Proceedings of COBEM 2003 17th International Congress of Mechanical Engineering
COBEM2003 - 2000 November 10-14, 2003, São Paulo, SP

ANALYSIS OF ALTERNATIVES FOR GAS TURBINES INLET AIR COOLING

Waldir Guirardi,
Maua Institute of Technology
Praça Mauá 01, 09580-900
S. Caetano do Sul, SP, Brasil
Phone: +55 11 4239-3401
Fax: +55 11 4239-3041
Email: waldir.g@uol.com.br

Roberto Peixoto (corresponding author)
Maua Institute of Technology
Praça Mauá 01, 09580-900
S. Caetano do Sul, SP, Brasil
Phone: +55 11 4239-3401
Fax: +55 11 4239-3041
Email: robertopeixoto@maua.br

Abstract: In the coming years, it is expected an increase of the electricity consumption In Brazil. This fact will collaborate to the diffusion of distributed generation of electricity strategy and to the search for optimized systems for energy conversion systems. The natural gas availability in Brazil is expanding, and has motivated investors for electricity generation using systems based on gas turbine associated with the production of heat/cold. The goal is to reach consumers with demand lower than 5 MW, such as commercial centers, condominiums, and also some industries.
Energy efficiency of combustion turbines, is negatively affected by the increment of the inlet air temperature. The entrance air-cooling in gas turbines can promote a more energy efficient and economical operation, and also a better performance in the environmental point of view.
This paper presents the influence of inlet atmospheric air conditions in gas turbine efficiency and operational parameters, and the benefits of air-cooling systems. It makes a comparison between two cooling technologies, evaporative cooling and refrigeration by absorption systems. The absorption system is driven by the heat contents of the turbine effluent gases.

Keywords: Combustion turbines, Evaporative cooling, Absorption cooling.

NOTATION

AC absorption cooler	CP compressor	DB dry bulb temperature [^oC]	EC evaporative cooler
GT Gas turbine	h specific enthalpy [kJ/kg]	m mass flow [kg/s]	P pressure, kPa
Q heat [KW]	RH relative humidity	s specific entropy [kJ/kg-C]	T temperature[^oC]
TB turbine	W Power [KW]	WB wet bulb temperature [^oC]	X LiBr concentration, %

Subscripts

0	dead state	a	air	abs	absorber	cc	combustion chamber			cond	condenser
cp	compressor	evap	evaporator	gen	generator	i	inlet	o	outlet	r	real
swb	saturation temperature on WB curve					t	turbine	w	water		

Greek symbols

η	energetic efficiency	η_{ex}	exergetic efficiency	ω	humidity ratio

1. Introduction

In Brazil there is a massive use of hydraulic sources for electricity generation. The hydro-electrical energy is generated in distant places due to the decrease of hydraulic sources near the big consumer centers. This fact creates the need for long transmission lines, and, as a consequence, the increase of losses and the reduction of the safety level for the energy supply.

Despite the periods of turbulence in the global economy, Brazil is expected to grow in the near future, what will increase the electricity consumption. This scenario associated with difficulties for hydraulic electricity generation, has raised the interest in the small-scale energy production.

The diffusion of the strategy electricity distributed generation as an alternative for the demand increasing, has originated the search for optimized cogeneration systems, with electricity and integrated heating and/or cooling production. This alternative is expected to expand for small and medium size consumers like commercial and residential centers, and also in some industries.

Natural gas availability in Brazil is increasing, either due to the discovery of new fields and to the possibility of importation from Bolivia and Argentina. This fact motivates the investors for electricity generation. Consumers with demand lower than 5 MW constitute the main market, and gas turbine is an available technology.

Energy efficiency of combustion turbines, is negatively affected by the increment of the inlet air temperature, and this generally occurs in the period of electricity consumption peak. During this period the cost of the bought energy is very high and it becomes necessary to increase the proper production. The entrance air-cooling in gas turbines can promote a more energy efficient and economical operation, and also a better performance in the environmental point of view.

This paper presents the influence of inlet atmospheric air conditions in gas turbine efficiency and operational parameters, and the benefits of air-cooling systems. It makes a comparison between two cooling technologies, evaporative cooling and refrigeration by absorption systems. The absorption system is driven by the heat contents of the turbine effluent gases

The study is based on the numerical simulation of a straightforward mathematical model developed using the conservation principles. The software EES was used for the simulation. Operational parameters, such as produced or consumed powers, fuel consumption, outflows, temperatures, etc were calculated. The simulations had varied the altitude, the air relative humidity and temperature using atmospheric data for two cities in Brazil Belém (north region) and Curitiba (south region). Curitiba being a relatively high city, distant of the sea and more to the south, has a colder and drier climate than Belém, that is a city to the level of the sea, close to a great river, and nearby to the equator.

