**Techno-economic analysis of Gas Turbine inlet air cooling for Combined Cycle power plant for different climatic conditions**

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**Abstract**

This paper reports on a techno-economical parametric analysis of an inlet air cooling system applied to an aero-derivative Gas Turbine (GT) for a combined cycle power plant (CC). The system is based on a cold water thermal storage charged nighttime by mechanical chillers; chilled water is used in the hottest day hours to cool inlet air to the compressor. Three cases have been analyzed, supposing the plant operating in different sites, Phoenix (AZ - USA), New Orleans (LA - USA) and Abu Dhabi (UAE), characterized by quite different climatic conditions. Particular attention was paid to the influence of storage volume and heat exchanger sizing on both energetic and economic performances. Results have been obtained by a numerical code specifically developed to size the inlet air cooling system and to model the whole power plant behavior over the entire year on one hour basis. A 55 MWe combined cycle power plant with a GE LM6000 gas turbine was assumed as a reference case. Operational hours and power output augmentation were higher in hotter climates; wet climates required huge thermal storages, thus increasing the investment cost. The best techno-economic performance is attained for sites with high temperature combined with low relative humidity, typical of desert areas. The parametric analysis showed that the size of cooling storage is a very important parameter for the economical revenue.

**Keywords:** Gas turbine, inlet air cooling, thermal energy storage.

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**Nomenclature**

C cash flow ($)

CC combined cycle

COP chiller coefficient of performance

E electricity production (Wh)

GT gas turbine

IC inlet air cooling

IRR internal rate of return (%)

NPV net present value ($)

P net power (W)

PBT payback time (y)

Q heat (W)

r discount rate (%)

R.H. relative humidity (%)

ST steam turbine

$T_{amb}$ ambient temperature (°C)

$\eta$ plant efficiency (%)

$\eta_{incr}$ incremental cycle efficiency (%)

## 1. Introduction

It is well known that Gas Turbine (GT) performances highly depend on inlet air conditions: in particular, power output strongly decreases as inlet air temperature increases. An ambient temperature increase of 10°C gives rise to a simple cycle Gas Turbine (GT) power output decrease of about 5% to 13% and to a cycle efficiency reduction of about 1.5% to 4%, depending on the GT model. When considering the Combined Cycle (CC) power plant, a similar power output decrease (about 5 to 8%) takes place [1].

Bianchi et al. [2], but also Wang and Braquet [3], give a comprehensive picture and a critical evaluation of technologies available for GT power augmentation. Among the different strategies available to compensate for power reduction at high ambient temperatures, the adoption of inlet air cooling systems (IC) appears a very interesting choice in almost all CC configurations. Barigozzi et al. [4] have shown that inlet cooling system performance (and thus the achievable profits) is strongly correlated with key gas turbine characteristics. In particular, aero-derivative gas turbines, compared to heavy-duty machines, achieve higher performance. Power augmentation technologies also improve plant flexibility as they can be used to have a better control of daily cycle operating strategy, to match the grid demand for profit optimization and to give access to the frequency regulation market.

Different inlet air cooling technologies are available nowadays for GT power augmentation. They can be classified into two main categories: water evaporation systems and heat transfer systems. Water evaporation systems include evaporative cooling, inlet fogging and even over-spray systems. Heat transfer systems instead are diminishing inlet air temperature directly by mechanical or absorption chillers or indirectly by cold thermal storage systems.

In water evaporation systems, a certain amount of demineralized water is mixed with air and starts to evaporate decreasing the stream temperature. According with Ehyaei et al. [5], with such a technology, the minimum achievable temperature is limited to the ambient wet bulb temperature. Lower air temperatures and therefore larger power outputs can be attained by heat transfer systems based on chillers. This is because inlet air temperature only depends on heat exchange surface and coolant temperature (for water typically 4°C) [6]. A drawback of indirect cooling systems is the increase of inlet losses for heat exchangers introduction that gives rise to a slight GT loss of performance. Compression or absorption chillers or even an inverse refrigeration Brayton cycle coupled with the GT simple cycle, as proposed by Chen et al. [7], can be used. Palestra et all. [8] have shown that the use of a cold thermal storage can produce significant economic benefits as it allows to increase the electric power production when ambient temperature and electricity price are high, while shifting chillers electrical consumption during night time, when energy is cheaper. Another advantage is the improved possibility to control power output independently on the ambient conditions. In this way the difference between the energy that has been programmed to be sold and the one that is effectively sold to the market is reduced and the revenues optimized. Al-Ibrahim and Varnham [9] made a comparative analysis of different inlet air cooling systems for GT power augmentation in the Saudi Arabia context. They concluded that wetted media evaporative cooling is better suited for hot and dry climates, where inlet air temperature values about 20°C can be reached with relatively low cost. But in areas where water availability is a problem, mechanical refrigeration with cold thermal storage seems to be the best solution. Dos Santos et al. [10] also made a comparison between evaporative cooling and absorption chiller direct cooling, underlying the limits of applicability of evaporative cooling in wet climates.

