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Thermo-Economic Analysis of Inlet Air Cooling In Gas Turbine Plants

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Abstract

Gas turbine power plants are widely used for power generation in the world; they are low cost, quick to install and engender stability with regard to electricity grid variations. They are nevertheless negatively impacted by ambient temperature: on hot days power demand increases while gas turbine power falls. An 18% decrease in efficiency occurs at ambient temperature 40°C due to lower air density and the resulting increase in compressor specific work.

Inlet cooling methods are used to cool inlet air to boost the power loss on hot days. In this paper chiller cooling and evaporative cooling are studied thermally and economically for a 264 MW gas turbine plant located at Korymat, southern Egypt. The results indicate that the annual power gained by chiller cooling is 117,027 MWh, and the net cash flow is $3,787,537 while the annual power gained by evaporative cooling is 86,118 MW, and the net cash flow is $4,503,548.

Keywords: Gas turbine, Inlet cooling, Power enhancement

1. Introduction

Gas turbine plants are used for electricity production in many countries around the world because of their low capital cost, short synchronization time of 30 minutes (time required for gas turbine to reach the base load from zero speed), stability with regard to the electricity grid and availability in many countries, including in this case Egypt. Total electricity generated by gas turbine plants in Egypt is about 7001 MW [1] from different gas turbine models and capacities varying from 25 MW to 260 MW. On hot days in summer, the ambient temperature reaches 40°C and the gas turbine power output decreases by 18% from rated capacity, leading to total power lost from gas turbine plants of about 1440 MW while the electric load increases to maximum due to air conditioning and ventilation.

It is therefore necessary to enhance the achievable gas turbine power output by cooling the inlet air. There are two main inlet cooling types: (i) evaporative or fogging cooling, and (ii) chiller cooling—electrical or absorption.

Evaporative systems cool the inlet air by evaporating water into the air stream. The water evaporation causes the air temperature to drop. Low humidity climates are suitable for use of 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 just over 90%. Secondly, the difference between wet and dry

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bulb temperatures in the outer section of the evaporative system is recommended not to be under 1°C.

Two working fluids are used in the chiller system, the first one is the refrigerant in the refrigeration machine which consists of a compressor,evaporator, condenser and expansion device. The refrigerant is used in this cycle to cool secondary fluid, usually chilled water, which is pumped through an air-water heat exchanger located at the gas turbine inlet to cool air coming into the compressor.

The chiller system has the advantage of being able to reduce the air temperature to 5°C, but it involves 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, the second was an off-peak ice harvester system in Lincoln, Nebraska (USA) in 1992 [2].

Hall et al. [3] documented the performance of a 36 MW CT plant in which a chilled water-based storage refrigeration system was tasked with cooling inlet air. The cooling system was able to reduce the air temperature from an ambient 35°C to 7°C, thus enhancing plant performance by 10%.

Zamzam and Al-Amiri [4] examined the potential use of employing CTIAC refrigeration 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 that evaporative cooling be used where a peak-power increase between 8% and 15% is required at high temperatures, and refrigeration cooling where a sustainable increase of 10–25% is required.

De Lucia et al. [5] concluded that evaporative inlet-cooling could enhance power by 2–4% a year depending on the weather.

An exergy,economic and environment analysis was done [6] for a 4900 kW absorption chiller integrated with a 159 MW gas turbine unit located at Bushr-Iran. The analysis shows that the gas turbine's power increases from 137 MW to 153 MW during the hottest month (August) when the inlet air was cooled from 37°C to 15°C. Moreover, efficiency rose from 33.4% to 43.2%

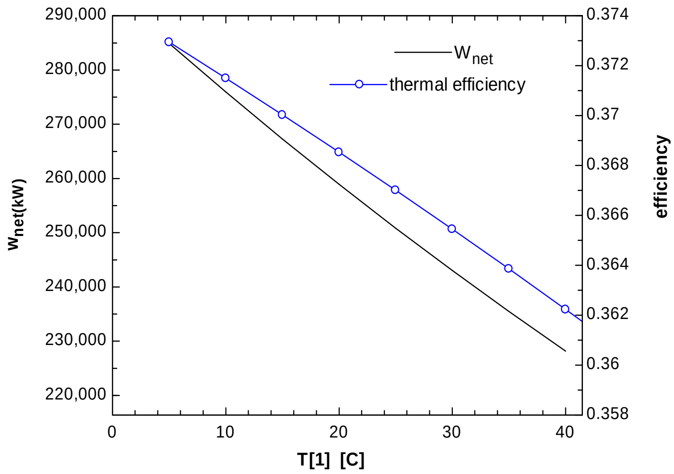

Figure 1: Gas turbine work and efficiency versus ambient temperature

The present study focuses mainly on the followings items, which are the specific contribution of the current paper in this subject:

- Thermodynamic modeling of the Korymat gas turbine power plant in Egypt.
- Chilled water system analysis, including capacity calculation.
- Evaporative cooling analysis.
- Gas turbine performance parameters with and without cooling.
- Economic analysis for two cooling systems based on site conditions.

