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Thermo-Economic Analysis of Gas Turbines Power Plants with Cooled Air Intake

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Abstract: Gas turbine (GT) power plants operating in arid climates suffer from a decrease in power output during the hot summer months. Cooling the intake air enables the operators to mitigate this shortcoming. In this study, an energy analysis of a GT Brayton cycle coupled to a refrigeration cycle shows a promise of increasing the power output with a slight decrease in thermal efficiency. A thermo-economic algorithm is also developed and applied to the Hitachi MS700 GT open cycle plant at the industrial city of Yanbu, the Kingdom of Saudi Arabia (latitude 24°05” N and longitude 38° E). The results show that the power output enhancement depends on the degree of chilling the air intake to the compressor. Moreover, maximum power gain ratio is 15.46% whilst a slight decrease in thermal efficiency is of 12.25% for this case study. The study estimates the cost of the needed air cooling system. The cost function takes into consideration the time-dependent meteorological data, operation characteristics of the GT and air cooler, the operation and maintenance costs, interest rate, and lifetime. The study also evaluates the profit of adding the air cooling system for different electricity tariff.

Keywords: Gas Turbine, Power Boosting, Hot Climate, Air-Cooling, Mechanical Refrigeration

1. Introduction

High electricity demand during summer is a challenge for the local utilities in Saudi Arabia and neighboring countries. Air conditioning is a driving factor for electricity demand and operation schedules for these countries. In the Kingdom of Saudi Arabia (KSA), the utilities employ gas turbine (GT) power plants (present capacity 14 GW) to meet the peak load. Unfortunately, the power output and thermal efficiency of GT plants decrease in the summer because of the high inlet air temperature. The high air temperature at the GT intake decreases the air mass flow leading to the less power output generated [1]. For an ideal GT open cycle, the decrease in the net power output is approximately 0.4 % for every 1 K increase in the ambient air temperature [1-2]. Therefore, cooling the air intake to improve the GT performance receives considerable attention. Both direct methods (e.g., evaporative cooling) and indirect methods (e.g., mechanical vapor compression cooling) for cooling the air intake have been studied [1-10]. In the evaporative cooling method, the air intake is cooled off by contacting a cooling fluid such as atomized water sprays, fog, or their combination [2-3]. This method is suitable for GT power plants operating in dry and hot regions [1, 4-8]. The evaporative cooling leads to higher plan efficiency compared to vapor compression cooling when used in geographical regions having low ambient relative humidity and temperature [1]. The evaporative cooling provides low capital and operation cost and reliability, required moderate maintenance, and reduces the NOx content in the exhaust gases [9]. The main disadvantages of evaporative cooling method are low operation efficiency and large quantities of water consumption. Furthermore, the impact of the non-evaporated water droplets in the air stream could damage the compressor blades [10]. Replacing the water sprays by fogging system eliminates erosion problem [11-15].

The mechanical vapor compression [8, 16] and absorption refrigerator machines [17-20] are the two widely used approaches for indirect cooling. These cooling methods overcome the constraint set by the relative humidity of intake air. Direct cooling methods can reduce the air temperature below the ambient wet bulb temperature (WBT); however, extensive air chilling leads to ice formation as ice crystals or as solidified layer on compressor entrance [21]. The indirect cooling methods have gradually gained popularity over the evaporative cooling. For instance, 32 GT units have been outfitted with mechanical air chilling systems in Riyadh, Saudi Arabia. In general, application of the mechanical air-cooling increases the net power and reduces the thermal efficiency on the other hand [7].

New cooling methods are also available in the literature [22-27]. Farzaneh-Gord et al. [22] and Zaki et al. [23] proposed the use of a reversed Brayton refrigeration cycle for cooling the air intake to enhance the performance of GT. According to the results of the study [23], the air intake temperature could be lowered below the ISO standard with the power output increase up to 20% and decrease in the thermal efficiency of 6%. Jassim et al. [24] performed the exergy analysis of the system and showed the maximum improvement is 14.66% due to the components irreversibility. Khan et al. [25] proposed cooling the turbine exhaust gases and feeding them back to the compressor inlet with water harvested from the combustion products. Erickson [26-27] suggested a power fogger cycle that is a combination of waste heat driven absorption air-cooling and a water injection system.

