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COSTS ASSESSMENT IN A HAT CYCLE

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Abstract. *The project aims to analyze the investment and electricity generation costs of a humidified air turbine (HAT) cycle for microturbine application. The analysis target the power levels, efficiency and optimum design operation point 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 parameters of variation were the outlet temperature of the combustion chamber (900 to 1400ºC), and the pressure ratio of the compressor and the turbine (9 to 13). Graphics we plotted for thermal efficiency, specific power, exergetic efficiency, unit exergetic cost and electricity generation cost of the cycle related to the cycle pressure ratio, thus enabling us to evaluate the best design operation point. This 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 hot path turbine components, reaching 1.078 MW of net cycle power on the simulation. Therefore, a detailed analysis of the cost of generation we made considering the fuel, the O&M and the investment costs. The investment cost of all components achieve 685,525.05US$ and the system showed the electricity generation cost about 25.60 R$/MWh, which is very competitive in the Brazilian electricity market.*

Keywords: *Gas Microturbine, Humidified Air Turbine, HAT cycle, Cost.*

1. INTRODUCTION

The assessment of the electricity generation and investment cost of an electrical generation plant is one of the most important factors in Energy Engineering, considering its feasibility and application. Owing to the increase in energy consumption, new generation plants are designed based on thermodynamic and economic parameters, in order to achieving the maximum power at the lowest generation cost. The HAT cycles for microturbine application is an interesting cost-benefit solution because provide high thermal efficiency, high power and require fewer components compared to the combined cycles, decreasing the total investment cost.

Therefore, several authors have studied the thermal performance deviation in gas turbines operating with a humid gas cycle in the last years. Lazzaretto and Segatto (2002) made an article aiming to optimize a traditional humid air cycle by improving the heat exchangers (after coolers, intercoolers and economizers), and elevating the total efficiency of the plant. The system was simulated using the Aspen Plus software, and for the calculus was performed by Fortran software starting from the data considered as excellent for the components of the heat exchangers' net. Through the research, they obtained a maximum efficiency of 54.9% using a pressure ratio of 20. Wan et al (2010) realized 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). Also Gate Cycle™ software was used for 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 the article, they concluded that the after cooler 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 they implemented in a laboratory in Shangai using the SIMATIC S7-300 PLC from Siemens Technology for control and measurement. Using the Matlab, was implemented a software called SJGT to realize the calculus which found from the results that maintaining the temperature of 665ºC on the inlet, it increases 9.5 kW on the total power. Therefore, the efficiency of the gas turbine were improved without restriction of the high temperature of the inlet. Roosen, Uhlenbruck and Lucas (2002) made a deep cost analysis for a combined power plant cycle. It was used the

Pareto technical numeric optimization in an evolution strategy. The cycle pressure ratio and turbine inlet temperature were one of the main parameters for the optimization. The Pareto optimization has proved its applicability in power plant cycles with defined parameters. Ameri and Enadi (2012) developed a tool using Matlab aiming to create a thermodynamics model through exergetic (physical and chemical) analysis and a exergoeconomic analysis in order to estimate the costs of generation and exergy destruction. The authors highlighted that the sectors, which happen the worst exergetic efficiency and the largest destruction of exergy, was the ones using combustion chambers. The temperature on the inlet of the gas turbine is inversely proportional to the generation costs, as indicated by the exergoeconomic analysis. Carrero et al (2014) made an economic analysis of a power plant, which used a microturbine for generation of heat and energy (CHP), and operating 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 water and additional electricity.

2. METODOLOGY

The modeling cycle was performed in the software GateCyle^{TM}. This software has a library of most required components of the cycle, performing iterations based on the thermodynamic, heat transfer and fluid mechanics equations, according to the input data. Two important parameters varied in this model: the turbine inlet temperature and the cycle pressure ratio. The analysis of results specified the design point based on net cycle efficiency (thermal efficiency), exergetic efficiency, specific work, exergetic unit cost, total investment cost and generation cost. Details about the thermal schematics, data for simulation and procedures for simulations are as follow.