Cool thermal storage, in addition to standard direct cooling system components, requires a tank. As cooling media, both chilled water and ice can be used. If water is chosen, the amount of stored energy depends on the difference between the warm water, coming from the air coils, and the cold water produced by the chillers. Among the different technologies available to keep separate warm and cold water, the most used is stratified chilled water storage. Such an application is simple and, compared to standard cooling systems, does not require any extra component but the storage tank. However, a huge volume is necessary for the tank, especially if the difference of temperature between chilled water and return water is small. Another possibility is ice harvester storage systems. To store cool energy, this second option uses the ice latent heat. Ice is stored into a tank in two phase equilibrium with liquid water. Ice harvesters produce ice on the external surface of an evaporator; when a certain thickness is reached, ice is harvested by heating the evaporator and leaving the sheets of ice falling by gravity into the storage tank. The heat required by the harvesting process is generally provided by the hot refrigerant gas taken from the chiller compressor.

Cold thermal storage selection can lead to very different economical results. In fact, the choice of the technology highly influences the investment cost: an ice storage with ice harvester machines is much more expensive than a system based on stratified chilled water requiring a much greater storage volume but a traditional and cheaper chiller. A power output increment over 25% has been reported by Al Bassam and Al Said [11] for an ice harvester system applied to a simple cycle power plant operating in Saudi Arabia, while Ameri et al. [12] for a plant in Iran have shown that the high investment cost made the choice unprofitable. In fact, the larger power output increase, given by ice thermal storage, was not enough to compensate for the higher capital cost. Similar conclusions were drawn by Palestra et al. [8] for a 127 MWe CC power plant operating in two different climatic scenarios (northern and southern Italy).

To evaluate cost-effectiveness of an Inlet Cooling system, a detailed investigation of CC operation integrated with IC system is necessary. The reason is the great influence of each site’s climate and plant configuration on the thermodynamic and on the economic final result. Gareta et al. [13] have shown, for the Spanish climate and power market scenario, that small direct cooling systems perform better than indirect ones with large thermal storage. Yokoyama and Ito [14] developed a numerical code to study inlet cooling with ice storage applied to GT cogeneration plants: the most relevant effect was the reduction of peak electricity demand obtained through an increase of capital cost and natural gas consumption. Chacartegui et al. [15] compared different IC systems applied to a combined cogeneration power plant. They showed that the introduction of a thermal storage coupled with an electrically driven chiller does not give relevant advantages with respect to the direct cooling solution; however, the thermal storage system allowed for a larger Net Present Value. Another interesting application in the field of cold thermal storage systems is the use of phase-changing materials (PCM), although this technology requires further development and investigations for a GT inlet air cooling implementation (Bedecarrats [16]).

In the present study, an in-house code developed for modeling GT inlet air cooling and supercharging [1,4,8] was used to evaluate the thermodynamic performances and the economics of a CC plant based on GE LM6000 GT model. The behavior of the whole system (combined cycled power plant with inlet air cooling and cold thermal storage) is examined for a hot and dry location (Phoenix, AZ, USA) typical of desert areas, a temperate and wet one (New Orleans, LA, USA) and a hot and wet climate (Abu Dhabi, UAE). The economic performance is evaluated for the three cases, assuming energy tariffs of the actual US, even for Abu Dhabi. As design choices heavily influence investment profitability, a parametric analysis, by independently varying the thermal storage and the heat exchanger size, has been also carried out.

## 2. Reference power plant and design operating conditions

The reference power plant of the present study is a 55.5 MW combined cycle based on GE LM6000PF GT and a two-level pressure bottoming steam cycle coupled with an air cooled condenser (fig. 1). GE LM6000PF engine was selected due to its high sensitivity to inlet air temperature variation and because of its wide diffusion worldwide. The main features of the selected GT at ISO condition and of the whole CC power plant are reported in Table 1.