1.1. *Effect of Ambient Temperature on Gas Turbine Output Power*

Gas turbine is usually designed at ISO conditions which meet ambient temperature and relative humidity of 15°C and 60% respectively. But if the ambient temperature increases to 40°C in summer, gas turbine power deceases to 80 % rated. Fig. 1 shows gas turbine output power and thermal efficiency versus ambient temperature for a 264 MW gas turbine.

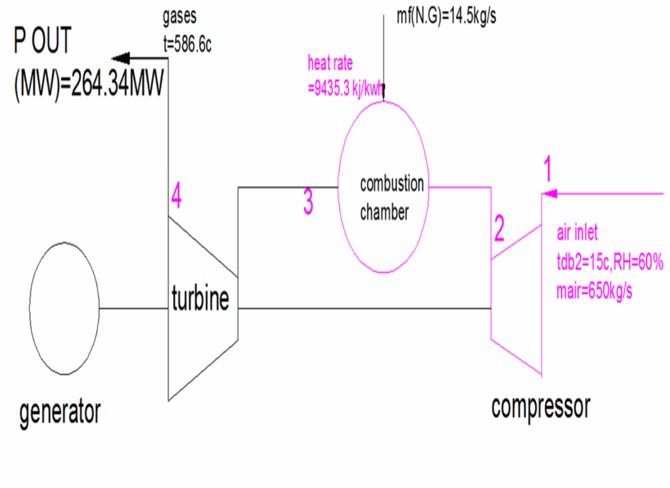

Figure 2: Gas turbine components

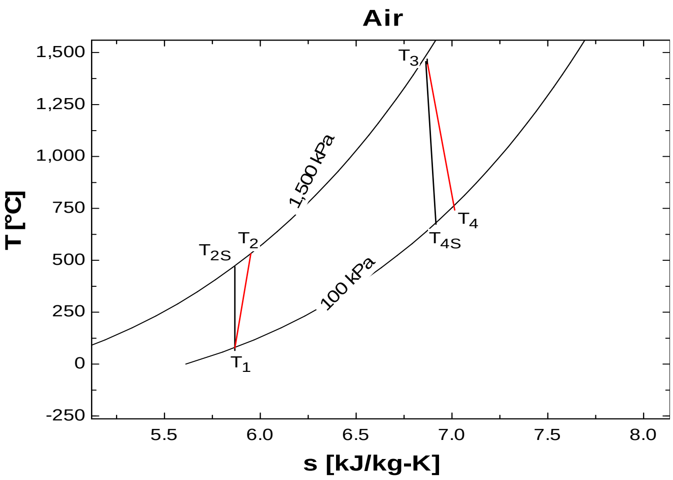

Figure 3: T-S diagram for gas turbine process

2. Thermo Dynamic Modeling of a Gas Turbine [7]

Basically, gas turbine power plants consist of four components including: the compressor, combustion chamber (CC), turbine and generator. A schematic diagram of a simple gas turbine is shown in Fig. 2. Fresh atmospheric air is drawn into the circuit continuously and energy is added by 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 compressor efficiency and turbine efficiency are represented as η_c and η_t respectively. The ideal and actual processes on the temperature-entropy diagram are represented in black and red lines respectively, as shown in Fig. 3.

