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INFLUENCE OF AIR PROPERTIES ON GAS TURBINE OPERATION UNDER OFF-DESIGN CONDITIONS

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Abstract. This work presents a study on the influence of atmospheric air conditions on the behavior of a gas turbine operating outside the reference conditions. With the computer model developed and validated, the performance of the Alstom turbine model GT11N2, which generates 115.4 MW on ISO conditions, was simulated. It operates in the Piratininga Thermoelectric Plant, located in the metropolitan region of the city of São Paulo. Using the data of temperature and relative humidity for each season of the year, the performance of this turbine operating without a heat recovery steam generator (HRSG) was analyzed by thermodynamic analysis, examining the values of generated power, heat rate and efficiency. As a way of improving performance, two air cooling technologies were simulated: mechanical compression and absorption chillers. Based on the obtained results, it was possible to observe that, for the case study analyzed, the use of cooling technologies are interesting proposals, from the energy point of view, but the increase in power and energy production is not much higher than the operation without air cooling technologies.

Keywords: absorption chiller, gas turbine, mechanical compression chiller, turbine inlet cooling

1. INTRODUCTION

Mass flow through gas turbines can be assumed relatively constant, considering that the mass flow of fuel is much smaller than the mass flow of air. Their power and efficiency vary significantly according to the pressure, temperature, and relative humidity of the air at the place where they are installed. (Guardi W., 2008)

Due to this influence, when a manufacturer indicates the properties of a turbine, these are presented under conditions specified by ISO 2314 (International Standards Organization ISO 2314 - Gas Turbines - Acceptance Tests: 1989), which indicates the requirements of the reference environment. These conditions are dry bulb temperature of 15 ° C, relative humidity (RH) of 60 %, wet bulb temperature of 7.2 °C and atmospheric pressure 101,325 kPa, and in this operating condition, the head losses at the inlet and outlet of the turbine must be disregarded. (Lora, and Nascimento, 2004)

According to Omidvar (2001), as observed in Fig. 1, considering a temperature of 25°C, heavy-duty turbines have a 10 % drop in power and about 5 % in the mass of flue gases, as well as a 3 % increase in the heat-rate and exhaust temperature of the flue gases. On the other hand, if the temperature of the incoming air in the turbine compressor is 5 °C, the increase in power and flue gases is around 5 %, while the decrease in heat rate and flue gas temperature can be approximated to 2 % and 3 %, respectively. Cooling the intake air enables the turbine to operate at temperatures of 15 °C or even lower, which increases power generation, and efficiency.

The humid air has a lower specific mass than the dry air, which reduces the mass flow, the power output, and also the heat-rate. For smaller values of specific humidity, the correction factor, that takes into account the off-design operation, for power is higher than one, which increases the power generated. Under these conditions, the heat-rate correction factor is less than one, decreasing its value and thus increasing efficiency. On the other hand, for high values of specific humidity, the correction factor for power is less than unity, decreasing the power generated and the heat-rate correction factor is higher than one, increasing its value and thus penalizing efficiency. (Lora and Nascimento, 2004)
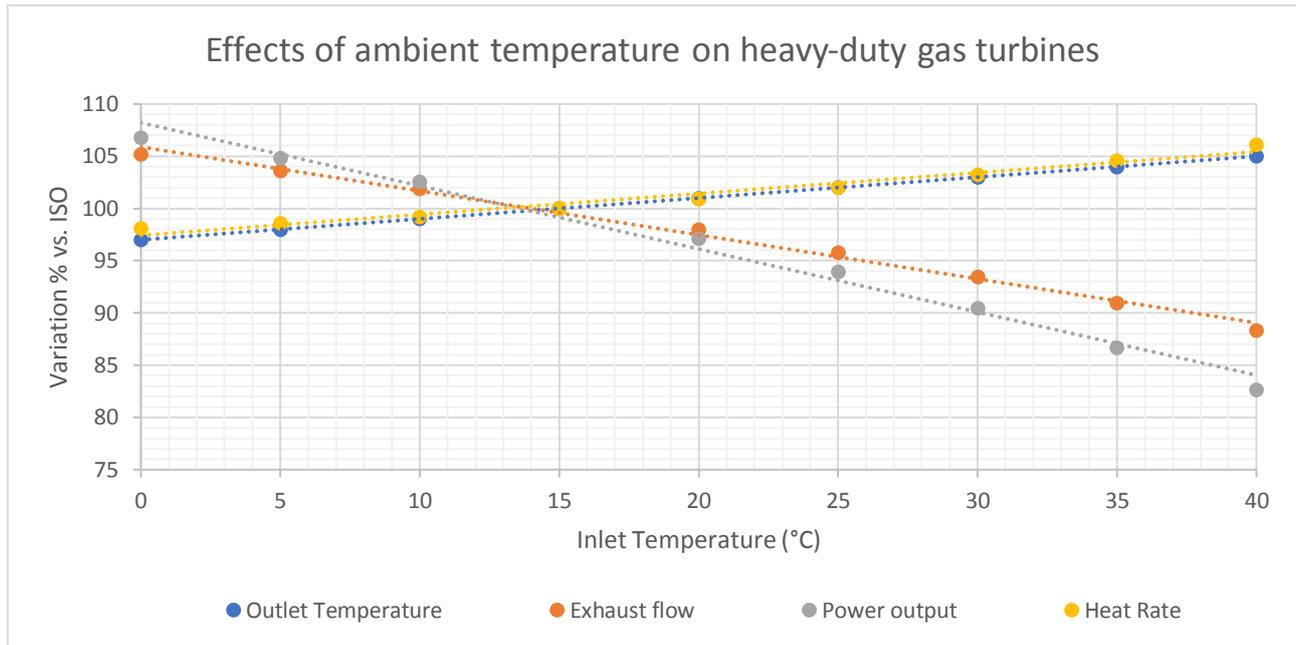
Figure 1. Effects of ambient temperature on heavy-duty turbines
(Adapted from Omidvar, 2001)