2.1 GateCycle^{TM} Model

The Fig. 1 shows the thermal cycle modeled in the software GateCycle^{TM}. The air suffers a first compression (C) and then enters in the after cooler (AC) in order to transfer 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). Later, 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 in the cycle, which it is possible to reuse. 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.

The inputs data for calculations we adopted from Wan, K. et al (2010). In the design analysis, the turbine inlet temperature varied within the range 900-1400°C, while pressure ratio range between 9 to 13.

Table 1 presents the main data for simulation. The turbine inlet temperature and total pressure ratio changed in order to determine the maximum cycle performance.

Table 1. Inputs data and computational assumptions and parameters from Wan, K. et al (2010)

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
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

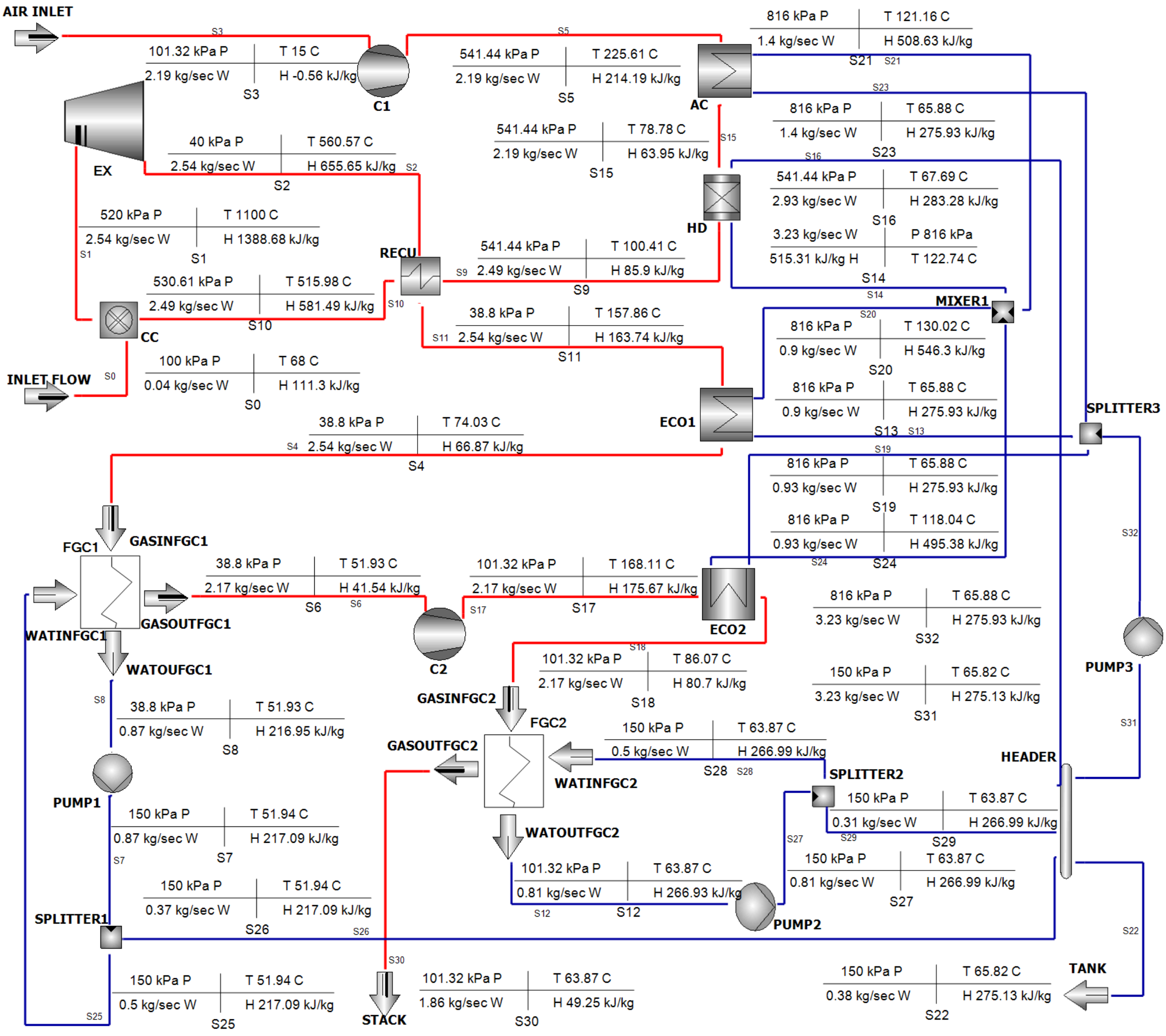

Figure 1. Simulation model developed in GateCycle™

2.2 Cycle parameters

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

$$\dot{w}_{Total} = \dot{w}_{ex} - \sum \dot{w}_c - \sum \dot{w}_p \tag{1}$$

Where, $\dot{w}_{Total}$ is the net cycle power of the cycle, in MW or kW, $\dot{w}_{ex}$ is the power in the expander, in kW, $\sum \dot{w}_c$ is the total power of the compressors, in kW, and $\sum \dot{w}_p$ is the total power of the pumps, in kW.

$$w_{ex} = \frac{\dot{w}_{ex}}{\dot{m}_g} \tag{2}$$

Where, w_{ex} is the expander specific work, in kJ/kg air, and $\dot{m}_g$ is the mass flow of gas, in kg/s.

$$\eta_{Total} = \frac{\dot{w}_{Total}}{\dot{Q}_{in}} \tag{3}$$

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

$$\eta_{ex} = \frac{\dot{w}_{Total}}{\dot{m}_{fuel} \times e_{fuel}} \tag{4}$$

