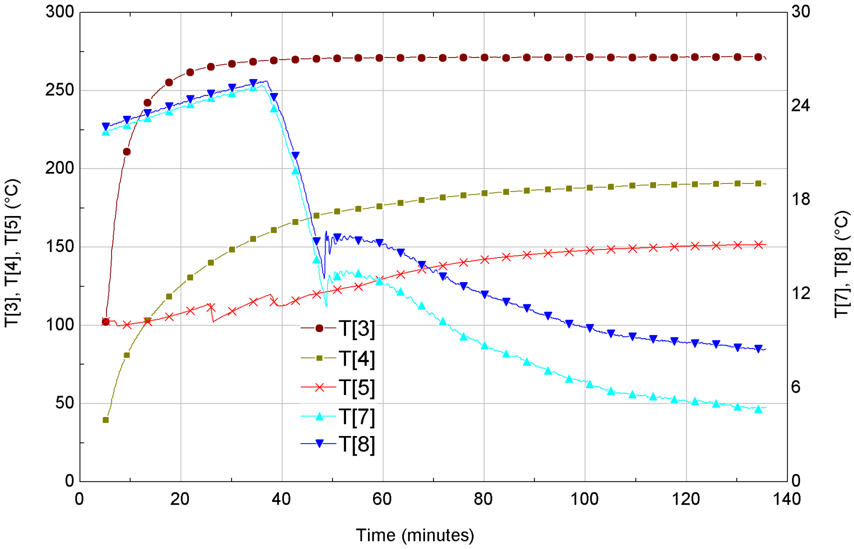

Figure 3. Thermal behavior for Mode 1

In the first 50 minutes, some adjustments were done to the plant to certify a proper operation. Air is purged from the HRSG, which explain drops in T[5]. Changes were done to the brine pipeline in order to set the fancoils to consume the cooling power produced (drops in T[7] and T[8]). After 135 minutes the system reaches the steady state condition. The minimum temperature reached was about +5°C. The brine temperature did not reached a lower value due to high cooling power consumption by the fan coils. The gauge pressure in the HRSG stabilized at approximately 4 bar, generating steam around 152°C.

Figure 4 presents the heat recovered and the cooling power for Mode 1.

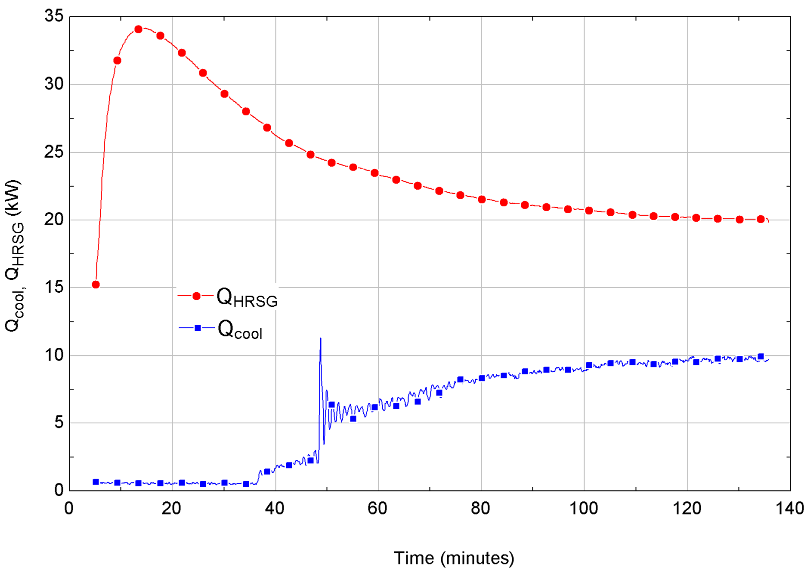

Figure 4. Heating power recovered and cooling power produced for Mode 1

When the microturbine starts, the exhaust gases is still in a low temperature and little heat is removed from the exhaust gases. As they become hotter, the heat recovered increases, reaching a maximum value. From that point on, the water pipeline temperature goes up and the temperature difference between the streams became smaller, making the heat recovered to decrease. By minute 40th, the temperature of the steam reaches 120°C (Fig. 3), and the system starts to produce cooling power. That is the minimum temperature required for separation of ammonia from ammonia-water solution. The maximum cooling power observed was 10 kW, lower than the 13.3 kW nominal cooling power in case of direct firing burning system. At the steady state condition, the COP was found 0.49. For this mode, efficiencies of about 23% and 31% for the microturbine and the cogeneration plant have been found, respectively