Generally, even though thermal analyses of GT cooling are abundant in the literature, a few economic evaluations of implementing the air intake cooling methods have been considerably investigated. Such evaluations should account for the variations in the ambient conditions (temperature and relative humidity), the fluctuations in the fuel, electricity prices, and the interest rates. Therefore, the selection of a cooling technology (evaporative or refrigeration) and the sizing out of the equipment should base on not only the result of thermal analysis but also the cash flow. There are some outstanding studies focusing on the economic aspects of the cooling methods in literature. Gareta et al. [28] presented a computational algorithm to calculate the yearly additional power gain for combined cycle GT along with the economic feasibility for some cooling technologies. Chaker et al. [13] studied the economic potential of using evaporative cooling for GTs in USA. Yang et al. [17] did the same for combined GT in KSA while Hasnain et al. [29] and Shirazi et al. [30] examined the use of ice storage methods for GTs' air cooling in KSA. Investigations showed that the efficiency of inlet fogging was superior for the intake temperatures of 15-20oC, though it results in a smaller profit than inlet chilling.

This study presents a thermal and economic analysis of a GT system fitted with an external chilled water loop. The analysis accounts for the changes in the thermodynamics parameters as well as the economic variables (e.g., profitability, cash flow, and the lifetime of the system) for GT and the cooling components. The objective of this study is to assess the importance of using a coupled thermo-economic analysis in the selection of cooling system and operation parameters. The developed thermo-economic algorithm is then applied to the Hitachi MS700 GT, an open cycle plant, at the industrial city of Yanbu, KSA (latitude 24o 05" N and longitude 38o E). Finally, the cost analysis that is based on 10% interest rate and three-year payback period of the water chiller is presented to determine the economic feasibility of using water chiller as an air cooler.

2. Thermodynamics Analysis

2.1. Gas Turbine Cycle Analysis

Figs.1a and b, respectively, show a schematic of a simple open GT Brayton cycle coupled to a refrigeration system and its T-S diagram. The power cycle consists of a compressor, a combustion chamber, and a turbine. The cooling system consists of a refrigerant compressor, air cooled condenser, throttle valve, and an evaporator. The chilled water from the evaporator passes through a cooling coil mounted at the air compressor entrance as shown in Fig. 1a. In this figure, the dotted line indicates a fraction of the electricity produced by the turbine and used to power the compressor and the pumps. Fig. 1c presents the T-S diagram of the refrigerant cycle by states a, b, c, and d.

As shown in Figs. 1a and b, the processes 1-2s and 3-4s are isentropic. Assuming air behaves as an ideal gas, the temperatures and pressures are related to the pressure ratio (PR) by

$$\frac{T_{2s}}{T_1} = \frac{T_3}{T_{4s}} = \left[\frac{P_2}{P_1}\right]^{\frac{k-1}{k}} = PR^{\frac{k-1}{k}} \tag{1}$$

where k is the specific heat ratio.

The net power output of the GT with the mechanical cooling system is

$$\dot{W}_{net} = \dot{W}_t - (\dot{W}_{comp} + \dot{W}_{el,ch}) \tag{2}$$

The first term of the RHS is the power produced by the turbine due to expansion of hot gases of mass flow rate $\dot{m}_t$ as

$$\dot{W}_t = \dot{m}_t\, c_{pg} \eta_t \left(T_3 - T_{4s}\right) \tag{3}$$

In this equation, $\dot{m}_t$ is the total gas mass flow rate from the combustion chamber given in terms of the fuel air ratio $f = \dot{m}_f / \dot{m}_a$ and the air humidity ratio at the compressor intake ω_1 ($kg_w/kg_{dry\ air}$) at state 1(Fig. 1a) as

$$\dot{m}_t = \dot{m}_a + \dot{m}_v + \dot{m}_f = \dot{m}_a(1 + \omega_1 + f) \tag{4}$$

The compression power for the humid air between states 1 and 2 is

$$\dot{W}_{comp} = \dot{m}_a\, c_{pa}\left(T_2 - T_1\right) + \dot{m}_v\left(h_{v2} - h_{v1}\right) \tag{5}$$

where hv2 and hv1 are the enthalpies of saturated water vapor at the compressor exit and inlet states, respectively; $\dot{m}_v = \dot{m}_a \omega_1$ is the mass of water vapor.

