PERFORMANCE ANALYSIS OF A COMBINED CYCLE FOR ELECTRICITY GENERATION WITH GAS TURBINE COMPRESSOR INLET AIR COOLING

F. Rodrigues, nano_r221@hotmail.com[1]
P. R. Wander, Prwander@unisinos.com[2]

[1] Mechanical engineering undergraduate course, Unisinos, 950 Unisinos Ave, São Leopoldo, 93022-750, Brazil
[2] Mechanical engineering graduate program, Unisinos, 950 Unisinos Ave, São Leopoldo, 93022-750, Brazil

Abstract: The increase in demand for electric power generation and the search for greater reliability in the Brazilian energy sector are obtaining great prominence in recent decades. Thermoelectric plants have gained increased attention because they have more flexibility and the possibility of installation near consumption areas, different from hydroelectric plants, which are the main source of electricity generation in Brazil. Thus, to enable the use of non-renewable energy sources with less environmental impact, a model was designed in EES software for the simulation of possible gains in electricity generation and efficiency in a combined cycle through the use of an inlet air cooling system for the gas turbine compressor using a direct expansion mechanical vapor compression refrigeration system. This happens because the performance of the gas turbine operation is greatly influenced by ambient temperature. To validate this model, comparisons were made with actual data collected from the processes of a thermal power plant operating in combined cycle in the metropolitan region of Porto Alegre, Brazil. To simulate the cooling system, technical data from an air conditioner manufacturer were used to generate an equation that calculates the cooling capacity and electric power according to, at the same time, the wet bulb temperature in the evaporator and the dry bulb temperature at the condenser. It was found that adding the cooling to the air entering the gas turbine compressor there is an average increase in overall efficiency of the combined cycle of 1.82% and a power generation increase of 5.37%, attaining 260 MW with the use of Fuel Oil for Electric Turbine (OCTE). This represents, for this plant, a possible increase of more than 70 GWh in annual capacity of electric power generation.
Keywords: Gas Turbine. Combined Cycle. Energy. Thermal Systems.

1. INTRODUCTION

Brazil is a country that has abundant water resource, then the main source of electricity generation is hydropower, which, according to the National Energy Balance (EPE 2015) report, corresponded to 65.2% of the domestic electricity supply in 2014. But with technological developments, efficiency and thermal reliability increases, generation scenario in Brazil is changing gradually.

By the early 2000s, the Brazilian energy matrix was basically composed of hydropower, which makes the system very vulnerable to hydrological conditions. In 2001, the lack of rain coupled with the low reliability of the transmission systems forced the government to implement a rationing system so that reservoir levels were resettled.

Since then, the growth of the thermoelectric participation in power generation is remarkable, according to the National System Operator (ONS, 2015) the average monthly thermal generation in 2014 was 10301.75 GWh, while in 2000 this figure was only 1252.57 GWh/month, indicating a growth of 722% . However, the hydroelectric generation grew by only 17% in the same period, ranging from 27965.56 GWh/month, in 2000, to 32715.41 GWh/month in 2014.

Because it is a non-renewable energy source, often the thermal energy becomes much more expensive compared to others. So, to extract the maximum performance in thermal power plants, project optimizations, improvements in equipment, materials, transmission lines are made and also the combination of different generation sources.

Being a country with a tropical climate, Brazil has high temperatures most part of the year, which ends up harming the thermodynamic cycle , as the efficiency and the energy generated by a gas turbine is directly influenced by the pressure, temperature and humidity in compressor air inlet. The temperature has the greatest influence on the energy generation, the higher the temperature the lower will be the generated power. (Carvalho Junior, 2012).

For these reasons, a study in the performance analysis of a combined cycle plant operating with the installation of a cooling system at the gas turbine compressor inlet is performed.

2 LITERATURE REVIEW

2.1 Combined Cycle

According to Çengel (2001) gas turbines operate in the Brayton thermodynamic cycle. In this cycle, the ambient air enters at the gas turbine, it passes through a compressor to get pressure increase. Fuel is then injected into the high-pressure air and ignited in the combustion chamber. The combustion's hot gas mass flows to the turbine section and produces the work in turbine shaft to drive the generator shaft and so generate electricity. Part of the generated work is also used to drive the compressor. The hot gas leaves the turbine, and in a combined cycle, it passes through a heat recovery steam generator (HRSG) to recover part of its wasted heat. The generated steam is used in a steam turbine to generate electricity, by the Rankine cycle.

Kelhofer et. al (1999) states that when two thermal cycles are combined in a generation plant, the efficiency that can be achieved is higher than the simple cycle. By combining two cycles, the cycle that works at a high temperature is called topping cycle and the cycle that works with the topping cycle rejected temperature is called bottoming cycle. The most developed kind of combined cycle is between the Brayton and Rankine cycle.

Boyce (2010) states that with the use of a combined cycle, the thermal performance can reach values very close to 60%. In the most useful combinations the gas turbine produces about 60% of total power while the steam turbine corresponds to others 40%. The thermal performance of gas and steam turbines is between 30% and 40%. The world record performance for a combined cycle plant belongs to a Germany installation, which uses a single shaft combined cycle turbine SGT5-8000H, by Siemens manufacturing, where the cycle amounted to 60,75% overall efficiency in 2011 (SIEMENS, 2011).