The compressor compression ratio is given by

$$Rp = \frac{p_2}{p_1} \tag{1}$$

Compressor isentropic efficiency in the range of 80% to 90% is expressed as:

$$\eta_c = \frac{T_{2S} - T_1}{T_2 - T_1} \tag{2}$$

So compressor exit temperature (T_2) is given by:

$$T_2 = T_1 \cdot \left(1 + Rp^{\frac{\gamma_a - 1}{\gamma_a}}\right) \tag{3}$$

Compressor work is obtained from:

$$W_c = C_{pa} \cdot (T_2 - T_1) \tag{4}$$

or

$$W_c = \frac{C_{pa} \cdot T_1 \left(Rp^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right)}{\eta_m \cdot \eta_c} \tag{5}$$

Combustion chamber analysis:

By applying energy balance on the combustion chamber:

$$\dot{m}_a \cdot C_{pa} \cdot T_2 + \dot{m}_f \cdot LHV = \left(\dot{m}_a + \dot{m}_f\right) \cdot C_{pg} \cdot T_3 \tag{6}$$

The air to fuel ratio is expressed as

$$F = \frac{\dot{m}_a}{\dot{m}_f} \tag{7}$$

The exhaust temperature of turbine (T_4) is obtained from the relation:

$$T_4 = T_3 \cdot \left[1 - \eta_t \cdot \left(1 - \frac{1}{RP^{\frac{\gamma_g - 1}{\gamma_g}}}\right)\right] \tag{8}$$

$$T_4 = T_3 \cdot (1 - \eta_t) \cdot RP_g \tag{9}$$

Where $RP_g = \left(1 - \frac{1}{RP^{\frac{\gamma_g - 1}{\gamma_g}}}\right)$

Turbine shaft work is given by:

$$\begin{aligned} W_t &= C_{pg} \cdot (T_3 - T_4) \\ W_t &= C_{pg} \cdot T_3 \cdot \eta_t \cdot \frac{RP_g}{\eta_m} \end{aligned} \tag{10}$$

The net work output from the gas turbine is obtained as:

$$W_{net} = W_t - W_c \tag{11}$$

The output power from gas turbine unit is expressed as:

$$P\,(MW) = \left(\dot{m}_a + \dot{m}_f\right) \cdot W_{net} \tag{12}$$

The specific fuel consumption is given as:

$$sfc = \frac{360^\circ}{W_{net}} \tag{13}$$

The heat added to the combustion chamber is calculated from:

$$Q_{add} = C_{pg} \cdot (T_3 - T_2) \tag{14}$$

or

$$Q_{1add} = C_{1pg} \cdot \left[T_{13} - T_1\,(1) \cdot \left(1 + RP_g\right)\right] \tag{15}$$

Gas turbine thermal efficiency is obtained from:

$$\eta_{th} = \frac{W_{net}}{Q_{add}} \tag{16}$$

The unit heat rate is expressed as:

$$HR = \frac{360^\circ}{\eta_{th}} \tag{17}$$

The thermodynamic modeling of the gas turbine with design data was solved by the (EES) Engineering Equation Solver to get the calculated parameters for the selected gas turbine.

3. Korymat Gas Turbine Power Plant

The Korymat 750 MW combined cycle power plant consists of two gas turbine units 2×264 MW and a steam turbine with 250 MW, manufactured by Siemens (V94.3A2), and installed and commissioned in 2008. The units' operating fuels are: (i) natural gas as main fuel, the specification listed in Table 2, and (ii) liquid fuel as secondary fuel. There are two steam turbines 2×627 MW commissioned in 1996, and the turbine data are shown in Table 1. The design of the inlet cooling systems depends on the ambient conditions of the site. A cooling system designed for hot humid sites differs from one designed for hot and dry sites, so the weather conditions should be studied to determine the weather profile. In the present study Korymat power plant south of Cairo was considered, and the temperature profile for 2009 presented to get the average temperature every month. Fig. 2 and Fig. 3 show the temperature and relative humidity profiles during one day per month [9]. Fig. 4 shows 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.

Table 1: Gas turbine design data

Item	Rate
Gas turbine output, MW	264.344
Air inlet temperature (ISO), °C	15
Relative humidity, %	60
Average air mass flow rate, kg/s	650
Ambient pressure, bar	1.013
Exhaust gases temperature, °C	586.2
Exhaust gases flow rate, kg/s	666
Heat rate, kJ/kWh	9435.4
Gas lower heating value, kJ/kg	47040
Compression ratio	15
Inlet temperature to turbine, °C	1350
Fuel gas mass flow rate, kg/s	14.59
Efficiency, %	37

Table 2: Natural Gas specification [8]

CH_4	C_2H_6	C_3H_8	C_4H_{10}	CO_2	N_2
88.1%	6.55%	2.74%	0.2%	1.89%	0.21%

Fig. 5 shows the relative humidity variation over the year as per day every month, the maximum, minimum and average are: 82%, 14% and 48% respectively.

