ANALYSIS OF A HUMID AIR CYCLE IN GAS TURBINE

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Abstract. *The project aims to analyze the costs and the generation of electrical energy from a humidified air gas turbine cycle. A research was made on the variations of humidified air cycles proposed for gas turbines applications. The analysis target the power levels, efficiency and optimum operating points in order to choose which enables to the higher power generation. A numerical simulation performed in the cycle in software Gate Cycle™ checked the environmental parameters that influence the cycle performance. The main points of variation were the outlet temperature of the combustion chamber, as well as the pressure ratio of the compressor and the turbine. Graphics were plotted for thermal efficiency and the specific power of the cycle related to the cycle pressure ratio using turbine inlet temperatures from 900 to 1400ºC, thus enabling us to evaluate the best point of operation. In addition, the use of the gas saturator and the recuperator was fundamental to increase the mass flow of air in the system, which influences the global efficiency of the cycle. The condenser of contact was essential for the reuse of the water in the operation of the cycle and in the additional production of drinking water. The performed analysis made it possible to study the performance of the system, with the ambient temperature ranging from 5 to 30°C. The design point was chosen for a turbine inlet temperature of 1100ºC and a cycle pressure ratio of 13, in which a higher efficiency (51.93%) is reached with a medium exergetic cost (2.001) without compromising the physical and metallurgical properties of the system. The saturator obtained a gain of 0.3022 kg/s on airflow, increasing the power of the cycle and thus reaching 1.078MW on the simulation. The fluid gas condensers obtained surplus of 0.6792 kg/s of potable water. The system showed generation cost of 39.83 R$/MWh, which is competitive in the Brazilian electricity market. Regarding the off-design analysis, the increment on the environment temperature results on higher generated power, unit exergetic cost and exhaust temperature, and also a depletion of the exergetic and the energetic efficiency caused by the rise of the exhaust energy*.

Keywords*: Gas Turbine, Condenser of contact, Thermal efficiency.*

1. INTRODUCTION

Microturbines are playing an increasing role in the distribution of power systems due to their small size, the provided environmental benefits and the reduction of O & M costs. However, their low electrical efficiency, about 30% (LHV), does not allow for competition directly in the case of internal combustion engines of the same power. In order to raise the competitiveness of microturbines, some measures can be taken, such as a substantial increase in inlet temperature in the turbine. The efficiency of microturbines can be improved also with humidification of the working fluid with heat recovery at the outlet of the microturbine. An advanced and promising energy generation is the use of a moist air cycle in conjunction with microturbine gas because of its high efficiency, low cost of specific investment, high power and low emissions rate.

2. LITERATURE REVIEW

Thermal performance deviation in gas turbines operating with a humid gas cycle was studied by several authors during last year. Lazzaretto and Segatto (2002) made an article aiming to optimize a traditional humid air cycle by improving the heat exchangers (aftercoolers, intercoolers and economizers), and elevating the total efficiency of the plant. The system was simulated using the Aspen Plus software, and the calculus was performed by Fortran software tarting from the data considered as excellent for the components of the heat exchanger net. Through the research a maximum efficiency of 54.9% was obtained using a pressure ratio of 20. Parente, Traverso and Massardo (2003) analyzed the influence of the use of a saturator in a humid air cycle by studying the thermodynamics factors to improve efficiency. To do so, two models of gas saturator were made: the first model, SAT, which was more complex, considered both the mass and energy balance. The second one, CT, was less complex, so that it could look like a humidifying tower**.** The simpler CT model could be modeled under high pressure between 10 to 15 bar, but it is necessary to implant a more accurate system in order to avoid the oversizing of the saturator. Wan et al (2010) made a study comparing an ordinary humid air cycle and one inverted Brayton with humid air for gas microturbines. The models were implanted using gPROMS (program used to introduce and execute process models in stationary or dynamic states of any complexity). In addition, the Gate Cycle™ software was used for the modeling standard gas on a gas turbine cycle. BAHAT cycle reached an efficiency of 44.39% and HAT cycle showed 41.57%. Nyberg and Thern (2012) analyzed distinct configurations of humid air cycle using IPSEpro for calculus and simulation. From this article, we concluded that the aftercooler and the recuperator improved the cycle's efficiency. Wei and Zang (2013) executed a research about low pressure and temperature on the gas turbine inlet in a humid air cycle, in order to improve its effectiveness. To do so, a real system was implemented in a laboratory in Shangai using the SIMATIC S7-300 PLC from Siemens Technology for control and measurement. Using the Matlab, a software called SJGT was implemented to realize the calculus, which found from the results that maintaining the temperature of 665ºC on the inlet, it increases 9.5kW on the total power. Therefore, the efficiency of the gas turbine can be improved without restriction of the high temperature of the inlet. Ameri and Enadi (2012) developed a software with Matlab aiming to create a thermodynamics model through exergetic (physical and chemical) analysis and also a exergoeconomic analysis in order to estimate the costs of generation and exergy destruction. The authors highlighted that the sectors where the worst exergetic efficiency and the largest destruction of exergy occur were the ones using combustion chambers. The temperature on the inlet of the gas turbine is inversely proportional to the generation costs, as the exergoeconomic analysis indicates. Carrero et al (2014) made an economic analysis of a power plant, which uses a microturbine for generation of heat and energy (CHP), and operates a humid air cycle (mHAT). Initially, it was found that the usage of a system such as the CHP or mHAT depends on the prices of natural gas and the electricity. In addition, the fuel represents an amount of 74 to 90% of the total costs, which includes natural gas, operation and maintenance, as well as water and additional electricity.