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.

$$k^* = \frac{1}{\eta_{ex}} \tag{5}$$

Where, k^* is the specific exergetic cost.

2.3 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:

$$C_{gen} = Tax \times \left(\left(C_{inv} / \dot{w}_{ex} \right) \times \left(AF / OH \right) + C_{O\&M} \right) + C_{NG} \tag{6}$$

$$AF = \frac{i \times (1+i)^n}{(1+i)^n - 1} \tag{7}$$

$$C_{NG} = \frac{\dot{m}_{fuel}}{\rho_{fuel}} \times \dot{w}_{Total} \times T_{GS} \tag{8}$$

In equations 5 to 7, *Tax* is the exchange rate at R$/US$; C_{inv} is the total investments costs, in US$; *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$/kWh; *i* i is the long-term interest rate; *n* is lifetime, in years; C_{NG} is the cost of natural gas, in R$/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. 2 below, except the total investment costs.

Table 2. Cost analysis parameters

Parameter	Unit	Value	Reference
Tax	R$/US$	2.342	Brazilian Central Bank (2013)
Tax_2	€/US$	0.726	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)

The equation of investment cost is as follows:

$$C_{inv} = \sum C_{invc} + C_{invex} + C_{invcc} + C_{invrecu} + Tax_2 \left(\sum C_{invhe} + \sum C_{invp} \right) + \sum C_{invdche} \tag{9}$$

Where, $\sum C_{invc}$ is the total investment cost of compressors, in US$; C_{invex} is the investment cost of gas turbine, in US$; C_{invcc} is the investment cost of combustion chamber, in US$; $C_{invrecu}$ is the investment cost of recuperator, in

US$; Tax_2 is the exchange rate at €/US$; $\sum C_{invhe}$ is the total investment cost of heat exchangers (economizers and aftercoolers), in US$; $\sum C_{invdche}$ is the total investment cost of direct contact heat exchangers (flue gas condensers and saturator), in US$; $\sum C_{invp}$ is the total investment cost of pumps, in US$.