4. Chiller System Inlet Cooling

In a chilled water system, water is first cooled in the water chiller then pumped to the water cooling coils in terminals in which air is cooled and dehumidified after flowing through the coils, the chilled wa-

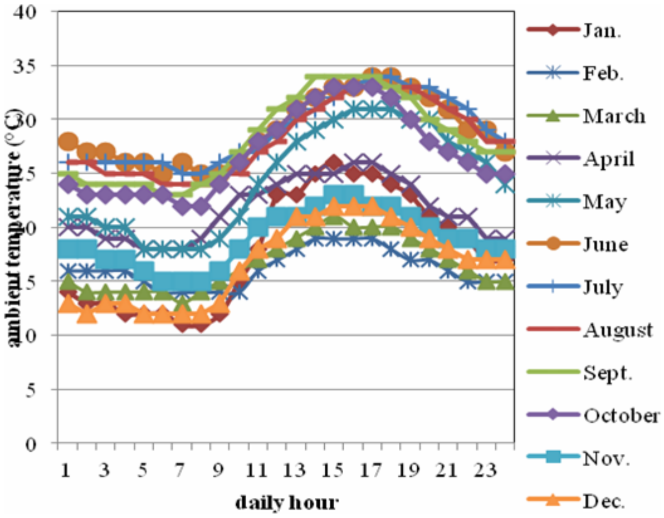

Figure 4: Hourly ambient temperature through the year

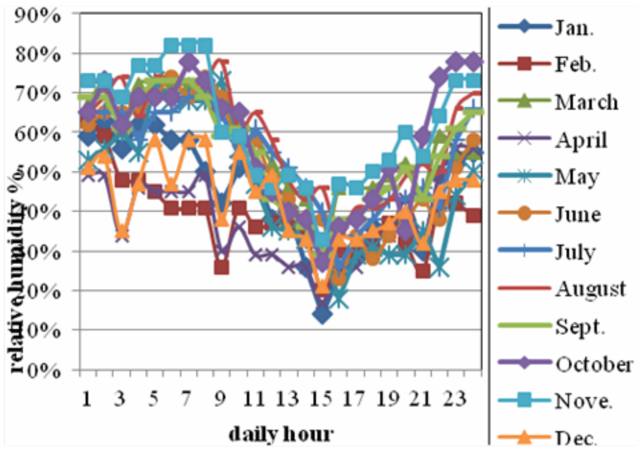

Figure 5: Hourly relative humidity through year

ter increases in temperature up to 15.6 to 18.3°C and then returns to the chiller. Chiller operation is based on the refrigeration cycle and understanding this cycle is necessary in the refrigeration cycle, in which air 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.

A flow diagram of the chiller system is presented in Fig. 4. The system consists of a chiller package unit with design capacity and chilled water pumps are used to circulate the chilled water between the chiller unit and cooling coil. Service water pumps are used to supply water from the River Nile to the chiller condenser to cool the refrigerant in the refrigeration cycle. The main compact heat exchanger (water–gas) is used for the heat exchange between hot air and chilled water, this heat exchanger being located in the gas turbine air intake and after air filter modules.

4.1. Chiller Sizing Optimization and Calculations

It is observed that the best efficiency is obtained closer to the gas turbine design temperature (15–18°C) rather than the lowest inlet air temperatures (5–7°C). Obviously, a lower size of cooler is required to obtain 15°C than 7°C. In addition, cash flows are higher with small coolers than with large ones (both mechanical or heat absorption), due to the improved efficiency and reduced maintenance. If the economic situation changes to one favorable to electricity production using natural gas as fuel, high-power chillers (sized to maximize improvement) will exhibit better economic behavior than lower-power ones. However, lower-power coolers in a better economic trend will increase the cash flow and enjoy better economic parameters than expected. Consequently, when energy markets are uncertain, it is recommended to size the cooling system to approach design point rather than to obtain maximum power enhancements [10].