TABLE 1

The cycle is rated with an efficiency of about 54%. Steam is produced at two pressure levels: 12.1 kg/s at 400°C / 60 bar and 3.6 kg/s at 220°C / 10 bar. With a design condenser pressure of 0.034 bar at ISO condition the steam turbine gross power is rated 13.77 MW. The low design condensing pressure was dictated by the need to guarantee a good condensing performance even at the high ambient temperature levels characterizing the selected installation sites. In fact, the oversized condenser heat transfer surfaces allowed to get a steam turbine back pressure of about 0.17 bar in the hottest operating condition ($T_{amb}$ = 45°C and 22% R.H.).

The GT inlet air cooling system has been already analyzed by the authors in previous studies comparing different storage systems [8] and different GT models [4]. In the present analysis the plant layout shown in Fig. 1 has been assumed. Chilled water is produced by using centrifugal compressor chillers driven by AC motors. During off-peak hours chillers are accumulating cooling capacity into a storage system based on stratified chilled water tank. Chiller COP (Coefficient of Performance) at nominal ISO conditions was assumed equal to 5.5. COP then varies depending on ambient condition. But it has to be noted that chillers only operate during nighttime, i.e. with favorable climatic conditions for heat rejection. Of course, COP values differ from site to site, with the lowest COP values reached in Abu Dhabi, also due to the high R.H., and the highest in Phoenix.

When the inlet cooling system is turned on, the chilled water is pumped into cross flow heat exchangers installed in the GT inlet casing just downstream of the filter section. Inlet and outlet water temperatures are 4°C and 8°C respectively, as required by thermal stratification. The pinch-point between water and cooled air stream has been set to 2°C. A drainage system, particularly significant in wet climates, is included in order to separate the condensed humidity from the gas stream before entering the compressor. Chillers have been finally supposed to be cooled by dry, mechanically driven towers, so their performance will be affected by ambient conditions.

FIGURE 1

## 3. Simulation method and assumptions

An in house developed Matlab® computer code was used to carry out automatically the IC system design and to model the performance of the whole plant (i.e. the combined cycle and the inlet air cooling system) for any possible operating condition. The structure of the simulation code is presented in Figure 2. Once the CC power plant configuration and the site location are defined, the first step of the integrated inlet cooling system simulation consists of providing all the input data required for the automatic design of inlet air cooling system components. These input data can be organized in three main subcategories:

1. climatic data;
2. reference CC power plant performance database;
3. user defined input data and control strategy, including the economic assumptions.

Two US sites and an Arabic Gulf location were selected in order to assess the influence of ambient temperature and relative humidity on the inlet air cooling system: Phoenix (AZ, USA), representative of a hot and dry climate, New Orleans (LA, USA) with a warm and wet climate and finally Abu Dhabi (UAE) with a wet and very hot climate. The METEONORM database from the TRNSYS® weather library provided the meteorological data for the annual simulations for the selected site locations. Figure 3 shows the comparison between the three sites climate in terms of temperature and relative humidity distributions. In each plot the number of hour per year of each one temperature degree and 10% relative humidity steps are compared. In particular, the average temperature value recorded over the year in New Orleans is about 19.9°C against 22.5°C of Phoenix and 26.7°C of Abu Dhabi. Conversely, New Orleans average relative humidity is about 73.9% against 58.1% of Abu Dhabi and 36.3% of Phoenix. The different climatic conditions are expected to influence the design of the heat exchanger and its performance as well, due to quite different water condensation rates in the wetter and hotter climate.

The CC power plant performance database has been previously built up for all possible operating conditions over the year by means of Thermoflex® commercial software. For each hour of the year simulations have been carried out for all the possible GT inlet air temperatures resulting from the cooling system (air temperature, relative humidity (R.H.) and pressure due to heat transfer coils). Thermoflex© model was previously tuned to match manufacturer performance data. Figure 4 shows the comparison between manufacturer and simulated trends versus ambient temperature of GE LM6000PF power output, heat rate, exhaust gas mass flow and temperature. The influence of R.H. on GT performance was also considered, as at high ambient temperature GT power output increases with rising R.H..

