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Assessment of Power Augmentation from Gas Turbine Power Plants Using Different Inlet Air Cooling Systems

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Abstract

In this paper, the influence of air cooling intake on the gas turbine performance is presented. A comparison between using different cooling systems, i.e., evaporative and cooling coil, is performed. A computer simulation model for the employed systems is developed in order to evaluate the performance of the studied gas turbine unit, at Marka Power Station, Amman, Jordan. The performance characteristics are examined for a set of actual operational parameters including ambient temperature, relative humidity, turbine inlet temperature, pressure ratio, etc. The obtained results showed that the evaporative cooling system is capable of boosting the power and enhancing the efficiency of the studied gas turbine unit in a way much cheaper than cooling coil system due to its high power consumption required to run the vapor-compression refrigeration unit. Nevertheless, it provides full control on the temperature inlet conditions regardless of the relative humidity ratio.

Keywords: Power augmentation; temperature control; evaporative cooling; cooling coil; gas turbine

Abbreviations

CEGCO	Central Electricity Generation Company
GE	General Electric
GT	Gas Turbine
ISO	International Standard Organizations
PLC	Programmable Logic Controller
SPBP	Simple Payback Period
TET	Turbine exit temperature
TIT	Turbine inlet temperature

Symbols

C_{pa}	Specific heat for air (kJ/kg. °C)
C_{pg}	Specific heat of the flue gas (kJ/kg. °C)
COP	Coefficient of performance of the mechanical chiller
DBP	Design back pressure (kPa)
D_f	Diesel fuel High Heating Value (MJ/kg)
Eff	Evaporative cooling effectiveness
f	Fuel to air mass ratio
h	Specific enthalpy
k	Specific heat ratio
$\dot{m}_a$	Air mass flow rate (kg/s)
$\dot{m}_f$	Fuel mass flow rate (kg/s)
$\dot{m}_{tot}$	Total mass flow rate (kg/s)
$\dot{m}_w$	Water mass flow rate (kg/s)
P_{atm}	Atmospheric pressure (kPa)
P_{base}	Power output in base case (kW)
$\Delta P_{combustion}$	Pressure drop in the combustor (kPa)
ΔP_{intake}	Intake pressure drop (kPa)
Q_{coil}	Cooling load on the coil cooler (kW)
Q_{in}	Heat input to the combustor (kW)
r_p	Pressure ratio
T	Temperature (K)
w	Specific humidity (kg water/kg air)
W_{ac}	Power consumed by the coil cooling (kW)
W_C	Compressor work (kW)
W_{net}	Net power produced (kW)
W_t	Turbine work (kW)
ϕ	Relative humidity
$\eta_{combustion}$	Combustion efficiency
$\eta_{compressor}$	Compressor isentropic efficiency
η_{th}	Thermal efficiency
$\eta_{turbine}$	Turbine isentropic efficiency

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1. Introduction

Gas turbines (GT) have been used for electricity generation in most countries around the world. In the past, their use has been generally limited to generating electricity in periods of peak electricity demand. Gas turbines are ideal for this application as they can be started and stopped quickly enabling them to be brought into service as required to meet energy demand peaks. However, due to availability of natural gas at relatively cheap prices compared to distillate fuels, many countries around the world, e.g. Jordan, use large conventional GTs as base load units, while small ones to meet any shortages in available electricity supplies occurring during an emergency or during the peak load demand periods. Such systems, especially those operating in an open or simple cycle have the disadvantage of being least efficient and so the unit cost of generated electricity is relatively high. For example, in Jordan, gas turbines used as peaking units consumed about $35x10^3$ tonnes of diesel fuel, but supplied less than 111 GWh, i.e, 1.3% of electricity generated in 2005 [1]. The average efficiency of GT peaking plants in Jordan over the last five years was in the range of 20-28% [2,3]. Such low efficiencies can be attributed to many reasons, such as, operation mode, poor maintenance, engine size and age. Unit cost of produced electricity and gaseous emissions that would otherwise arises from conventional generators could be reduced by employing a hybrid system that uses a renewable energy source, such as, solar energy [4-6], or by using advanced technologies, such as, regenerative cycle, combined cycle and power augmentation. The latter is achieved by cooling the air at the compressor intake which helps increasing the density of air flowing into the GT plant and thereby increasing the generated power from the engine.