3. METHODOLOGY

The modeling cycle was performed in the software GateCycle^{TM} where two analysis were made: design and off-design. In the design analysis, the turbine inlet temperature varied within the range 900-1400°C, and the cycle pressure ratio also varied within the range 9-13. Analyzing the results, the performance for the design point based on specific net power output and net cycle efficiency was specified. Off-design analysis was performed varying the ambient temperature from 5 to 30°C. Details about the thermal schematics, data for simulation and procedures for simulations are explained below.

3.1. Cycle parameters

In order to analyze the cycle regarding its effectiveness and its viability, there are some equations governing the system, giving an idea of the behavior of the cycle as shown below:

$$\dot{w}_{Total} = w_{Ex} - \sum w_C - \sum w_P \tag{1}$$

Where,$\dot{w}_{Total}$ is the netpower of the cycle, in MW or kW, w_{Ex} is the power in the expander, in kW, $\sum w_C$ is the total power of the compressors, in kW, and $\sum w_P$is the total power of the pumps, in kW.

$$\eta_{Total} = \frac{\dot{w}_{Total}}{\dot{Q}_{in}} \tag{2}$$

Where, $\eta_{Total}$is the thermal efficiency of the cycle, and $\dot{Q}_{in}$ the heat rate at the inlet of the combustion chamber on kW.

$$\eta_{ex} = \frac{\dot{m}_{fuel} \cdot e_{fuel}}{\dot{W}_{Total}} \tag{3}$$

Where, η_{ex} is the cycle exergetic efficiency, $\dot{m}_{fuel}$ is the mass flow of the fuel, in kg/s, e_{fuel} is the specific exergy of the fuel (49366), in kJ/kg, computed according to Lozano and Valero (1986) at 15°C, 1 bar and 60% of relative humidity.

$$k^* = \frac{1}{\eta_{ex}} \tag{4}$$

Where,k^* is the specific exergetic cost.

3.2. BAHAT GateCycle^{TM} Model

The figure below shows the thermal cycle in software GateCycle^{TM} where we can see that the air suffers a first compression (C) and then enters the aftercooler (AC) in order to lose heat and go to the saturator (HD), which is responsible for increasing the mass flow of the wet air. Then, it goes in the recuperator (Recu) to increase temperature an also enthalpy before the combustion chamber (CC). Continuing the cycle, the combustion products go to the expander (EX), which is responsible to decrease the pressure to a sub-atmospheric value (40 kPa).Then, the combustion products decrease temperature in the recuperator (Recu), the economizer (Eco1) and in the fluid gas condenser (FGC1) in which, additionally, it is decreased the water content that can be reused in the cycle. After that, the combustion products suffer a second compression and return to the atmospheric pressure following another economizer (Eco2) and another fluid gas condenser in order to decrease temperature and water content before the exhaust.