Below are described the equations for investment costs of compressor (C_{invc}), turbine (C_{invt}), combustion chamber (C_{invcc}) and recuperator ($C_{invrecu}$), in US$. (BEJAN, TSATSARONIS E MORAN 1996)

$$C_{invc} = \left(C_{11} \times \dot{m}_g / C_{12} \times \eta_c\right) \times \left(p_{out} / p_{in}\right) \times ln\left(p_{out} / p_{in}\right) \tag{10}$$

$$C_{invcc} = \left(C_{21} \times \dot{m}_g / C_{22} - \left(p_{out} / p_{in}\right)\right) \times \left[1 + e^{C_{23} \times T_{it} - C_{24}}\right] \tag{11}$$

$$C_{invt} = \left(C_{31} \times \dot{m}_g / C_{32} - \eta_{ex}\right) \times ln\left(p_{in} / p_{out}\right) \times \left[1 + e^{C_{33} \times T_{it} - C_{34}}\right] \tag{12}$$

$$C_{invrecu} = C_{41} \times \left(\dot{m}_g \times \left(h_{in} - h_{out}\right) / U \times \Delta T_{ml}\right)^{0.6} \tag{13}$$

Where , C_{11}, C_{12}, C_{21}, C_{22}, C_{23}, C_{24}, C_{31}, C_{32}, C_{33}, C_{34} and C_{41} are system constants; η_c is the compressor isentropic efficiency; p_{out} and p_{in} are the outlet and inlet pressure of the component, respectively, in kPa; T_{it} is the turbine inlet temperature, in K; η_{ex} is the expander isentropic efficiency; h_{in} and h_{out} are the specific enthalpies of recuperator, in kJ/kg; U is the global heat rate coefficient, in W/m²K; ΔT_{ml} denotes a log mean temperature difference in recuperator. These system constants are shown in Tab. 3 below.

Table 3. System constants (BEJAN, TSATSARONIS E MORAN 1996)

Constant	Unit	Value
C_{11}	US$/(kg/s)	71.1
C_{12}	-	0.9
C_{21}	US$/(kg/s)	46.08
C_{22}	-	0.995
C_{23}	1/K	0.018
C_{24}	-	26.4
C_{31}	US$/(kg/s)	479.34
C_{32}	-	0.92
C_{33}	1/K	0.036
C_{34}	-	54.4
C_{41}	US$/m$^{1.2}$	4122
U	W/(m^2·K)	18

Below are described the equations for investment costs of economizers and after cooler heat exchangers (C_{invhe}) and pumps (C_{invp}), in €. (QUOILIN et. al., 2011)

$$C_{invhe} = 190 + 310 \times A_s \tag{14}$$

$$C_{invp} = 900 \times \left(\dot{w}_p / 300\right)^{0.25} \tag{15}$$

Where A_S is the heat exchanger superficial area, in m².

Below is described the equation for investment costs of saturator and flue gas condenser, both of them can be considered as a direct contact heat exchanger ($C_{invdche}$) due to the production similarities. (UNITED STATES ENVIROMENTAL PROTECTION AGENCY, 2002)

$$C_{invdche} = 337.5 \times C_{bi}^{0.56} \quad (16)$$

Where, C_{bi} is the basic instrumentation cost, extracted from United States Enviromental Protection Agency (2002).

3. RESULTS AND DISCUSSION

We analyzed the relations of net cycle efficiency; specific work; exergetic efficiency; exergetic unit cost; total investment cost and the generation cost versus cycle pressure ratio. These parameters were simulated for a turbine inlet temperature range of 900ºC to 1400ºC and 9 to 13 for the cycle pressure ratio, a summary of the results in the maximum and minimum points for each combined values are shown in Tab. 4:

Table 4. Summary of results at extremes

Parameter	Turbine Inlet Temperature [ºC]	Cycle Pressure Ratio [-]	
		9	13
NET Cycle Efficiency [%]	900	46.49	45.54
	1400	57.19	57.59
Exergetic Efficiency [%]	900	44.82	43.82
	1400	55.03	55.41
Specific Work [kJ/kg air]	900	318.88	349.31
	1400	629.92	709.19
Exergetic Unit Cost [kW/kW]	900	2.23	2.28
	1400	1.82	1.80
Generation Cost [R$/MWh]	900	27.82	26.98
	1400	25.08	24.38
Total Investment Cost [US$]	900	601173.06	644598.99
	1400	642805.41	688893.34