In Fig. 6 ambient air at inlet conditions (T_1 = 40°C,RH = 35%) enters the coil and the cooling process is 1–1'–2, while air temperature is reduced to the dew point temperature, condensation is started and maintained until air outlet conditions reach T_2 = 15°C, RH = 100%. The coil cooling load is expressed as [11];

$$CCL = \dot{m}_a \left[(h_2 - h_1) - h_{fg} (w_1 - w_2) \right] kW \qquad (18)$$

The auxiliary power consumption by chiller auxiliaries is represented as power consumption by chiller circulation and service water pumps. The chiller parasitic load (power consumption by chiller is calculated as [12]:

$$Q_h = 0.7 \cdot CCL\, kW \qquad (19)$$

The hydraulic power of the centrifugal pump is expressed as:

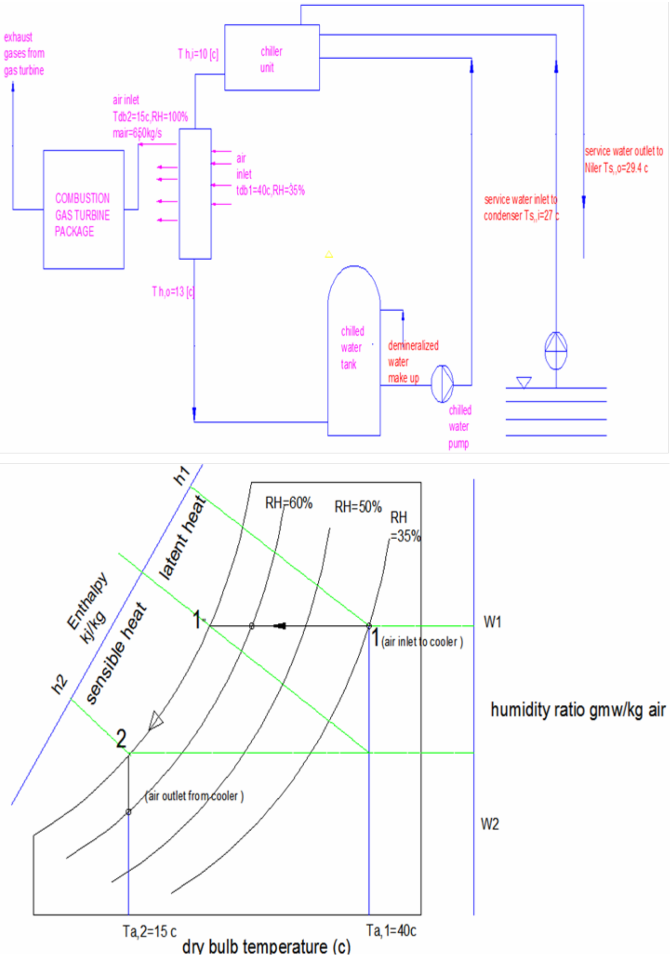

Figure 6: Chilling cooling process

$$P_p = \frac{\gamma Q H}{\eta} W \tag{20}$$

5. Evaporative cooling system

5.1. System Description

Evaporative coolers are used in gas turbine intake air coolers and residential applications Fig. 7 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 entry of the water into 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.

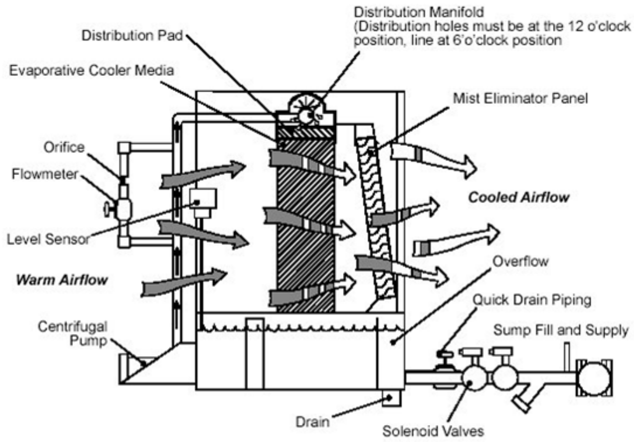

Figure 7: Evaporative cooler section

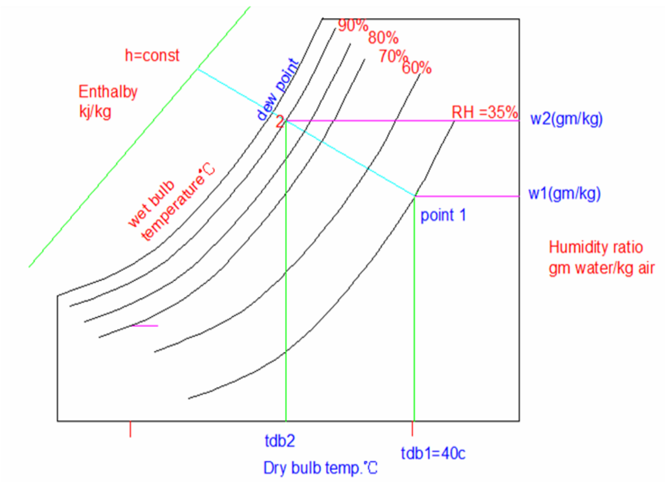

Figure 8: Evaporative cooler actual process

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, which is because of the temperature differences between the water and air, and the latent heat of evaporation [13].