It should also be considered the altitude of the site since this will have a direct influence on the atmospheric pressure and, therefore, on the turbine performance, less dense air reduces the mass flow and output power proportionally. A linear correction factor must be applied to multiply the power at ISO conditions (at sea level) to account for the site elevation. This correction reaches the value of 0.92 for a site with an altitude of 600 meters above sea level, which corresponds to a power decrease of 8 % for this elevation. (Lora and Nascimento, 2004)

Santos and Andrade (2012) compare the influence of two different air cooling systems (mechanical compression and absorption chillers), for a situation where the temperature is 36 °C and relative humidity at 18 %. The turbine without any cooling system generates 33.17 MW. With a mechanical chiller of COP 4.5, there is an increase of 4.5 MW, and for an absorption chiller, there is an increase of 5.3 MW, both cooling the air up to 10 °C. No pressure loss was considered. In the same paper, for the mechanical compression systems, it is only necessary to supply electric power and water for the heat exchange in the condenser. Using this system, it was possible to observe that in an environment with a temperature of 30 °C and relative humidity 18 %, with a refrigeration system with COP 4.0 cooling the air to the temperature of 10 °C, there was a gain of 13.5 % in power produced.

Popli et al. (2013) emphasize that absorption chillers require only a tenth of the electrical energy consumed by mechanical chillers. For a turbine that generates 7 MW in an environment of 55 °C and relative humidity of 80 %, using three one stage lithium bromide-water absorption chillers, with individual capacity of 4.1 MW and COP of 0.77, the air was cooled to 10 °C, which caused an increase of 23.3 % in net power and 13 % inefficiency.

In the face of the above, the main objective of this paper is to study the influence of atmospheric air on the behavior of a gas turbine operating outside the reference conditions. As a second objective, the most suitable of air treatment technology for the specific operating conditions of a gas turbine will be determined by the development of a routine to simulate the behavior of gas turbine operating outside the ISO condition.