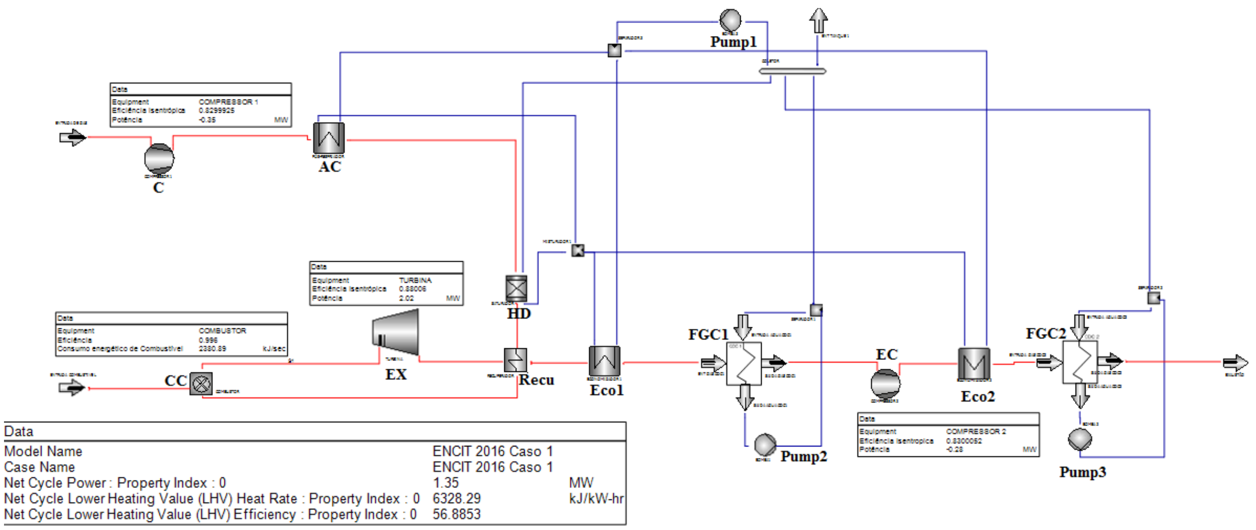


Figure 1. Simulation model developed in GateCycle™.

Table 1 and Tab. 2 present the main data for simulation. Some values were represented as a range, because they were used to determine the maximum cycle performance. The values mentioned were extracted from Wan, K. et al (2010), one of the main articles used for the development of this research.

Table 1.Inputs Data

Input parameters	Unit	Value
Entering air mass flow rate	kg/s	2.19
Entering air pressure	bar	8.16
Entering water mass flow rate	kg/s	3.27

Table 2. Computational assumptions and parameters

Input parameters	Unit	Value
Turbine inlet temperature, (range)	ºC	900-1400
Combustion efficiency	%	99.6
Isentropic efficiency of air compressor and exhaust compressor	%	83
Isentropic efficiency of expander	%	88
Isentropic efficiency of pumps	%	83
Total pressure ratio (range)	-	9 - 13
Hot side pressure loss of AC, Eco1, Eco2, HD, FGC1 and FGC2, % (of inlet pressure)	%	1
Cold side pressure loss of AC, Eco1, Eco2, HD, FGC1 and FGC2, % (of inlet pressure)	%	3
Heat loss of AC, Eco1, Eco2, FGC1 and FGC2	%	1
Hot side pressure loss of Recu, % (of inlet pressure)	%	3
Cold side pressure loss of Recu, % (of inlet pressure)	%	2
Heat loss of Recu	%	1
CC pressure loss (% of inlet temperature)	%	4

3.3. Cost Analysis

In order to evaluate if the generated power value is compatible with the existing tariffs in the Brazilian electricity market, the generated electricity (C_{gen}) cost calculation model in R$/kWh is as follows:

$$C_{gen} = Tax \cdot \left(C_{inv} \cdot \frac{AF}{OH} + C_{O\&M} \right) + C_{NG} \tag{5}$$

$$AF = \frac{i \cdot (1+i)^n}{(1+i)^n - 1} \tag{6}$$

$$C_{NG} = \frac{\dot{m}_{fuel}}{\rho_{fuel}} \cdot \dot{w}_{total} \cdot T_{GS} \tag{7}$$

In equations 5 to 7, Tax is the exchange rate at R\$/US\$; C_{inv} is the investments costs of the turbine in US\$/kW; AF is the amortization factor; OH is the operation hours per year in h/year; $C_{O\&M}$ is the cost of operation and maintenance in US\$/kWh; i is the long-term interest rate; n is lifetime in years; C_{NG} is the cost of natural gas in R\$/kWh; ρ_{fuel} is the density of natural gas in kg/m^3; T_{GS} is the charge for supply of natural gas R\$/m^3. These parameters are shown in Tab. 3 below.