Fig. 8 explains the evaporative cooling process, while the air inlet conditions at 40°Cand RH 35% passing through cooling media which is 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 90%. Line 1–2 on the psychometric chart represents the cooling process, and the cooling temperature depends on the air relative humidity, as the outlet temperature is expressed by:

$$T_{a2} = T_{a1} - N \cdot (T_{a1} - T_{wb}) \ ^\circ C \quad (21)$$

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, also the difference between power with and without cooling is calculated:

$$\Delta P = P_2 - P_1 \, (kW) \quad (22)$$

The water vaporization flow rate is expressed by the relation:

$$R = \frac{V\,(w_2 - w_1)\,\rho_a}{\rho_w} \, m^3/s \quad (23)$$

the blow down rate can be determined from the blow down ratio which is:

$$E = \frac{the\,blow\,down\,rate}{evaporation\,rate} = \frac{B}{R} \quad (24)$$

This ratio will be calculated from the water hardness—E curve. Water hardness of 150 ppm was used for this purpose [14] and the ratio (E) was equal to 4, so the blow down rate is given by

$$B = 4 \cdot R \quad (25)$$

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

$$Q_w = B + R \left(m^3/s\right) \quad (26)$$

6. Economic analysis for inlet cooling systems

The basis of most design decisions is economic. Designing a system that functions properly is only one part of the engineer's task. The system must also be economical and show an adequate return on investment.

The comparative study of chiller system inlet cooling and evaporative cooling will use the Total Annual Cost Method, [15] to gauge the annual cost of alternative designs. The total annual cost generally consists of five terms: capital investment, fuel cost, O&M cost, electricity consumption cost and treated water cost. The relation is expressed by:

$$TAC = C_i \cdot (AFCR) + C_f + C_O + C_E + C_w \quad (27)$$

where: TAC—Total annual cost ($/year), C_f—Annual fuel cost ($/year), C_O—Annual O&M cost ($/year), C_i—Capital Investment ($), C_E—electricity consumption cost by system ($/year), ($AFCR$)—Annual Fixed Charge Rate, C_w—Treated water cost ($/year).

The chiller capital investment is expressed as per every incremental power increase as 500 $/kWh [16] while it is 50 $/kWh for evaporative cooler. The $AFCR$ factor is in the range of 10.5%–20% so it is selected to be 15%.

The natural gas unit price is taken as about 1.25 mmBTU [17] (as per Egypt's energy market) so the total fuel price is:

$$(C_f) = C_n \cdot Q_g \ \$/year \quad (28)$$

The chiller operation maintenance cost is taken as 6% [18] of capital cost, and 4% for the evaporative cooler since it has smaller components.

The electric power consumption price per year is estimated based on unit price 0.07 $/kWh

$$CE = 0.07 \cdot CCL \ (kW) \quad (29)$$

The net cash flow income per year is

$$(C_{f,net}) = C_{Fi} - TAC \ \$ \quad (30)$$

Income cash flow (C_{Fi}) is expressed as:

$$C_{Fi} = C_e \cdot P \ (\$/year) \quad (31)$$

The Payback Period is the time required to recover the cost of an investment and is calculated as:
$P_B = investment/\,(cash\,flow/year)$

7. 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 the real climatic circumstances that prevailed during 2009 at Korymat, south of Cairo, Egypt. The power plant performance

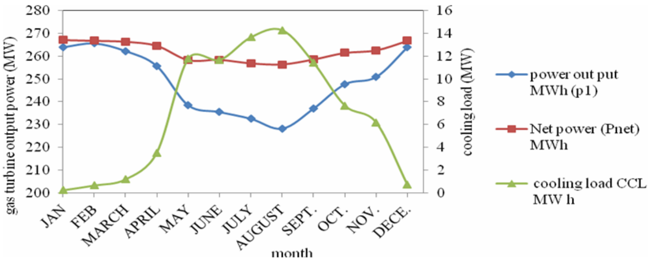

Figure 9: Relation between cooling load and gas turbine power output (MWh) with and without chilling cooling