4.2. Mode 2 – Steam, 25 kW_{el} and cooling storage system;

In Mode 2, the cooling power is stored in the cooling storage system. The test was carried out for maximum and minimum gauge pressure set to 2.9 bar and 2.7 bar respectively. Figure 5 shows the temperature profiles for the thermocouples T[3], T[4], T[5], T[7], T[8] placed at the corresponding points indicated in Fig. 1 and Fig. 2

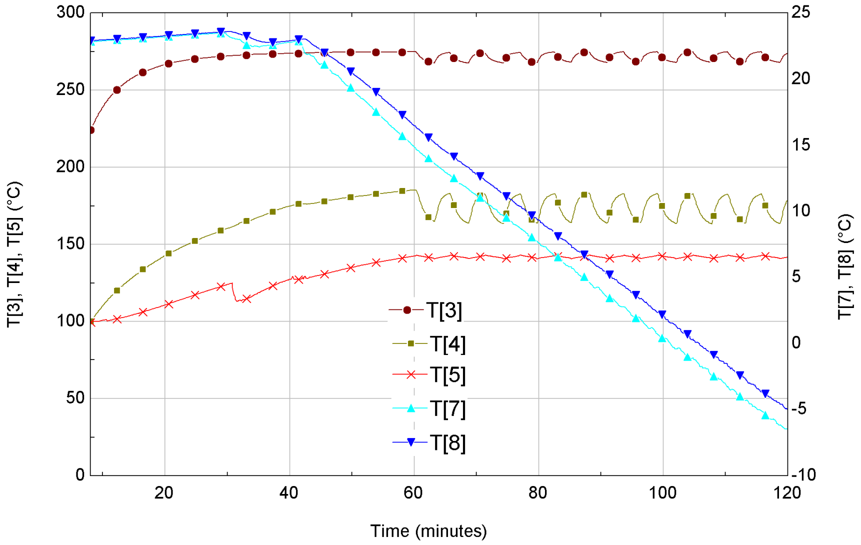

Figure 5. Thermal behavior for Mode 2

As shown in Figure 5, the exhaust gases temperatures (T[3] and T[4]) and the HRSG temperature (T[5]) increase until 60 minutes of the operation. From that point on, the diverter valve starts to control the heating power recovered. T[5] is kept between 141°C and 143°C, the corresponding saturation temperatures at the minimum and maximum gauge pressures set, respectively. T[3] and T[4] present cyclical behavior due to the redirecting of the exhaust gases between the chimneys (Figure 1). The outlet brine temperature, T[7], drops from 23°C to -6.5°C. The steady state is not fully reached because the cooling power is not being entirely stored in the cooling storage system. For this reason, the brine temperatures have a tendency to keep in decreasing and do not stabilize.

Figure 6 shows the heating power supplied to the HRSG and the corresponding cooling power produced by the chiller.

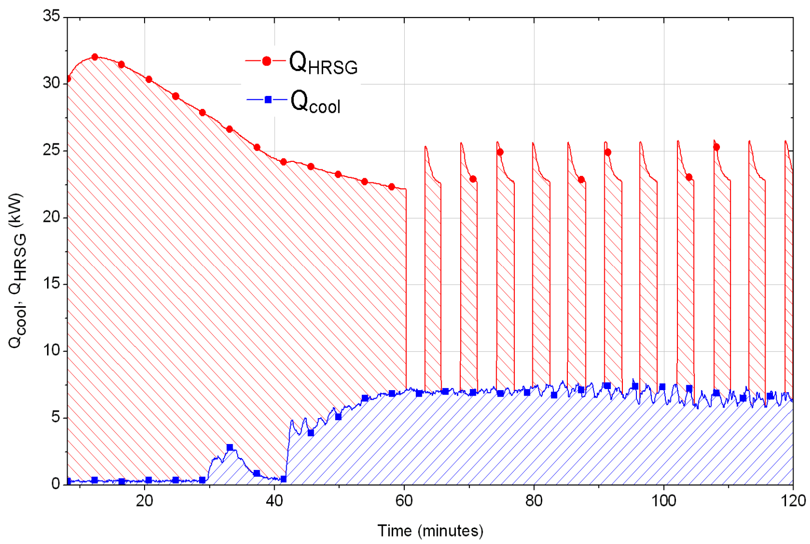

Figure 6. Heating power recovered and cooling power produced for Mode 2

The cogeneration plant takes about 60 minutes to reach the full operation in this mode. During the full operation, the heating power supplied is intermittent. In the second hour, from 60 up to 120 min, the total supplied heating power was calculated to be approximately 11.3 kW (see shadow area below Q_{HRSG} in Figure 4), producing an equivalent cooling power of about 6.9 kW. These results lead to a potential COP of 0.61.