Different geographical regions have different climatic conditions, i.e, ambient temperature and relative humidity ratio. For example, the weather in deserts (hot and dry) is different from that of coastal regions in which humidity is very high. In Jordan, during the summer season, the ambient temperature may reach as high as 37 °C, or even higher in July and August when peak demand occurred due to increasing demand on electrical energy for air-conditioning and ventilation systems. The ambient temperature has a strong influence on the gas turbine performance [7,8]. Generally, unlike the heat rate, the net power output form a gas turbine decreases with the increase of the ambient air temperature. This is due to reduced net power output, which is directly proportional to the air mass flow rate; net power produced decreases when the ambient air temperature increases. Practically, a 25% loss of the rated power capacity of the gas turbine at ISO conditions as the ambient temperature reaches 40 °C is reported [9]. On the other hand, during summer the demand for electrical energy rises up. Therefore, it is necessary to enhance GT power output, which can be achieved through cooling the air just before it enters the gas turbine's compressor. There are two main ways that may be used to cool the inlet air: these include (i) evaporative or fog cooling; and (ii) mechanical cooling and thermal storage. Mechanical cooling may be coil cooling system or absorption chiller cooling [10, 11, 12, 13]. Such techniques can boost the power out by about 30%, when supplying the inlet air temperature at approximately 10 °C.

In related open literature, many researchers studied different cooling methods to enhance the performance of GT plants operating at high conditions of ambient temperature. Johnson [14] presented a discussion of the theory and operation of evaporative coolers for industrial gas turbine installations. Calculations of parameters to predict the performance of evaporative cooler were discussed, in addition to installation, operation, feed water quality and the causes and prevention of water carry-over. Ondryas et al., [15] investigated the impact of using chillers at the air intake to boost gas turbines power in cogeneration plants during high ambient temperatures. Three types of chillers i.e, absorption, vapour-compression and thermal energy storage were studied. Motive energy for the chillers is steam from the GT exhaust for the absorption system and electricity for vapour-compression chillers. Description of the chilled water distribution in the inlet air system was provided and the overall economics of the power augmentation benefits was investigated. It had been reported that air chilling could be effectively used to boost power during high temperatures and the benefit from peak power production could outweigh the needed investment. In another paper by Mercer [16], it was reported that chillers utilizing thermal storage systems would increase the GT power output by 25% during peak periods. Evaporative coolers, on the other hand, give an increase of about 10-15%. Water fogging is another cooling method, which would enhance power out put by 10% to 20% [8, 10]. It was reported by researchers and published information by manufactures of such systems that a gain of 1% in the power output is attainable for every 1.5-2.0 °C drop in the inlet air temperature using water chillers. The performance of the gas turbine using an intake cooler for time varying annual climatic conditions had also been evaluated [17]. Using an evaporative cooler enhanced the plant output by 2-3%, while chilled water coolers would increase the power output by 5-7%, depending on the climatic conditions. Kolp et al., [18] presented an analysis for the effect of various forms of inlet air heating, cooling and supercharging on a 40 MW GE LM6000 gas turbine. It was found that reduction by 28 °C of inlet air temperature would increase power out put by about 30% and reduce the heat rate by approximately 4.5%. De Paepe and Dick [19] presented a technological and economical analysis of different types of condensers for water recovery in steam injected GT. It was shown that injecting steam into the cycle increases its power output and efficiency at relatively low capital costs, i.e. short pay back periods varies from 1.5 to 9.5 years depending on employed system and desired configuration. However, the high rate of water consumption is the major drawback of such systems.

In this paper, the influence of air cooling at the intake of the compressor on the performance of an actual GT engine operating at Marka Power Plant, East of Amman, has been theoretically investigated using prevailed climatic conditions during 2005 [20]. The effect of two air cooling methods, i.e, evaporative and coil cooling systems, were examined. Each of the studied two coolers modifies air temperature and humidity differently and each system has different cooling capacity that would limit the minimum attainable temperature at the compressor inlet. It is not the aim of this paper to discuss other issues related to off-design operation, compressor or turbine design and cooling systems; rather to provide guidance information concerning the proposed cooling systems to deduce so that its overall performance can be compared with the base-case conditions without inlet air-cooling.

2. Theory and performance modelling

In their continuous planning for load growth, electricity utilities search for the most economic generation schemes. But this will be subject to a number of constraints, such as, the type of fuel available, peak-to-base demand ratio and compliance with national environmental standards. To assess the behaviour of a power plant over its expected ranges of operation, appropriate mathematical models which can predict the performance under both design-point and off-design or part-load operating conditions have been developed. In this investigation, the performance analysis of two air cooling systems is discussed and compared with the base-case scenario without inlet air cooling. For the two cooling systems, the main operating variable is the intake temperature at the inlet of compressor, while TIT and TET remain invariant. Figure 1 shows a simple sketch of the studied GT engine. The gas turbine engine consists of a compressor, a combustor and a turbine, in addition to a cooling system. Two types of air cooling systems, i.e., evaporative and cooling coil, are investigated in this study. The evaporative cooling system is taken as water spraying, modelled as an adiabatic saturator, and the used cooling coil resembles a typical refrigeration cycle. The performance of the power plant is examined with each air cooler.