2. GAS TURBINE

Basically, combustion turbines or gas turbines (GT) are thermal engines, under the point of view of the Thermodynamics, for production of mechanical energy using the energy of the gases produced during the fuel combustion. The Brayton cycle, using air as working fluid is the ideal thermodynamic cycle that serves of model for the GT. This cycle is shown in Fig. (1a). It consists basically of compressor, heater, turbine and cooler. The Brayton cycle is formed by four ideal processes, initiating by an isentropic compression ($1\rightarrow2$), followed by isobaric heating ($2\rightarrow3$) and posterior isentropic expansion ($3\rightarrow4$). The fourth stage is the air isobaric cooling of ($4\rightarrow1$). A real GT consists of compressor, turbine and combustion chamber. The last stage (effluent gases cooling) is made in the atmosphere, without the existence of a heat exchanger. The real engine does not operate in closed cycle, therefore, air does not return to the equipment being discarded with the combustion gases. Figure (1b) shows the scheme of a GT and Fig. (2) shows the diagram h x s for air in the Brayton cycle.

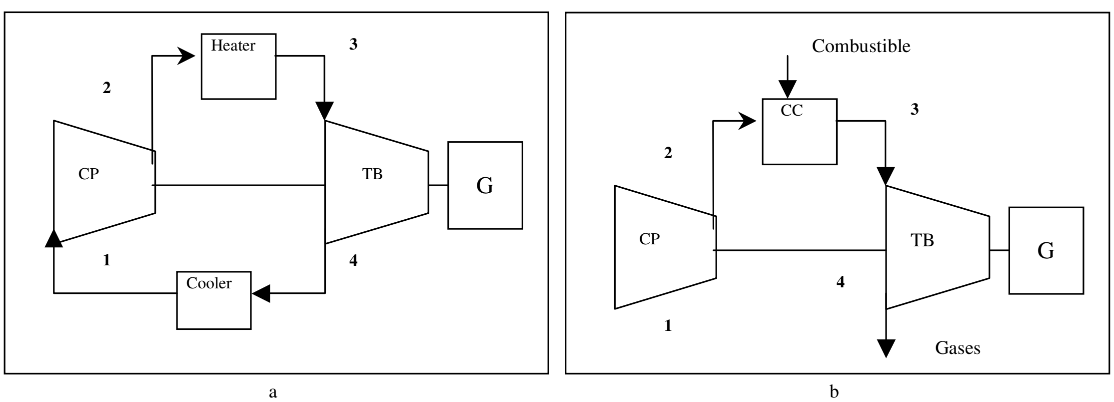


Figure 1 – a) Basic scheme of Brayton cycle. b) Basic scheme of gas turbine

2.1 Gas turbine analyzed.

There are several types of GT configurations, varying from the basic design to regenerative, with reheat and intercooling. The study was developed considering the basic design GT, which is the most used for power below 5 MW.

In a real GT the processes are irreversible and compression and expansion have a significant entropy increase as it is showed by the dotted lines in Fig. (2).

The GT mathematical modeling is based on the conservation equations, mass and energy balances. To simplify the analysis the following assumptions were adopted:

1. Closed cycle with humid air as a working fluid, based on Brayton cycle, with irreversible processes (as described above)
2. Steady state conditions.

3. Negligible pressure drop in the pipes
4. Negligible heat losses to the surrounding.

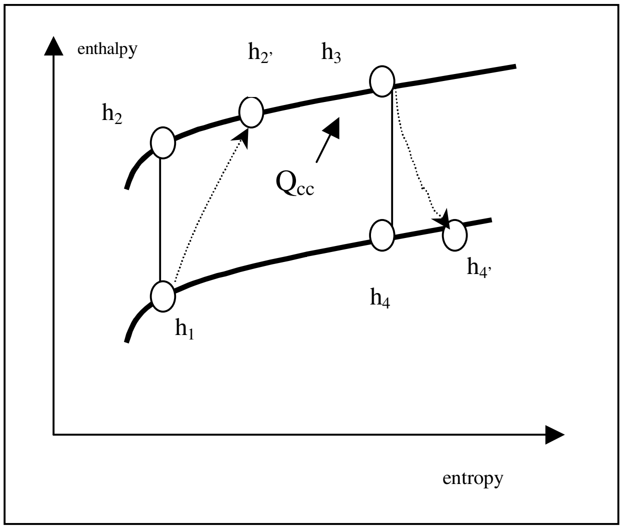


Figure 2 – Mollier diagram for air

The equations are:

Compressor

$$m_a h_1 = m_a h_{2'} + W_{cp}$$

Combustion chamber

$$m_a h_{2'} + Q_{cc} = m_a h_3$$

Turbine

$$m_a h_3 = m_a h_{4'} + W_t$$

The efficiency of the real equipment is calculated by:

$$\eta_r = \eta_{cc} m_a \frac{(h_3 - h_4)\eta_t - \dfrac{h_2 - h_1}{\eta_{cp}}}{Q_{cc}}$$

Where the efficiencies are defined by:

Combustion chamber:

$$\eta_{cc} = \frac{Q_{cc}}{Q_r}$$

$Q_r =$ Fuel energy

Compressor:

$$\eta_{cp} = \frac{h_2 - h_1}{h_{2'} - h_1}$$

Turbine:

$$\eta_t = \frac{h_3 - h_{4'}}{h_3 - h_4}$$

3. INLET AIR COOLING

Two alternatives for inlet air-cooling were studied: evaporative cooling and absorption cooling systems driven by the heat contents of the turbine effluent gases.

3.1. Evaporative Cooling (EC)

When not saturated air enters in direct contact with water, part of the water evaporates, increasing the air relative humidity and decreasing its temperature. This is a process where the variation of the air temperature and humidity is caused only by the water evaporation.

It is considered that the process is adiabatic, not having heat exchange with the ambient. In the theoretical process, the air is considered saturated at the exit, and its temperature is called adiabatic-saturation temperature that is very close to the wet bulb temperature WB. In the real process, the air final temperature can be around 1 ^oC higher than the WB.

The mathematical modeling for the EC system simulation was developed using the mass and energy balance equations:

$$m_{ai}h_{ai} + m_w h_w = m_{ao}h_{ao}$$

$$m_{ai} + m_w = m_{ao}$$

$$\omega_{ai} + \frac{m_w}{m_{ai}} = \omega_{ao}$$

The EC effectiveness is defined by the equation below and it is adopted equal to 0,85 in the mathematical modeling.

$$\eta_{EC} = \frac{T_i - T_o}{T_i - T_{swb}}$$

In Fig. (3a) it is presented a scheme of an evaporative cooling system and Fig. (3b) shows the psychometric chart of evaporative cooling of the humid air.

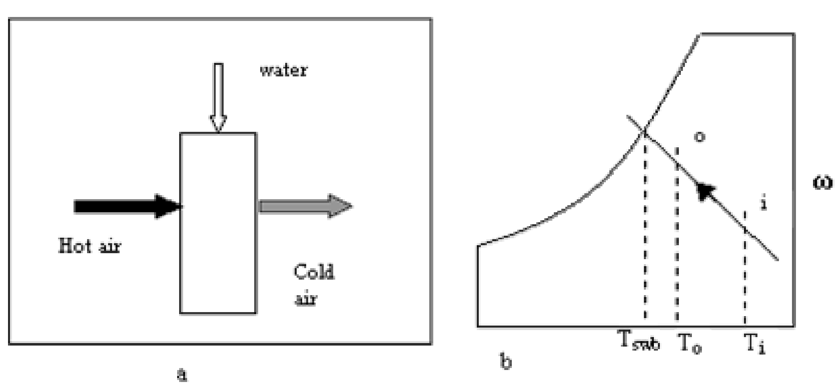


Figure 3 – a) Evaporative Cooling (EC) scheme. b) Evaporative cooling diagram of psychometric chart.

3.2. Absorption Cooling (AC)

AC is a cooling system that works based on the absorption cycle which transfers heat from a cold source to a hot source. The cycle utilizes two fluids: an absorbent and a refrigerant. Figure (4) shows AC where, inside the hatched line, it is presented the system for the refrigerant pressure rising. The main characteristic of this refrigeration system is to be driven basically by heat and not by work as it occurs in a vapor compression cycle (there is a small conumption of work to move the solution pump). In the generator, a concentrated solution of refrigerant receives heat, what provokes the formation of refrigerant overheated vapor that is directed to the condenser. Liquid refrigerant from the condenser passes through the expansion device and enters the evaporator where it vaporizes creating the cooling effect. Refrigerant liquid/vapor is moved to the absorber where it dissolves in the concentrated solution that comes from the generator.