2. BASE CASE AND CONSIDERATIONS

To verify the influence of the air properties on the behavior of a gas turbine, the overall performance of a turbine was analyzed. Thermodynamic analyzes were performed, and also, according to the presented literature, correction factors were obtained and were used in routines developed in the Engineering Equation Solver (EES) software, computational models were developed to determine the behavior of the turbine and also of the technologies of air cooling. As a base case, the Alstom GT11N2 - 60 Hz turbine that operates at the Piratininga Thermoelectric Plant, located in the metropolitan region of the city of São Paulo, was studied. Tab. 1 presents the properties of this turbine under ISO conditions

Table 1. ISO proprieties for Alstom GT11N2 – 60 Hz turbine (Gas Turbine Handbook. 2015)

Parameter	ISO value
Power	115.4 MW
Heat Rate	10620.2 kJ/kWh
Efficiency	33.9 %
Pressure Ratio	15.9
Exhaust gas mass flow	400 kg/s
The temperature of the exhaust gases	526.1 °C

The altitude of the site where the power plant is installed was obtained through a topographic map and was fixed at a value of 739 meters. Silva (2018) made a survey of ambient temperature and relative humidity for each hour, every day of the year for the city of São Paulo for 2017. These data were grouped according to the seasons. The mean temperature and relative humidity of each season, as well as their standard deviation (σ) were obtained and can be observed in Tab.2.

Table 2. Temperature and Relative Humidity according to the seasons of the year for the city of Sao Paulo in 2017

Season	Mean Temperature (°C)	Standard Deviation for Temperature (°C)	Mean Relative Humidity (%)	Standard Deviation for RH (%)
Summer	23.26	± 3.44	71.03	± 14.63
Fall	19.32	± 3.41	71.38	± 13.86
Winter	18.21	± 4.60	63.57	± 19.96
Spring	21.30	± 4.09	66.99	± 16.83

The head loss of the incoming air was determined by Santos (2012) and Şahim (2005). The pressure drop of the filters was assumed to be 98 Pa; for the air cooler, the value considered was 150 Pa, such that the total pressure drop considered in the simulation was 248 Pa. It is essential to notice that, according to Şahim (2005), the air side head loss should not exceed 254 Pa at the inlet. The head loss of the gases at the turbine outlet, associated with the use of the absorption chiller, was assumed to be 1960 Pa and the relative humidity of the air at the exit of the air cooler was considered 90 %.

The Chiller Carrier - Variable -Speed Water-Cooled Liquid Chillers - 30XW-V, model 1470 was used for modeling the chillers for the mechanical compression. This system has a cooling capacity of 1461 kW and COP of 5.38. Similarly, by consulting datasheet of leading manufacturers of absorption refrigeration systems, the absorption chiller was modeled according to the Trane Classic Absorption Series Absorption Chiller; Single-Stage Hot Water or Steam-Fired Absorption Water Chillers, model 420. Such chiller has a single stage and works with the lithium bromide/water pair. It can operate either with hot water or steam as a source of heat. It has a cooling capacity of 1402 kW and COP of 0.64. Fig. 2 shows the configuration of the turbine working with mechanical chiller and Fig. 3 indicates the configuration using the absorption chiller as air treatment.
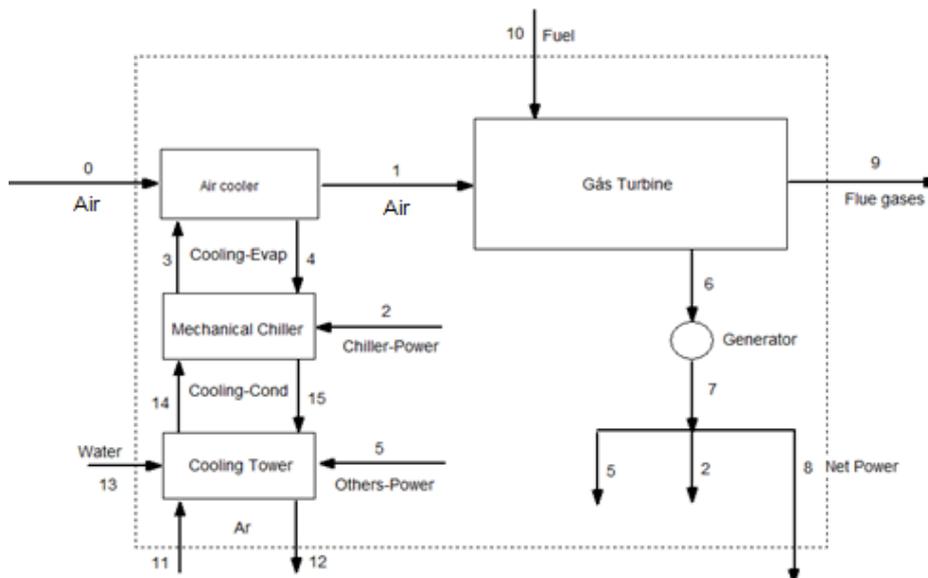