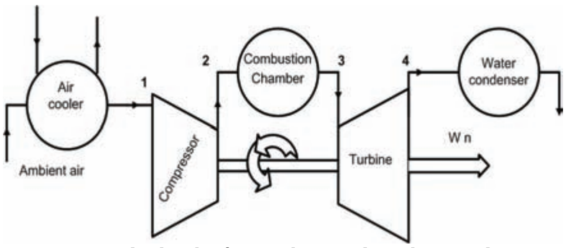
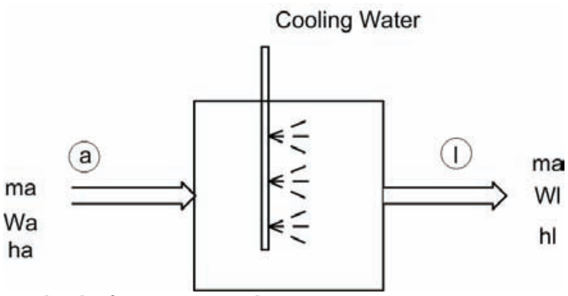


Figure 1: Simple sketch of gas turbine cycle with air cooler

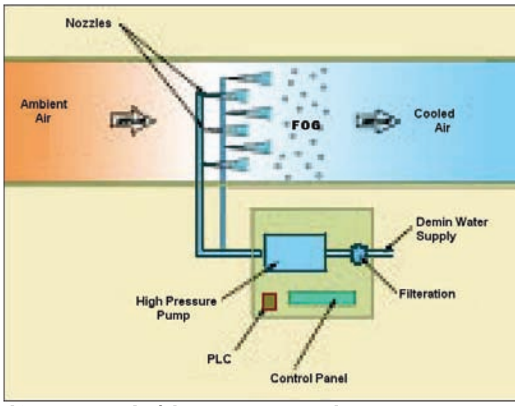
The thermal power analysis of the GT plant is performed taking into consideration the compressor and turbine efficiencies., effectiveness of evaporative cooling unit and coefficient of performance of mechanical chillers. Pressure losses in the compressor intake and combustor and turbine exhaust ducts are also considered as well as the variations of the specific heats of air and combustion products. These calculations were carried out by means of a specially designed computer programme. This model is based also on energy and flow matching of the turbomachinery components, i.e., the components are aerodynamically coupled along the flow satisfying mass continuity. They are coupled by the engine shaft, so energy balance exists. The working fluid passing through the compressor is assumed to be an ideal mixture of air and water vapour, while that passing through the turbine is assumed to be an ideal mixture of combustion gases and all of which are assumed to behave as ideal gases. The main points of calculation procedure for the proposed system, reference to Figure 1, in are presented below.

3. Air Cooling Systems

The water spraying cooler is modelled as an adiabatic saturator that delivers air to the compressor with about 100% relative humidity, i.e., saturated air. The air temperature drops to the wet bulb temperature as a result of passing through the water spraying cooler. The air exit temperature is dictated by both the ambient air temperature and relative humidity. A simple sketch of the spray cooler and control system is shown in Figure 2a.



(a) Sketch of evaporative cooling system



(b) PLC control of the evaporative cooling system

Figure 2: Cooling system

A typical evaporative cooling system consists of a series of high pressure pumps that are mounted on a skid, PLC based control system with temperature and humidity sensors, and an array of fog nozzles installed in the inlet air duct as shown in Figure 2b. Sensors are provided to measure relative humidity and dry bulb temperature. Special programming codes use these measured parameters to compute the ambient wet bulb temperature and the wet bulb depression, i.e. the difference between the dry bulb temperature and the wet bulb temperature. They quantify and control the amount of evaporative cooling that is possible with the ambient conditions. The system turns on or off fog cooling stages to match the ability of the ambient conditions to absorb water vapour. The control system also monitors pump skid operating parameters, such as, water flow rates and operating pressure, and provides alarms when these parameters are outside acceptable ranges.

Ambient air enters the cooler at T_a and ϕ_a. Adding an adequate quantity of water to air stream in the spray cooler raises the air moisture content and decreases its temperature. If the spray cooler is assumed large and insulated, then such cooler resembles the classical adiabatic saturation process. Analysis of this process is available in the thermal engineering literature. Applying mass and energy balances to the cooling system yields:

$$C_{pa}(T_1 - T_a) = \omega_1 h_{fg1} + \omega_a (h_{ga} - h_{l1}) \qquad (1)$$

Where ω_a and ω_1 are the humidity ratios, i.e. the ratio of the mass of water vapour to the mass of air, before and after the cooler, respectively. In general, ω is related to the water vapour pressure at saturation temperature. A simple sketch of the coil cooling is shown in Figure 3. This air cooler operates in a different way than the water spray cooler; however, the temperature and relative humidity of fluid leaving the cooler depend on the coil temperature and relative humidity of ambient air. Ambient air enters the coil cooling at T_a and ϕ_a. Air passing over the outer surface of the coil experiences a drop in temperature and possibly a decrease in specific humidity, ω. The coil temperature can be adjusted to allow air to reach a certain desired temperature. In this case, the cooling load to be removed using cooling coil can be estimated; hence, the power input to the associated refrigeration system can also be evaluated.