2.2 Gas Turbine Inlet Air Cooling

Çengel (2001) explains that the compressor power consumption depends on the ambient temperature, and states that for every 3 °C of decrease in ambient temperature the power consumed by the compressor decreases by 1%, increasing the efficiency of the cycle. Sa (2011) demonstrated in tests on gas turbines that each 1 °C increase in ambient temperature, above ISO conditions, involves the reduction of 0.1 % in thermal efficiency of the tested turbines. The temperature effect on the gas turbine output variables can be seen in fig. 1.

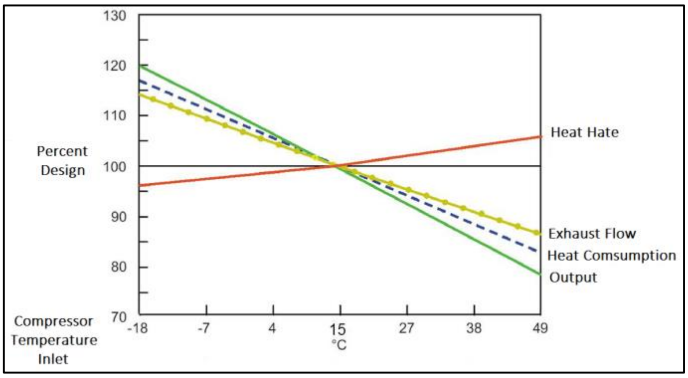

Figura 1 - Inlet Air Compressor Temperature Effect in the Power Output, Heat Rate and Exhaust Gas Flow of a Gas Turbine GE MS7241(FA). GE (2003)

As mentioned, the compressor inlet air temperature is one of the most influent aspect in the efficiency and power delivered by the gas turbine. This is because the air specific volume is proportional to its temperature, thus decreasing the air entering mass flow in warmer ambient conditions, since the turbine is a machine that works at constant volume.

According to Kehlhofer et al (1999) as the air specific volume is higher at elevated temperatures the compressor mass flow is reduced, so the gas turbine mass flow will also affect the amount of energy passing through the HRSG.

The conditions set for the design of gas turbines are ISO conditions (ambient temperature of 15 °C , relative humidity of 60% and air pressure of 101.3 kPa) (GE, 2003) . These conditions are rarely found in tropical countries, such as Brazil. According to INMET (2015) data for the state of Rio Grande do Sul, more precisely in the Porto Alegre region, the annual average measured in 1961-1990 is 19.5 °C and maximum average temperatures are around 24.8 °C with peaks of 30.2 °C in summer, which means a considerable difference to the design conditions.

According to Boyce (2010) to perform the compressor inlet air cooling some methods can be used, but the most common are the evaporative method; absorption cooling method or mechanical vapor compression with thermal energy storage method.

Jonsson an Yan (2005) classified the available inlet air cooling systems into the following groups:

- Evaporative Media Coolers: Water is distributed over saturated evaporative cooling media through which the air passes to be humidified;
- Spray inlet coolers or fogging systems: Water is injected into the air through nozzles, located in the compressor inlet, and creates a fog of small water droplets.
- Mechanical vapor compression or absorption chillers, where a heat exchanger cools the inlet air. Chillers can increase the gas turbine power output by 15e20% and the efficiency by 1e2% (i.e. if gas turbine exhaust gas energy is recovered). A chiller can cool the inlet air regardless of the ambient conditions; however,the specific investment cost is much higher than for evaporative media and spray coolers.

The cooling method selected for this study was the method of cooling by mechanical vapor compression.

The general configuration of this type of cooling can be seen in Fig. 2a, while the cooling process in a psychrometric chart is observed in Fig. 2b. One advantage of this method is that it does not depend on air moisture condition to cool, being more effective in humid regions, also promoting greater decrease in air specific volume compared to the evaporative method.

The mechanical refrigeration by direct expansion of a compressed liquid occurs in an evaporator, which is composed of coils and is installed in the gas turbine inlet filter house. Ammonia is an excellent coolant and can be used in these cases, but one should be careful with possible fluid leakage, requiring the installation of leak detectors in the inlet air compartment.