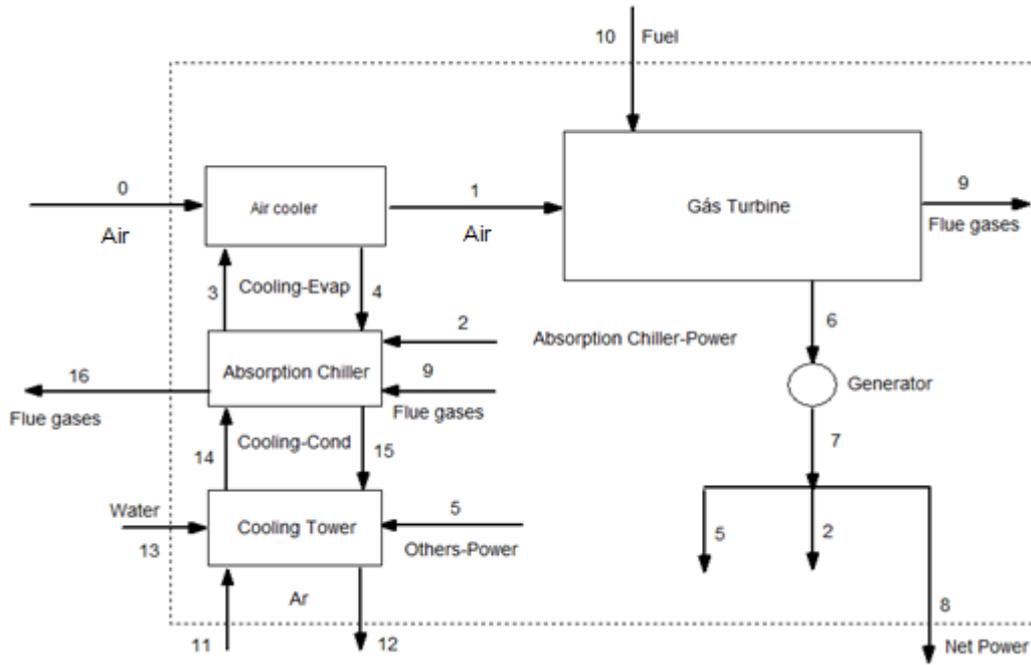Figure 2. Configuration for turbine and mechanical chiller (Authors)

Figure 3. Configuration for turbine and absorption chiller (Authors)

For both cases, it was considered that the generator had an efficiency of 96 %. The temperature of the water entering the condenser was 30 °C and the exit temperature was 35 °C, heading to the cooling tower; in the evaporator, it enters at 12 °C and leaves at 7 °C. Chilled water at 7 °C is pumped through a radiator, responsible for cooling inlet air. The pump efficiency was adopted as 87 %.

For the calculation of the number of machines required, the energy needed to cool the incoming air up to 15 °C was compared with the energy available in the chosen chiller. Since the number of chillers operating can only be an integer, it was considered the largest integer closest to the calculated one. Then, according to ASHRAE (1999), functions were used to correct chiller power as a function of load, and all conditions of chillers operation were amended. The following equations were used in this work for the control volumes at steady state.

Overall mass balance: Eq. (1).

$$(\sum \dot{m}_i)_{out} = (\sum \dot{m}_i)_{in} \tag{1}$$

Energy balance: Eq. (2).

$$\dot{Q} + (\sum \dot{m}_i h_i)_{in} = (\sum \dot{m}_i h_i)_{out} + \dot{W} \tag{2}$$

The thermal efficiency of the cycle: Eq. (3).

$$\eta_{cycle} = \frac{W_{net}}{m_{fuel} \times LHV_{fuel}} \tag{3}$$

The exergy efficiency of the cycle: Eq. (4).

$$\eta_{exergy} = \frac{Ex_{product}}{Ex_{fuel}} \tag{4}$$

The exergy of product of the plant was considered the net power, and the total exergy of fuel of the plant was calculated by Eq. (5), where Szargut et al. (1988) provided values of the exergy to energy ratio, Φ, by Eq. (6).