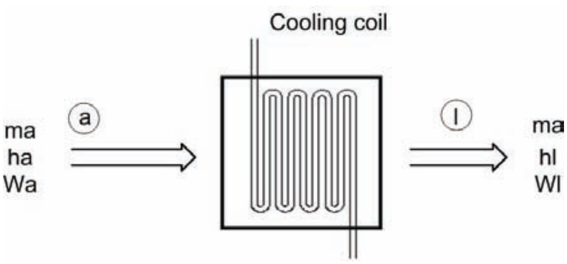


Figure 3: A Typical cooling coil

The cooling load (Q) removed from the air flowing at ambient conditions into the power plant can be estimated using the first law of thermodynamics as follows:

$$Q_{coil} = \dot{m}_a C_{pa}(T_1 - T_a) \qquad (2)$$

Where C_{pa} is the specific heat of the dry air at constant pressure, and T_1 is the temperature at the compressor inlet which should not be lower than 5°C in order to avoid condensation and freezing conditions. Assuming that this load is removed using a typical refrigeration machine having a fixed coefficient of performance, the power needed to operate such machine is estimated from the following relation:

$$W_{ac} = \frac{Q_{coil}}{COP} \qquad (3)$$

Where COP is the coefficient of performance of the cooling coil of the employed chiller. Then, the output net power of gas turbine will be reduced as follows:

$$W_{net} = (W_t - W_c) - W_{ac} \qquad (4)$$

4. Gas turbine unit

In this study, the selected actual engine, GT 6001 B (PG 6541-B) from General Electric Power Systems, with a nominal rating of 20 MW_e, is an open cycle and a single-shaft gas turbine. This engine is owned and operated, at present as a standby and a peaking unit at Marka Power Station in Jordan, by CEGCO and firing diesel fuel. In order to undertake a design-point analysis for the chosen GT engine, a set of available practical data were used, as summarized in Table 1. Such data were taken from the technical manuals of CEGCO and GE [21, 22]. However, a standard gas turbine cycle is considered for the present analysis in order to make it easier for results comparison.

Table 1: Technical specification of selected gas turbine engine

Power plant	Marka Power Station
General electric model	PG5341
Site	Marka, Amman East
Cycle	Single shaft, simple cycle, industrial engine
Pressure ratio	12
Inlet pressure	93 kPa
Air flow rate	70 kg/s
Power output	20 MW
Altitude	735 m
Compressor	Stages: 17 Speed: 5100 rpm Type: axial flow, heavy duty
Turbine	Stages: 2 Speed: 5100 rpm
Inlet turbine temperature	1350 K
Fuel	Light distillate fuel (Diesel)

4.1. Compressor

The intake pressure drop (ΔP_{intake}) is taken to be 1 kPa, the intake temperature is the same as the ambient temperature in the base-case scenario, specific heat ratio for air $\kappa_a = 1.4$, and specific heat for air $C_{pa} = 1.005$ kJ/kg.°C. For a pressure ratio r_p, the pressure of the fluid leaving the compressor can be determined from the following equation:

$$P_2 = r_P P_1 \qquad (5)$$

Where inlet pressure entering the compressor is $P_1 = P_{atm} - \Delta P_{intake}$

The compressor isentropic efficiency can be evaluated using the following empirical equation:

$$\eta_{compressor} = 1 - \frac{[0.09 + (r_p - 1)]}{300} = 0.87 \qquad (6)$$

The isentropic outlet temperature leaving the compressor is determined from the following equation:

$$T_{2s} = T_1 (r_p)^{\frac{k_a - 1}{k_a}} \qquad (7)$$

The isentropic temperature rise is determined from the following equation:

$$T_{sr} = T_{2s} - T_1 \quad (8)$$

The actual temperature rise in the compressor is calculated from the definition of isentropic efficiency:

$$T_{ar} = \frac{T_{sr}}{\eta_s} \quad (9)$$

Then, the actual outlet temperature leaving the compressor is:

$$T_{2a} = T_1 + T_{ar} \quad (10)$$

The actual work consumed by the compressor is given by:

$$W_c = \dot{m}_a C_{pa} T_{ar} \quad (11)$$

4.2. Combustor

At a specific heat of the flue gas $C_{pg} = 1.15$ kJ/kg.°C, outlet temperature from the combustor T_3, combustion efficiency assumed to be $\eta_{combustion} = 0.99$, and a pressure drop ($\Delta P_{combustion}$) in the combustor assumed to be 48 kPa, then, outlet pressure from the combustor is determined from the following equation:

$$P_3 = P_2 - \Delta P_{combustion} \quad (12)$$

Fuel mass flow rate is determined from the following equation:

$$\dot{m}_f = \frac{q_{in}/D_f}{\eta_{combustion}} \quad (13)$$

Where diesel fuel high heating value is assumed to be around 42 MJ/kg and fuel/air ratio (f) is determined from the following equation:

$$f = \frac{\dot{m}_f}{\dot{m}_a} \quad (14)$$

Then, heat input to the combustor can be estimated from energy balance across the combustor:

$$Q_{in} = \dot{m}_a C_{pg} (T_3 - T_{2a}) \quad (15)$$

4.3. Turbine

The power produced by the turbine is determined assuming a specific heat ratio of exhaust gas $k_g = 1.332$, invariable turbine inlet temperature T_3, inlet pressure P_3, and design back pressure $DBP = 1$ kPa. The turbine isentropic efficiency can be evaluated using following empirical equation:

$$\eta_{turbine} = 0.9 - \frac{[(T_3/P_4) - 1]}{250} = 0.86 \quad (16)$$

Outlet pressure at the turbine's exit is:

$$P_4 = P_{atm} + DBP \quad (17)$$

The expansion ratio is determined from the following equation:

$$r_{p_1} = \frac{P_4}{P_3} \quad (18)$$

The isentropic outlet temperature leaving the turbine is determined from the following equation:

$$T_{4s} = T_3 (r_{p1})^{\frac{k_g - 1}{k_g}} \quad (19)$$

The isentropic temperature drop is determined from the following equation:

$$\Delta T_{4s} = T_3 - T_{4s} \quad (20)$$

The actual temperature drop is obtained from the definition of turbine's isentropic efficiency:

$$\Delta T_{4a} = T_{4S} \eta_{turbine} \quad (21)$$

The actual outlet temperature leaving the turbine is determined from the following equation:

$$T_{4a} = T_3 - \Delta T_{4a} \quad (22)$$

The total mass flow rate is given by:

$$\dot{m}_{tot} = \dot{m}_a + \dot{m}_f \quad (23)$$

The work produced from the turbine is determined by the following equation:

$$W_t = \dot{m}_{tot} C_{pg} T_{4a} \quad (24)$$

The power output obtained from the gas turbine power plant is:

$$W_{net} = W_t - W_c \quad (25)$$

The thermal efficiency without cooling intake air before being introduced to the gas turbine unit is:

$$\eta_{th} = \frac{W_{net}}{q_{in}} \quad (26)$$

5. Results and analysis

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 from CEGCO, taking into account real climatic circumstances prevailed during 2005 at Marka, east of Amman, Jordan. 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. The analysis is restricted to the design point performance of GT engine and off-design operation is not considered in this study.

Figure 4 shows the effect of ambient temperature on power output and thermal efficiency of the gas turbine power plant in the standard case.

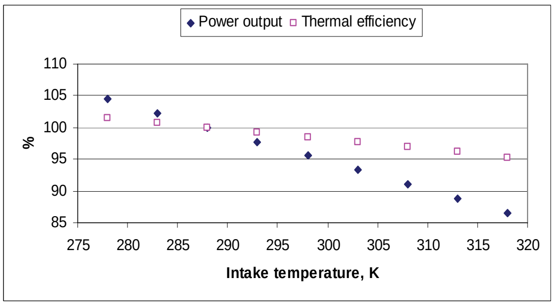


Figure 4: Thermal efficiency and power output versus intake temperature

As can be seen, thermal efficiency decreases as the inlet air temperature increases, where the drop in efficiency is significant at higher temperatures. It is clear that efficiency and power output are inversely related with intake air temperature; higher ambient temperature will result in reduced power output and, consequently, the plant's thermal efficiency shrinks. But it should be noted that temperatures of less than about 5 °C should not be allowed in the intake system in order to avoid difficulties arising from icing. Computed power output and efficiency with varying days of the year are shown in Figures 5 and 6 respectively.