The last term in Eq. 2 ($\dot{W}_{el,ch}$) is the power consumed by the cooling unit for driving the refrigeration machine electric motor, pumps, and auxiliaries. The thermal efficiency of a GT coupled to an air cooling system is

$$\eta_{cy} = \frac{\dot{W}_t - (\dot{W}_{comp} + \dot{W}_{el,ch})}{\dot{Q}_h} \tag{6}$$

Substituting for T4s and $\dot{m}_t$ from Eqs. 1 and 4 into Eq. 3 yields

$$\dot{W}_t = \dot{m}_a (1 + \omega_1 + f) c_{pg} \eta_t T_3 \left(1 - \frac{1}{PR^{\frac{k-1}{k}}} \right) \tag{7}$$

The turbine isentropic efficiency ηt can be estimated using the practical relation recommended by Alhazmy et al. [7] as

$$\eta_t = 1 - \left(0.03 + \frac{PR - 1}{180} \right) \tag{8}$$

Relating the compressor isentropic efficiency to the changes in temperature of the dry air and assuming that the compression of water vapor behaves as ideal gas, the actual compressor power becomes

$$\dot{W}_{comp_{air}} = \dot{m}_a \left[c_{pa} \frac{T_1}{\eta_c} \left(PR^{\frac{k-1}{k}} - 1 \right) + \omega_1 \left(h_{v2} - h_{v1} \right) \right] \tag{9}$$

The compression efficiency η_c can be evaluated using the following empirical relation [7]:

$$\eta_c = 1 - \left(0.04 + \frac{PR - 1}{150} \right) \tag{10}$$

The heat balance in the combustion chamber (see Fig. 1a) gives the heat rate supplied to the gas turbine cycle as

$$\dot{Q}_h = \dot{m}_f \, NCV \eta_{comb} = \left(\dot{m}_a + \dot{m}_f \right) c_{pg} T_3 - \dot{m}_a c_{pa} T_2 + \dot{m}_v \left(h_{v3} - h_{v2} \right) \tag{11}$$

Introducing the fuel air ratio $f = \dot{m}_f / \dot{m}_a$ and substituting for T2 in terms of T1 into Eq. 11 yields

$$\dot{Q}_h = \dot{m}_a T_1 \left[(1 + f) c_{pg} \frac{T_3}{T_1} - c_{pa} \left(\frac{PR^{\frac{k-1}{k}} - 1}{\eta_c} + 1 \right) + \frac{\omega_1}{T_1} \left(h_{v3} - h_{v2} \right) \right] \tag{12}$$

The simple expression for f is selected according to Alhazmy et al. [8] as

$$f = \frac{c_{pg} \left(T_3 - 298 \right) - c_{pa} \left(T_2 - 298 \right) + \omega_1 \left(h_{v3} - h_{v2} \right)}{NCV \eta_{comb} - c_{pg} \left(T_3 - 298 \right)} \tag{13}$$

In this equation, hv2 and hv3 are the enthalpies of water vapor at the combustion chamber inlet and exit states, respectively, and can be calculated from Dossat [32]:

$$h_{vj} = 2501.3 + 1.8723 T_j ; \quad j \textit{ refers to state } 1 \textit{ or } 3 . \tag{14}$$

The four terms of the gas turbine net power and efficiency in Eq. 2 ($\dot{W}_t, \dot{W}_{comp}, \dot{W}_{el,ch}, \dot{Q}_h$) depend on the air temperature and relative humidity at the compressor inlet whose values are affected by the type and performance of the cooling system. The chillers' electric power $\dot{W}_{el,ch}$ is calculated in the following account.

2.2. Refrigeration Cooling System Analysis

For this analysis, the inlet air is cooled by using a cooling coil placed at the compressor inlet bell mouth. The chilled water from the refrigeration machine is the heat transport fluid as shown in Fig. 1a. The chiller's total electrical power can be expressed as the sum of the electric motor power ($\dot{W}_{motor}$), the pumps ($\dot{W}_P$), auxiliary power for fans, and control units ($\dot{W}_A$) as

$$\dot{W}_{el,ch} = \dot{W}_{motor} + \dot{W}_P + \dot{W}_A \tag{15}$$

In this equation, $\dot{W}_A$ which is estimated to be from 5% to 10% of the compressor power is the input power to the auxiliary equipment, such as the condenser fans, control system, and so on. In this study, an air cooled condenser is used and 10% of the power required to drive the compressor motor is estimated for the cycle auxiliaries ($\dot{W}_A = 0.1 \, \dot{W}_{motor}$). The second term in Eq. 15 is the pumping power which is related to the chilled water flow rate and the pressure drop across the cooling coil as