Table 3. Cost Analysis Parameters

Parameter	Unit	Value	Reference
Tax	R\$/US\$	2.342	Brazilian Central Bank (2013)
OH	h/year	8030	-
i	% p.y.	5	Long-term interest rate
N	year	20	-
ρ_{fuel}	kg/m^3	0.7	-
T_{GS}	R\$/m^3	0.057	COMGAS (2013)
$C_{O\&M}$	US\$/kWh	0.00444	NREL (2013)
C_{inv}	\$/kW	750	GTW Handbook (2013)

4. RESULTS AND DISCUSSION

For the design analysis, we analyzed the relations of net cycle efficiency; specific work; exergetic efficiency and the exergetic unit cost versus cycle pressure ratio. The results we plotted are described in section 4.1. After we defined the design point, we carried out the off-design analysis and showed the results in section 4.2.

4.1. Design Analysis

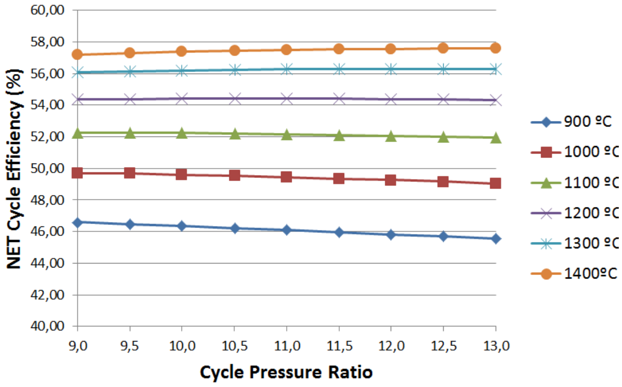
In Fig. 2, it is possible to observe that the curves for the 1200°C temperature denote a shift line in the system. As the cycle pressure ratio increases for several turbine inlet temperature the same trend in results with decreasing thermal efficiency [Fig. 2 (A)] and exegetic efficiency [Fig. 2 (C)] it is observed. Please, observe the opposite behavior for specific work [Fig. 2 (B)] and exergetic unit cost [Fig. 2 (D)]. The growth of exergetic unit cost means a greater destruction of exergy in the system, thereby even if the system presents a high specific work with increased pressure, there is a bigger destruction of exergy and it will be a lag for the work generated.

Under temperatures of 1300ºC to 1400ºC, there is an increase on thermal and exergetic efficiencies of the cycle as the cycle pressure ratio rises, which it is also noted for the exergetic unit cost. However, there is a physical and a metallurgical limitation under these temperatures for durability, as we can see that the rising of the temperature on the turbine will compromise the lifetime of the system causing higher maintenances expenses. This affects directly the costs analysis that we will comment later. Considering those aspects, the chosen system will operate under 1100ºC and cycle pressure ratio of 13. According to the analysis of the implantation based on the energetic costs, this point of operation leads to a better use of the resources without exceeding the physical limitation of the system. Moreover, it is possible to generate a relatively high power, with high thermal and exergetic efficiency. Table 4 shows the performance parameters at the selected design point.

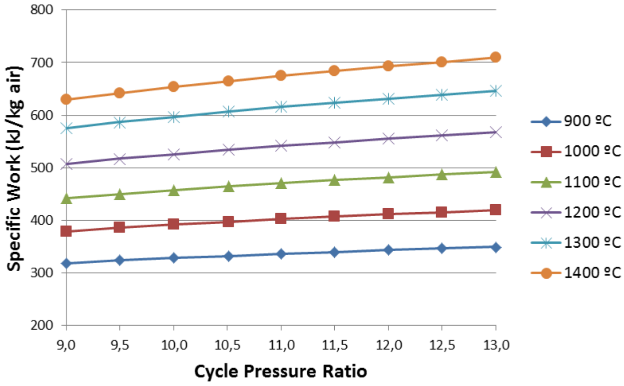
Table 4. Parameters for maximum performance cycle

Parameter	Unit	Value
Net Cycle Power	MW	1.0780
Specific Work	kJ/kg air	492.180
NET Cycle Heat Rate	kJ/kWh	6931.980
NET Cycle Efficiency	%	51.930
Exergetic Efficiency	%	49.970
Exergetic Unit Cost	kW/kW	2.001

