Thermo-economic analysis of inlet air cooling in gas turbine plants

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Gas turbine power plants are widely used for power generation in the world; it has low cost, low time of installation, and stable with electricity grid variations. But it is very affected by ambient temperature, in hot days power demand is increased while gas turbine power decrease with percent 18% when ambient temperature increase to 40°C because air density is lower and compressor specific work increases. Inlet cooling methods used to cool inlet air to boost the power loss in hot days. Chiller cooling and evaporative cooling is studied thermally and economically for 264 MW gas turbine plant located at Korymat – South of Egypt, the results indicates that the gas turbine annual power gained by cooling by chiller is 117027 MWh, and the net cash flow is 3787537 $, while the gas turbine annual power gained by cooling by evaporative is 86118 MW, and the net cash flow is 4503548 $

Keywords: gas turbine, inlet cooling, power enhancement

INTRODUCTION

Gas turbine plants are used for electricity production in many countries around the world because it has low capital cost short synchronization time which it is 30 minutes (time required for gas turbine to reach the base load from zero speed), stability with electricity grid, and due to gas availability in many countries like Egypt. Total electricity generated by gas turbine plants in Egypt about 7001 MW (Egyptian Electricity Holding Company (2009/2010). from different gas turbine model and capacities varying from 25 MW to 260 MW in hot days in summer the ambient temperature reaches to 40 °C, the gas turbine power output decreases by 18 % from rated capacity, this lead to total power lost from gas turbine plants about 1440 MW while the electric load increases to maximum due to conditioning and ventilation.

Therefore it is necessary to enhance the gas turbine power output which can be achieved by cooling the inlet air. Mainly there are two inlets cooling types

(i) Evaporative or fogging cooling.
(ii) Chiller cooling electrical or absorption.

Evaporative systems cool the inlet air by pulverizing water into the air stream. The water evaporation causes the air temperature to decrease; Locations of low humidity climate are suitable to use this cooling technology. Two considerations must be taken into account. Firstly, the maximum relative humidity that it is possible to reach with an evaporative system is hardly over 90%. Secondly, the difference between wet and dry bulb temperatures in the outer section of the evaporative system is recommended not to be under 1°C.

In chiller system, two working fluids will be used, first on is refrigerant in the refrigeration machine which consists of compressor, evaporator, condenser, expansion devise the refrigerant used in this cycle to cool secondary fluid (usually water) the chilled water temperature exit then pumped through (air-water) heat exchanger located at gas turbine inlet to cool coming air to the compressor. The chiller system has advantages where the air temperature can be cooled to lower degrees reach to 5 °C but it has very high capital cost.

The first application of combustion turbine inlet air cooling (CTIAC) was a direct air conditioning system for a plant in Battle Creek, Michigan (USA) in 1987-88, and the second was an off-peak ice harvester system in Lincoln, Nebraska (USA) in 1992 (McCracken, 1994). Hall et al.,
1994, documented the performance of a 36 MW CT plant in which a chilled water-based storage refrigeration system issued to cool the inlet air. The cooling system was able to reduce the air from an ambient temperature of 35 °C to 7 °C, thus enhancing plant performance by 10%. Zamzam and Al-Amiri, 2002 examined the potential use of employing refrigerated CTIAC systems in the United Arab Emirates. They used wet-bulb and dry-bulb weather data to determine characteristic design conditions of three Emirates: Al-Ain, inland arid, very hot and relatively dry; Abu Dhabi, coastal Arabian Gulf, very hot and humid; and Fujairah, coastal Oman Gulf, hot and very humid. For given inlet air temperatures they determined annual gross energy increase, average heat-rate reduction, cooling load requirement, and net power increase. For viability, they recommended an inlet air temperature of 15–25 °C They recommended evaporative cooling be used where a peak-power increase between 8% and 15% is required at high temperatures, and refrigerated cooling where a sustainable increase of 10-25% is required. De Lucia et al., 1995, concluded that evaporative inlet-cooling could enhance power by 2–4% a year depending on the weather.

**Effect of Ambient Temperature on Gas Turbine Output Power:**

Gas turbine usually designed at ISO conditions which met ambient temperature and relative humidity 15 °C and 60% respectively at this conditions the gas turbine output power and thermal efficiency has 100% rated. But if the ambient temperature increases to 40 °C in summer gas turbine power deceases to 80 % rated Figure.1 indicates the gas turbine output power and thermal efficiency versus ambient temperature for 264 MW gas turbine.