$$\dot{W}_P = \dot{m}_{cw} v_f \left(\Delta P \right) / \eta_{pump} \tag{16}$$

The isentropic compression process (a-b_s) is the minimum energy utilized by the compressor, as depicted in Fig. 1c. The actual chiller power includes losses due to mechanical transmission, inefficiency in the drive motor converting electrical to mechanical energy, and the volumetric efficiency [32]. In general, the compressor electric motor work is related to the refrigerant enthalpy change as

$$\dot{W}_{motor} = \frac{\dot{m}_r \left(h_b - h_a \right)_r}{\eta_{eu}} \tag{17}$$

The subscript *r* indicates refrigerant and η_{eu} is known as the energy used factor, $\eta_{eu} = \eta_m \eta_{el} \eta_{vo}$. The quantities on the right hand side are the compressor mechanical, electrical, and volumetric efficiencies, respectively. η_{eu} is usually determined by manufacturers and depends on the type of the compressor, the pressure ratio (P_b/P_a), and the motor power. For this analysis, η_{eu} is assumed as 85%.

Cleland et al. [31] developed a semi-empirical form of Eq. 17 to calculate the compressor's motor power usage in terms of the temperatures of the evaporator and condenser in the refrigeration cycle, T_e and T_c, respectively, as

$$\dot{W}_{motor} = \frac{\dot{m}_r \left(h_a - h_d\right)_r}{\dfrac{T_e}{\left(T_c - T_e\right)}\left(1-\alpha\, x\right)^n \eta_{eu}} \tag{18}$$

In this equation, α is an empirical constant that depends on the type of refrigerant and x is the quality at state d in Fig. 1c. The empirical constant is 0.77 for R-22 and 0.69 for R-134a [31]. The constant n depends on the number of the compression stages and n = 1 for a simple refrigeration cycle with a single stage compressor. The nominator of Eq. 18 is the evaporator capacity $\dot{Q}_{e,r}$ and the first term of the denominator is the coefficient of performance of an ideal refrigeration cycle. Eqs. 2, 5, and 18 could be solved for the power usages by the different components of the coupled GT-refrigeration system and the increase in the power output as function of the air intake conditions. This thermodynamic performance analysis is coupled to a system economic analysis described later.