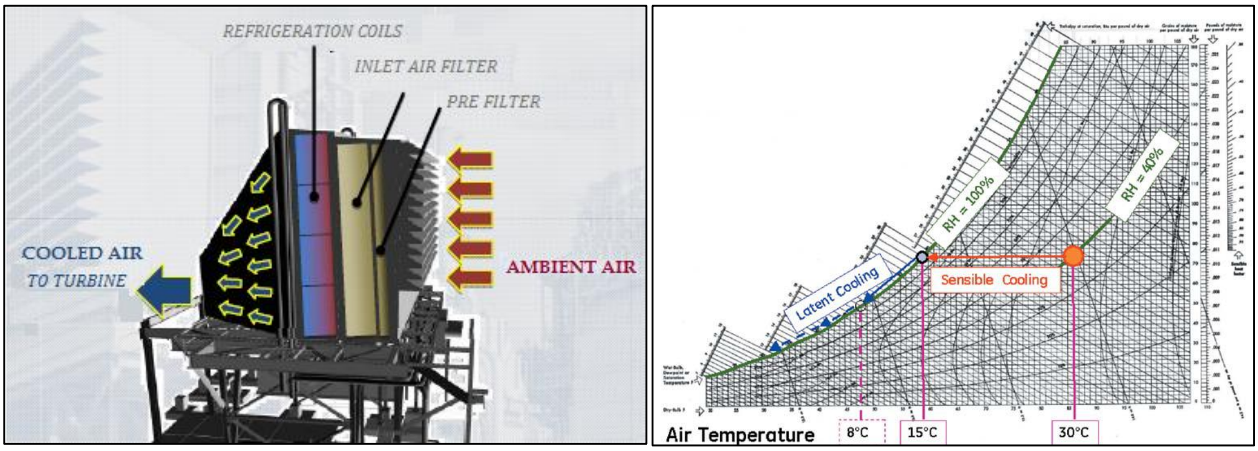

Figure 2 – a) Inlet air cooling installation. Stellar-energy (2012); b) Mechanical vapor compression refrigeration effect. GE (2008)

3 MAIN EQUIPMENTS

3.1 The Thermoelectric Plant

The plant under study is located in the metropolitan area of Porto Alegre, Rio Grande do Sul, Brazil. The installation schematic diagram, as with the addition of the cooling system, is shown in Fig. 3a, which has a GE MS7001FA gas turbine, an ANSALDO HRSG (Heat Recovery Steam Generator) with three pressure levels and a Siemens SST- 900RH steam turbine. The gas turbine is responsible for the production of 182.4 MW at ISO conditions, as mentioned before, operating with OCTE (Oil Fuel for Electric Turbine) and the steam turbine is responsible for producing 83.00 MW. The plant physical installation arrangement is similar to that in Figure 3b.

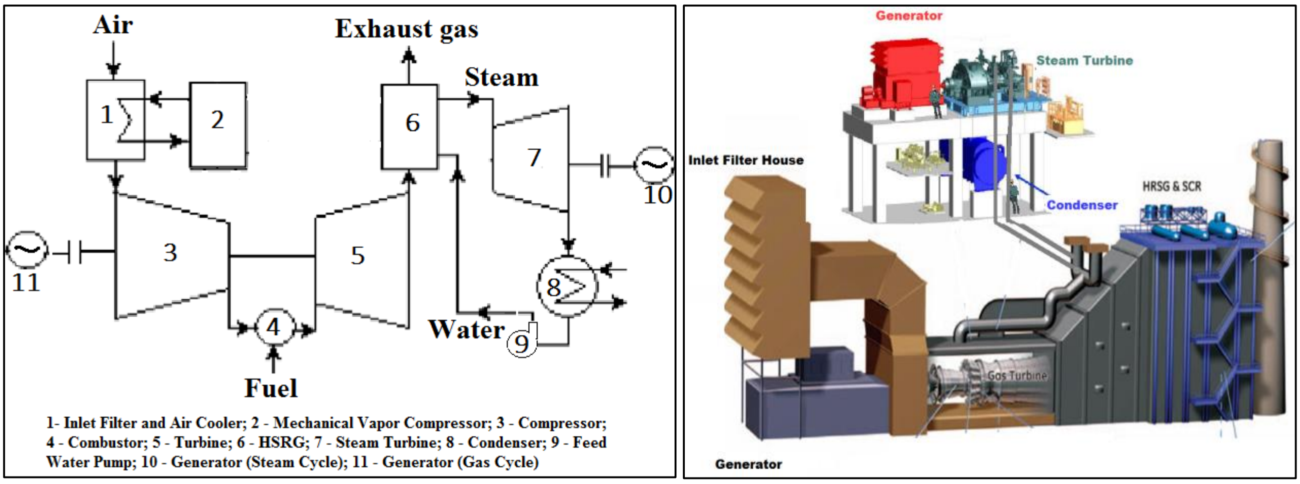

Figure 3 – a) Schematic diagram of Cooled Combined Cycle; b) General Combined Cycle Plant Installation. Adapted from SIEMENS (2009)

4. METHODOLOGY

To attend the objective of this work a thermodynamic model was designed for the calculation of efficiency and power generated with and without gas turbine compressor inlet air cooling. The model considers the plant operating only at base load (maximum power), condition that corresponded to 95% of the operation in 2014, and use of OCTE (Oil Fuel for Electric Turbine).

Efficiency calculations were made using computer modeling in the *EES* (Engineering Equation Solver) software and collection of the actual cycle operation data in a thermal power plant in the metropolitan region of Porto Alegre, Rio Grande do Sul, except for the inlet air cooling in which literature data were used.

The methodology to obtain the results was through the following steps:

- Survey of the equations for the model construction;
- Acquisition of actual data system operation;
- Insertion and model arrangement in *EES* software;

- Model validation;
- Determination of efficiency and generated power by the cycle.