$$Ex_{fuel} = \Phi \times LHV_{fuel} \tag{5}$$

$$\Phi = 1.0334 + 0.0183 \times (H/C) - 0.0694 * (1/N_C) \tag{6}$$

For exhaust gas mass flow was considered the following composition of gases, in mass fraction: 0.7126 nitrogen; 0.1773 oxygen; 0.0699 carbon dioxide; 0.0227 water vapor and 0.0175 argon (the mass fraction describe the relative amount of a given species). Moreover, this composition was used to calculate the energy exhaust gas in Eq. (7), where it was considered the exhaust gas mass flow, the specific heat of exhaust gas and the difference between the temperature of exhaust gas and the temperature reference of 25°C. The specific heat of exhaust gas was obtained from the product of mass fraction of each gas in exhaust gas by the specific heat of each gas in the temperature of exhaust flow (Eq. (8)).

$$Q_{exhgas} = m_{gas} \times cp_{exhgas} \times \left(T_{exhgas} - T_{ref}\right) \tag{7}$$

$$cp_{exhgas} = \left(\sum_1^n x_{i,exhgas} \times cp_{i,exhgas}\right) \tag{8}$$

3. RESULTS

A mathematical model was developed to correct the properties of the gas turbine when working outside the ISO conditions. The final value of each parameter is obtained by multiplying the ISO value by the correction factors obtained, also considering the values of the head loss. Tab. 3 summarizes the corrections factors.

Table 3. Correction factors for heavy-duty gas turbine

Independent Variable	Parameter	Correction Factor
Ambient Temperature	Outlet gases temp.	CF_gas.out = 0.002*T_amb + 0.97
	Outlet gases mass flow	CF_gas.mf = -3E-05*(T_amb)² - 0.003*T_amb + 1.0497
	Power	CF_power = -6E-05*(T_amb)² - 0.0036*T_amb + 1.061
	Heat Rate	CF_HR = 3E05*(T_amb)² + 0.0008*T_amb + 0.9825
Altitude	Power	CF_power = -0.0002*(alt) + 1.002
Absolute Humidity	Power	CF_power = -0.1443*ω + 1.001
	Heat Rate	CF_HR = 0.3746* ω + 0.9976

Patel (2000) brings an interesting study of the relationship of ambient temperature in the parameters studied. Pressure drop was allowed at the outlet of 1.8 kPa. Using his data in the validation, the results shown in Tab. 4 are obtained. The values of power and efficiency were collected in the paper the values of calculated power and calculated efficiency were obtained with the developed model, using the same conditions observed in each case. As the relative error is lower than 5 %, the model was considered valid.

Table 4. Relative error obtained with the model developed

Turbine	Temperature (°C)	Power (MW)	Efficiency (%)	Calculated Power (MW)	Calculated Efficiency (%)	Relative Error: Power (%)	Relative Error: Efficiency (%)
GT10B	15	23.10	33.00	22.69	32.67	1.77	1.00
	35	20.10	32.11	19.64	31.06	2.29	3.27
	40	18.90	31.80	18.69	30.54	1.11	3.96
Frame 5 (3571 PA)	15	25.20	28.00	24.76	27.72	1.75	1.00
	35	21.40	26.40	21.42	26.36	0.09	0.15
	40	20.40	25.99	20.39	25.91	0.05	0.04

After the validation, the model was applied to the Alston turbine, considering the altitude, temperature, and humidity at Piratininga location. The results can be seen in Tab. 5. The power reaches 89.3 MW in summer and 92.3 MW in winter, which represent decreases of 22.6 % and 20.0% in power output respectively. As expected, in the summer, the biggest drop was observed, because of the highest temperatures. The annual energy generated corresponds to 785 GWh, a decrease of 225 GWh, when compared to ISO condition (a drop of 22.3 %).

When there are chillers associated with the air filter, the pressure drop at the entrance is 248 Pa. There is no pressure drop at the output. It was also considered the power consumed by fans and pumps, besides the power consumed by the chiller chosen. The air in the inlet is cooled to 15 °C with a relative humidity of 90 %. Tab. 6 presents the results, it can be seen that the net power reaches only 94.5 MW in the best scenario, it represents a decrease of 18.06 % when compared with ISO conditions. However, when the results of Tab. 6 are compared to those in Tab. 5 it is possible to highlight the increase of the net power and energy production (about 3.3 %) and the energy and exergy efficiency (1.15 %).