FIGURE 2

FIGURE 3

Heat exchanger design is automatically performed on the basis of the assumed temperature value for air cooling in the most severe ambient condition occurring on site. In the present study a limit value of 10°C for the coils air exit temperature has been assumed in the heat exchangers design. Daily system's operation was limited to eleven hours in order to limit the storage volume required. Storage tank volume is selected indeed by considering the cooling demand during on-peak hours, while chiller is sized in order to restore completely, during off-peak hours, the stored cold energy. In the design phase, the user may decide to undersize the cooling storage and/or the air coils in order to limit installation costs. In this case, GT inlet air temperature values higher than the set point can take place in the hottest hours and stored cooling water can be fully consumed before the 11 hours operation. In the simulations of the three cases related to different sites a 50% under sizing was assumed both for the air coils and the storage volume (Table 2).

In the usage of stored water, an optimization of control strategy is also performed by varying the number of operational hours or the inlet air temperature in order to maximize the income of the single day operation. Table 2 summarizes the code input data.

FIGURE 4

Input data include all the information required for the economical analysis. Capital costs have been estimated on the basis of data found in the literature [17] or directly provided by cooling system components manufacturers. Table 3 shows components price considered in the analysis. Note that soil cost was not considered. Finally, for the economic evaluation the US market was considered and the following assumptions have been adopted:

- an average energy price of 0.07 US$/kWh based on the electric US market results for the year 2010 (www.crea121online.it/biblioteca/Il%20mercato%20elettrico%20in%20USA.pdf, 08/01/2015) from which an hourly energy price distribution was derived (Fig. 5);
- a natural gas cost of 0.38 US$/Sm³;
- O&M expenses of 2.5% of total installation cost;
- investment return over 11 years;
- a 7% discount rate, i.e. the rate used to discount future cash flows to the present value;
- a 91% power plant availability, i.e. about 8000 h/y of CC plant operation.

All the input data defined, the code automatically performs the design of all main components (air coils, storage tank, chillers, cooling towers, piping and pumps); a simulation of the plant all over the year is then carried on, allowing to define the operating conditions of combined cycle with and without inlet air cooling on a one hour basis. All relevant parameters are made available at the end of the simulation for every hour of every day in the year. Information like the larger amount of electric energy produced by the combined cycle with the inlet air cooling system, the related natural gas consumption increase and the chillers energy consumption during night time are typical outputs. All these data are used to compute capital costs and cash flows, from which indicators of profitability like pay-back time (PBT), net present value (NPV) of the investment and internal rate of return (IRR) are calculated as follows:

$$NPV = \sum_{n=0}^{N} \frac{C_n}{(1+r)^n} \qquad (1)$$

where $C_n$ is the net cash flow at year n, r is the discount rate (7%) and N is the time period for investment return (11 years). NPV is thus a function of both discount rate r and investment time period N. IRR is then computed as the rate of return r for which NPV function is zero. PBT is instead computed as the year (n) at which the cumulative cash flow (NPV) equals the capital costs.

TABLE 2

TABLE 3

FIGURE 5

## 5. Results and discussion

Plant simulations have been carried out for a whole year on a one hour basis for the three locations. Daily results for two representative days (July and January) are reported to show the influence of the inlet air cooling sizing on the most relevant thermodynamic parameters and CC performances. Then annual CC plant performances and economic analysis as well are presented for the considered site locations. As design choices heavily influence investment profitability, a parametric analysis has been finally carried out, by independently varying thermal storage and heat exchanger size.

### *5.1 IC system configuration*

5.1.1 Inlet air cooling component sizing

Table 4 reports the most significant data derived from the automatic design of inlet air cooling system components, i.e. air coils, storage tank and chillers. Cooling systems have been sized in order to guarantee 50% of the cooling demand during the hottest day of the year. For all the considered locations minimum air temperature at GT inlet was assumed equal to 10°C, while chilled water temperatures have been set 4°C at inlet and 8°C at outlet. Maximum heat load in the specific site depends on the temperature and relative humidity in the hottest day of the year. Phoenix and Abu Dhabi are characterized by almost the same maximum temperature of about 45°C, but Abu Dhabi has a much higher R.H.. New Orleans maximum temperature is much lower, about 33°C, but its R.H. is even higher than in Abu Dhabi. As a consequence, heat exchanger surface resulted to be wider for air coils applied to the hottest and wet climate of Abu Dhabi and lower for the US locations, with New Orleans showing the lowest value. This is because both sensible and latent heat contributions to cooling load are higher at Abu Dhabi. At New Orleans latent heat contribution is relevant but not the sensible one, while the latter dominates in Phoenix. For Abu Dhabi a larger heat exchange surface is required, so tube sheet number is increased and/or fins spacing reduced. The final result is that higher pressure losses on the air side occur for Abu Dhabi, followed by Phoenix and New Orleans.