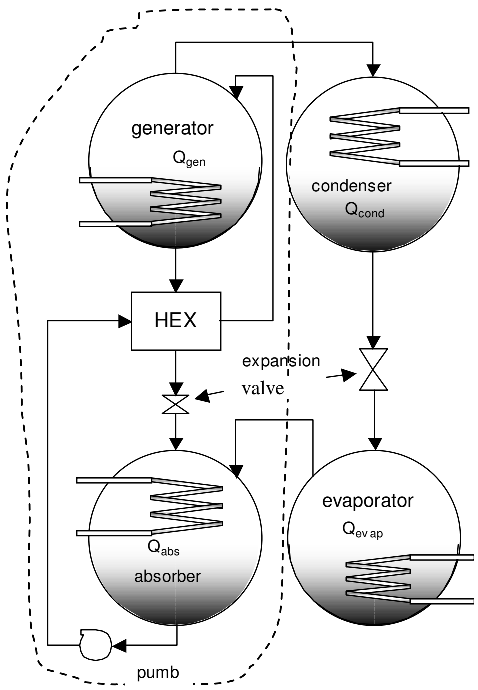


Figure 4 – Absorption cooling (AC) scheme.

Fig. (4) shows the presence of a internal heat exchanger used to increase the coefficient of performance preheating the solution that leaves the absorber. The mathematical modeling of the absorption cycle was developed through the use of mass and balance equations based on Herold ET all (1996).

Generator

$$m_{igen} = m_{ogen} + m_w$$

$$m_{igen} h_{igen} + Q_{gen} = m_{ogen} h_{ogen} + m_w h_{wgen}$$

Condenser

$$m_w h_{wgen} - Q_{cond} = m_w h_{wcond}$$

Water expansion valve

$$h_{wcond} = h_{wev}$$

Evaporator

$$m_w h_{wev} + Q_{evap} = m_w h_{wevap}$$

Absorber

$$m_{igen}h_{oabs} = m_{ogen}h_{iabs} + m_w h_{wevap}$$

Internal heat exchanger

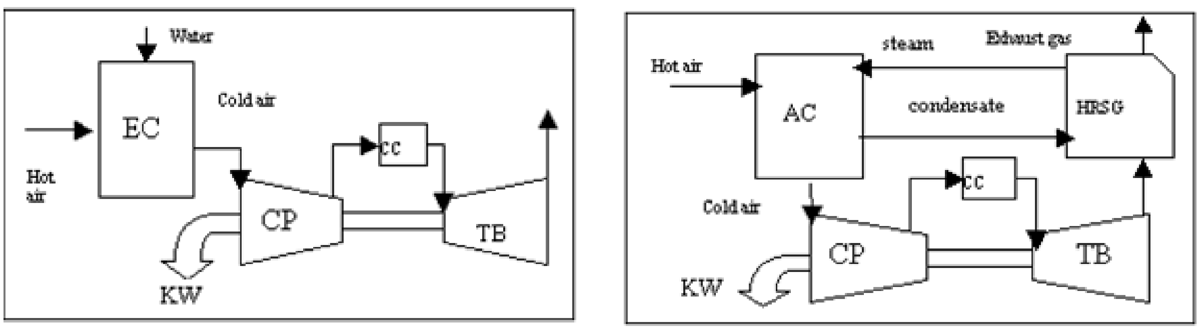
$$m_{igen}\left(h_{igen} - h_{oabs}\right) = m_{ogen}\left(h_{ogen} - h_{iev}\right)$$

Solution expansion valve

$$h_{iev} = h_{iabs}$$

3.3. Association of gas turbine with air-cooling systems.

Figure (5a) shows the GTEC system formed by GT associated with EC. Figure (5b) shows the association of a GT with an AC called GTAC. This system, that is more complex than the GTEC, involves the use of a heat recovery steam generator (HRSG) using GT exhaust gases. This component is necessary due to the fact that commercially available heat recovery absorption systems are driven by steam or hot water.