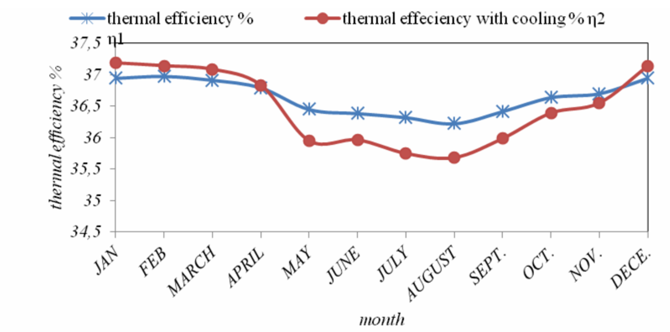

Figure 10: Thermal efficiency of the gas turbine with and without an integrated chiller system

is characterized by the plant efficiency and net power output, as well as water mass flow rate in the case of evaporative cooling, which are estimated based on actual values of given variables, i.e. temperature, relative humidity and gas turbine engine characteristics. Fig. 9 shows the effect of variation of chiller cooling load on gas turbine output power without and with cooling versus months, as the power output increased by cooling during January and February months slightly by 1.21% and 0.448%. Since the air temperature was about 17°C, the difference between ambient temperature and target cooling temperature (15°C) is small, so the net power output, which is equal to the difference between power output with and without cooling, is also small. During hot months in the year—March/April until September—net power output increased by cooling, as the percentage of power increased gradually to reach the maximum in August at about 12.36%, also the extra power due to using the chiller is 117,027 MW in this year, and the average increase in power output is 5.66%.

Gas turbine thermal efficiency after using the

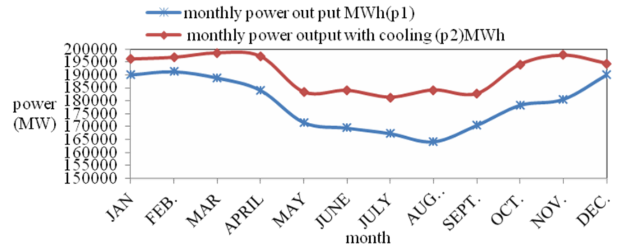

Figure 11: Relation between Power output (P_1) without cooling-power output (P_2) with cooling by evaporative

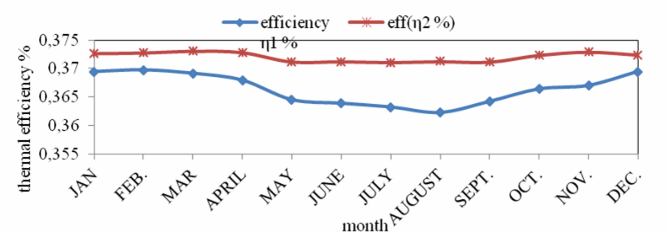

Figure 12: Thermal efficiency without cooling (η_1)-Thermal efficiency with cooling (η_2) by evaporative cooling

chiller system increased in the cold months as shown in Fig. 10. The maximum efficiency in the month of December was about 0.51% as the weather is cold and the cooling system is more efficient in this season. Due to the ambient temperature being around 20°C, the percentage increase in fuel is small and the heat addition is less, resulting in higher efficiency.

In the period May to September efficiency decreased slightly through use of the chiller system. As shown in Fig. 10, the average decrease in efficiency was about -0.5%. By using evaporative cooling, gas turbine power output increased for all months based on the selected weather profile for the Korymat site but the percentage increase varied from month to month depending on the climate (temperature, relative humidity).

As shown in Fig. 11 the power increase through evaporative cooling was at its lowest in February with 2.3% and its highest in August with 12.07%. This is because the cooling effect (T_{a1}–$T_{Wb,2}$) is larger in August than in February. The average percentage increase is about 6.84 %, and the total power gained by using evaporative cooler in this year was 86,118 MW, when an evaporative cooler is available through 60% of the year (5256 hours operation under selected site conditions).

