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Analysis of Gas Turbine Performance with Inlet Air Cooling Techniques Applied to Brazilian Sites

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Abstract: *For geographic regions where significant power demand and highest electricity prices occur during the warm months, a gas turbine inlet air cooling technique is a useful option for increasing output. Inlet air cooling increases the power output by taking advantage of the gas turbine's feature of higher mass flow rate, due the compressor inlet temperature decays. Industrial gas turbines that operate at constant speed are constant-volume-flow combustion machines. As the specific volume of air is directly proportional to the temperature, the increases of the air density results in a higher air mass flow rate, once the volumetric rate is constant. Consequently, the gas turbine power output enhances. Different methods are available for reducing compressor intake air temperature. There are two basic systems currently available for inlet cooling. The first and most cost-effective system is the evaporative cooling. Evaporative coolers make use of the evaporation of water to reduce the gas turbine inlet air temperature. The second system employs two ways to cool the inlet air: mechanical compression and absorption. In this method, the cooling medium flows through a heat exchanger located in the inlet duct to remove heat from the inlet air. In the present study, a thermodynamic analysis of gas turbine performance is carried out to calculate heat rate, power output and thermal efficiency at different inlet air temperature and relative humidity conditions. The results obtained with this model are compared with the values of the condition without cooling herein named of Base-Case. Then, the three cooling techniques are computationally implemented and solved for different inlet conditions (inlet temperature and relative humidity). In addition, the gas turbine was tested under different cooling methods for two Brazilian sites, and comparison between chiller systems (mechanical and absorption) showed that the absorption chiller provides the highest increment in annual energy generation with lower unit energy costs. On the other hand, evaporative cooler offered the lowest unit energy cost but associated with a limited cooling potential.*

Keywords: *Gas turbine, Turbine inlet cooling, TIC, Evaporative cooling, Chiller absorption.*

LIST OF SYMBOLS AND NOMENCLATURE

Abbreviations

ISO	International Organization for Standardization
TIT	Turbine Inlet Temperature
TIC	Turbine Inlet Cooling

Symbols		***Units***
C_p	Specific heat at constant pressure	[kJ/kg°C]
COP	Coefficient of performance of the mechanical chiller	[-]
LHV	Lower fuel heat value	[kJ/kg]
h	Specific enthalpy	[kJ/kg]
HR	Heat rate	[kJ/kWh]
$\dot{m}$	Mass flow rate	[kg/s]
P	Pressure	[Pa]
ΔP	Pressure drop	[Pa]
$\dot{Q}$	Heat transfer rate	[kW]
r	Compressor pressure ratio	[-]
T	Temperature	[°C]
Tb	Dry-bulb temperature	[°C]
Tw	Wet-bulb temperature	[°C]
$\dot{W}$	Power output	[MW]
SFC	Specific fuel consumption	[kg/kWh]

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ε	Evaporative cooling effectiveness	[-]
ϕ	Relative humidity	[-]
γ	Specific heat ratio	[-]
η	Efficiency	[-]
ω	Specific humidity	[kg_{water}/kg_{air}]

Subscripts

0	Ambient air
01, 02, 03, 04, 05 ,06	Points denoted in Fig. 1 and 2
a	Air
avg	Average
C	Compressor
MC	Mechanical chiller
CL	Cooling load
f	Fuel
g	Flue gas
in	Input
N	Net
t	Turbine
T	Total
th	Thermal efficiency
w	water

INTRODUCTION

The gas turbine is composed of a compressor that supplies air at high pressure to the combustor, which provides flue gas at high pressure and temperature turbine (Punwani, 2003). These engines are of constant-volume and their power output is directly proportional and limited by the air mass flow rate. As the compressor has a fixed capacity for a given rotational speed and volumetric flow rate of air, their volumetric capacity remains constant. Therefore, the mass flow rate of air enter into the gas turbine varies with their specific mass, which means that it depends on the temperature and the relative humidity of the ambient air (American Society of Heating, 2008).

Gas turbine performance is critically limited by the predominating ambient temperature, mainly in hot and dry regions. It occurs because the power output is inversely proportional to the ambient temperature (Nasser and Kalay, 1991). The temperature drop provides an augment in the air density and consequently elevates air mass flow rate; this behavior increases the power output and efficiency at about 0.7% per degree Celsius for heavy duty gas turbine (Zunig, 2005).

As gas turbine has been widely used for power generation and the demand for electricity is highest during periods of elevated ambient temperatures. This factor leads to an increase in power plant peaking capacity. In the Saudi Electric Company's (SEC), for example, approximately 42% of the annual energy is generated by combustion turbines, and during the summer these turbines suffer a 24% decrease in their capacity, due to ambient temperature up to 50°C (Al-Ibrahim and Varnham, 2010).

Nowadays, one of the main methods employed to increase the gas turbine efficiency is the cooling of the compressor intake air, inducing the turbine inlet cooling (TIC). Two main commercially available TIC options are: (i) evaporative media cooling and (ii) mechanical or absorption chiller systems.

Several literature works have extensively investigated the TIC effect on the gas turbine performance enhancement. Al-Ibrahim and Varnham (2010) reported that refrigerative cooling can uses mechanical or electrical vapor compression refrigeration equipment. Equipment and O&M costs for mechanical chillers are cheaper than absorption systems, but capital costs are higher and parasitic power requirements can be 30% of the power gain.

Jaber *et al.* (2006) studied the influence of air cooling intake on the gas turbine performance by comparing two different cooling systems, evaporative and cooling coil. Their results showed that the evaporative cooling and chiller system present similar improvement in the power output, about 1.0-1.5 MW, but the cooling coil of the mechanical chiller consumes more energy to run the vapor-compression refrigeration unit and the overall plant performance decrease.

Alhazmy and Najjar (2004) compared two different techniques of air coolers, water spraying system and cooling coils, and the results were analyzed for a specific set of operational and design conditions. The spray coolers are less expensive than chiller coils. However, it is extremely affected by ambient temperature and relative humidity. This method is capable of reducing the ambient air temperature by 3-15°C, producing power augmentation by 1-7%, and increasing the efficiency by 3%. Chiller cooling offers full control over the air intake conditions, yet it has high parasitic power consumption and this power is removed from the gas turbine output. Their results also showed that although the cooling coil during cold and humid conditions (T = 25°C and ϕ = 80%) improves the power output by 10% and 18% for hot and humid conditions (T = 50°C and ϕ = 80%), the demand operational power reduces the net power by 6.1% and 37.6%, respectively.

Nasser and El-Kalay (1991) suggested the use of a simple Li Br/water absorption system to cool inlet air of gas turbine in Bahrain. This technique is able to enhance the power output by 20% during the summer. Their results showed that a drop temperature of 10°C, when the ambient temperature is 40°C, can increase the power output by 10%.