The models were considered valid by comparing the real data process output, according to the conditions.

4.1 Cooling Calculation Method

As mentioned above, the cooling method adopted in this work is the vapor mechanical compression with direct expansion. To each gas turbine, manufacturers perform a specific refrigeration system design, so the data are not available for consultation. The solution adopted was to use the technical data from a commercial air cooling unit found at manufacturer’s catalog.

The machine chosen was the highest capacity available from Carrier (2015), model 40VX60H + 38EX_20 + 38EX_20 + 38EX_20, which has an evaporator unit and three condenser in series with a maximum cooling capacity of 211 kW. The commercial model does not comply with the cooling capacity needed, then it was assumed an extrapolation factor of 20 for the equipment data behavior.

Table 1 shows the cooling unit behavior as a function of the ambient wet bulb and dry bulb temperature.

Table 1 – Part of Cooling capacity and Electric Power Consumption by Refrigeration Unit

			60 TR Fixa - High Air Flow (40VX60H + 38EX_20 + 38EX_20 + 38EX_20)											
Air flow at Evap. (m^3/h)			52992											
DBT evap. inlet (°C)			22				24.35				26.7			
WBT evap. Inlet (°C)			12.4	13.6	14.8	16	14.1	15.3	16.5	17.7	15.8	17	18.2	19.4
DBT Extheral ambient (°C)	20	CT	181544	181590	183279	189152	189731	186587	191694	197740	197921	193656	200268	206563
		CS	181544	181322	176732	160606	189731	181317	184439	168461	197921	193329	191948	176049
		PEC	42053	42056	42255	42910	42952	47222	43183	43870	43876	48265	44154	44871
	25	CT	176271	175696	176785	182095	183669	183705	184885	190418	191717	186892	193168	198930
		CS	176271	175696	172602	157514	183669	183440	180268	165345	191717	186891	187858	172981
		PEC	46055	46070	46194	46833	46974	46976	47115	47784	47910	52462	48083	48777
	30	CT	169352	169338	169830	174571	177139	177118	177640	182559	184983	184963	185955	190779
		CS	169352	169338	167512	154153	177139	177118	175217	162006	184983	184963	182258	169673
		PEC	50291	50291	50317	50934	51200	51199	51231	51872	52141	52139	52253	52860

Ambient humidity has little influence on gas turbine performance, unlike the cooling system where it is very important. As shown in Tab. 1 there are a substantial increase in the total cooling capacity (CT), which is sensible capacity plus latent capacity, with increasing wet bulb temperature, while the sensible capacity has the opposite behavior. This must be considered in the cooling system model, as for the compressor inlet air cooling moisture removal will not be considered, since the removal of sensible heat is the most important factor for this study.

For the cooling system modeling was necessary to establish a relationship between the amount of Total Heat (CT) and Sensible Heat (CS) removed in a certain temperature so that the model was closer to the real condition. This was done according to eq. 4.1, where the result is an average relative capacity of the total capacity and sensible capacity.

$$CAP_{Rel,média} = \frac{\frac{CT\ (WBT,DBT)}{CS\ (WBT,DBT)}}{nCAP} \tag{4.1}$$

Where CT (WBT,DBT) is the total cooling capacity due to WBT (Wet-Bulb Temperature) and DBT (Dry-Bulb Temperature), CS (WBT , DBT) is the sensible cooling capacity due to WBT and DBT and nCAP is the total number of capacities.

With $Cap_{Rel,média}$ defined is possible to correct the values for the cooling capacity due to WBT and DBT, according to eq. 4.2.

$$CAP\ (WBT,DBT) = \frac{CT\ (WBT,DBT)}{CAP_{Rel,média}} \tag{4.2}$$

After defining the cooling capacity due to the WBT and DBT, the drafting of a new cooling capacity table with the behavior for each temperature was required. With the obtained data in this correction, it was possible to make a new table and by using the "Regression" tool Excel (Microsoft) data analysis the coefficients of the correction curves were calculated for CAPFT (Cooling Capacity in Temperature Function) and PECFT (Electrical Power Consumption in temperature Function), which are defined by eq. 4.3 and eq. 4.4.

$$CAPFT = a + [b.\ WBT] + [c.\ WBT^2] + [d.\ DBT] + [e.\ DBT^2] + [f.\ WBT.\ DBT] \tag{4.3}$$

$$PECFT = g + [h.\ WBT] + [i\ .WBT^2] + [j.\ DBT] + [k.\ DBT^2] + [l.\ WBT.\ DBS] \tag{4.4}$$

Part of the table prepared to define an equation that represents the capacity curve as a function of temperature can be seen in Tab. 2, where the yellow line indicating the reference standard temperature with WBT of 19.4 °C and DBT of 35 °C .