The main features of storage tank and chiller for all the considered cases are also reported in Table 4. According to the previous analysis, a huge storage volume is necessary in wet climates (Abu Dhabi and New Orleans), due to the strong contribution of latent heat. A shift from a wet to a dry climate, at constant ambient temperature (i.e. comparing Abu Dhabi to Phoenix) results in less than half the storage volume. Chillers are sized accordingly to restore nighttime the daily exploited chilled water. Plant location climate is thus shown to have a significant influence on size and performances of all cooling system components.

TABLE 4

5.1.2 Thermodynamic results

Yearly simulations on a one hour basis have been carried out for the three site locations, both with and without the IC system in operation. To discuss the influence of different site climates, two daily ambient temperature profiles with the corresponding R.H. have been selected, one representative for July and one for January (Fig. 6). Note that for the July day R.H. is progressively increasing when moving from Phoenix to Abu Dhabi and finally to New Orleans. For the typical January day, New Orleans and Abu Dhabi show similar R.H. peak values, with the latter characterized by a higher variability.

Figure 7 shows air temperature at compressor inlet during plant operation over the two considered days. As can be observed, the minimum air temperature level is different for the two days, depending on ambient temperature and cooling system capacity. An optimization routine provides indeed the inlet air temperature set-point that maximizes daily revenues. Typically, during the hottest day, an increment of inlet air temperature grants the inlet air cooling system to remain operational all along the peak period. If a lower inlet air temperature had been chosen, the thermal storage would have been exhausted in advance, obliging the inlet air cooling system to be turned off.

Figure 8 reports chilled water available in the storage. In the summer day about 80% of total stored chilled water is consumed. This value is typical of stratified chilled water systems, where useful thermal stored energy is always below 80-90% of the total. Thus it results that stored cooling energy is completely exhausted for all the considered climates in the typical day of July.

FIGURE 6

FIGURE 7

FIGURE 8

FIGURE 9

An interesting data for sites located in desert environments is the fresh water production that can be obtained from moisture condensation. Figure 9 shows each hour the amount of condensed water from the air stream crossing the air coils. As expected, New Orleans location is the one characterized by the larger amount of condensed water, followed by Abu Dhabi. In both these sites air humidity is condensed both in summer and winter time, with a peak production of about 4 $m^3/h$ as a maximum. Shifting attention to Phoenix, a much smaller amount of water is condensed in summer time, while no condensation takes place in winter.

Figure 10 reports the power outputs and power output augmentation for the two selected typical days and the considered site locations. When IC system goes into operation CC power output undergoes an increase of 7-9 MW (about 15%) in July while only of 2 MW in January, whatever the site location. Power augmentation progressively increases along the day hours up to about 10 MW for New Orleans and 14-15 MW for the two other locations. Of course the inlet air cooling system never allows to restore the winter performance during summer day because of the assumption of limiting both storage volume and air coils surface in order to satisfy only 50% of the maximum cooling demand in the year.

Figure 11 reports the daily distributions of CC plant efficiency variation for the considered cases, being plant efficiency variation defined as the difference between the net CC plant efficiency with and without IC system in operation:

$$\Delta\eta = \eta_{CC,IC} - \eta_{CC} \qquad (2)$$

Efficiency variation takes into account for IC system components consumption, i.e chillers and pumps, when in operation. During daytime the CC plant improves significantly its efficiency for all the sites (up to 1% in summer time for Phoenix and Abu Dhabi). The worst case is the one of New Orleans where moderate temperatures and wet conditions only allow an increase in daytime of 0.5%. During the night, when IC system is off and chillers are in operation to restore the cold water storage, auxiliaries consumption increases, resulting in a decreased efficiency. This is more evident in summer period (down to -1%), while in winter time the efficiency reduction during the night is very small because obviously cold water exploitation is much less. Data of Fig. 11 demonstrate that installing an IC system in a dry and hot climate, like that one of Phoenix, allows for the best efficiency augmentation. A wet climate (like the one of Abu Dhabi and New Orleans) instead produces a stronger efficiency reduction during the night because it requires a huge cold thermal storage to match the higher cooling load due to latent heat of the humid air stream entering the compressor. This is also due to the use of cooling towers to cool the chiller condenser; as a result higher humidity translates in a higher chiller condensing temperature and thus in a lower chiller COP.