In the current study, the effect of different compressor intake air cooling systems (evaporative media cooling and chiller systems) is theoretically studied for a single shaft gas turbine. Results for power output and gas turbine thermal efficiency are also obtained to allow selecting a preliminary cooling method as a function of gas turbine operating ambient conditions. The gas turbine performance including these inlet cooling methods is also tested for two Brazilian sites with different climate conditions.

GAS TURBINE CYCLE

Figure 1 illustrates the studied combustion single shaft gas turbine cycle. The model employed to simulate the air thermodynamic states (from 0 to 6) is also schematized. Pressure and temperature calculations for each point are also determined, as shown in the Figure 1.

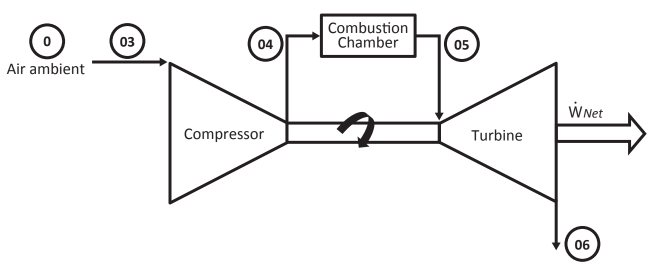


Figure 1. Schematic of the typical gas turbine cycle.

The compressor inlet temperature is equal to ambient temperature once that pressure drop at inlet and exhaust ducts is neglected, resulting that:

$$P_0 = P_{03} \tag{1}$$

The air and combustion products are assumed to behave as ideal gases because the main objective in this study was the comparison among the inlet cooling methods.

The pressure of the air leaving the compressor (P_{04}) is calculated as:

$$P_{04} = r \cdot P_{03} \tag{2}$$

Using the polytropic relations for gas ideal and knowing the isentropic efficiency of compressor, the discharge temperature (T_{04}) can be calculated as:

$$T_{04} = \frac{T_{03}}{\eta_c}\left[\left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_{03} \tag{3}$$

The compressor power can be estimated using the first law of thermodynamic as follows:

$$\dot{W}_C = \dot{m}_a \cdot C_{pa,avg}\left(T_{04} - T_{03}\right) \tag{4}$$

where $\dot{m}_a$ is the air mass flow rate and $C_{pa,avg}$ is the specific heat of the dry air at constant pressure, determined as a function of the average temperature across the compressor (Alhazmy and Najjar, 2004).

Assuming a pre-defined combustor pressure drop, the combustion chamber discharge pressure (P_{05}) can be calculated as:

$$P_{05} = P_{04} - \Delta P_{Combustor} \tag{5}$$

The heat delivered by combustor chamber is determined from energy balance in it:

$$\dot{Q}_{in} = \dot{m}_a \cdot C_{pg,avg} \cdot \left(T_{05} - T_{04}\right) \tag{6}$$

where $C_{pg,avg}$ is the flue gas specific heat calculated as function of the average temperature across the combustion chamber (Alhazmy and Najjar, 2004).

By knowing the fuel gas lower heat value (*LHV*), the natural gas mass flow rate is defined as:

$$\dot{m}_f = \frac{\dot{Q}_{in} / LHV}{\eta_{Combustor}} \tag{7}$$

where $\eta_{Combustor}$ is the combustor chamber efficiency.

The discharge temperature of the gas that leaving the turbine can be written as:

$$T_{06} = T_{05} - \eta_t \cdot T_{04}\left[1 - \left(\frac{1}{P_{05}/P_{06}}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{8}$$

where η_t is the turbine isentropic efficiency and P_{06} is the turbine exhaust gas pressure.

Hence, the power produced from the turbine is equal to:

$$\dot{W}_t = \dot{m}_T \cdot C_{pg,avg}\left(T_{05} - T_{06}\right) \tag{9}$$

where $\dot{m}_T$ is the total mass flow rate; it is composed of fuel and air mass flow rate:

$$\dot{m}_T = \dot{m}_a + \dot{m}_f \tag{10}$$

and $C_{pg,avg}$ is the flue gas specific heat calculated as function of the average temperature across the turbine (Alhazmy and Najjar, 2004).

Lastly, the net power obtained from the gas turbine is given by:

$$\dot{W}_N = \dot{W}_T - \dot{W}_C \tag{11}$$

The specific fuel consumption is determined as:

$$SFC = \frac{3600 \cdot \dot{m}_f}{\dot{W}_N} \tag{12}$$

Another important gas turbine parameter is the heat rate (HR), calculated as:

$$HR = SFC \cdot LHV \tag{13}$$

The thermal efficiency of the gas turbine is determined by the following equation:

$$\eta_{th} = \frac{3600}{SFC \cdot LHV} \tag{14}$$

INLET AIR COOLING SYSTEMS

Figure 2 shows a simple sketch of the system herein studied, which consists of a standard gas turbine power plant and an intake air cooler. The gas turbine power plant consists of compressor, combustion chamber and turbine. In this work, three different inlet air cooling techniques are proposed for analysis: evaporative cooling, absorption and mechanical chiller.

Gas turbine performance will be evaluated with each cooling method and compared with values of the Base-Case (without any cooling system). The working fluid passing through the compressor is the air, and it assumed to be an ideal gas, while in the turbine the working fluid are the flue gases. A brief description of each inlet cooling technology will be presented in the subsequent section.

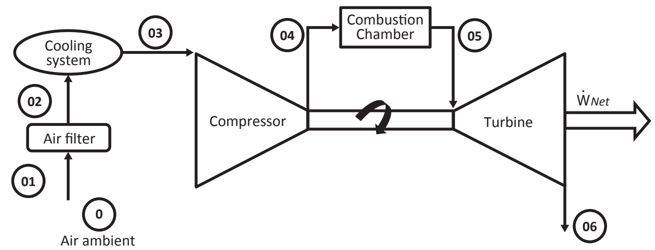


Figure 2. Representation of the gas turbine cycle with cooling system.

Evaporative cooling

Evaporative cooling is most appropriated cooling system to hot dry areas, because it utilizes the latent heat of vaporization to cool ambient temperature from the dry-bulb to the wet-bulb temperature (Al-Ibrahim and Varnham, 2010). The process employed by this cooling method convert sensible heat in latent heat, being the ambient air cooled by evaporation of the water from wet surface of the panel (cooling media) to the air (Amell and Cadavid, 2002). In Fig. 3, a typical evaporative cooling system is presented.