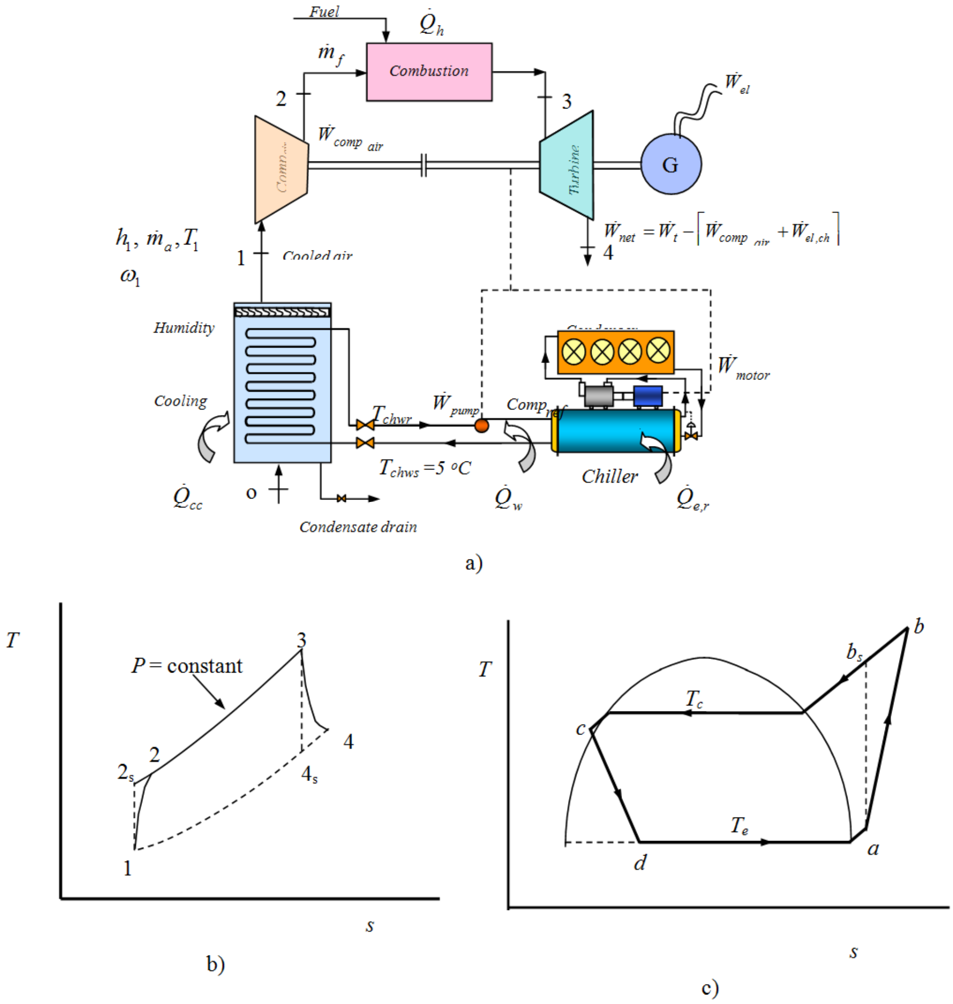

Fig. 1. *a). Simple open type gas turbine with a chilled air-cooling unit; b) T-S diagram of an open type gas turbine cycle; c) T-S diagram for a refrigeration machine.*

3. Economics Analysis

The increase in the power output will add to the revenue of the GT plant. However, this increase will partially be offset by the increase in capital cost associated with the installation of cooling system, personnel, and utility expenditures for the operation. For a cooling system including a water chiller, the increase in expenses involves the capital installments for the chiller (C^c_{ch}) and cooling coil (C^c_{cc}), the annual operational annual expenses. The latter is a function of the operation period top and the electricity rate. If the chiller consumes electrical power $\dot{W}_{el,ch}$ and the electricity rate is Cel ($/kWh) then the total annual expenses are

$$C_{total}\ (\$/y) = a^c\left[C^c_{ch} + C^c_{cc}\right] + \int_0^{t_{op}} C_{el}\, \dot{W}_{el,ch}\ dt \tag{19}$$

where $a^c = i_r\left(1+i_r\right)^n \Big/ \left(\left(1+i_r\right)^n - 1\right)$ is the capital-recovery factor.

The chiller's purchase cost which is related to the chiller's capacity, $\dot{Q}_{e,r}$ (kW or ton/day), can be estimated from vendors or mechanical equipment cost index. For a particular chiller size and methods of construction and installation, manufacturers usually give the capital cost as

$$C_{ch}^{c} = \alpha_{ch} \times \dot{Q}_{e,r} \tag{20}$$

where αch is a multiplication cost index in \$/kW. For simplicity, the maintenance expenses are assumed as a certain fraction αm of the capital cost of the chiller; therefore, the total chiller capital cost is given as

$$C_{ch}^{c}(\$) = \alpha_{ch}\left(1+\alpha_{m}\right)\dot{Q}_{e,r} \tag{21}$$

Similarly, the capital cost of a particular cooling coil is given by manufacturers in terms of the cooling capacity that is directly proportional to the total heat transfer surface area (Accm2) Kotas [34] as

$$C_{cc}^{c}\ (\$) = \beta_{cc}\left(A_{cc}\right)^{m} \tag{22}$$

In this equation, βcc and m depend on the type of the cooling coil and material. For this study and the local KSA market, βcc = 30000 and m = 0.582 are recommended. Substituting Eqs. 21 and 22 into Eq. 19 together with the assumption that the chiller power is an average constant value and the electricity rate is time independent for simplification, the annual total expenses for the cooling system become

$$C_{total}\ (\$/\text{y}) = a^{c}\left[\alpha_{ch}\left(1+\alpha_{m}\right)\dot{Q}_{e,r} + \beta_{cc}\left(A_{cc}\right)^{m}\right] + t_{op}C_{el}\dot{W}_{el,ch} \tag{23}$$

In Eq. 23, the heat transfer area Acc is used to evaluate the cost of the cooling coil. An energy balance for both the cooling coil and the refrigerant evaporator, taking into account the effectiveness factors for the evaporator εeff,er and the cooling coil εeff,cc, gives

$$A_{cc} = \frac{\dot{Q}_{cc}}{U\,\Delta T_{m} F} = \frac{\dot{Q}_{e,r} \times \varepsilon_{eff,er} \times \varepsilon_{eff,cc}}{U\,\Delta T_{m} F} \tag{24}$$

where U is the overall heat transfer coefficient for the chilled water-air tube bank heat exchanger. Gareta et al. [28] suggested a moderate value of 64 W/m2 K and recommended the factor F to be 0.98.