a b

Figure 5 – a) GTEC scheme. b) GTAC scheme

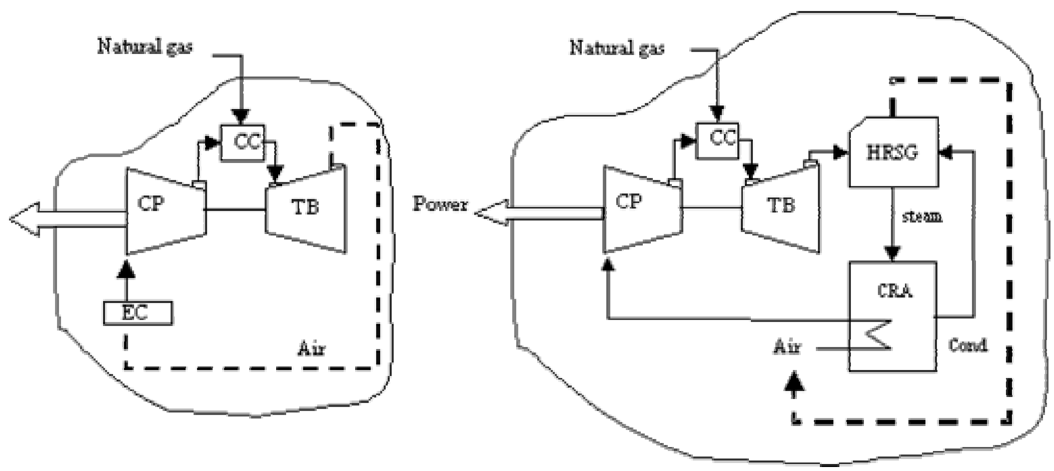


Figure 6 – Control volumes

3.4. Exergetic analysis.

A simplified exergetic analysis was developed considering a control volume that contains all the equipment and the entire atmosphere, whose conditions of air are different from that one in the equipment inlet. This assumption means that all heat exchanges needed to return the air to its original conditions are inside of the control volume, as shown in Fig. (6). In this case we have an adiabatic system, and the exergetic efficiency can be defined as:

$$\eta_{ex} = \frac{oulet\_exergy}{inlet\_exergy} = \frac{W}{Q\left(1 - \frac{T_0}{T_{cc}}\right)}$$

It was considered in the study the following parameters that characterize the *dead state* for the exergy analysis:

T_0 = 293,15K P_0 = 101,325 kPa UR_0 = 50 % X_0=50 % T_{cc} = 1273,15 K

4. RESULTS AND DISCUSSION

As it is showed in Fig. (7), the GT energy production is affected by the location where it is installed, due to the ambient air different thermodynamic conditions. The power is directly related to the air inlet pressure in the compressor, and indirectly to the temperature and relative humidity. Thus, keeping the dry air bulb temperature constant, the produced power decreases with the increase of the relative humidity. This influence is more pronounced at highest temperatures.

The altitude of the GT installation affects the barometric pressure and the air relative humidity and, consequently, the produced power. This can be observed in Fig. (8) where the GT power produced in Curitiba is lower compared to the power produced in Belém, for the same conditions of DB. The reason for that is the lower air density in Curitiba, due to the lower barometric pressure. The lower RH in Curitiba reduces this effect. As the volumetric flow is the same for both cities, the air mass flow is lower in Curitiba and the produced power is lower too. The specific-energy produced in Curitiba is greater but the power is lower.

Figure (9) shows the influence of the cooling system in the turbine performance. The power increase obtained with GTEC in Curitiba is greater than in Belém. This is due to the lower relative humidity in Curitiba, what makes the air-cooling more effective. GTAC has better performance, considering power production, than GTEC. The reason is the lower inlet air temperature and absolute humidity obtained by the GTAC. The influence of the GTAC is much greater in the city of Belém than in Curitiba. The reason is that the temperatures obtained after cooling in the two cities is nearly equal, but the temperature decrease is larger in Belém. The change in produced power is caused by climatic conditions variation in the cities, as can be seen in Fig. (9). Figure (10) shows the temperature decrease using EC.

Figure (11) shows the GTAC power variation. In contrast to the GTEC, the higher is the ambient air temperature and the relative humidity the larger is the increment in the power production. In the same way, for the GTEC the power production is directly affected by barometric pressure.

Another GTAC advantage is the exhaust gases temperature that is lower than the obtained in GTEC, due to the use of the exhaust gas energy to generate the vapor for the absorption cycle. Figure (12) shows the exergetic efficiency variation caused by reduction of the temperature and humidity of air with the use of GTEC and GTAC.

CONCLUSIONS

The air-cooling in the entrance of gas turbines is a good option for power production increase mainly during the period of energy consumption peak.

The performance improvement obtained with the cooling of the entrance air depends on the process used. Hotter and more humid places have better results with the GTAC than colder and driery regions. GTEC presents better results in dry than in humid places.

The increase of power obtained with the GTAC varies between 8 to 37 % more than the presented by GTEC.

Based on the simplified exergy analysis performed, the exergetic losses with the GTAC are 20 % lower than the values presented by GTEC. One improvement in the GTAC system is the exclusion of the boiler for vapor, or hot water, generation. In this case, using an absorption system that would employ directly in its generator the turbine exhaust gases, as analyzed by Varani, 2001.