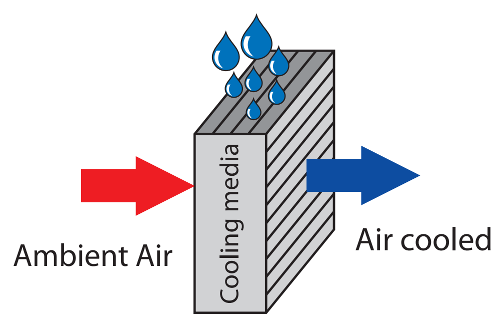


Figure 3. Schematic representation of the evaporative coil (cooling media).

The inlet air temperature after cooling process (Fig. 2) can be calculated as:

$$T_{03} = Tb_{02} - \varepsilon\left(Tb_{02} - Tw_{02}\right) \tag{15}$$

The evaporated water mass flow associated with the evaporative cooling is given by:

$$\dot{m}_w = \dot{m}_a \cdot \left(\omega_{02} - \omega_{03}\right) \tag{16}$$

where $\dot{m}_a$ is mass flow rate of air, and ω_{02} and ω_{03} are the air specific humidity in the inlet and outlet of the evaporative system, respectively. It is well known that as the water evaporates, the mineral content (calcium carbonate, magnesium, sodium, salts etc.) of the remaining water increases in concentration. Therefore, the water that is drained from cooling equipment to remove mineral build-up is called "blow-down" water or "bleed" water. Eq. (16) not considers the blow down water because typically only a very small stream of bleed-off water is necessary for proper equipment operation and high purity or demineralised water supply is used (Al-Ibrahim and Varnham, 2010).

The cooling load associated with the evaporative cooling system results:

$$\dot{Q}_{CL} = \dot{m}_a \cdot C_{pa,avg} \cdot (T_{02} - T_{03}) \tag{17}$$

where $\dot{m}_a$ is the air mass flow rate and $C_{pa,avg}$ is the specific heat of the dry air at constant pressure, determined as a function of the average temperature across the evaporative system (Alhazmy and Najjar, 2004).

Absorption and mechanical chiller systems

Another alternative to provide air cooled to the gas turbine is the cooling chiller mechanism. Figure 4 shows a typical architecture used in the heat exchanger of the chiller systems. There are two currently available chiller options to cool the compressor intake air: mechanical refrigeration and absorption cooling.

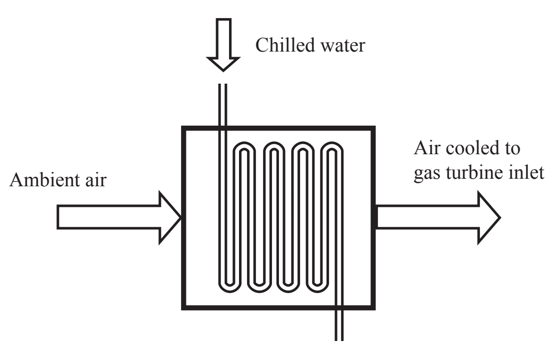


Figure 4. Schematic representation of the chiller coil.

The power required to drive the absorption chiller is usually obtained by the recovery of the heat from turbine exhaust gases, and the chilled water is passed through a heat exchanger to cool the ambient air temperature.

Absorption systems in power plants can use lithium - bromide or ammonia - water combination (Singler *et al.*, 2001). In these cycles, the cooling load removed from the air flowing at ambient conditions into the power plant can be calculated applying the first law of thermodynamics as follows:

$$\dot{Q}_{CL} = \dot{m}_a \cdot [(h_{02} - h_{03}) - h_{w,03} \cdot (\omega_{02} - \omega_{03})] \tag{18}$$

where h_{02} and h_{03} are the air enthalpy in the inlet and outlet of the evaporative system, respectively.

Water parcel present in the air will be condensed in the heat exchanger and may be used at the plant in the other process. This quantity of water can be estimated by:

$$\dot{m}_w = \dot{m}_a \cdot (\omega_{02} - \omega_{03}) \tag{19}$$

which is the same equation employed to determine the evaporated water in the evaporator (Eq. 16).

In a mechanical compression system (De Lucia *et al.*, 1996), a liquid such as water or secondary refrigerant is cooled for purposes of refrigeration. A disadvantage, usually associated with this method, is the high input of power to drive the compressor section and various distribution pumps (Singler *et al.*, 2001). Installation and maintenance costs tend to be higher than the other two inlet cooling methods systems previously described. The above equations (Eq. 18 and Eq. 19) for the condensed water mass flow rate and cooling load are also valid to the mechanical chiller system.

The power needed to drive the mechanical chiller (parasitic losses) is estimated according the relation:

$$\dot{W}_{MC} = \frac{\dot{Q}_{CL}}{COP} \tag{20}$$

where COP is the coefficient of performance of the mechanical chiller and this value is fixed.

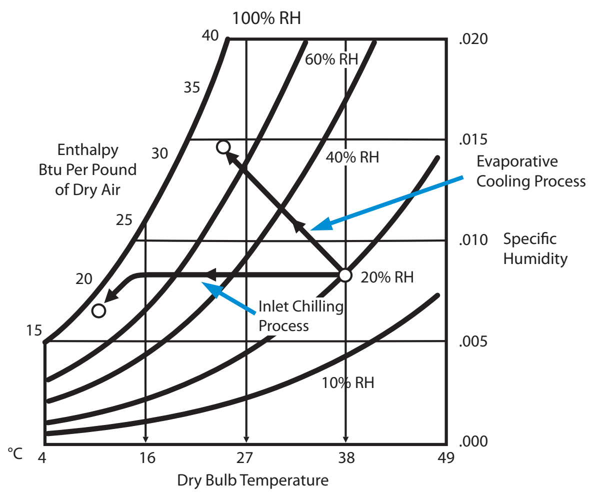
The power required to drive the mechanical chiller is consumed of the output power produced by the gas turbine. Then, the gas turbine power net is given by:

$$\dot{W}_N = \dot{W}_t - \dot{W}_C - \dot{W}_{MC} \tag{21}$$

For this cooling method, the gas turbine power output is decrease by the power consumption by mechanical chiller. On the other hand, the absorption chiller system has a very small parasitic loss which can be considered negligible (Mohanty and Paloso, 1995).

Chillers cooling differ of the evaporative systems because they are not limited by the ambient wet-bulb temperature. The attainable temperature is restricted only by the capacity of the chilling machine to produce coolant and the ability of the coils to transfer heat.

During the process of cooling, the air follows a line of constant specific humidity, until the saturation point is reached, and then the water of the air begins to condense (Brooks, 2000), as shown in Fig. 5. More details about absorption and mechanical chiller typical architectures can be found in Kakaras *et al.* (2006) and De Lucia *et al.* (1994).



Adapted from Brooks (2000).

Figure 5. Psychrometric processes: evaporative and chiller cooling systems.