It is possible to observe that there is a decreasing behavior for the thermal efficiency and exergetic efficiency with the increase of cycle pressure ratio for low turbine inlet temperatures. The low turbine inlet temperature will lead to low enthalpy values of the work fluid at combustion chamber inlet because the low regenerated heat in recuperator, it results in the raising of the fuel consumption and the fall of the thermal efficiency. Due to the rising the specific fuel exergy is fixed, there is a decrease of exergetic efficiency; consequently, the exergetic unit cost also rises. On the other hand, for high turbine inlet temperature values the thermal efficiency and exergetic efficiency grow, because the enthalpy of work fluid increases due to the high turbine inlet temperature, therefore a bigger part of regenerated heat in recuperator. This thermal energy enter in the combustion chamber, reducing the fuel consumption, with a positive impact in thermal efficiency, exergetic efficiency and a negative impact in exergetic unit cost.

The specific work will rise if the turbine inlet temperature and pressure cycle ratio improve. Then, the work fluid will have a higher specific enthalpy achieving a bigger capacity for power generation per air mass flow unit.

With the rising of turbine inlet temperature and cycle pressure ratio, there is an increase in the total investment cost, once it is necessary the use of robust components to resist high temperatures and pressures. In addition, there is a decrease in the generation cost related to the rise of turbine inlet temperature and cycle pressure ratio. This is due to the fact, of the turbine generates more work, and a better reuse of energy in recuperator, decreasing the fuel consumption. Even with an increase of the total investment cost, the generation cost values go down because of the cycle operation.

However, the operation of a system 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 components. Considering these aspects, we selected 1100ºC of turbine inlet temperature and analyzed the variation of cycle pressure ratio. Thermal efficiency and exergetic efficiency charts were made (Fig. 2), specific work and exergetic generation cost (Fig.3), generation cost and total investment cost (Fig. 4), versus the cycle pressure ratio.