More work is being carried out at Instituto Mauá de Tecnologia in order to improve the mathematical modeling and also to include other variables, including the system cost.

REFERENCES

AMELL, A, A., CADAVID, F.J., “Influence of the relative humidity on the air cooling thermal load in gas turbine power plant”, Applied Thermal Engineering 22 (2002) 1529-1533.

BASSILY,A.M., “Effects of evaporative inlet and aftercooling on the recuperated gas turbine cycle”.Applied Thermal Engineering 21 (2001) 1875-1890.

HEROLD, K.E., RADERMACHER, R., KLEIN, S.A, “Absorption Chillers and Heat Pumps”. CRC Press, Inc 1996.

EES – “Engineering Equation Solver”, F-Chart Software, Middleton, Wisconsin, 2002.

GOULART, S.V.G, LAMBERTS, R, FIRMINO, S, “Climatic data of 14 Brazilian cities”. ABRAVA, 1998.

MONE, C.D.,CHAU, D.S.,PHELAN, P.E., “Economic feasibility of combined heatand power and absorption refrigeration with commercially available gas turbines”, Energy Conversion and Management 42 (2001) 1559-1573.

NAJJAR, Y.S.H., "Enhancement of performance of gas turbine engines by inlet air cooling and cogeneration system", Applied Thermal Engineering Vol. 16 Nº. 2 (1996) 163-173.
PERRY, R.H, GREEN, D.W., "Perry's Chemical Engineer's Handbook". 7 th. ed., McGraw Hill Companies, 1997.
TALBI, M.M., AGNEW, B., "Exergy analysis: an absorption refrigerator using lithium bromide and water as the working fluids", Applied Thermal Engineering 20 (2000) 619-630.
VARANI, C. M. R., "Energetic and Exergetic Evaluation of an Lithium Bromide /Water Absorption Refrigeration System Utilizing Natural Gas". D. Sc. Thesis, Federal University from Paraíba, 2001.

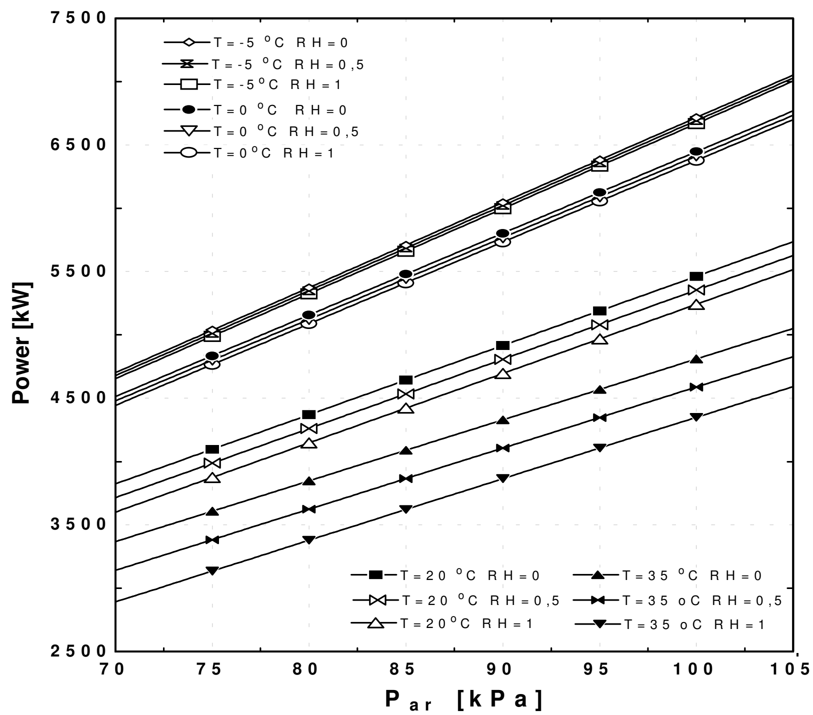


Figure 7 – Influence of inlet air pressure, temperature and relative humidity in the GT power.