Table 2 – Part of the data used to obtain the CAPFT – Capacity of refrigeration depending of WBT and DBT temperatures

WBT (°C)	WBT²	DBT (°C)	DBT²	WBT.DBT	CAP [kW]	CAPFT
12.4	153.76	20	400	248	203.445	0.998
12.4	153.76	25	625	310	197.536	0.969
...	...	...	...	...	...	...
18.2	331.24	35	1225	637	199.741	0.98
18.2	331.24	40	1600	728	191.081	0.937
...	...	...	...	...	...	...
19.4	376.36	30	900	582	213.794	1.049
19.4	376.36	35	1225	679	203.772	1
19.4	376.36	40	1600	776	193.545	0.949

A similar table was done for the Electric Power Consumption as a function of Temperature, as can be seen in Tab.3.

Table 3 - Part of the data used to obtain the PECFT - Electric Power Consumption depending of WBT and DBT temperatures

WBT (°C)	WBT²	DBT (°C)	DBT²	WBT.DBT	PEC [kW]	PECFT
12.4	153.76	20	400	248	42.053	0.737
12.4	153.76	25	625	310	46.055	0.807
...	...	...	...	...	...	...
18.2	331.24	25	625	455	48.083	0.842
18.2	331.24	30	900	546	52.253	0.916
...	...	...	...	...	...	...
19.4	376.36	30	900	582	52.860	0.926
19.4	376.36	35	1225	679	57.038	1
19.4	376.36	40	1600	776	61.198	1.072

To determine the amount of heat to be removed from the ambient air to compressor inlet, eq. 4.5 is used.

$$\dot{Q}_{resf}=CAPFT.(20).(203.772) \tag{4.5}$$

Where CAPFT is the cooling capacity correction factor according to temperature, 20 is the commercial machine extrapolation factor and 203.772 is the reference cooling capacity in kW at the standard manufacturer temperature, that is 19.4 °C WBT and 35 °C DBT.

Thus, it's possible to determine the enthalpy in which air will enter the compressor, h_{ar1}, through eq. 4.6 to subsequently obtain T_{ar1} temperature.

$$\dot{Q}_{resf}= \dot{m}_{ar}.(h_{ar,amb}-h_{ar1}) \tag{4.6}$$

Where $\dot{m}_{ar}$ is the air mass flow in kg/s, $h_{ar,amb}$ is the ambient air enthalpy in kJ/kg and h_{ar1} is the cooled air enthalpy in kJ/kg.

To determine the electric power consumed at each temperature condition may be used eq. 4.7.

$$\dot{W}_{resf}=PECFT.\ (20).(57.038) \tag{4.7}$$

Where PECFT is the correction factor of the electric power consumption as a temperature function, 20 is the commercial machine extrapolation factor and 57.038 is the electric power consumed at the standard temperature manufacturer in kW.

According to Carvalho Junior (2012), the cold air temperature limit is 7 °C to prevent the ice formation at the compressor inlet. Therefore, this is the lower limit value adopted for this work. To attend this, the cooling system will be in operation only when ambient temperatures are above 17 °C. Its design was conceived in order to operate the most part of the time ensuring that with the air cooling, there is no icing formation at the compressor inlet and has a high annual utilization factor.

4. MODEL RESULTS x REAL DATA

When comparing power curves for recorded temperatures as the basis for gas turbine model preparation, the values obtained were very close, with maximum error of 1.66%, the difference between them can be seen in tab. 4 and in Fig. 4. Measured data were taken from the database and the comparison temperatures range is made in a period of 24 hours, with a 30 min interval between the measurements.

Table 4 - Gas Turbine model validation with use of OCTE

Ambient Temperature (°C)	Real Data Power Output (MW)	Model Power Output (MW)	Deviation %
19	174.447	173.951	1.10
20	173.320	172.821	0.59
21	172.202	171.698	0.43
22	171.091	170.584	0.25
23	169.989	169.478	0.16
24	168.894	168.379	0.61
25	167.806	167.288	0.83
26	166.727	166.205	0.59
27	165.655	165.129	0.81
28	164.590	164.061	0.92
29	163.533	163.000	1.09
30	162.483	161.947	1.30
31	161.440	160.901	1.66

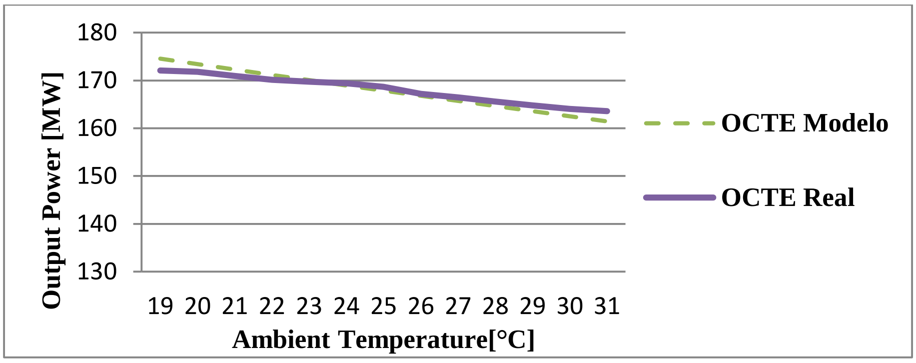

Figure 4 – Output Power depending on Ambient Temperature for Model and Real Data

For the HRSG model, there is a difference between the real and the model temperatures, what is caused by changes at the actual process that had to be taken in the boiler modeling to make possible the calculation. However, the biggest difference is on the medium-pressure economizer heat exchanger measurement, that does not affect the results, since the most relevant differences between the model and the real process temperatures have a maximum value of 1.50%. These small differences obtained allows considering the model as valid.