**THERMO DYNAMIC MODELING OF GAS TURBINE**

Basically, the gas turbine power plants consist of four components including the compressor, combustion chamber (CC), turbine and generator. A schematic diagram for a simple gas turbine is shown in figure. 2. The fresh atmospheric air is drawn into the circuit continuously and energy is added by the combustion of the fuel in the working fluid itself. The products of combustion are expanded through the turbine which produces the work and finally discharges to the atmosphere.

It is assumed that the compressor efficiency and the turbine efficiency are represented $\eta_c$ and $\eta_t$ respectively. The ideal and actual processes on the temperature-entropy diagram are represented in black and red line respectively as shown in figure.1. The thermo dynamic modeling of gas turbine with design data solved by the (EES) Engineering Equation Solver to get the calculated parameters for the selected gas turbine. The results of the model are presented in the graphs below.

**KORYMAT GAS TURBINE POWER PLANT**

Korymat 750MW combined cycle power plant consists of two gas turbine units 2*264 MW and steam turbine with 250 MW, manufactured by Siemens V94.3A2, it were be installed and commissioned on 2008, the units fuel operated is natural gas as main fuel and liquid fuel as
second fuel, there are two steam turbines 2*627 MW commissioned at 1996. The turbine data shown below in Table 1. Design of inlet cooling systems is depend on the ambient conditions of the site; the cooling system designed for hot humid sites differ for another one designed for hot and dry so that the weather conditions should be studied to determine the weather profile. In the present studying Korymat power plant South of Cairo have been considered, the temperature profile for year 2009 presented to get the average temperature every month. Figure 2 and 3 indicate the temperature and relative humidity profiles during one day per month. (Weather underground conditions, 2009). Figure 2 indicates the hourly ambient temperature through one day per month over one year, the maximum temperature in July is 35°C, the minimum in Jan. is 11°C, and the average is 23°C. Figure 3 indicates the relative humidity variation over the year as per day every month, the maximum, minimum and average are: 82%, 14% and 48% respectively.

**CHILLER SYSTEM INLET COOLING**

In a chilled water system, water is first cooled in the water chiller. It is then pumped to the water cooling coils in terminals in which air is cooled and dehumidified after flowing through the coils, the chilled water increases in temperature up (15.6 to 18.3°C) and then returns to the chiller. Chiller operation is based upon the refrigerating cycle and understanding this cycle is necessary in the refrigerating cycle, air is passing over the cooling coils raises the water temperature which is circulated through the evaporator. The air passes through the chiller coils, raises the temperature of the liquid refrigerant to its boiling point and evaporates it into a gas.

Flow diagram of the chiller system is presented in Figure 4, the system consists of chiller package unit with design capacity, chilled water pumps is used to circulate...
the chilled water between chiller unit and cooling coil. Service water pumps are used to supply the River Nile water to chiller condenser to cool the refrigerant in the refrigeration cycle. Main compact heat exchanger (water – gas) is used to heat exchange between hot air and chilled water, this heat exchanger located in gas turbine air intake and after air filters modules.

**Chiller Sizing Optimization and Calculations:**

It is observed that the best efficiency is obtained closer to gas turbine design temperature ($15\text{-}18^\circ\text{C}$) rather than for the lowest inlet air temperatures ($5\text{-}7^\circ\text{C}$). Obviously, a lower cooler size is required to obtain $15^\circ\text{C}$ than $7^\circ\text{C}$. In addition, cash flows are higher with small coolers than with high-sized ones (both mechanical or heat absorption), due to the efficiency improvement and maintenance reduction. If the economic situation changes to a favorable electricity production using natural gas as fuel, the high-power chillers (sized to maximize improvement) will have better economic behavior than the lower-power ones. However, lower-power coolers in a better economic trend will increase the cash flow and make better economic parameters than expected. In consequence, when energy markets are uncertain, it is recommended to size the cooling system to approach design point rather than to obtain the maximum power enhancements (Gareta et al).

In Figure 5, ambient air at inlet conditions ($T_1=40^\circ\text{C}$, $RH=35\%$) enter to the coil the cooling process is $1\rightarrow 1\rightarrow 2$ while air temperature is reduced till reach to the dew point temperature, the condensation is started till air outlet conditions reach to $T_2=15^\circ\text{C}$, $RH=100\%$. 