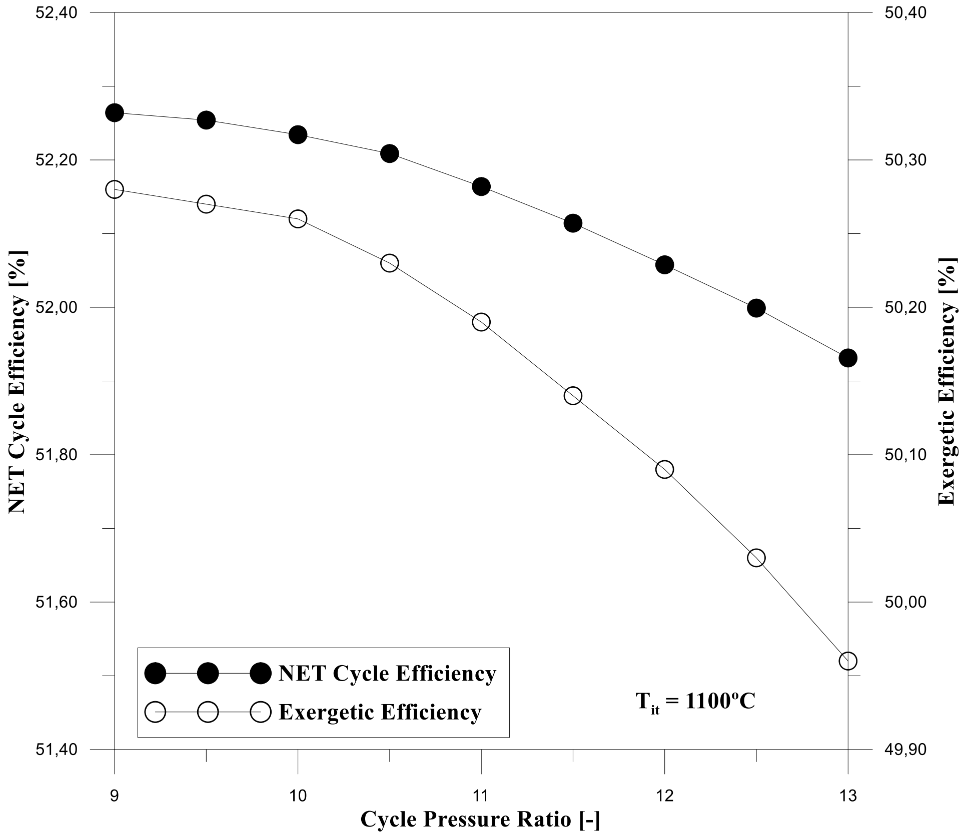

Figure 2. Thermal and exergetic efficiencies

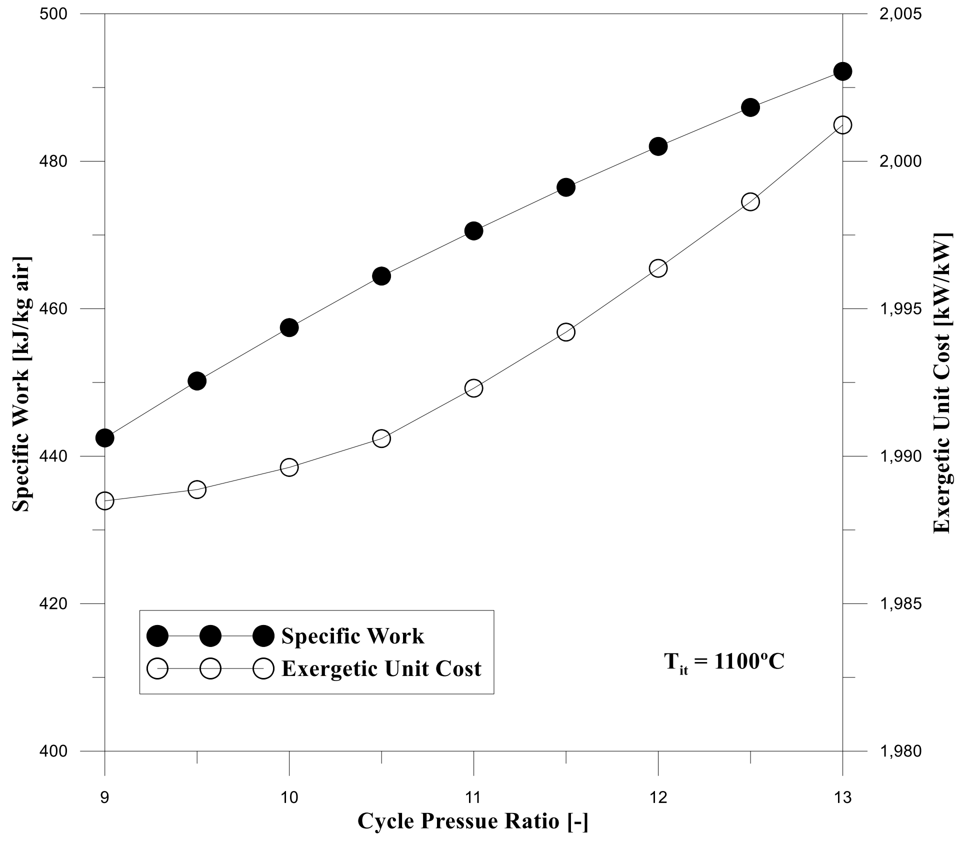

Figure 3. Specific work and exergetic unit cost

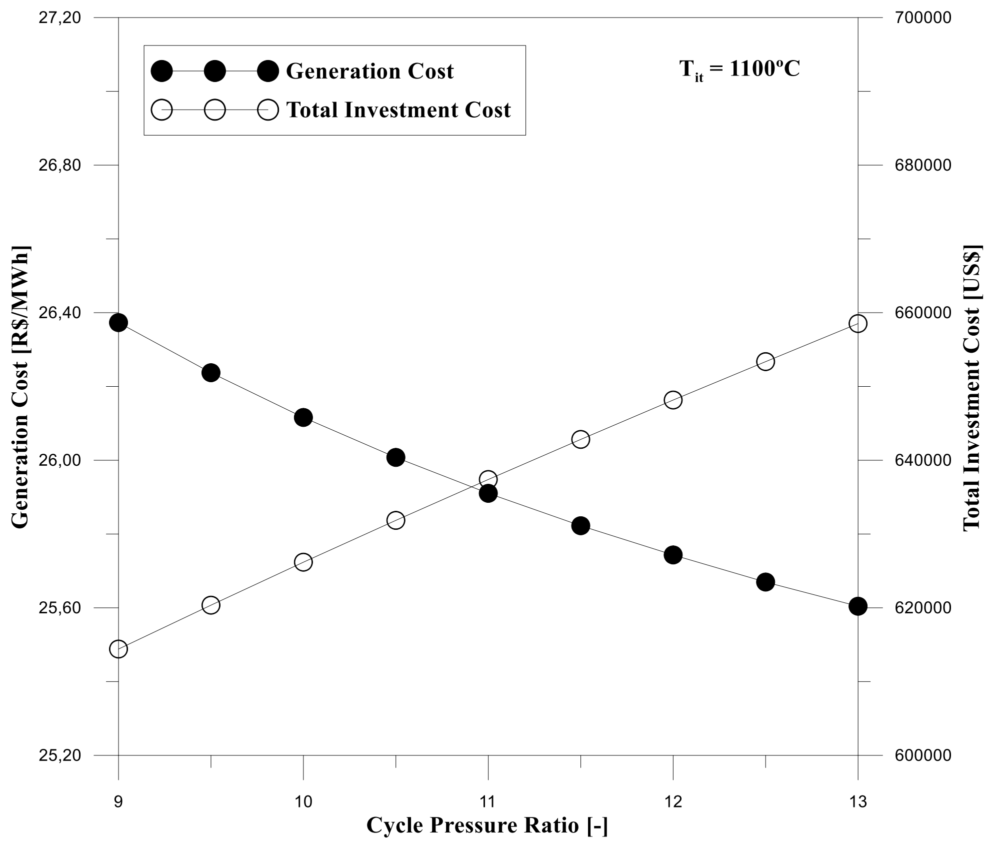

Figure 4. Generation cost and total investment cost

Meanwhile, an increase of the cycle pressure ratio results on the rising of turbine work. This way there is a better reuse of fluid energy during the expansion process. However, it is necessary a caution when increasing this parameter because high values of pressure ratio affects in a negative way the turbine components and increasing the maintenance costs. For this reason, it was chosen a cycle pressure ratio of 13, where even at the lowest values of thermal efficiency and exergetic efficiency it is possible to achieve lower generation cost. Table 5 shown the maximum performance parameters.

Table 5. Parameters for maximum performance cycle

Parameter	Unit	Value
NET Cycle Power	MW	1.078
Specific Work	kJ/kg air	492.180
NET Cycle Efficiency	%	51.930
Exergetic Efficiency	%	49.970
Exergetic Unit Cost	kW/kW	2.001
Generation Cost	R$/MWh	25.601
Total Investment Cost	US$	685,525.05

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. Heat exchangers like the economizers and the flue 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. This factors combined made the generation cost reach 25.601 R$/MWh, which is lower than the previously estimated, 39.83 R$/MWh by Arrieta et al. (2016) and, accordingly Brasil (2014), lower than average Brazilian tariff, so the system looks competitive in the actual Brazilian electricity market.

4. CONCLUSION

The air humid cycle operating with a microturbine made it possible to improve the electricity generation efficiency. In this sense, the results allow us to come to the following conclusions:

- The optimum design point was chosen with the turbine inlet temperature of 1100ºC, where it was possible to obtain high values of thermal efficiency (51.93) and exergetic efficiency (49.97) without compromising the useful life of the components. Likewise, the choice of cycle pressure ratio of 13 took into account the high specific work (492.18 kJ/kg air) and lower generation cost (25.60 R$/MWh);
- The cycle achieved lower fuel consumption due to the use of the gas saturator the recuperator, increasing the net power and producing a bigger gas mass flow at high temperatures inside the combustion chamber;
- The system showed a total investment cost of 685,525.05 US$ and generation cost of 25.60 R$/MWh, which is competitive in the Brazilian electricity market.

5. ACKNOWLEDGEMENTS

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