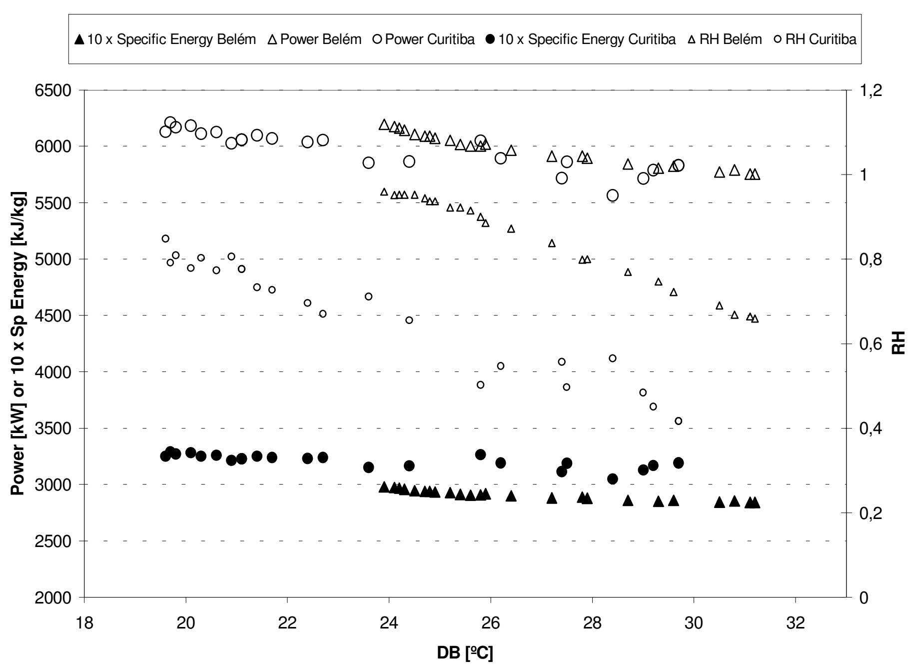


Figure 8 –Installation site influence in the GT power.

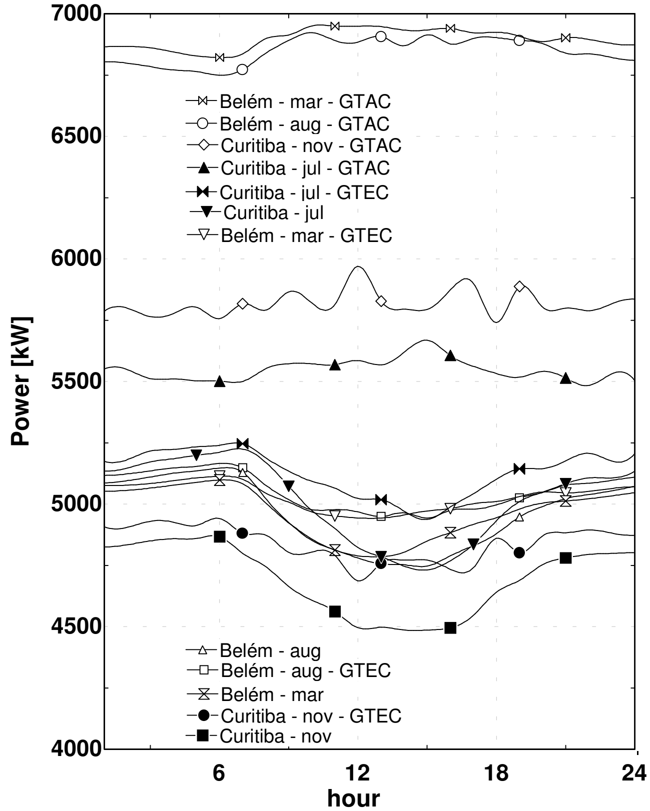


Figure 9 – Influence of the cooling-system in GT power during the day.

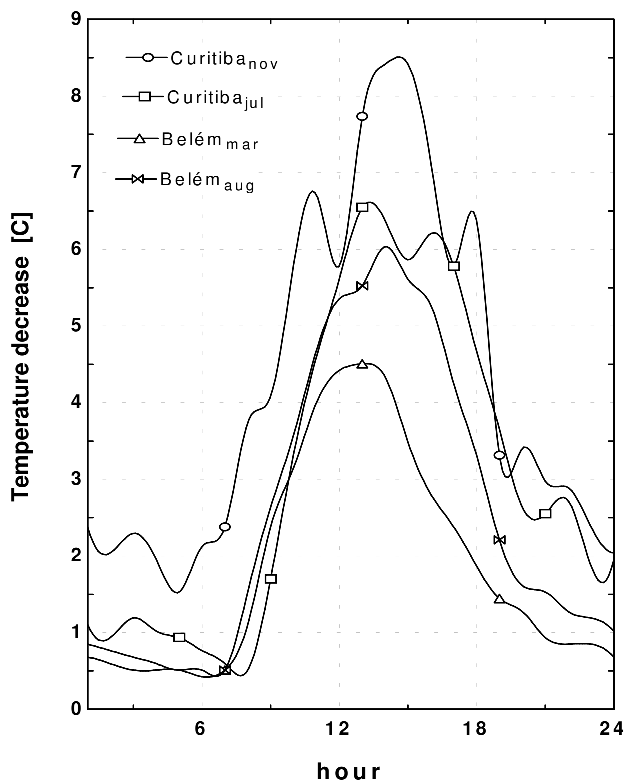


Figure 10 - Temperature decrease using EC in Belém and Curitiba.