The steam turbine model turbine resulted a value 2.29% below the actual equipment electric power, which is about 83.00 MW, so it can be considered valid, whereas the difference is relatively small. The model also showed a difference in the generated power with gas turbine compressor inlet air temperature variation, this occurs as a result of the recovery boiler performance, as mentioned above, then the results tendency were satisfactory.

4. OVERALL RESULTS

The results described in this chapter were obtained from the EES designed model and the input data from the actual process such as compressor suction and discharge pressure, the turbine inlet temperature, compressor inlet volumetric air flow, air/fuel ratio, HRSG steam flow and pressure, etc.

The model validation for different compressor inlet temperatures allow considering valid the cooling method. To determine the system's behavior over a full year, dry bulb and wet bulb temperatures of a typical year in the region of Porto Alegre, RS, were chosen. The typical year for a region is generated by a TRY (Test Reference Year) file, in which data is generated by the elimination of years with extreme temperatures over a long term database, until remain only a single year called the reference year.

Through Fig. 5 is possible to see the great influence of ambient temperature on the behavior of electrical power generation. The Combined Cycle Trend curve graphically represents this influence, which can be observed since the higher the ambient temperature, the smaller is the generation capacity. Power output is inversely proportional to the ambient temperature, thus, at the ends, where the warmer periods of the year are presented, it is lower than at the center of the curve, which represents the period of low temperatures. This fact occurs due to the air density decrease with temperature increase, as explained in the previous chapter.

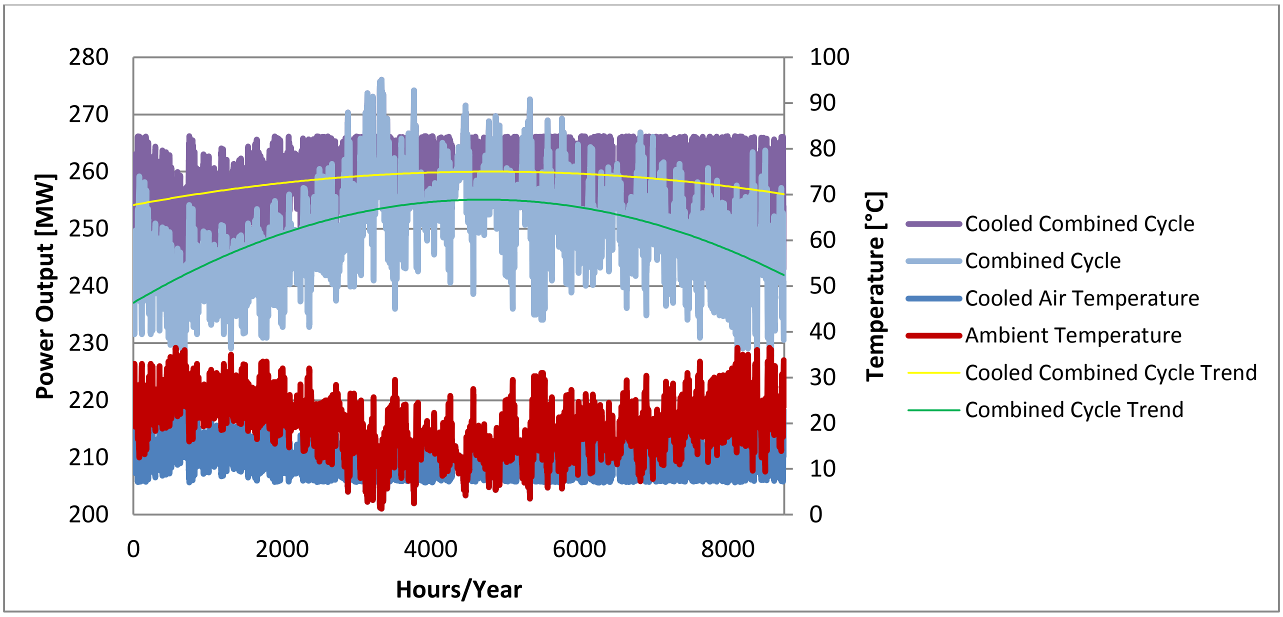

Figure 5 - Chart Performance of Electric Power and Ambient Temperature depending on Annual Hours.

Another important point is that the temperature and therefore the power variation are high on the same day. It's normal the occurrence of high temperature variations during the day in the studied area, morning and night with lower temperatures than the afternoons.

By adding the cooling system there is a scenery change. This system allows the compressor inlet temperature to be lower than the ambient temperature. The representation of this effect is shown in Fig. 5 by the Cooled Combined Cycle Trend. As can be seen, the effect of air cooling generates uniform power output curve, with little variation from center to edges.

Another highlight is the upper limit of the power with cooling system, which is limited by the minimum temperature cooling allowed by the refrigeration system, since a real equipment was used for the study, bringing the model closer to an industrial application. Table 5 show a stratification of the key values found in the simulation.