Table 5. Turbine parameters corrected for each season

Parameters	ISO	Summer	Fall	Winter	Spring
Power (MW)	115.4	89.3	91.6	92.3	90.5
Energy (GWh)	1011[*1]	192.9[*2]	198.0[*2]	199.3[*2]	195.5[*2]
Heat Rate (kJ/kWh)	10620	10836	10736	10706	10782
Efficiency (%)	33.9	33.2	33.5	33.6	33.4
Exhaust gas mass flow (kg/s)	400	386	392	394	389
Temperature flue gas (°C)	526	535	531	530	533
Energy Exhaust gas (MJ/s)	201.1	209.5	211.4	211.8	210.5

[*1] – Annual power production of the gas turbine (Power x 24 hours per day x 365 days per year).
[*2] – Three months of power production of the gas turbine (Power x 24 hours per day x 30 days per month x 3 months).

Table 6. Turbine parameters when operating with mechanical chillers

Parameters	Summer	Fall	Winter	Spring
Turbine Power (MW)	94.6	94.6	94.6	94.6
Chiller Power consumption (MW)	1.37	0.52	0.10	0.80
Net power (MW)	93.2	94.1	94.5	93.8
Energy (GWh)	201.4	203.2[*1]	204.1[*1]	202.6[*1]
Heat Rate (kJ/kWh)	10,637	10,637	10,637	10,637
Efficiency (%)	33.8	33.8	33.8	33.8
COP	1.55	1.54	1.27	1.53
Number of chillers	6	2	1	3
Exhaust gas mass flow (kg/s)	400	400	400	400
Temperature flue gas (°C)	526.1	526.1	526.1	526.1
Energy Exhaust gas (MW)	213.5	213.5	213.5	213.5
Exergy efficiency (%)	30.9	31.2	31.3	31.1
Flue gas exergy (MJ/s)	90.2	91.9	92.4	91.0

[*1] – Three months of power production of the gas turbine (Power x 24 hours per day x 30 days per month x 3 months).

When the absorption chiller is associated with the turbine, the head loss both at the inlet and at the outlet was considered. The same head loss value is maintained at the inlet, while a drop of 1.96 kPa at the outlet is considered. The air in the inlet is cooled to 15 °C and relative humidity of 90 %. The thermal energy consumption was calculated by multiplying the number of chillers used by the amount of heat required in the generator of each chiller. Tab. 7 shows the values for each season of the year, obtained in the simulation of gas turbine associated with the absorption chillers. Analyzing the results, it can be seen that the net power reaches only 94.6 MW in the best scenario, which represents a drop of 18.4 % when compared to ISO conditions. Comparing annual energy produced with the mechanical chiller (811 GWh year) to that produced with absorption chiller (817 GWh year), the difference represents more than 0.71 %.

Table 7. Turbine parameters when operating with absorption chillers

Parameters	Summer	Fall	Winter	Spring
Turbine Power (MW)	94.6	94.6	94.6	94.6
Chiller Power consumption (MW)	0.04	0.01	0.01	0.02
Thermal energy consumption (MW)	64.9	8.3	0.5	19.6
Net power (MW)	94.5	94.6	94.6	94.6
Energy (GWh)	204.2[*1]	204.3[*1]	204.3[*1]	204.3[*1]
Heat Rate (kJ/kWh)	10,646	10,646	10,646	10,646
Efficiency (%)	32.5	32.5	32.5	32.5
COP	0.68	0.65	0.77	0.64
Number of chillers	6	2	1	3
Exhaust gas mass flow (kg/s)	400	400	400	400
Temperature flue gas (°C)	379.6	507.6	525.1	482.3
Energy Exhaust gas (MW)	148.6	205.2	213.0	193.9
Exergy efficiency (%)	31.3	31.3	31.3	31.3
Flue gas exergy (MJ/s)	51.9	86.7	92.1	78.9

[*1] – Three months of power production of the gas turbine (Power x 24 hours per day x 30 days per month x 3 months).

4. CONCLUSION

The performance of the turbine GT11N2, manufactured by Alstom, was simulated. This turbine operates in the Piratininga thermoelectric plant, located in the south of São Paulo city. Under ISO conditions, it produces 115.4 MW of power with an efficiency of 33.9 %.