A large heating power is lost to the atmosphere due to the low cooling storing capacity of the plant. Modifications are in progress to provide suitable cooling power consumption. Currently, efficiencies of about 24% and 30% to the microturbine and the cogeneration plant have been found, respectively.

4.3. Mode 3 – Steam, 20 kW_{el} and cooling storage system;

In Mode 3, the output power was set to 20kW_{el}. The test was carried out for maximum and minimum gauge pressure set to 2.9 bar and 2.7 bar respectively. Figure 7 shows the temperature profiles for the thermocouples T[3], T[4], T[5], T[7], T[8].

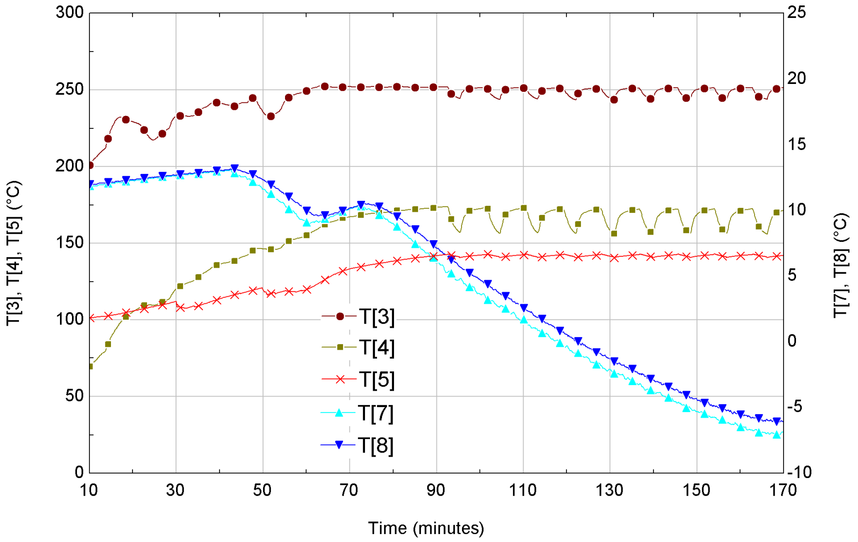

Figure 7. Thermal behavior for Mode 3

The thermal behavior for Mode 3 is similar of Mode 2 (Fig. 5), except for the lower temperatures of the exhaust gases, T[3] e T[4], due to the lower output power. The brine temperature, T[7], drops from 13°C to -7°C

Figure 8 shows the heating power supplied to the HRSG and the corresponding cooling power produced by the chiller.

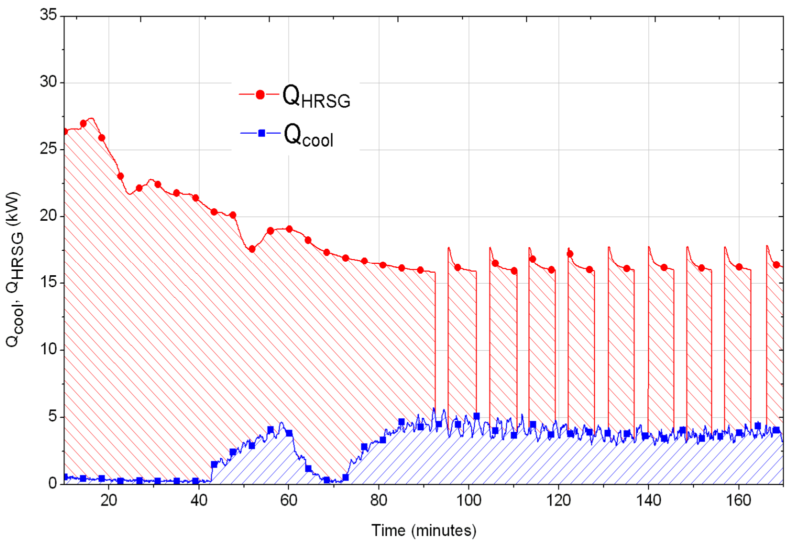

Figure 8. Heating power recovered and cooling power produced for Mode 3

Approximately 90 minutes was demanded to reach full operation in this mode. During the full operation, from 90 up to 170 min, the total supplied heating power was calculated around 10.8 kW (see shadow area below Q_{HRSG} in Figure 6), producing an equivalent cooling power of about 3.9 kW. These results lead to a potential COP of 0.36. Efficiencies of about 22% and 26% to the microturbine and the cogeneration plant have been found, respectively.