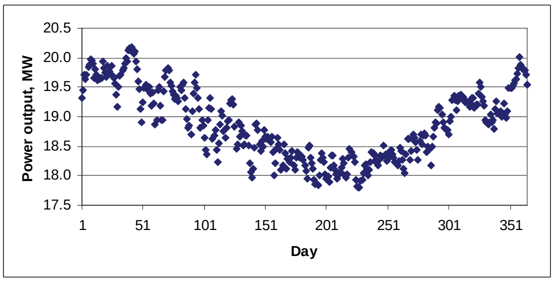


Figure 5: Power output versus day of the year

As expected, the higher the ambient temperature is the lower rate of generated power from the GT unit gets. Unfortunately, the minimum produced power is during the summer season, May-October This id (days 150-300), when peak demand occurred in Jordan. due to the increasing demand on electrical energy for air-conditioning and ventilation systems. Such case would make it more difficult for electricity providers to match the increasing demands during hot and dry summer. Thus, simple and cost-effective measures should be taken in order to meet consumers' needs. In this study, the basic assumption is that the effectiveness of the evaporative cooling system will be around 80%, which is reasonable for such large systems. According to weather data provided by the Meteorological Department, in Jordan, the average ambient temperature during hot days may exceed 35 °C with a low relative humidity ratio of about 30%. Such conditions are favourable for evaporative cooling systems. In such case, the inlet air temperature can be decreased by approximately 10-15°C to reach a wet bulb temperature of about 20-25 °C.

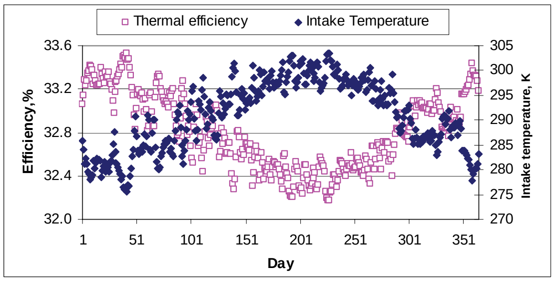


Figure 6: Thermal efficiency and intake temperature versus day of the year

The calculated thermal efficiency of the employed gas turbine engine versus the day of the year is shown in Figure 7. It is obvious that the efficiency during summer is lower than that incurred during winter, but there was a slight increase of around 1% in the obtained efficiency when compared with the results of the base case scenario for same GT engine. Increasing the effectiveness of the evaporative cooler by 10% to reach 90% would result in a slight increase in the produced power and final thermal efficiency.

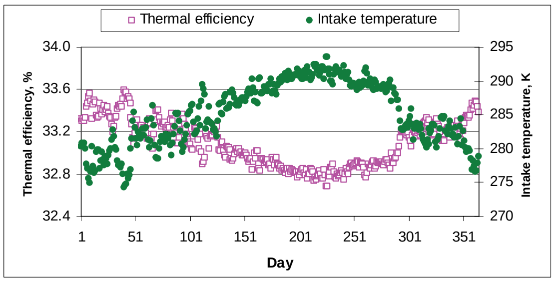


Figure 7: Thermal efficiency and intake temperature versus day of the year

Higher intake temperatures will lead to deterioration in the power output and thermal efficiency of the studied engine, which can be ratified by the increased work of compressor due to the fact that air viscosity is proportional to temperature. Thus, compressor consumes more effort to provide the same pressure rise; hence the net out put is reduced. Another important reason behind the increased power, in addition to decreasing the temperature of intake air, is the higher rates of mass flow across the engine because of added water. It is estimated that mass flow rate of water in the evaporative cooling system will range between 0.3-0.5 kg/s, i.e., less than 0.7% of the air mass flow rate. That is enough to bring the incoming air to near saturation conditions depending on climatic conditions on that particular day or time –see Fig. 8.

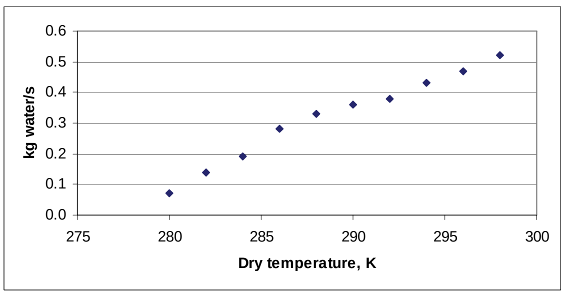


Figure 8: Water mass flow rate versus dry temperature

In the case a more effective evaporating cooling system, with effectiveness of 90% is used the water flow rate may reach an average of about 0.6 kg/s and the resultant net increase in the power out put is estimated to be around 1 MW, i.e., 5% of the rated power of the studied engine. However, it should be noted an that the function of evaporating cooling system is cooling the air by humidifying it, which is an adiabatic process. Thus, water flow rate will follow not only the dry bulb temperature, but more importantly, the relative humidity on a particular time. Thence, higher relative humidity ratios would result in lower amounts of water to reach saturation conditions, i.e., wet bulb temperature which is constant during the cooling process.