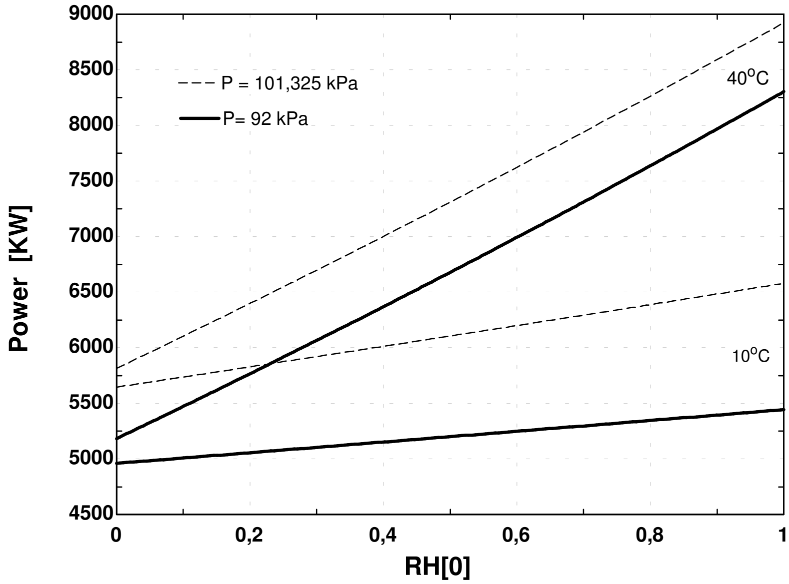


Figure 11 – Influence of the pressure, temperature and relative humidity in GTAC power.

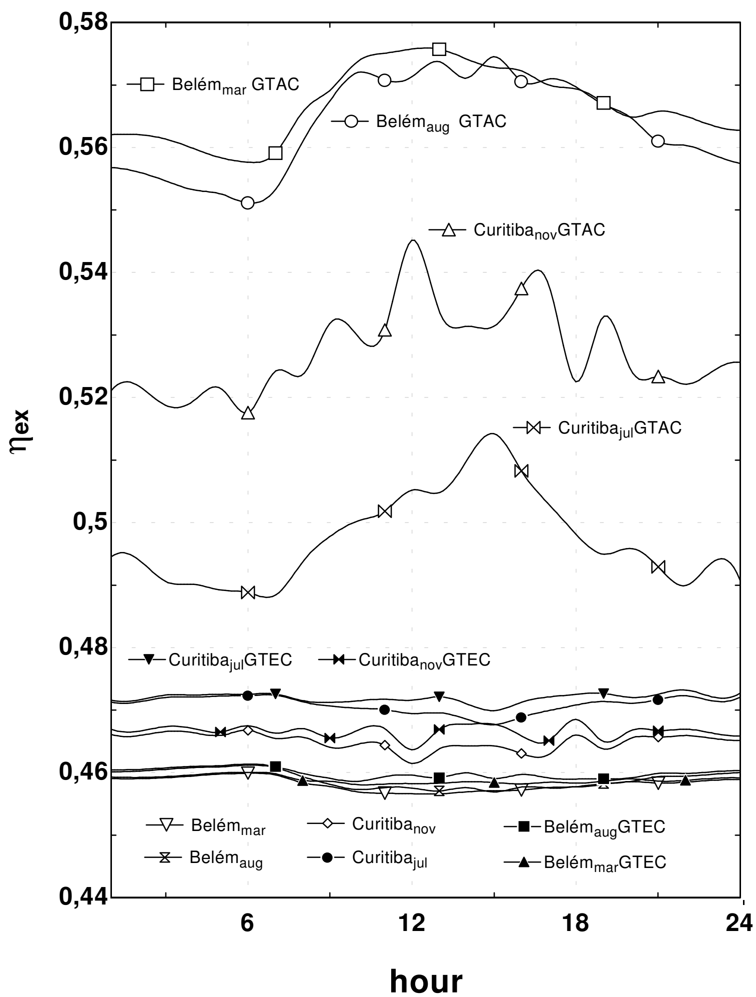


Figure 12 – Influence of the cooling-system in exergetic efficiency.