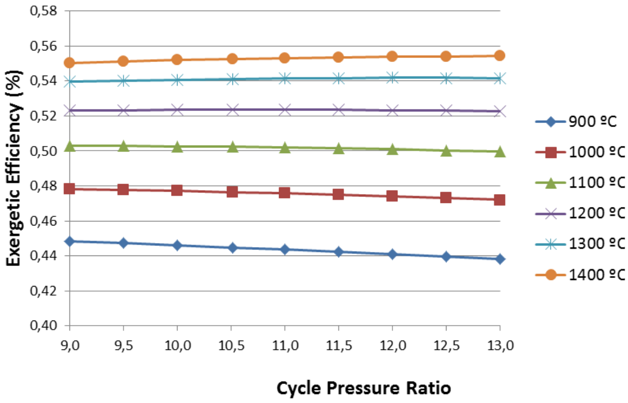
The usage of the saturator on the cycle promoted a gain on mass flow of 0.3022 kg/s of humid gas. This increase of mass throughout the recuperator causes an enthalpy rise of 495.9 kJ/kg that comes through the energy transfer from the exhaust gas of the turbine. The energy recovery through the usage of the saturator and with the recuperator contributes to the net power increase by producing a bigger gas mass flow at high temperatures inside the combustion chamber, with low fuel injection (0.0437 kg/s total).



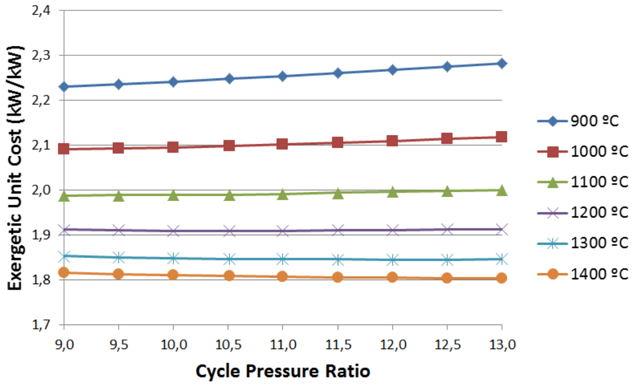
(A) Net cycle efficiency



(B) Specific work



(C) Exergetic efficiency



(D) Exergetic unit cost

Figure 2.Simulation results for design condition.

The heat exchangers like the economizers and the fluid gas condenser gives a better recovery of energy between the components of the cycle, being responsible for transferring energy from the airflow to the water that goes into the saturator. The economizer 1 and the fluid gas condenser 1 also contribute for lower temperatures of the combustion products that contribute for less required power for compressor 2. The usage of contact condensers also contributes for the reuse of fractions of water from the combustion products that was added on the saturator and from the burning process of natural gas. With that in mind, the fluid gas condensers offer a surplus of water mass flow of 0.6792 kg/s, which can be used in the cycle or for an external user.

Figure 3 shows that at the turbine inlet temperatures higher than 1200ºC, the increasing of temperature and pressure causes a falling on the generation costs. However, the usage of a system operating under these temperatures cause high thermal and mechanical wear of the components, which implies a reduction on the operation hours between maintenance and a higher need for replacing the hot gas path components. Considering these aspects, we selected 1100ºC of turbine inlet temperature and a cycle pressure ratio of 13 for design condition, with those values meeting the working hours cost of 39.83 R$/MWh. According to Brasil (2014), the value of the average tariff for supply of Brazilian companies for residential and industrial supply was higher than the estimate cycle costs in any operating mode shown in Fig. 3, so the system proves to be competitive in the Brazilian electricity market.

4.2. Off-Design Analysis

In order to analyze a system operating under different regions or climates, it was performed an on off-design analysis, varying the environment temperature and evaluating the variation of the performance parameters of the cycle related to a standard temperature of 15ºC that can be seen in Fig. 4.

Even when increasing the environment temperature causes a decreasing on the air density, which results in a smaller gas mass flow into the combustion chamber causing a fall in the generated power, the rising of environmental temperature also will increase the enthalpy of the gas flow at the combustor inlet, rising the generated power. However, the higher environment temperature will decrease the difference between the hot and cold currents, resulting in an insufficient heat exchange on the economizers, fluid gas condensers and recuperator, with reduction in thermal, and exergetic efficiencies of the cycle, caused by the rising of the exhaust gas energy.