The second simulated method is the cooling coil, i.e., employing a mechanical chiller, to pre-cool air intake. It is found that it will be possible to decrease intake air temperature by about 20 °C or even more if needed during hot summer days regardless of the relative humidity ratio. Figure 9 shows the effect of using cooling coil on the temperature of the intake air; and Figure 10 illustrates the net efficiency of the gas turbine power plant on each day of the year. As mentioned earlier, lower efficiencies often incurred during summer, which makes it an attractive option to augment power by employing chillers or other means for air pre-cooling.

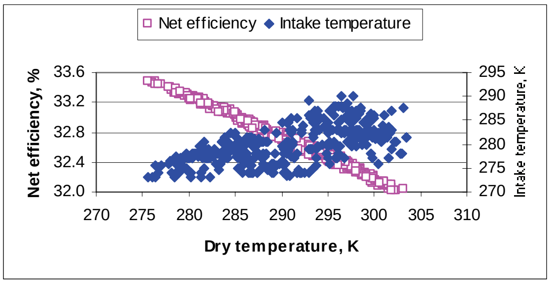


Figure 9: Net efficiency and intake temperature versus dry temperature

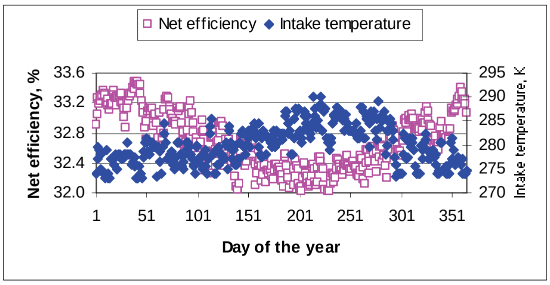


Figure 10: Net efficiency and intake temperature versus day

An actual increase in the net power produced from the plant is recorded by about 0.5-1 MW over that calculated in the base case scenario under similar operating conditions and assumed coefficient of performance (COP) of the chiller system equals two. The exact value is dependent on ambient temperature and COP of the employed chiller. For a higher COP, the augmented power and net power produced from the plant will be increased – see Figures 11 and 12. It can be seen from these figures that the effect of COP is significant at high ambient temperatures.

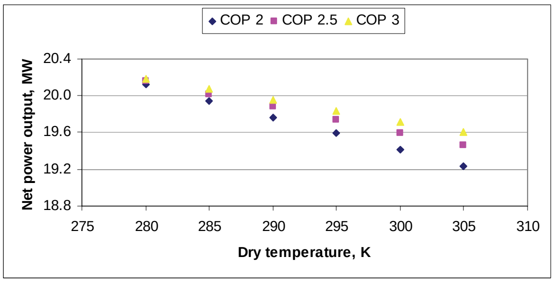


Figure 11: Net Power output versus air dry temperature for different COP

This is because a chiller with high COP will consume less energy to produce the same cooling effect, and consequently the plant's net efficiency is higher. Similar results were reported by independent researchers from different countries but not for Jordan [8,12-15,23]. Nevertheless, it is interesting to note when air temperature in the intake system is reduced, the net power output will be reduced as well, due to the fact that the employed chiller will consume more energy to bring the temperature down to the desired level.

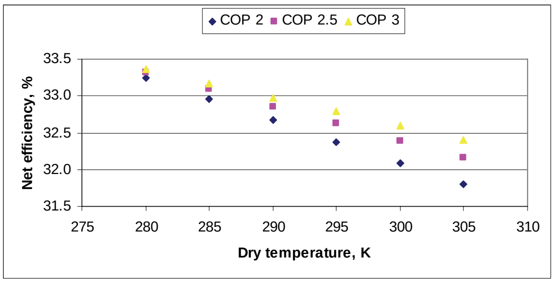


Figure 12: Net efficiency versus air dry temperature for different COP

The computed results of the different operating scenarios revealed that the net power output from the gas turbine plant varied significantly when intake air was cooled down as compared with the base case without cooling – see Fig. 13. It is obvious that there is a slight difference between the employed two cooling systems, but the basic issue here is the augmented power from the gas turbine plant as a direct result of reducing air temperature at the compressor intake.

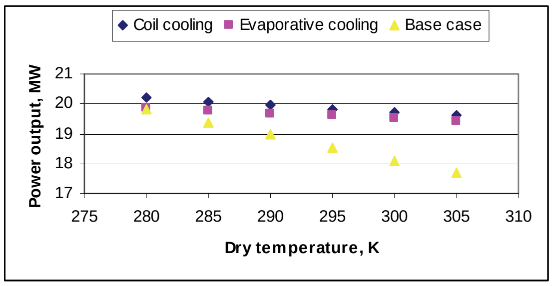


Figure 13: Estimated power output of different cooling systems (effectiveness of evaporative system is 0.8 & COP of chiller is 3)

Assuming an average dry bulb temperature of about 32 °C, and a temperature drop of about 10 and 13 °C when evaporative and coil cooling systems were used, then the augmented power would be around 1 and 1.5 MW, respectively.