Table 5 – Stratification of the main simulated values

	DBT [°C]	WBT [°C]	Compressor Inlet Air Temperature [°C]	Combined Cycle Power Output [MW]	Cooled Combined Cycle Power Output [MW]	Power Output Increase [MW]
Max Increase of Power Output	26.5	26.0	14.60	239.893	253.492	13.599
Min Increase of Power Output	36.5	22.4	26.23	227.219	239.164	11.945
Max Power Output	1.3	1.3	1.30	276.081	276.081	-

In various simulated conditions, the cooling system was more effective when the DBT was close to WBT. This effect was discussed by Magalhães Sobrinho (2011), and, according to him, the WBT influences the film coefficient responsible for the heat exchanger performance. In their tests the influence of relative humidity in the system COP showed that for a low humidity the performance of air conditioning system has a drop of up to 75 % in performance. In this study the influence is not as significant as the values found by Magalhães, but it is clear that the relative system behavior is the same, due to the relative humidity increase. The increase in capacity is caused by latent heat removal from humid air.

The use of compressor air inlet cooling has enabled the improvement in results due to increased specific air mass, i.e., the power consumed in the compressor is very close to the uncooled operation, but on the other hand, the mass flow is greater, increasing the turbine work and decreasing the compressor specific work. In the model the compressor work practically does not change, because a constant discharge pressure value was adopted, while in the real process the compressor discharge pressure decreases with increasing temperature. The effect in the work ratio reduction can be seen in Tab. 6.

Table 6 – Ratio between Compressor and Turbine Average Work With and Without Inlet Compressor Air Cooling

	NORMAL	**COOLED**
Work done by the Turbine [MW]	346.675	354.406
Work Consumed by the Compressor [MW]	167.509	167.496
Work Consumed by Refrigeration System [MW]	0.000	0.941
Work ratio	0.4832	0.4753

It can be seen that the compressor power consumption is virtually the same, but the work ratio is reduced when the air is cold, even if adding the power consumed by the cooling system, which leads to increased net power.

Another important fact to be analyzed with the cooling use is the comparison of the overall efficiency of Brayton, Rankine and Combined cycle. Table 7 shows the results obtained by the model, which is now considering the power consumed by the cooling system when used.

Table 7 - Global Efficiencies of Cycles

	NORMAL	**COOLED**
Global Efficiency Combined Cycle	0.5151	0.5212
Global Efficiency Brayton Cycle	0.3533	0.3596
Global Efficiency Rankine Cycle	0.3490	0.3483

It is possible to see the increase in efficiency of the Brayton cycle and combined cycle, but the Rankine cycle shows a small decrease. Increased efficiencies in combined cycle are directly related to higher mass flow passing through the turbine. According to Al-Ibrahim and Varnham (2010), on the base load the generation cycle capacity is defined by the amount of air mass that enters in the turbine. High mass flow rates of flue gas increase generation capacity, and also the mass flow of fuel. However, the increase in fuel consumption is less than the increase in power output by improving the efficiency of the Brayton Cycle and Combined Cycle accordingly. Kehlhofer (1999) states that the decrease in temperature compressor inlet slightly decreases the efficiency of the Rankine cycle, but due to its small participation in the process is not enough to influence the combined cycle efficiency.

Table 8 shows the average increase in overall efficiency obtained upon cooling in comparison to normal process.

Table 8 - Average Increase in Efficiency and Power Generation With Compressor Inlet Air Cooling

	[%]
Efficiency Increase	1.82
Power Generation Increase	5.37

Analysis of full data of plant performance with the use of cooling reveals that the highest efficiency increase, about 2.19%, occurs on days with high temperatures and high moisture levels. This occurs due to the fact that the cooling system capacity increase in humid days and gas turbine efficiencies decrease with higher compressor inlet temperature. So, the lowest rates of efficiency increase, about 1.61%, occur in the days with temperatures close to 20 °C and very low humidity.

As can be seen in Fig.6, for the whole year, the values are quite significant. The chart shows the generated energy increase in each month of the year, as measured in Fig. 5, especially for the hot months, where the gain can reach 9627 MWh in March. The annual increase in electricity generation with cooling use is 74570 MWh.

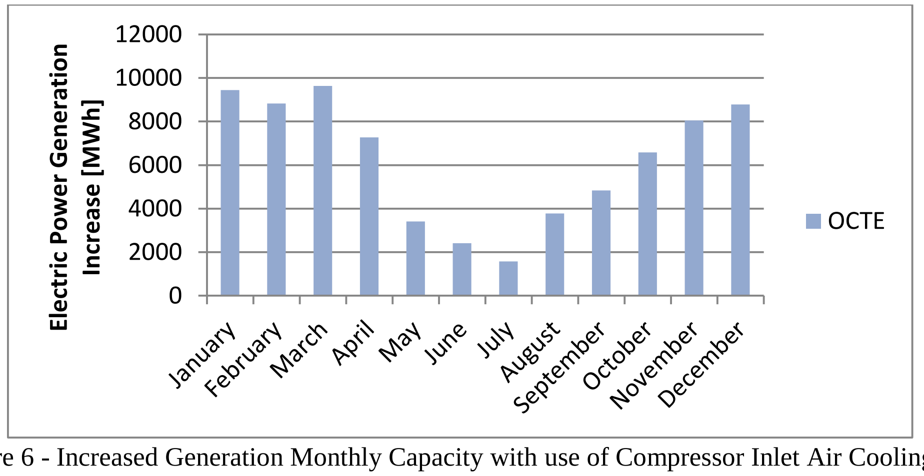

Figure 6 - Increased Generation Monthly Capacity with use of Compressor Inlet Air Cooling