The average temperature and relative humidity and their deviation were obtained for each season of the year for São Paulo. It was observed that in the year 2017, on average, these values were not so high, with the highest temperature value calculated being 26.7 °C and the lowest, 13.6 °C; the air humidity varied between 85.7 % and 43.6 %, values observed in summer and winter respectively. The filter system is responsible for a drop of 0.098 kPa on the inlet pressure.

Applying the relevant corrections for temperature, altitude and relative humidity, the decrease in power varied, as expected, between 89.3 MW in summer and 92.3 MW in winter, corresponding to reductions of 22.6 % and 20.7 %, respectively. The heat-rate variation was between 0.81 % in winter and 2 % in summer. When comparing the annual energy generated, a drop of 22.3 % was observed.

The mechanical chiller uses part of the power generated by the turbine for its operation, reducing the net power produced. There is an additional inlet pressure drop of 0.15 kPa, and there is no head loss at the output. When the mechanical chiller was associated with the turbine, produced 93.2 MW in the summer and 94.5 MW in the winter, an increase of 4.4 % and 2.5 %, respectively, when compared to the turbine without cooling. When comparing the energy generated, there was an increase of 3.3 %, generating 811,3 GWh.

When the absorption chiller was associated to the turbine, an additional 1.96 kPa pressure drop was computed at an outlet, this type of chiller consumes much less power from the turbine, instead, it uses the energy contained in the exhaust gases. The power produced was 94.5 MW in the summer and 94.6 MW in the winter, an increase of 5.8 % and 2.4 %, respectively, when compared to the situation without any cooling. The energy generated is 4.0 % higher than that generated without any air-cooling technology, reaching 817.1 GWh in the year studied. It was observed that the power generated with the use of mechanical chillers was smaller than the power generated with the absorption chiller. This is because of the higher electric power consumption of the compression cooling system.

Is necessary to highlight, that despite the low electricity consumption in absorption refrigeration chiller, its relatively small performance is attributed to the high head loss at the output and also more significant power consumption on the cooling tower pumps. It can also be noticed that water consumption is higher when the absorption chiller is used, reaching almost three times the amount of water used when operating with mechanical chiller.

Based on the obtained results, it was possible to observe that, for the case study analyzed, the use of cooling technologies are interesting proposals from the energy point of view, but the power gain and energy production are not much higher than the operating condition without air cooling technologies. It is necessary to improve the analysis, for example, obtaining more details about the gas turbine operation during the day or by the hour because it is possible that, in some periods the air temperature inlet of the gas turbine is much higher than the temperature averages used in this study.

It is necessary to make a second law analysis to identify the relevant irreversibilities for each system and propose possible improvements. An economic analysis would also be essential because the increase of the energy production between 3.3 % (25.6 additional GWh from turbine associated with mechanical chiller) and 4.0 % (additional 31.4 GWh from turbine associated with an absorption chiller) may be attractive alternatives from the economic point of view.

5. NOMENCLATURE

$cp_{i,exhgas}$	specific heat of each gas in the temperature of exhaust flow: kJ/kg-K
cp_{exhgas}	specific heat of exhaust gas: kJ/kg-K
η_{exergy}	exergy efficiency
Ex_{fuel}	total exergy of fuel of the plant: kW
$Ex_{product}$	total exergy of products of the plant: kW
h_i	enthalpy of the flow at stream "i": kJ/kg
H/C	the atomic ratio of hydrogen and carbon
LHV_{fuel}	Low Heating Value of the natural gas: kJ/kg (50,023 kJ/kg);
m_{fuel}	fuel consumption: kg/s
m_{exhgas}	the exhaust gas mass flow: kg/s
$\dot{m}_i$	mass flow at stream "i": kg/s
N_c	mean number of carbon atoms in the molecule
$\dot{Q}$	heat generated or rejected: kW
$\dot{Q}_{exhgas}$	energy exhaust gas: kW
T_{exhgas}	Temperature of exhaust gas: °C
T_{ref}	Temperature reference of 25°C
$\dot{W}$	power produced or consumed: kW

W_{net}	net power: kW
Φ	exergy to energy ratio for fuels
$x_{i,exhgas}$	mass fraction of each gas in exhaust gas

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