RESULTS AND DISCUSSIONS

In the present study, a single shaft gas turbine is numerically simulated operating with natural gas. Table 1 shows technical parameters for an arbitrary (non commercial) gas turbine unit used to evaluate both the performance of the Base-Case (without cooling intake air) and each inlet air cooling studied method. Inlet/outlet ducts pressure losses are also listed.

Initially, a Base-Case (case without any cooling method) was simulated employing the ISO conditions (T = 15°C and ϕ = 60%). After, the influence of ambient inlet temperature on gas turbine performance is verified, as shown in Fig. 6 (maintaining ϕ = 60%). The turbine inlet temperature was fixed at TIT = 1385.09 K.

Table 1. Technical specifications of the selected gas turbine engine.

Description	Adopted values
Cycle	Single shaft, simple cycle, industrial engine
Pressure ratio	11 [-]
Turbine inlet temperature	1385 [K]
Air flow rate	141 [kg/s]
Isentropic efficiency of compressor	85.4 [%]
Isentropic efficiency of turbine	86.8 [%]
Combustion efficiency	99.0 [%]
Inlet pressure loss	100 [mmH_2O]
Exhaust pressure loss	200 [mmH_2O]
Combustion chamber pressure loss	1.2 [%]
Fuel, LHV	Natural gas; 48,235.63 [kJ/kg]

It is known that, for an engine with constant speed rotational, the volumetric flow rate is approximately constant. Then, an increase in the ambient temperature reduces the specific mass and decreases mass flow rate. Consequently, the power output decays to conserve the volumetric flow rate, as show in Fig. 6.

Figure 7 presents the temperature decrease obtained using the evaporative cooling method as a function of the ambient temperature and ambient relative humidity fixed at 60%. Three different evaporative cooling effectiveness values were simulated showing that a larger temperature decrease is achieved when the effectiveness is higher, as expected.

A typical evaporative cooling effectiveness is ε = 0.90 that provides a temperature drop equal 6°C when the intake temperature is 34°C with an ambient relative humidity of 60%. As the wet-bulb temperature limits the application of this method, a

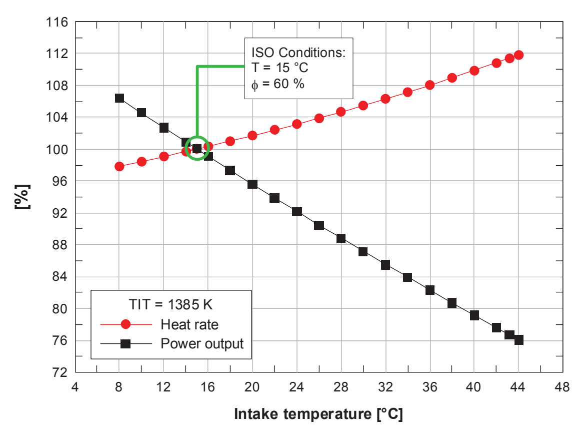


Figure 6. Effect of intake temperature on the gas turbine performance in comparison with ISO conditions.

lower ambient relative humidity condition has been also tested.

The gas turbine power output is presented in Fig. 8 for both Base-Case and evaporative cooling inlet conditions. Note that the power output obtained is lower at Base-Case conditions, when the intake air is not cooled. Furthermore, the ambient dryness affects the gas turbine performance, providing a higher power output level when the ambient relative humidity is lower ϕ = 18% in comparison with ϕ = 60%, as shown in Fig. 8.

This fact is associated with the essence of the evaporative cooling method. The ambient air passes by the cooling media following a constant enthalpy-line (Fig. 5), but the resultant temperature drop is limited by the intake air initial relative humidity.

When the evaporative cooling technique is employed (Fig. 9), the gas turbine thermal efficiency level is higher in comparison with the Base-Case, as occurred for the power output results. At ϕ = 60% and air intake temperature of 34°C, the air cooling process enhances the turbine ISO power output and thermal efficiency in 3.7% and 2.3%, respectively. For ϕ = 18% the power output and thermal efficiency increase 8.4% and 5.3% when compared with Base-Case values, showing that the lower intake air relative humidity elevates the evaporative cooling performance.

Numerical simulations also included the inlet chilling method for providing compressor intake air cooling. Figure 10 shows the temperature drop obtained employing both inlet cooling methods: evaporative and absorption chiller, at ϕ = 18% and ϕ = 60%.

When the absorption chiller is employed, the compressor inlet air temperature is independent of wet-bulb temperature, but there is a minimum acceptable value imposed by the compressor icing formation risk. At this work, a temperature of 10°C was adopted in absorption and mechanical compression chiller cooling methods. According to temperature drop results (Fig. 10), the absorption chiller method reaches a better cooling effect in comparison with evaporative cooling, mainly when the ambient intake temperature is higher, up to 20°C.

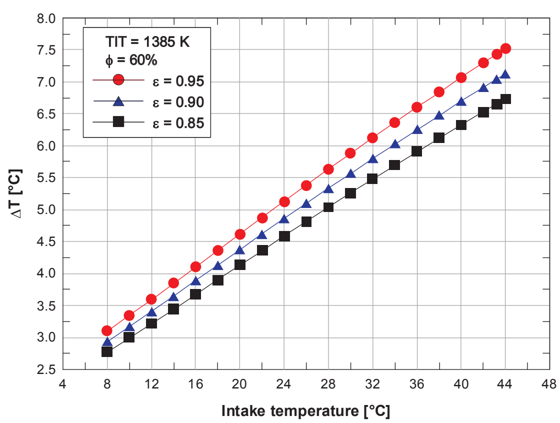


Figure 7. Effect of evaporative cooling effectiveness on the inlet temperature drop.

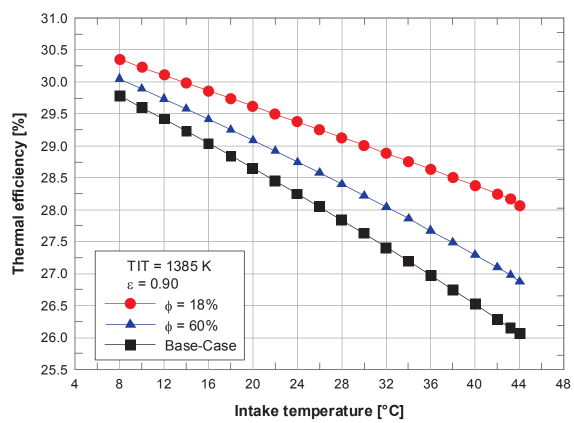


Figure 9. Effect of ambient intake temperature on the gas turbine thermal efficiency employing evaporative cooling system.