Gas turbine efficiency was increased by using

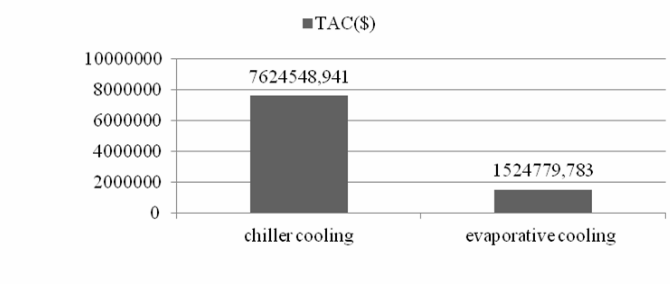

Figure 13: Total annual cost for chiller and evaporative cooling

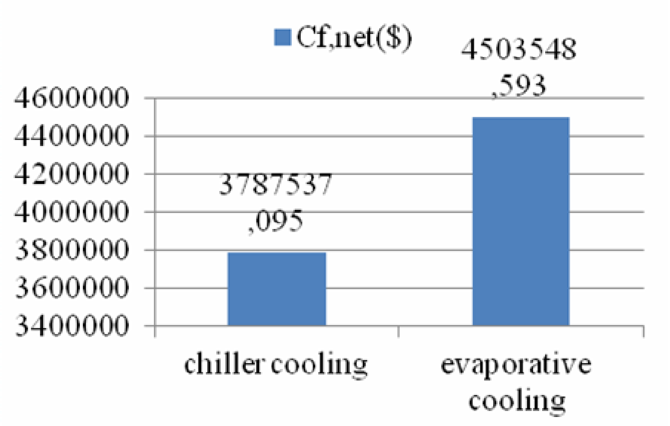

Figure 14: Net cash flow for chiller and evaporative cooling

evaporative inlet cooling as shown in Fig. 12. The maximum percentage increase was in August, at 2.4%, the minimum increase in December, at 0.79%, but this also depends on the cooling effect, as the average increase is 1.54%. Efficiency gives a good indication for specific fuel consumption.

Fig. 13 shows the total annual cost for chiller and evaporative cooler systems. It is noted that the annual cost for the chiller is higher than for the evaporative cooler, because the electricity consumption for the chiller is very high and the capital cost is also higher.

Fig. 14 indicates the net cash flow from electricity sales when the gas turbine is integrated with two cooling systems. It shows that the net cash flow is higher in the case of cooling by an evaporative cooler because the total annual cost of the chiller is higher due to its higher power consumption.

8. Conclusion

From the present study for gas turbine plant with 264 MW located at Korymat power plant, in southeast Egypt for selected weather data and after studying the effect of two inlet cooling technologies to cool inlet air for enhancement of gas turbine power loss at the summer peak, we can conclude that the total yearly gas turbine output power gained due to cooling by the chiller system is 117,027 MWh, the annual cost is $7,624,548.90 and 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 86,118 MWh, the total annual cost is $1,524,779.70, the net cash flow is $4,503,548.50, and the payback period is 0.66 year. Average gas turbine thermal efficiency of the plant in the case of using the chiller was 36.46% while with evaporative cooling it was 37.205%.

The evaporative cooler is effective for cooling inlet air for a gas turbine, it is more economical, has low maintenance and low electricity consumption, water is available from the River Nile and the capital cost is low.

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

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Nomenclature

ΔP Difference Between Gas Turbine Output (Without And With Cooling), MW

$\dot{m}_{a1}$ Air Mass Flow Rate At Inlet Of Cooler, kg/s

$\dot{m}_{f1}$ Fuel Mass Flow Rate Without Cooling, kg/s

η_2 Gas Turbine Efficiency With Cooling,

ρ_a Air Density, kg/m^3

ρ_w Water Density, kg/m^3

B Blow Down Rate, m^3/s

CCL Chiller Cooling Load, kW

E Blow Down Ratio

$h1$ Ambient Air Enthalpy At Cooler Inlet, kJ/(kg·K)

$h2$ Ambient Air Enthalpy At Cooler Outlet, kJ/(kg·K)

LHV Lower Heating Value For Natural Gas, kJ/kg

N Evaporative Cooler Efficiency,

P Gas Turbine Power Out Put, MW

$P2$ Gas Turbine Power Output With Cooling, MW

P_1 Gas Turbine Power Output Without Cooling, MW

Q_{add} Heat Added To Combustion Chamber, kJ/(kg·K)

Q_h Chiller Parasitic Load, kW

R Evaporative Cooler Vaporization Rate

$T_{a,1}$ Ambient Air Temperature At Inlet, °C

$T_{a,2}$ Ambient Air Temperature At Outlet, °C

$T_{wb,2}$ Wet Bulb Temperature Of Outlet Air, °C

V Air Volume Flow Rate, m^3/s

w_1 Humidity Ratio Of Air At Cooler Inlet, gm/kg

w_2 Humidity Ratio Of Air At Cooler Outlet, gm/kg