4.4. Mode 4 – Hot water, 10 kW_{el} and cooling storage system

In Mode 4, hot water is used to power the chiller's generator. The output power was set to 10 kW_e. Tests with higher output power were not possible to perform due to cavitation problems in the hot water pump. In this mode, the hot water thermocouples (T[5] and T[6]) measure the inlet and outlet temperatures in the HRSG. Figure 9 shows the temperature profiles for the thermocouples T[3], T[4], T[5], T[6], T[7], T[8].

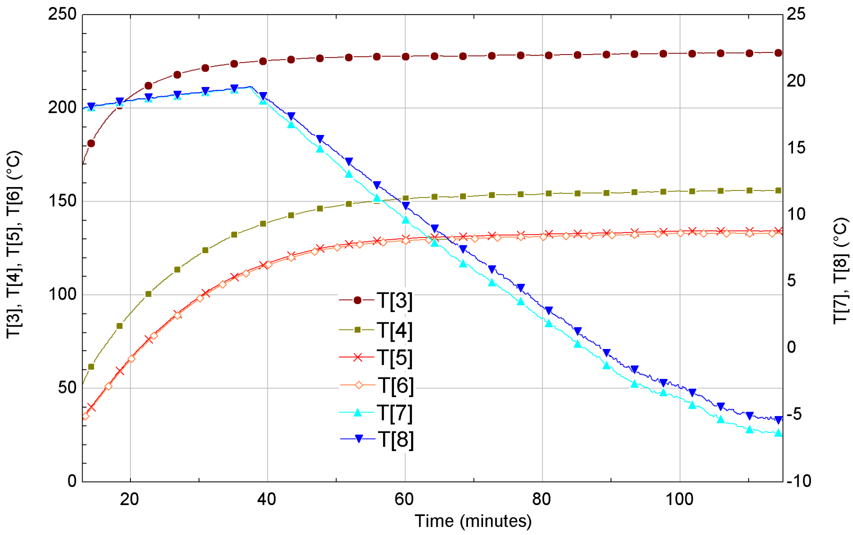

Figure 9. Thermal behavior for Mode 4

Working with hot water, no automatic devices is used to control the heat recovered rate. The temperature of the hot water pipeline (T[5]) stabilizes at approximately 130°C. It should be noted that T[6] (return of hot water from the chiller's generator) is not much different from T[5]. The minimum temperature obtained was -6°C at the end of the test.

Figure 10 shows the heating power recovered and the cooling power consumed. In addition, the heat supplied to the generator can be calculated in light of the measures of T[5], T[6] and m[5], and is also shown in Fig.10.

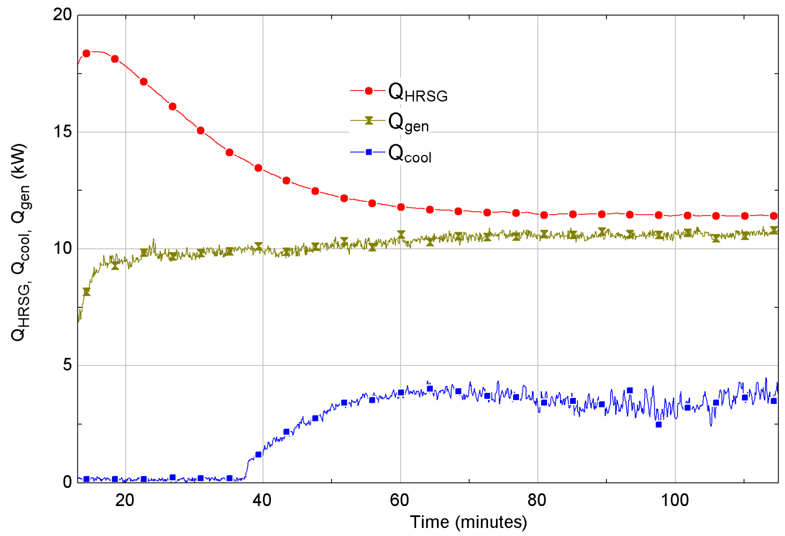

Figure 10. Heating supplied to the HRSG and to the generator and cooling power produced for Mode 3

In the steady state condition, the cooling power produced and the heating power recovered was about 4 kW and 11 kW, respectively. These values are lower than the results presented for Modes 1, 2 and 3 due to the lower mass flow rate of the exhaust gases for the output power of 10kW_{el}. Efficiencies of about 13%, 16% and 92% for the microturbine, the cogeneration plant and the HRSG have been found for this mode, respectively.