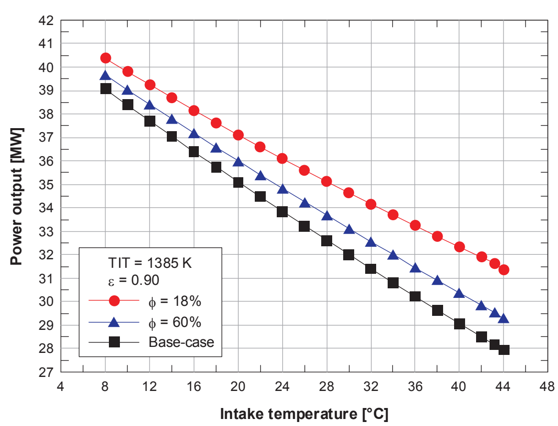


Figure 8. Effect of ambient intake temperature on the gas turbine power output using evaporative cooling.

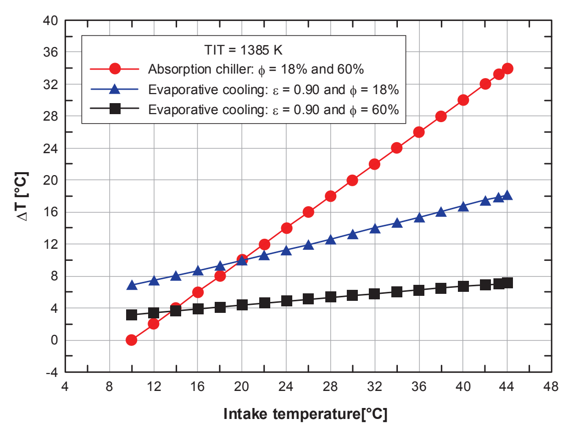


Figure 10. Comparison between evaporative and absorption chiller inlet cooling methods.

Figure 11 presents the gas turbine power output results obtained by inlet chilling technique at ϕ = 18% and 60%, and the Base-Case is also plotted. Note that the power output is independent of ambient temperature because the compressor inlet temperature is fixed at 10°C. Thus, the inlet compressor temperature is the same for all intake temperature (abscissa axis in Figs. 11 and 12). This fact implies that the absorption chiller can be employed in a wide range of ambient conditions when compared with evaporative cooling.

The power output gain is considerable when the absorption chiller is applied, mainly at higher ambient temperatures. For example, at T0 = 30°C, the increment is about 4.2 MW when compared with Base-Case value (without cooling). This same behavior is presented by the gas turbine thermal efficiency results, as shown in Fig. 12.

The thermal efficiency gain in comparison with Base-Case is lesser at low ambient temperatures, but this effect is more intense at higher ambient temperatures. When the ambient temperature is 30°C, the thermal efficiency increases 6.5% in comparison to Base-Case results. While for the absorption chiller the power output is constant (Fig. 11), the mechanical refrigeration chiller has the power output affected by the chiller coefficient of performance (COP), as shown in Fig. 13.

At lower intake ambient temperatures and ϕ = 60%, the COP variation is not remarkable, but its effect is more significant as the ambient temperature increases. For example, at T0 = 40°C and COP = 7.0, the power output is 4.2 MW and 0.9 MW larger than results obtained with COP = 2.0 and COP = 4.5, respectively.

It is important to remember that although a mechanical chiller system with high COP provides a relevant gas turbine power improvement, it requires also a high cost investment. So, at the present study, a mechanical chiller machine with COP = 4.5 satisfies a trade-off solution between power output gain and economic features (Farzaneh-Gord and Deymi-Dashtebayaz, 2011; Jaber *et al.*, 2007).

The COP influence on the thermal efficiency (Fig. 14) presents a behavior similar of obtained for the power output

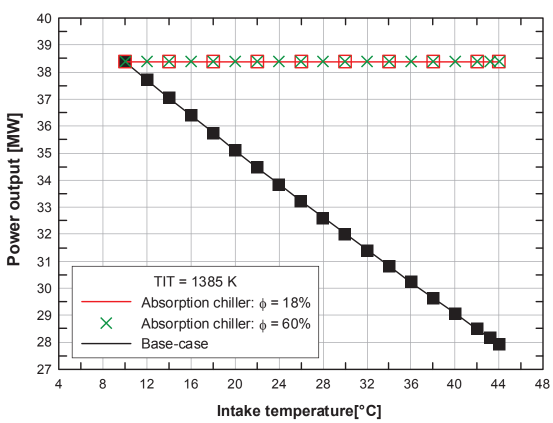


Figure 11. Effect of ambient intake temperature on the gas turbine power output absorption chiller cooling system.

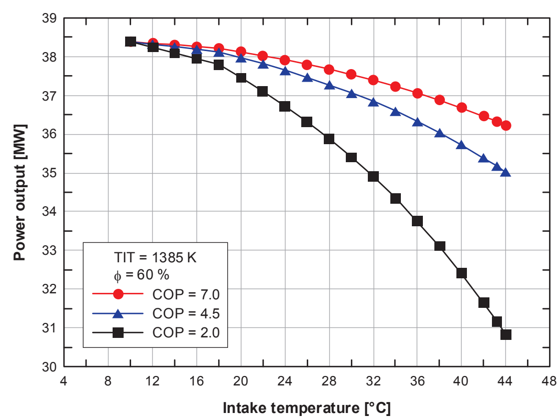


Figure 13. Effect of mechanical chiller coefficient of performance (COP) on gas turbine power output.

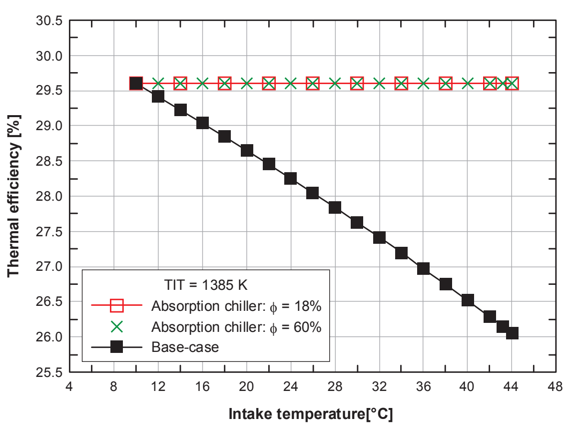


Figure 12. Effect of ambient intake temperature on the gas turbine thermal efficiency absorption chiller cooling system.