6. CONCLUSION

The model results with the cooling system on the compressor inlet represent a very attractive thermodynamic gain, with average increases in efficiency and power of 1.82% and 5.37%, respectively.

The estimated increase in power generation in the model reached more than 70 GWh with the implementation of gas turbine compressor inlet air cooling when considering the whole year, thus expanding the installed capacity of thermal power plant without major changes in its physical structure.

The cooling system choice by mechanical vapor compression and direct expansion was appropriate to the climate of the region, which has very humid and hot characteristics. Evaporative cooling would have a smaller effect due to the high humidity and the smaller air density increase of this process. The cooling capacity and the electrical power cooling system consumption are directly influenced by these two factors. The better performance of the cooling system occurs in the lowest DBT system's operation (17 °C) and relative humidity of 100%. As the region has an average DBT of 19.5 °C and a relative humidity of 82%, in the thermal point of view the system is suitable for deployment in the thermal power plant under study.

7. REFERENCES

Al-ibrahim, Abdulrahman M.; Varnham, Abdulhadi, 2010. "A review of inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia". Applied Thermal Engineering, V. 30, Pg. 1879–1888.

Boyce, Meherwan P, 2010. "Handbook for cogeneration and combined cyclepower plants". 2. Ed. New York, ASME.

Carrier, 2015. Catálogo Técnico Ecosplit Inverter – K. Agosto de 2015.

Carvalho Júnior, Manoel Lélio Martins de, 2012. "Resfriamento de ar de entrada em turbina a gás no parque gerador elétrico brasileiro". Tese de Doutorado. Universidade de São Paulo. Programa de Pós-Graduação em Energia. São Paulo. Disponível em: <http//:www.teses.usp.br>.

Çengel, Yunus A.; Boles, Michael A., 2001. "Termodinâmica", 3ª ed. Amadora (Portugal): McGraw-Hill.

EPE, Empresa de Pesquisa Energética, 2014.Balanço Energético Nacional 2014: Ano base 2013. Rio de Janeiro, RJ.

GE, General Electric Power Systems, 2003. "Manual de treinamento de operação de turbo-gerador a gás", Schenectady (EUA): GE.

GE, General Electric Oil & Gas, 2008. "Inlet Air Cooling", Folder, Florence (Italy): GE. Disponível em: < http://site.ge-energy.com/businesses/ge_oilandgas/en/literature/en/downloads/inletair_cooling.pdf>

INMET, Instituito Nacional de Meteorologia, 2015. "Normais climatológicas do Brasil 1961-1990". Disponível em: <http://www.inmet.gov.br/portal/index.php?r=clima/normaisClimatologicas>.

Jonsson M, Yan J, 2005. Humidified gas turbines - a review of proposed and implemented cycles. Energy 30 (2205):1013e78.

Kehlhofer, R. H.; Warner, J.; Nielsen, H.; Bachmann, R., 1999. "Combined Cycle Gas-Steam Turbine Power Plants. Ed. Pennwell, USA. 288 p.

Magalhães Sobrinho, P, 2011. "Avaliação do efeito da umidade relativa do ar no coeficiente de performance de um sistema de ar condicionado". Tese (Livre Docência) – Faculdade de Engenharia do Campus de Guaratinguetá, Universidade Estadual Paulista Júlio de Mesquita Filho, Guaratinguetá, 63 p.

ONS, Operador Nacional Do Sistema Elétrico, 2015. Histórico da operação: Gráfico de geração. Rio de Janeiro, RJ. Disponível em: <http://www.ons.org.br/historico/geracao_energia.aspx>.

Sa, Ashley De; Zubaidy ,Sarim Al, 2011. Gas turbine performance at varying ambient temperature, Applied Thermal Engineering, vol. 31, 14–15, pg 2735–2739.

SIEMENS. SST-900 industrial steam turbines Up to 250 MW. Siemens brochure. 2009. Disponível em: <http://www.energy.siemens.com/hq/en/fossil-power-generation/steam-turbines/sst-900.htm#content=Technical%20Data>.

SIEMENS. Efficiency Record of Combined Cycle Power Plant. Siemens Innovation news. 2011. Disponível em: <http://www.siemens.com/innovation/en/news/2011/efficiency-record-of-combined-cycle-power-plant.htm>.

STELLAR-ENERGY. Turbine Inlet Air Chilling. TIAC solutions, 2012, USA. Disponível em: <http://www.stellar-energy.net/media/18901/TIAC%20Solutions.pdf>

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.