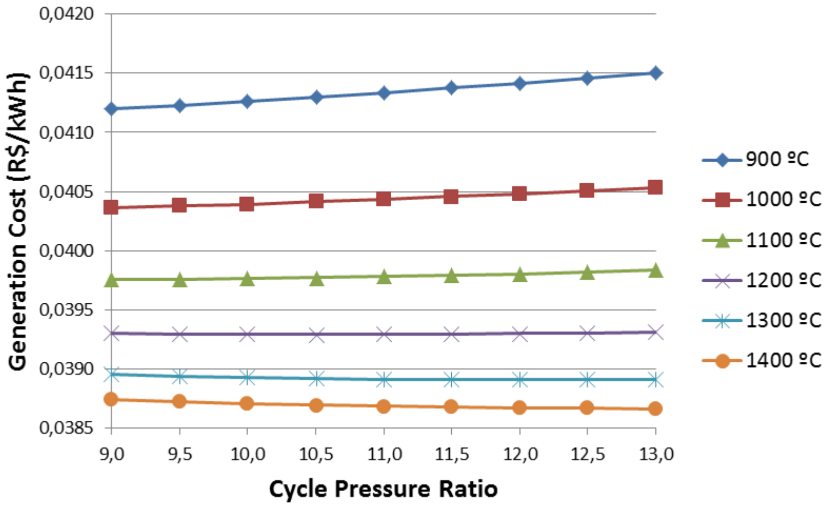


Figure 3. Generation Cost simulation for design point.

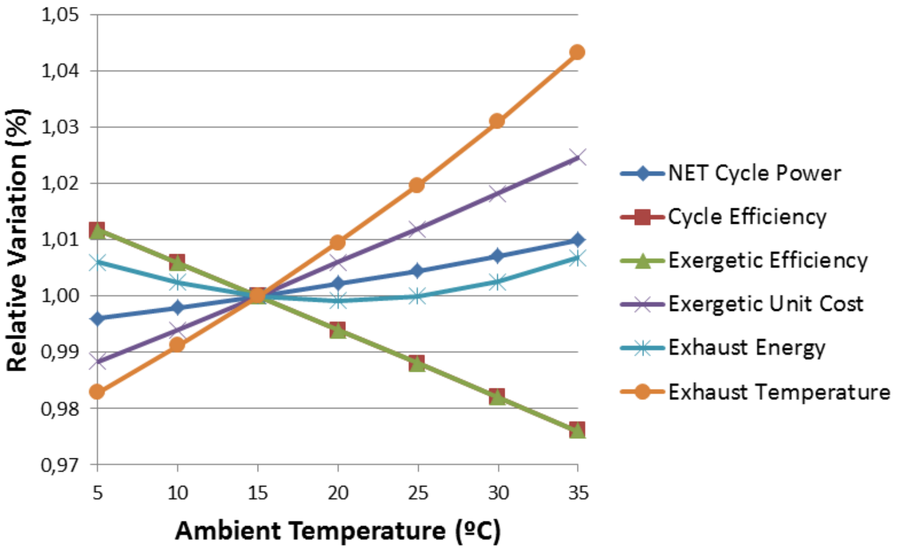


Figure 4.Off-design results for the variation of ambient temperature.

5. CONCLUSION

With the humid air cycle, it is possible to increase the efficiency of electricity generation on a microturbine, in this sense the results allow us to come to the following conclusions:

- The great point of operation is found under a temperature of 1100ºC and a pressure ratio of 13, where a higher efficiency (51.93%) is reached with a medium exergetic cost (2.001 kW/kW) without compromising the physical and metallurgical properties of the system;
- The saturator obtained a gain of 0.3022 kg/s on gas mass flow due to the use of aftercooler for cooling the compressed gas and heating the water that goes to the saturator, increasing the power of the cycle, and thus reaching 1.078 MW on the simulation.
- The recuperator promoted a gain of 495.9 kJ/kg on the cold current from the exhaust gas of the turbine decreasing in 0.0437 kg/s the fuel consumption.
- The fluid gas condensers obtained a surplus of 0.6792 kg/s of potable water;
- The system showed generation cost of 39.83 R$/MWh that is competitive in the Brazilian electricity market;
- Regarding the off-design analysis, the increment on the environment temperature resulted on higher generated power, unit exergetic cost and exhaust temperature, and also a depletion of the exergetic and the energetic efficiency caused by the rise of the exhaust energy.

6. ACKNOWLEDGEMENTS

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