FIGURE 10

FIGURE 11

To better appreciate the benefits of IC system, the power output augmentation due to inlet air temperature reduction has been calculated for all the year hours both when air cooling system is on or off. The power values shown in Fig. 12 are net, i.e. after deductions for IC system auxiliaries consumption. The negative values refer to off-peak hours when chillers are operating to recharge the thermal storage; they include chiller power and the gas turbine power reduction due to the presence of air coils. The high data dispersion at high ambient temperature is due to the variation of relative humidity. Power augmentation at high ambient temperatures (thus for Phoenix and Abu Dhabi locations) reaches 14-15 MW, corresponding to roughly 25-27% of the Combined Cycle power at ISO conditions. The power increase reduces down to about 9 MW (corresponding to 16% of $P_{CC,ISO}$) for the case of New Orleans.

When IC is off, power output decrease is never larger than 1.8 MW and for Phoenix (dry climate) it is always less that for the two other cases.

A useful parameter for the estimation of how efficiently peak energy is produced by this system is the incremental cycle efficiency defined as:

$$\eta_{incr,CC} = \frac{\Delta P_{cc,net}}{\Delta Q_{cc}} \qquad (3)$$

$\Delta P_{CC,net}$ is the net increment of power output and $\Delta Q_{CC}$ the extra heat provided by the fuel for the combined cycle with the inlet air cooling system in operation. $\eta_{incr,CC}$ is a significant parameter as it gives, for every hour, the reference marginal cost of generated peak energy by means of inlet air cooling. Thus it is possible to compare $\eta_{incr,CC}$ with the average efficiency of other typical peaking solutions (inlet fogging, supplementary firing, simple-cycle gas turbines etc). Figure 13 shows the incremental efficiency for the three locations, as a function of ambient temperature. Also reported is the reference CC efficiency at ISO condition (dotted red line). When ambient temperature is low (i.e. below 15-20°C), power output increment is small and efficiency is penalized by power losses due to air coils and auxiliaries. As a consequence, over-production of electric energy is obtained with an incremental cycle efficiency smaller than the reference value at ISO condition, excepted Phoenix. For higher ambient temperatures, $\eta_{incr,CC}$ increases even up to 1% more than the net efficiency at ISO condition, i.e. 53.77%. Note that a dry location like Phoenix always shows incremental efficiencies higher than the ISO reference level: this is due to the more efficient cooling performance, not penalized by the cooling load contribution due to latent heat.

FIGURE 12

FIGURE 13

<u>5.1.3 Annual performance results</u>

Monthly averaged values of electric energy overproduction, electric efficiency variation and incremental efficiency were calculated from simulation results on one hour basis. Figure 14 presents net monthly electricity overproduction ($\Delta E_{net}$) for the three cases. In hot climates (Phoenix and Abu Dhabi) electricity overproduction appears to be significantly higher than those of New Orleans. In the hottest months $\Delta E_{net}$ is about 3.5 GWh for both Phoenix and Abu Dhabi, while for the wet and temperate climate of New Orleans the gain is smaller (about 2.5 GWh), because auxiliaries consumption is higher and power overproduction is lower. In winter time the electricity overproduction is strongly reduced, especially for Phoenix, where it becomes even negative in February. This is because of the low ambient temperature that never allows the IC system to go in operation. In Abu Dhabi (but also in New Orleans) the IC system is always in operation, resulting in an overproduction that gets higher than 1.5 GWh (about 0.4 GWh in New Orleans).

Figure 15 shows the annual trend of the monthly average of $\eta_{incr,CC}$ for the three considered locations. In summer time all plant locations show almost the same level of the incremental efficiency, slightly higher than the ISO reference CC plant efficiency of 53.77%. In winter time both wet climates experience an efficiency reduction, particularly New Orleans.

FIGURE 14

FIGURE 15

Table 5 finally summarizes annual plant performance for the three selected site locations presenting maximum and average power augmentation $\Delta P_{CC,}$ electric energy overproduction $\Delta E$, auxiliaries extra consumption $\Delta E_{AUX}$ and fuel overconsumption $\Delta Q_{CC}$. Note that power augmentation has been computed only considering the IC system operational hours. $\Delta E$ and $\Delta Q$ values have been instead integrated all over the year. Finally, $\eta_{incr,GROSS}$ and $\eta_{incr,NET}$ have been calculated as the ratio between gross or net electric energy overproduction and fuel extra-consumption.