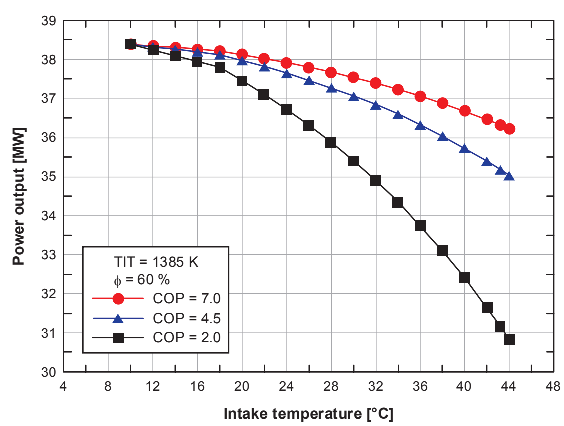


Figure 14. Effect of mechanical chiller coefficient of performance (COP) on gas turbine thermal efficiency.

results, that is, a thermal efficiency increase as the COP elevates. However, this effect is more pronounced when the COP changes from 2.0 to 4.5 in comparison to 4.5 to 7.0 COP value.

The ambient relative humidity influence on the gas turbine power output is plotted in Fig. 15, using a COP equal to 4.5. Results showed a power output enhancement when the relative humidity is lower (ϕ = 18%) in comparison with ϕ = 60%, but this behavior is more significant when the intake ambient temperature is superior to 20°C.

The gas turbine power output gain under ϕ = 18% operation is due to a higher temperature drop (sensible heat transfer) under unsaturated conditions in comparison with ambient relative humidity equal to ϕ = 60%. After the saturated condition is reached (ϕ = 100%), the cooling load is now represented by the water condensed process (latent heat transfer).

According to previously cited, evaporative cooling, absorption and mechanical chiller systems provide a better performance at lower ambient relative humidity

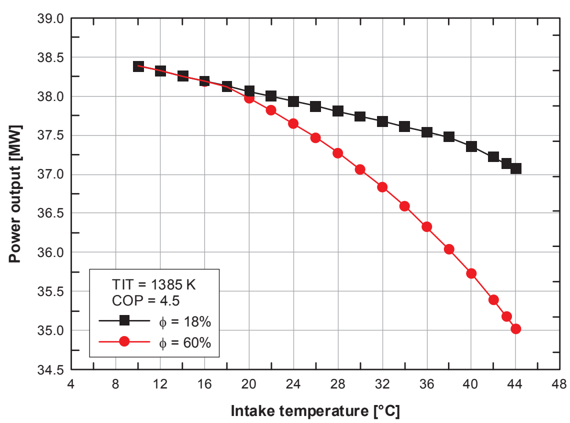


Figure 15. Effect of ambient intake temperature on the gas turbine thermal efficiency employing mechanical chiller cooling system.

(Figs. 8, 11 and 15). Thus, the mass flow water and cooling load comparative results plotted in Figs. 16 and 17 were obtained at ϕ = 18%.

It can be also observed that the evaporated mass flow water for the evaporative cooling system presents a quasi-linear water consumption distribution. On the other hand, there is no condensed mass flow water for the chiller systems until the intake ambient dew-point temperature is inferior to the pre-fixed compressor inlet temperature of these two cooling methods, which occurs at intake ambient temperature higher than 38°C, as depicted in Fig. 16.

The gas TIC methods have a cooling load represented by the sensible and latent heat transfer mechanisms (Fig. 5). For the evaporative system, the temperature drop is limited by the wet-bulb temperature of the intake air leading to a smooth growth in the sensible cooling load as the ambient intake temperature increases. Nevertheless, the cooling load obtained for absorption and mechanical chiller systems is subjected to the pre-specified compressor inlet temperature (10°C), as shown in Fig. 17.

At this case, when the intake ambient temperature is higher than 20°C and ϕ = 18%, both chiller systems require cooling loads superior to the evaporative cooling method due to a more intense temperature drop provided by the absorption and mechanical chiller methods. Besides this sensible cooling load, there is a latent heat transfer parcel regarding to the condensed mass flow water (Fig. 16).

Notice that for the absorption chiller technique, the cooling load required to cool the intake air temperature until 10°C comes from the gas turbine gases exhaust, and it is no computed in the power output results, as observed in Fig. 18, whereas the mechanical chiller method requires a cooling load that penalizes the gas turbine net power output.

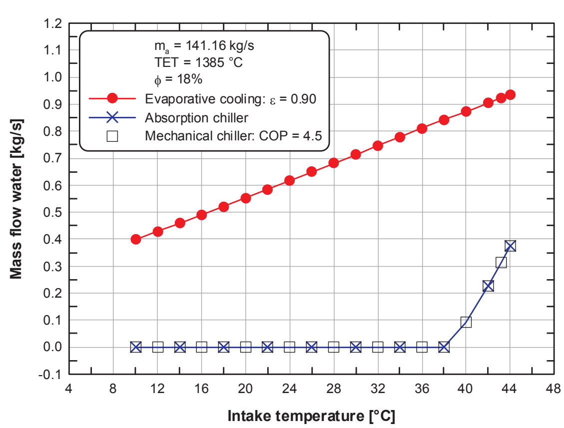


Figure 16. Effect of ambient intake temperature on the evaporated and condensed mass flow water.

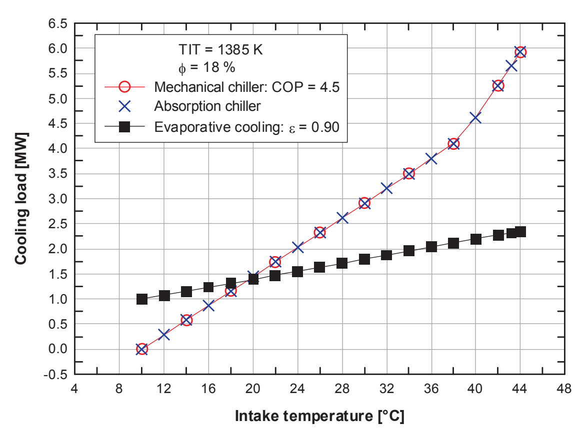


Figure 17. Effect of ambient intake temperature on the cooling load of evaporative and chillers cooling systems.

Figure 18 also shows that the advantage of compressor inlet cooling method in improving gas turbine performance, once all tested cooling systems results in power output augmentation in comparison with Base-Case (case without cooling). As shown in Fig. 19, this increase in power output and thermal efficiency results is less effective when the intake air temperature elevates for the three cooling methods.

At lower intake ambient temperatures, the two chiller technologies have a similar behavior (Figs. 18 and 19). However, as the inlet air temperature increases the mechanical chiller efficiency (which is determined by its COP value), it imposes a gas turbine reduction performance, while the absorption chiller provides a constant power output and thermal efficiency results.

Gas turbine performance was tested for two different geographic sites, as indicated in Table 2. The analysis is performed computing the average monthly maximum dry bulb temperature and average monthly relative humidity over 2006-2010 years (extracted from automatic meteorological station, according to INMET (2011)). At this procedure, the daily collected actual data (maximum temperature and average relative humidity) were treated to calculate an average monthly value over the analyzed 5 years.