In Fig. 2, which shows the different temperatures in the combined refrigerant, water chiller, and air cooling system, the mean temperature difference for the cooling coil (air and chilled water fluids) is

$$\Delta T_{m} = \frac{\left(T_{o}-T_{chwr}\right)-\left(T_{1}-T_{chws}\right)}{\ell n\left(\left(T_{o}-T_{chwr}\right)/\left(T_{1}-T_{chws}\right)\right)} \tag{25}$$

Equations 22 and 24 give the cooling coil cost as

$$C_{cc}^{c} = \beta_{cc}\left(\frac{\dot{Q}_{cc}}{U\,\Delta T_{m} F}\right)^{m} \tag{26}$$

where $\dot{Q}_{cc}$ is the thermal capacity of the cooling coil. The atmospheric air enters at To and ω0 and leaves the cooling coil at T1 and ω1 before reaching the compressor, as shown in Fig. 1a. Both T1 and ω1 depend on the chilled water supply temperature Tchws and the chilled water mass flow rate $\dot{m}_{cw}$. When the outer surface temperature of the cooling coil falls below the dew point temperature (corresponding to the partial pressure of the water vapor), the water vapor condensates and leaves the air stream. This process may be treated as a cooling-dehumidification process as shown in Fig. 3.

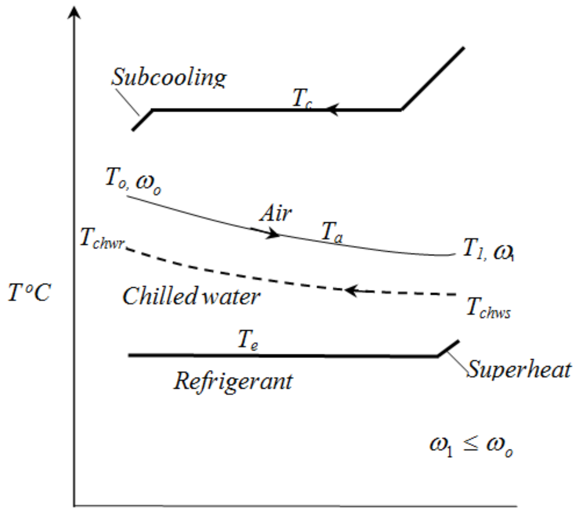

Fig. 2. *Temperature levels for the three working fluids, not to scale.*

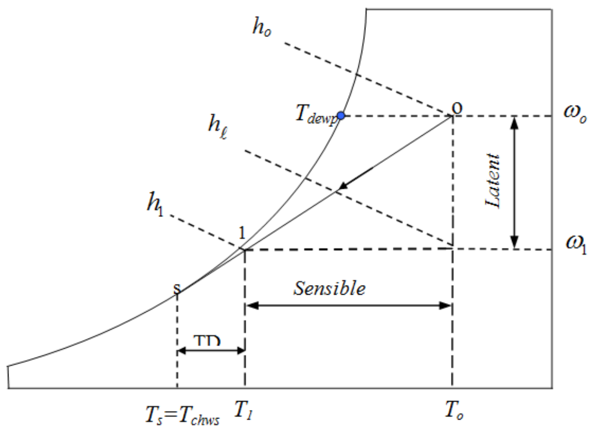

Fig. 3. *Moist air cooling process on the psychrometric chart.*

Steady state heat balance of the cooling coil gives $\dot{Q}_{cc}$ as

$$\dot{Q}_{cc} = \dot{m}_{a}\left(h_{o}-h_{1}\right) - \dot{m}_{w}h_{w} = \dot{m}_{cw}c_{pw}\varepsilon_{eff,cc}\left(T_{chwr}-T_{chws}\right) \tag{27}$$

where $\dot{m}_{cw}$ is the chilled water mass flow rate, $\dot{m}_{w}$ is the rate of water extraction from the air, and $\dot{m}_{w} = \dot{m}_{a}\left(\omega_{o}-\omega_{1}\right)$. It is usually a small term when compared to the first and can be neglected [35].