Hotter climatic conditions give the opportunity to keep IC system operational for a larger number of hours: in Abu Dhabi it stays in operation all along the year, while in New Orleans and particularly in Phoenix, it is not. Maximum and average power output augmentations during peak hours are also different: larger values are obtained in hot climates with a maximum of 28.5% for Phoenix. The best average value is 9.15 MW for Abu Dhabi, thanks to the greater number of operational hours. As a result, the gross electricity overproduction is much higher in Abu Dhabi (35.7 GWh/y) when compared to Phoenix (28.58 GWh/y) or even to New Orleans

(20.13 GWh/y). Incremental efficiency of gross energy overproduction is always above 54% for all the cases, with the highest value of 55.7% in Abu Dhabi: as already noted, this is generally better than the average efficiency of other technologies. As an example, if peak energy is produced with simple-cycle gas turbine it would present an efficiency of 36-45% depending on the GT model, while supplementary firing would operate with efficiency around 27%. If energy consumed by auxiliaries overnight is considered, net incremental efficiency decreases down to 48.1% for New Orleans, 50.4% for Abu Dhabi and 51.8% for Phoenix, that shows the best performance. However, it has to be taken into account that peak energy is generated in day hours with an efficiency of about 54-56% (with a lower marginal cost), while the 2-6% penalty for auxiliaries is due to energy consumed by night, i.e. when electric energy has a lower value.

TABLE 5

#### 5.1.4 Economic results

Economic evaluation of the investment has been assessed by using the data base of capital costs of the main items (Table 3), energy price and fuel cost based on the US market. Abu Dhabi location was included in this economic analysis, even if its actual market scenario would be different from the US one. This in order to investigate the influence of a wet and hot climate also from the economic point of view. Economic results given by the investment for all cases are summarized in Table 6. For wet climates, about 50-60% of total capital cost depends on the storage tank: this happens because of the huge volume required for air humidity condensation. The storage tank cost strongly reduces when considering Phoenix location. Air coils contribution to capital cost is always about 25%, but Abu Dhabi installation is more expensive, due to the presence of very high temperatures all over the year. Site climate strongly influence the cash-flows; a net positive annual revenue of about 1800 k$ for New Orleans has been found. This is almost doubled for Abu Dhabi location (3200 k$), with Phoenix lying in between (about 2400 k$). This difference is explained only by the site climate that is obviously more favorable for the hot climates, and especially where ambient temperature is always above 10°C, allowing the IC system to be in operation all over the year.

Economic results have been finally evaluated also calculating the Net Present Value (at 11 years, according with an investment return over the same 11 years) NPV, the Internal Rate of Return IRR and the Pay Back Time PBT of the investment. The economic analysis clearly indicates that the implementation of an inlet cooling system with cold water thermal storage always appears a profitable strategy, as NPV values are always positive. Moreover, profitability indexes appear to be strongly dependant on site location. A wet and temperate climate like New Orleans shows the worst performance: PBT is about 3 year, with an IRR of 37.6% and an NPV of 7.2 M$. Abu Dhabi results are much better, especially in terms of NPV (17.7 M$). But it is Phoenix location that allows to get the best performance: even if its NPV is lower than in Abu Dhabi (about 15.2 M$ against 17.7 M$ of Abu Dhabi), it provides the highest IRR of 97.5 % and the shortest PBT of about 1.1 years. All these parameters indicate that the implementation of an IC system with cold thermal storage in a CC power plant equipped with a GE LM6000PC GT on a hot and dry climate site location like Phoenix provides quite good economic results.

TABLE 6

### *5.2 Parametric analysis*

As design choices heavily influence investment profitability, a parametric analysis, by independently varying thermal storage volume and air coil size has been carried out. The total amount of cooling energy that can be stored and used daily is directly influenced by tank volume reduction or increase. The air coil size influences the achievable GT inlet air temperature, and also the IC system operational hours. These choices determine the characteristics of main components like chillers and cooling towers, also influencing the annual revenues, because extra-production of electricity is someway proportional to total consumed cooling energy. Therefore thermal storage volume size has been first varied for all cases taking the air coil at their reference sizing of 50%: the consequent variation of both thermodynamic and economical performances has been then analyzed. The same analysis was repeated by varying the air coil size, at a fixed tank storage size of 50%.

Figure 16 shows in a logarithmic scale the relationship between net electric energy overproduction and storage volume; reference cases (50% sizing) have been highlighted on every line with a filled symbol.

An increment of thermal storage volume allows for a bigger energy production, but one can note that first derivative of such a function is negative, thus a continuous growth of tank volume is going to give less benefits in terms of electric energy production. Depending on site location, after a certain volume is reached, the energy overproduction increases slightly. This result may be translated in terms of profitability. Significantly larger overproductions can be achieved in Abu Dhabi even though with larger storage volumes compared to Phoenix.