Table 2. Geographic sites.

Site	Campos, RJ, Brazil	Goiânia, GO, Brazil
Altitude [m]	25.0	770.0
Latitude [°]	- 21.72	- 6.64
Longitude [°]	- 41.34	- 49.22
Pressure [kPa]	101.25	92.8

Campos is a coast city, with higher average maximum temperatures occurring during the summer months (31-33°C) in December to March, and lower ones occurring in June to August (~27.5°C), as shown in Figure 20. The average relativity humidity level is high during all year, reaching almost 80% in April (Fig. 21).

On the other hand, Goiânia is a planned city located at the central region of the Brazil with maximum average

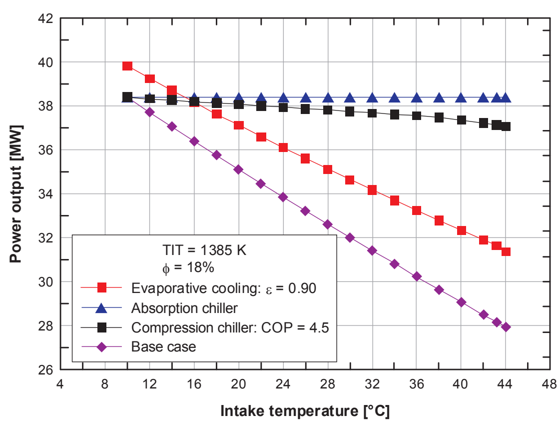


Figure 18. Comparison between the gas turbine power output for each simulated cooling techniques.

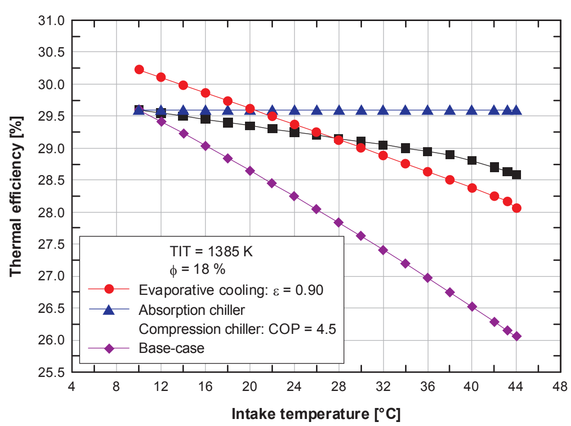


Figure 19. Comparison between the gas turbine thermal efficiency for each simulated cooling techniques.

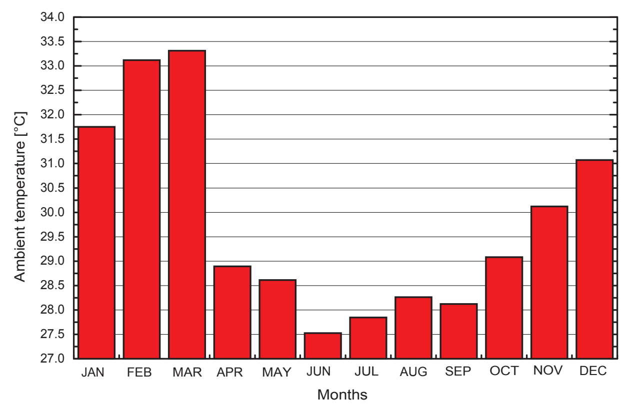


Figure 20. Campos's site: monthly average maximum temperature occurred over 2006-2010 years.

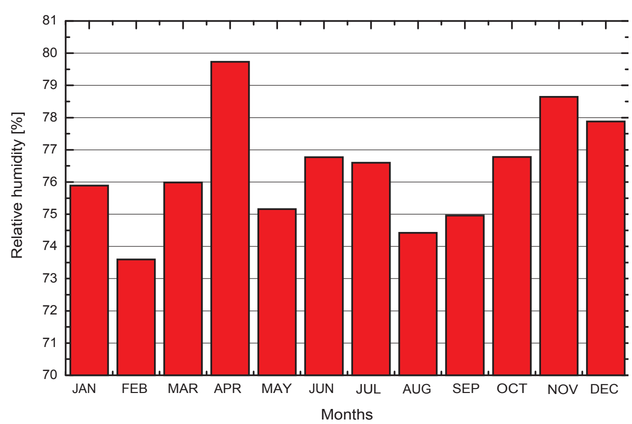


Figure 21. Campos's site: monthly average relative humidity occurred over 2006-2010 years.

temperatures (33-34°C) occurring during August to October. For the other months, the temperature is in the 30.5-31.5°C range, as presented in Fig. 22. Differently of Campos city, this site has lower average relativity humidity levels (Fig. 23). For example, the collected value in August was around 40% against 74% obtained for the Campos's site. This fact will be important when the cooling methods will be tested, mainly for the evaporative cooling which is strongly dependent from the relative humidity level.

As the water reservoirs level usually decreases from July to December in these geographic sites, the hydropower generation also is reduced. Consequently, the Brazilian agency named *Operador Nacional do Sistema Elétrico* (ONS) determines the thermal power plant operation during this period. At this context, the incremental electric energy generation and an analysis of the electricity cost will be herein conducted only for the second semester (over cited 2006-2010 meteorological data).

Figure 24 and Fig. 25 present the incremental electric energy generation provided when the cooling methods are included, for the Campos and Goiânia locations, respectively. Notice that, these values represent enhancement in relation to the energy generation produced by the gas turbine without any cooling.

Employing results of economic analysis presented in previous literature works (Al-Ibrahim and Varnham, 2010; De Lucia *et al.*, 1996; Gareta *et al.*, 2004), an economic study was also performed taking into account the cost of additional devices associated with each cooling method. Besides, the stationary power generation system is expected to have a high utilization factor, i. e., this system is assumed to operate 20 hours per day, 180 days per year, resulting in 3,600 hours per year.

Thus, the total incremental annual energy unit cost had been computed in Campos and Goiânia, considering the following contributions: capital cost associated with each inlet air cooling method, operation and maintenance (O&M) fixed and variable costs, and incremental fuel costs. Table 3 presents the increment in annual energy generation (computed by sum of the monthly contribution presented in Fig. 24 and Fig. 25),

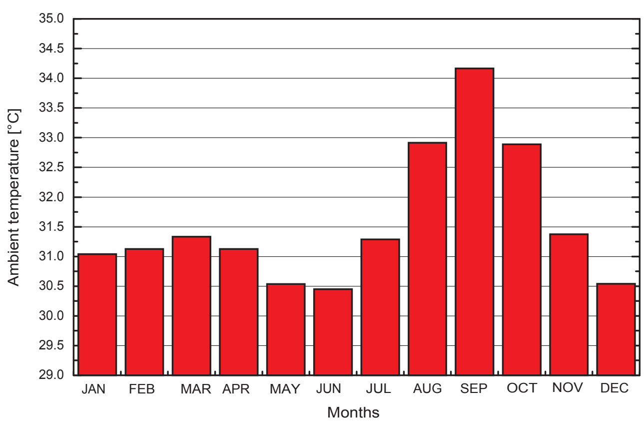


Figure 22. Goiânia's site: monthly average maximum temperature occurred over 2006-2010 years.