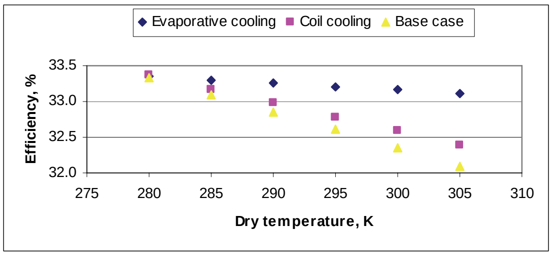


Figure 14: Predicted thermal efficiency of employed cooling systems (effectiveness of evaporative system is 0.8 and COP of chiller is 3)

This finding represents an improvement of about 5-7%, which is in full agreement with published work done elsewhere [8,12-15,23]. In terms of thermal efficiency, the evaporative cooling system is more efficient than coil cooling system as can be seen in Fig. 14. This can be attributed to the fact that in case of coil cooling system some of the generated power will be needed to run the refrigeration cycle, thus, the final efficiency of the gas turbine power plant drops slightly. But in the evaporative cooling system de-mineralized water is needed, instead of electricity to cool down the incoming air into the gas turbine, which is scarcely available in Jordan. This represents an important issue that should be taken into consideration when selecting most appropriate cooling system for particular power plant and location in Jordan or elsewhere. In the future, it is highly advisable to extend this study to include absorption cooling and thermal storage systems.

The expected annual net revenues resulting directly from excess power generation, in this study, is significant. To predict economics of such systems, a detailed cost-benefit analysis should be carried out. But at this stage, it is very difficult to estimate the required capital investment for such cooling systems due to many reasons. The most important is the size of the system and the actual site conditions. However, based on available published data in the open literature, it has been estimated that an approximate cost of simple evaporative cooling system is about 7-10 JD or 10-15 US$ per kW installed. This brings the total cost of the evaporative cooling system of between 140,000–200,000 JD. While, the approximate cost of industrial cooling coil system is about 700-1000 JD/ton Ref or 1000-1500 US$/ton Ref installed. This brings the total cost of the required chiller of between 400,000–575,000 JD. However, such capital costs are much cheaper than that required to install a new gas turbine unit with an average specific investment of about 300-500 US$ per kW [24].

Based on the previous analysis and calculated annual revenues, the predicted SPBP would be around 2 and 5 years for evaporative and cooling coil systems, respectively. A higher capital cost will lead to longer recovery periods; however, doubling the initial cost will increase the payback period by one year approximately. But it should be hated that higher temperatures will result in shorter payback periods. The obtained SPBP is considered very short from economic point of view and highly attractive for investments in such projects, especially when evaporative cooling system is considered.

To sum up, it can be said that using intake air cooling technologies for gas turbine power plants is an attractive option to increase power output and efficiency of these plants. Therefore, it is a main recommendation of this study for concerned governmental institutions and power generation companies to conduct detailed and specific studies for all gas turbine power plants in the Jordan in order to evaluate the precise costs and benefits of different cooling systems. Hence, the most appropriate cooling system can be selected and installed soon.

6. Conclusions

This article presents a detailed technical assessment for two cooling methods to cool the intake air before being introduced into the compressor. Although the performance of evaporative coolers is highly dependent on ambient temperature and humidity, they do operate efficiently during dry and hot climatic conditions such those prevailed in Jordan during summer. The obtained results shows that the evaporative cooling system appears to be capable of boosting generated power by about 5% and enhancing efficiency of the selected gas turbine unit in a way that is less expansive than a cooling coil. But providing needed water for the operation of such coolers may be a challenge in some regions, in Jordan.

Cooling coil gives a full control of the compressor inlet conditions regardless of ambient conditions; however, it demands a quite large operational power. During hot and dry conditions, the net power output produced from the GT unit increases by about 1.0-1.5 MW when cooling system is used over that calculated in the base case under similar operating conditions. The coil cooling increases the efficiency of the power plant by about 1 % or more depending on the operating conditions and the selected chiller system, because the power extracted for refrigeration must be included. Extracting this power from the gas turbine output reduces the overall plant performance, i.e., large deterioration in the efficiency occurs at high temperature due to the increasing cooling load. For a higher COP, the augmented power and hence net power, produced from the plant will be increased. Other means, such as, absorption cooling system is expected to yield more power since energy required to run the absorption cycle is taken freely from the turbine's exhaust but at a higher capital cost.

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