![Figure 3. Hourly relative humidity through year](image)

![Figure 4. Chilling cooling flow diagram](image)

![Figure 5. Chilling cooling process](image)
EVAPORATIVE COOLING SYSTEM

System Description:

Evaporative coolers are used in gas turbine intake air coolers and residential applications. The Fig. 6 shows a schematic arrangement of the main parts of a media evaporative cooling system. Water is sprayed upward from a header pipe into the top of an inverted half pipe and is deflected downward onto a distribution pad on top of the media. The distribution pad facilitates the water entry to the media by gravitational force and wets a very large surface area formed by several layers of the media. The excess water (not absorbed by the media and not evaporated by the hot air) is collected in the bottom part of the cooler and forwarded to a reservoir tank. The hot air passes through the media and evaporates the water up to the saturation point before entering the compressor. In this process, the temperature of the air drops by both sensible heat transfer because of the temperature differences between the water and air and the latent heat of evaporation (Energy, 2009).

Figure 7 explains the evaporative cooling process, while the air inlet conditions at 40°C and RH 35% when air passing through cooling media which already wetted with water the air relative humidity increases so the ambient dry bulb temperature decreases with constant wet bulb temperature until ambient relative humidity reaches to 90%. Line 1-2 on psychometric chart represent the cooling process the cooling temperature is depend on the air relative humidity where outlet temperature expressed by:

\[ T_{a2} = T_{a1} - N(T_{a1} - T_{wb}) \, ^\circ C \]  (4)

The cooler effectiveness is selected to be 90%. Gas turbine output power is calculated at cooling temperature to estimate the annual gas turbine power gained by cooling according to average ambient conditions the difference between power with and without cooling calculated

\[ \Delta P = P_2 - P_1 \, (kW) \]  (5)

The heat addition at new cooling temperature calculated as

\[ Q_{add2} = \dot{m}_r \ast (C \cdot V) \, kW \]  (6)

Gas turbine thermal efficiency getting from:

\[ \eta_2 = \frac{P_2}{Q_{add2}} \, \% \]  (7)

Water vaporization flow rate expressed by relation

\[ R = \frac{V(w_2 - w_1) \rho_a}{p_w} \, m^3/s \]  (8)
the blow down rate can be determined from the blow down ratio which equal to:

$$E = \frac{\text{the blow down rate}}{\text{evaporation rate}} = \frac{B}{R}$$  \hspace{1cm} (9)

This ratio will be calculated from water hardness- E curve the water hardness equal to 150 ppm as used in this purposes (Energy, 2002). the ratio (E) equal to 4 so the blow down rate given by:

$$B = 4 \times R$$  \hspace{1cm} (10)

Total water consumption ($Q_w$) equal to the sum of the evaporation rate and blow down rate:

$$Q_w = B + R \text{ (m}^3/\text{S)}$$  \hspace{1cm} (11)

**RESULTS**

In order to establish a systematic comparison between the effects of the two coolers, the performance of the gas turbine unit is examined for a restricted set of operational and design conditions of an operating GT unit taking into account real climatic circumstances prevailed during 2009 at Korymat, South of Cairo, Egypt. The power plant performance characterized by the plant efficiency and net power output, as well as water mass flow rate in the case of evaporative cooling, are estimated based on actual values of given variables, i.e. temperature, relative humidity and gas turbine engine characteristics. Fig.8 shows the effect of variation of chiller cooling load gas turbine output power without and with cooling versus months the power output increased by cooling during January and February months slightly by 1.21 % and 0.448 % because the air temperature about 17 °C, the difference between ambient temperature and target cooling temperature (15°C) is small so that the net power output which is equal the difference between power output with and without cooling is small.

During hot months in the year March, April, till September the net power output increase by cooling the percentage of power increasing gradually to reach the maximum in August reach about (12.36%) the extra power due to using the chiller is 117,027 MW in this year the average increase in power output is 5.66%.

The gas turbine thermal efficiency after using the chiller system is increased in cold months as indicated in Figure.9, the maximum efficiency in December month about 0.51% where the weather is cold and the cooling system will be efficient more in this season the ambient temperature around 20 °C the percent increase in fuel is small, the heat addition is less so that efficiency increase.

In months, May, June, July, August, and September the efficiency will be slightly decrease with using chiller system as shown in Fig.9 the average decrease in efficiency about -0.5%. By using evaporative cooling to cool gas turbine inlet air the gas turbine power output will be increased for all months based on the selected weather profile for Korymat site but the percent increase varying from month to another depend on the climate (temperature, relative humidity).