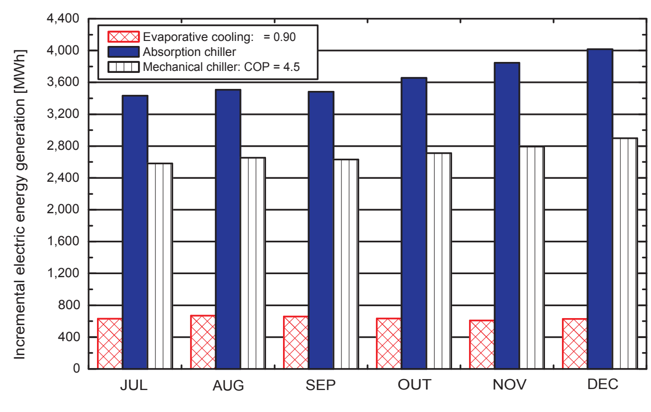


Figure 24. Campos's site: monthly incremental electric energy generation provided by the tested cooling methods.

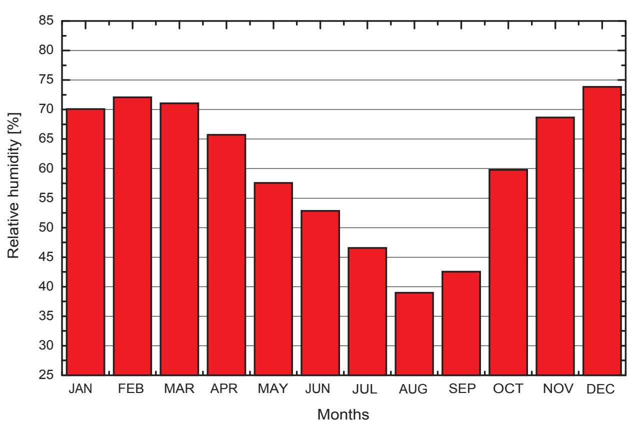


Figure 23. Goiânia's site: Monthly average relative humidity occurred over 2006-2010 years.

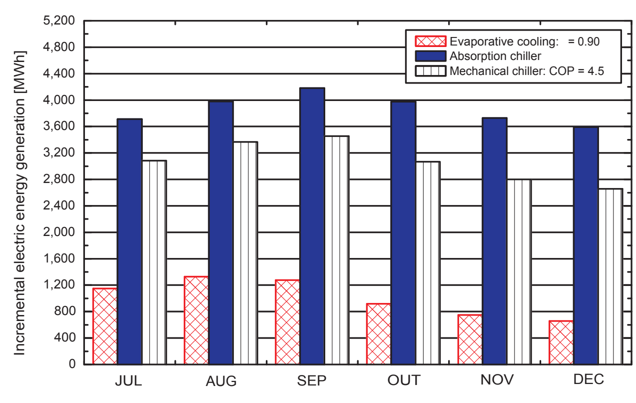


Figure 25. Goiania's site: monthly incremental electric energy generation provided by the tested cooling methods.

fuel cost and total energy unit cost obtained for Campos and Goiânia's sites, employing the three studied cooling methods.

The evaporative cooling provides monthly augmented energy generation lesser than 800 MWh for Campos's site (Fig. 24) resulting in a annual contribution of about 5,000 MWh as shown in the third column of Table 3. The absorption chiller system produced close to four times more electric energy than the power plant with evaporative cooling system (~22,000.0 MWh). For the mechanical chiller method, the yearly incremental electric energy generation is around 16,000.0 MWh in the Campos site (Table 3, third column) due to power extraction from the gas turbine to drive the vapor compression equipment.

On the other hand, Fig. 25 shows that the incremental monthly electric energy production at Goiania's site is higher than Campos city (Fig. 24) when the evaporative system is employed due to the lower relative humidity occurred at the first site. Although this cooling method offers the lowest total energy unit cost (Table 3), the evaporative option has an inferior capacity of increment in annual energy generation when compared with the other two cooling methods.

Results showed that for both locations, the best cooling option is the absorption chiller system, which provided simultaneously a superior value of the increment in annual energy generation and a low energy unit cost in comparison with mechanical chiller method.

CONCLUSIONS

The variation of the ambient air temperature showed that at ISO conditions, an increase of 21°C in the intake temperature generates a reduction of 11.46% in the gas turbine power output and of 7.18% in the cycle thermal efficiency (Fig. 6). These results indicate that TIC technologies are primordial to ensure the gas turbine load capacity, even at severe ambient conditions. The evaporative cooling method provided a limited temperature drop because it depends on the ambient wet-bulb temperature while the chiller systems offer a large cooling effect (Fig. 17).

In order to perform a comparison between evaporative and chiller cooling systems, the gas turbine performance was tested under different operational conditions (ambient temperature and relative humidity). At T_0 = 36°C and ϕ = 18%, the Base-Case (without cooling) gives a power output of 33.17 MW (Fig. 18). An increment of 3.2 MW is obtained when the evaporative is used. On the other hand, the power output is augmented in 4.5 MW and 5.3 MW for the mechanical and absorption chiller systems, respectively.

It is important to observe that any cooling system require additional components. For example, the absorption chiller needs a heat recovery device to utilize the gas exhaust energy. However, these added components present an inferior cost when compared with a large simple-cycle gas turbine engine (Najjar, 1996).

Thus, the best cooling alternative must take into account several factors as gas turbine parameters, power plant installed capacity, load operation type, site location, climatic conditions, desired cooling potential, and economic feasibility. At this context, this work showed that considering two specified Brazilian's site and economic features, the absorption chiller presented higher increment in annual energy generation with a lower unit energy cost.

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Table 3. Increment in annual gas turbine parameters.

City	Cooling method	Increment in annual energy generation [MWh]	Increment in annual fuel cost [$/MWh]	Increment in energy unit cost [$/MWh]
Campos	Evaporative	5,048.57	9.96	12.65
	Absorption chiller	21,943.26	37.05	43.28
	Mechanical chiller	16,268.38	49.97	56.45
Goiânia	Evaporative	8,582.36	29.33	32.02
	Absorption chiller	23,166.46	37.01	43.24
	Mechanical chiller	18,429.30	46.52	53.00

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