4.5. Results for measurements and performance parameters

Measurements of mass flow obtained in the tests and the related specific heats are presented in Table 1 .

Table 1. Mass flow rate and specific heat

Mode	m_2 (kg/s)	m_3,m_4 (kg/s)	m_5,m_6 (kg/s)	m_7,m_8 (kg/s)	cp_{eg} [kJ/kg-K]	cp_{hw} [kJ/kg-K]	cp_b [kJ/kg-K]
1	0.0023	0.242	-	0.67	1.03	-	3.80
2	0.0022	0.240	-	1.10	1.03	-	3.78
3	0.0019	0.197	-	1.09	1.02	-	3.78
4	0.0016	0.151	1.99	0.99	1.02	4.27	3.78

The electrical, heating and cooling power and the main assessment parameters calculated for the system are presented in Table.

Table 2. Output power and assessment parameters

Mode	W_{el} (kW)	Q_{HRSG}(kW)	Q_{cool}(kW)	COP (-)	η_{MT} (%)	η_{cog} (%)	η_{HRSG} (%)
1	25	20.1	9.9	0.49	23	31	-
2	25	11.3	6.9	0.61	24	30	-
3	20	10.8	3.9	0.36	22	25	-
4	10	11.5	3.6	0.31	13	17	92

The presented results show that the microturbine's efficiency reduces with the output electric power, as expected accordingly to Capstone System Manual, 2001. Also, the usable thermal power available is reduced, due to the lower temperature (T[3]) and mass flow rate of the exhaust gases, as shown in Table 1 and 2.

The COP was greatest for Mode 2, however part of the heat contained in the exhaust gases is lost to the environment due to the heat recovery control. The highest cooling power produced was found in Mode 1, related to the high HRSG pressure.

Mode 4 presented the smallest value for the efficiency of the microturbine and COP. In the steady state condition, part of the heat recovered from exhaust gases is lost in its way to the generator ($Q_{gen} < Q_{HRSG}$), probably due to losses in the heat exchanger and the hot water pipeline, as observed in Figure 10. HRSG's efficiency was found 0.92 for a electrical output power of 10kW. This parameter shows that COP could be improved.

In Mode 1, a great amount of heat is recovered in the absence of any control devices. In the end of the test, the HRSG stabilizes at approximately 4 bar. At that point, all the heat recovered from the exhaust gases is supplied to the generator, and no heat is stored in the HRSG. For this particular cogeneration plant, this should be the pressure desired in full operation to maximize the use of the microturbine thermal energy. Nevertheless, the exhaust gases temperature downstream of the heat exchanger (T[4]=190°C) shows that considerable heat is still available for cogeneration, and another HRSG should be considered in addition of the existing one.

The tests showed the decrease of the microturbine's efficiency with lower output power, as expected. Greater electrical power provided higher heat recovery rate into the HRSG and higher cooling power production by the absorption chiller, as shown in Table 1. Small scale cogeneration plants should not be designed to work in partial load, in order to attain enhanced efficiencies.

A workbench with electrical resistances is being installed inside the storage cooling system in order to evaluate cooling power production and COP for different outlet brine temperatures. Also, different HRSG pressures are being tested systematically to evaluate quantitatively COP's performance. An analysis to evaluate the uncertainty of the results is in progress and will be reported in a further work.

5. CONCLUSIONS

Experimental results for a small scale cogeneration plant were reported. Steam and hot water were used to power the generator of an -10°C absorption chiller. The results obtained for the best test were 31% and 23% for the cogeneration and microturbine efficiency, respectively and 0.49 for the COP. The maximum output electrical power obtained was 25 kW and cooling power produced was about 10 kW (2.8 RT). For cogeneration efficiency improvement, another HRSG should be considered in addition of the existing one, in order to recover the heat available in the exhaust gases downstream of the heat exchanger, due to their high temperature (T[4]=190°C).

The microturbine's efficiency was observed to decrease when working in partial load, as expected. The heat recovery rate and the cooling power produced were greater for full load operation. The tests performed showed that COP is greater for higher output electrical power and HRSG pressures. Furthermore, improvements in the HRSG could lead to an increase in the COP, as efficiency of HRSG was found 0.92. Small scale cogeneration plants should not be designed to work in partial load, in order to attain enhanced efficiency for the microturbine and the COP for absorption chillers.

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