In Eq. 27, the enthalpy and temperature of the air leaving the cooling coil (h_1 and T_1) can be calculated from

$$h_1 = h_o - CF\left(h_o - h_s\right) \tag{28}$$

$$T_1 = T_o - CF\left(T_o - T_s\right) \tag{29}$$

where CF is the contact factor of the cooling coil. It is defined as the ratio between the actual air temperature drop to the maximum at which the air theatrically leaves at coil surface temperature T_s = T_{chws} and 100% relative humidity. Substituting h_1 from Eq. 28 into Eq. 27 gives

$$\dot{Q}_{cc} = \dot{m}_a\left[CF(h_o - h_{chws}) - (\omega_o - \omega_1)h_w\right] \tag{30}$$

Eqs. 24 and 30 yield

$$\dot{Q}_{e,r} = \frac{\dot{m}_a\left[CF\left(h_o - h_{chws}\right) - \left(\omega_o - \omega_1\right)h_w\right]}{\varepsilon_{eff,er} \times \varepsilon_{eff,cc}} \tag{31}$$

Eqs. 24, 30, and 31 give the cooling water flow rate, cooling coil capacity, and the evaporator capacity in terms of the air mass flow rate and properties.

Refrigeration Cooling System Analysis

Combining Eqs. 23 and 24 and substituting for the cooling coil surface area, pump, and auxiliary power give the cost function in terms of the evaporator capacity $\dot{Q}_{er}$. Total annual cost is given as

$$C_{total} = \left\{ \begin{array}{l} a^c\left[\alpha_{ch}\left(1+\alpha_m\right)\dot{Q}_{er} + \beta_{cc}\left(\dfrac{\dot{Q}_{er} \times \varepsilon_{eff,er} \times \varepsilon_{eff,cc}}{U\,\Delta T_m F}\right)^m\right] + \\ t_{op}\,\dot{Q}_{er} C_{el}\left[\left(\dfrac{1.1\left(T_c - T_e\right)}{T_e\left(1-\alpha x\right)^n \eta_{eu}}\right) + \left(\dfrac{\varepsilon_{eff,er}\, v_f\left(\Delta P\right)}{c_{p,w}\,\Delta T_{ch,w}\eta_{pump}}\right)\right] \end{array} \right\} \tag{32}$$

The first term in Eq. 32 is the annual fixed charges of the refrigeration machine and the surface of air cooling coil, while the second term is the operation expenses that depend mainly on the electricity rate. The motor power has been increased by 10% to account for the auxiliaries' consumption. If the water pump's power is considered as infinitesimally small compared to the compressor power, the second term of the operation charges can be dropped. If the evaporator capacity $\dot{Q}_{er}$ is replaced by the expression in Eq. 31, the cost function, in terms of the primary parameters, becomes

$$C_{total} = \left[\frac{\dot{m}_a\left[CF\left(h_o - h_{chws}\right) - \left(\omega_o - \omega_1\right)h_w\right]}{\varepsilon_{eff,er} \times \varepsilon_{eff,cc}}\right] \times \left\{ \begin{array}{l} a^c\left[\alpha_{ch}\left(1+\alpha_m\right) + \beta_{cc}\left(\dfrac{\varepsilon_{eff,er} \times \varepsilon_{eff,cc}}{U\,\Delta T_m F}\right)^m \left(\dfrac{\dot{m}_a\left[CF\left(h_o - h_{chws}\right) - \left(\omega_o - \omega_1\right)h_w\right]}{\varepsilon_{eff,er} \times \varepsilon_{eff,cc}}\right)^{m-1}\right] + \\ t_{op}\, C_{el}\left[\left(\dfrac{1.1\left(T_c - T_e\right)}{\left(T_e\right)\left(1-\alpha x\right)^n \eta_{eu}}\right) + \left(\dfrac{\varepsilon_{eff,er}\, v_f\left(\Delta P\right)}{c_{p,w}\,\Delta T_{ch,w}\eta_p}\right)\right] \end{array} \right\} \tag{33}$$

4. Evaluation Criteria of GT-cooling System

In order to evaluate the feasibility of a cooling system coupled to a GT plant, the performance of the plant is examined with and without the cooling system. In general, the net power output of a complete system is

$$\dot{W}_{net} = \dot{W}_t - \left(\dot{W}_{comp} + \dot{W}_{el,ch}\right) \tag{34}$$