As indicated in Figure. 10 the power increases by using evaporative cooling in February month with percent 2.3 % which is smallest percent the maximum percent increase in August 12.07 % this is because the cooling effect ($Ta1-TWb$,2) in August is larger than February. The average
percent increase about 6.84 % the total power gained by using evaporative cooler in this year is 86118 MW , when evaporative cooler is available through 60% along the year (5256 hours operation under selected site conditions).

The gas turbine efficiency with using evaporative inlet cooling increases by cooling with evaporative cooling as indicated in Figure.11 the maximum percent increases in August give 2.4% but the minimum increasing in December which give 0.79% this also depend on the cooling effect, the average increase is 1.54% the efficiency give a good indication for specific fuel consumption. Figure.12 indicates the total annual cost for chiller and evaporative cooler system it is noted that the annual cost for chiller is higher than evaporative cooler, because the electricity consumption for chiller very high and the capital cost also higher. Figure.13 indicate the net cash flow from electricity sales when gas turbine integrated with two cooling systems, it is indicated that the net cash flow is higher in case of cooling by evaporative cooler because the total annual cost of chiller is higher due to electric power consumption by chiller.

**CONCLUSION**

From the present studying for gas turbine plant with 264 MW located at Korymat power plant –South East of Egypt for selected whether data and after studying the effect of two inlet cooling technologies to cool inlet air for enhancement gas turbine power loss at summer peak we can conclude that the total yearly gas turbine output power gained due to cooling by chiller system is 117027 MWh, the annual cost for is 7,624,548.9 $, the net cash flow from the plant is 3,787,587 $ and the payback period is 3.3 years. While the yearly gas turbine power gained due to using evaporative cooling is 86118 MWh, the total annual cost is 1,524,779.7 $, the net cash flow is
for gas turbine, it’s more economical, low maintenance, low electricity consumption, River Nile water availability, and low capital cost.

Other cooling technologies may be studied for gas turbines, such that absorption cooling, fogging system and compressor inter cooling.

REFERENCES


APPENDIX

After modeling the unit the gas turbine network output as relation of ambient temperature given by:

\[ W_{net} = 294002 - 1858.9 \times T_1 + 5.2969 \times T_1^2 \text{ (kW)} \]

Thermal efficiency of the plant given by

\[ \eta_{th} = 0.37435 - 0.000279 \times T_1 - 5.95311 \times 10^{-7} \times T_1^2 \]

Air mass flow rate expressed as:

\[ \dot{m}_a = 671.522 - 2.4539 \times T_1 + 0.008653 \times T_1^2 - 0.0000226 \times T_1^3 \text{ (kg/s)} \]

Fuel mass flow rate

\[ \dot{m}_f = 14.9227 - .05453 \times T_1 + 0.00019 \times T_1^2 \text{ (kg/s)} \]

The heat addition:

\[ Q = 782550 - 4304.69 \times T_1 + 1.7380 \times T_1^2 \text{ (kJ/kg)} \]

NOMENCLATURE

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
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<tbody>
<tr>
<td>CCL</td>
<td>Chiller Cooling Load(kW)</td>
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<tr>
<td>h1</td>
<td>Ambient Air Enthalpy At Cooler Inlet(kJ/kg.K)</td>
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<tr>
<td>h2</td>
<td>Ambient Air Enthalpy At Cooler Outlet (kJ/kg.K)</td>
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<tr>
<td>( \dot{m}_{h1} )</td>
<td>Air Mass Flow Rate At Inlet Of Cooler (kg/s)</td>
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<td>N</td>
<td>Evaporative Cooler Efficiency (%)</td>
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<td>P</td>
<td>Gas Turbine Power Output (MW)</td>
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<tr>
<td>P2</td>
<td>Gas Turbine Power Output With Cooling (MW)</td>
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<td>( \Delta P )</td>
<td>Difference Between Gas Turbine Output Without And With Cooling (MW)</td>
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<td>Q_{add}</td>
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<td>Fuel Mass Flow Rate Without Cooling (kg/s)</td>
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<td>LHV</td>
<td>Lower Heating Value For Natural Gas(kJ/kg)</td>
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<td>Q_{h}</td>
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<td>Wet Bulb Temperature Of Outlet Air(°C)</td>
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<td>E</td>
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<td>Humidity Ratio Of Air At Cooler Outlet(gm/kg)</td>
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