Figure 17 reports the investment pay-back time as a function of thermal storage size. It can be observed that carefully sizing the IC system, PBT may decrease significantly. For example, a storage 25% sizing in New Orleans will result in half the PBT.

Figure 18 shows Net Present Value at 11 years as a function of total capital cost of the investment for all the locations. One can note that in Phoenix, i.e. with hot and dry climate, with a capital cost half of that needed in Abu Dhabi, almost the same NPV can be obtained. Note also that maximum NPV is obtained for all location sites with smaller tank volumes, close to 40% storage sizing, i.e. with a further under sizing with respect to the reference case of 50%. However, a storage volume reduction results in a small penalty in terms of NPV, but in a not negligible gain in terms of PBT.

Such an approach was repeated also for the sizing of the air coils, with the objective to further increase profitability of the investment. The air coil sizing was varied between 30% and 90%. Air coil contribution to the overall cost is lower and always below 22%, therefore their impact on profitability parameters resulted to be smaller if compared to storage volume. An air coil size increase always translates into a PBT increase, whatever the considered site location. NPV variation is quite small, showing a maximum about 50%. An air coil reduction will result in a small or even no penalty in terms of NPV, but also in a negligible gain in terms of PBT, of the order of a month. This leads to the conclusion that a 40% to 50% air coils sizing is always a good choice, independently from the local climatic conditions.

FIGURE 16

FIGURE 17

FIGURE 18

## 6. Conclusions

Stratified chilled water cool thermal storage technology, applied to inlet air cooling of combined cycle power plant has been analyzed and compared for three sites characterized by very different combinations of ambient temperature and relative humidity annual profiles. An in-house code previously developed was used to automatically size all IC system components and to predict the impact of the introduction of IC system in the different climatic conditions. Climatic conditions of site locations proved to be highly influent: thus, in a quite hot climate like Phoenix or Abu Dhabi it is possible to obtain almost the double the gross electric energy overproduction than in New Orleans. Relative humidity has a strong impact mainly on the sizing of the storage tank, almost doubling the investment cost in Abu Dhabi and New Orleans. Economical benefits of inlet cooling system strongly depends on design parameters: a sensitivity analysis showed that, by properly sizing the thermal storage, better results can be achieved. Hot and dry climate like that of Phoenix is particularly advantageous for inlet air cooling adoption; net present value of the investment is quite large and pay-back time can be reduced to 1 year, with an appropriate sizing of thermal storage. Abu Dhabi location, even if characterized by the highest NPV values, shows a larger PBT, due to the larger cost of the storage tank. New Orleans gives the worst results when compared to the other sites, even if a proper sizing of the storage tank could result in a PBT of about 2 years. Air coil sizing was instead shown to play a negligible role in the techno-economic analysis. An air coil under sizing of about 40-50% always gives better performance from a techno-economical point of view, whatever the site location. Finally, even the techno-economical analysis presented in this paper refers to three specific site locations, at least the technical results can be extended to any other location worldwide with similar climatic condition. For example, a desert location like Riyadh is expected to give results similar to Phoenix, making inlet air cooling systems with cold thermal storage an attractive solution also for this region.

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**Figure captions**

Fig. 1. CC, cool thermal storage and inlet air cooling system layout.

Fig. 2. Simulation tool structure.

Fig. 3. Ambient temperature and relative humidity yearly distributions.

Fig. 4. Ambient GTs a) Power output, b) efficiency, c) exhaust gas mass flow rate and d) exhaust gas temperature vs. ambient temperature.

Fig. 5. Average electric energy price.

Fig. 6. Selected ambient a) temperature and b) R.H. profiles (January: plain symbols - July - hollow symbols).

Fig. 7. GT inlet air temperature.

Fig. 8. Chilled water consumption.

Fig. 9. Condensed water.

Fig. 10. CC a) power output and b) power augmentation.

Fig. 11. CC efficiency variation.

Fig. 12. CC Power augmentation versus ambient temperature.

Fig. 13. CC incremental efficiency versus ambient temperature.

Fig. 14. Net electric energy overproduction.

Fig. 15. CC incremental efficiency.

Fig. 16. Net electric energy overproduction as a function of storage volume.

Fig. 17. Pay-back time of the investment as a function of storage volume.

Fig. 18. Net Present Value as a function of total capital cost.