The three terms in Eq. 34 are functions of the air properties at the compressor intake conditions (T1 and ω1), which in turn depend on the performance of the cooling system. This analysis considers the power gain ratio (PGR), a broad term suggested in [8] that takes into account the operation parameters of the GT and the associated cooling system

$$PGR = \frac{\dot{W}_{net,with\,cooling} - \dot{W}_{net,without\ cooling}}{\dot{W}_{net,without\ cooling}} \times 100\% \tag{35}$$

For a stand-alone GT, PGR = 0. Thus, PGR gives the percentage enhancement in power generation by the coupled system. The thermal efficiency of the system is an important parameter to describe the input-output relationship. The thermal efficiency change factor (TEC) proposed in [8] is defined as

$$TEC = \frac{\eta_{cy,with\,cooling} - \eta_{cy,without\ cooling}}{\eta_{cy,without\ cooling}} \times 100\% \tag{36}$$

Both PGR and TEC can be easily employed to assess the changes in the system performance, but are not sufficient for a complete evaluation of the cooling method.

To investigate the economic feasibility of retrofitting a gas turbine plant with an intake cooling system, the total cost of the cooling system is determined (Eq. 32 or Eq. 33). The increase in the annual income cash flow from selling the additional electricity generation is also calculated. The annual energy electricity generation by the coupled power plant system is

$$E\ (\text{kWh}) = \int_0^{t_{op}} \dot{W}_{net}\,dt \tag{37}$$

If the gas turbine's annual electricity generation without a cooling system is Ewithout cooling and the cooling system increases the power generation to Ewith cooling, and then the net increase in revenue due to the addition of the cooling system can be calculated from

$$Net\ revenue = \left(E_{with\ cooling} - E_{without\ coolig}\right)C_{els} \tag{38}$$

The profitability due to the coupled power plant system is defined as an increase in revenues due to the increase in electricity generation after deducting the expenses for installing and operating the cooling system as

$$Profitability = (E_{with\ cooling} - E_{without\ cooling})C_{els} - C_{total} \tag{39}$$

The first term in Eq. 39 gives the increase in revenue whilst the second term gives the annual expenses of the cooling system. The profitability could be either positive or negative, which, respectively, means an economic insensitivity for adding the cooling system and an economic disadvantage, despite the increase in the electricity generation of the plant.

5. Results and Discussion

The analysis and economic feasibility are applied for the performance of the HITACH 700 model GT plant with a water chiller air cooling system. This plant, which has been already connected to the main electric grid, is located at the Industrial City of Yanbu (latitude 24o 05' N and longitude 38o E). The specifications of the GT plant are described in Table 1. The water chiller capacity is selected on basis of the maximum annual ambient temperature. On August 18th, 20xx, the dry bulb temperature (DBT) reached 50oC at 2:00 PM and the relative humidity was 84% at dawn time. The recorded hourly variations in the DBT (To) and RHo and the values are, respectively, shown in Fig. 4 and Table 2. The evaporator capacity of the water chiller (ton refrigeration) given in Eq. 30 is function of the DBT and RH. Fig. 5 shows that if the chiller is selected based on the maximum DBT = 50oC and RH = 18% (the data at 1:00 PM), its capacity would be 2200 tons. Another option is to select the chiller capacity based on the air maximum RH (RH = 0.83 and To = 28.5oC), which results in 3500 tons. It is more accurate, however, to determine the chiller capacity for the available climatic data of the selected day and to determine the maximum required capacity, as shown in Fig. 6. For the weather conditions at Yanbu city, a chiller capacity of 4200 tons is selected.

The hourly performance parameters of the GT plant, with and without cooling system (Eqs. 35 and 36), are calculated and compared. All thermo-physical properties are determined to the accuracy of the EES software [33]. The results show that the cooling system decreases the intake air temperature from To to T1 and increases the relative humidity to RH2 (Table 2). The chilled air temperature T1 is calculated from Eq. 29, assuming contact factor of 0.5 and a chilled water supply temperature of 5oC. Using the data in Table 2, the solution of Eqs. 35 and 36 gives the daily variation in the PGR and TEC (Fig. 7). There is certainly a potential benefit of adding the cooling system when there is an increase in the power output all the time; the calculated average for the design day is 12.25 %.

The PGR follows the same pattern of the ambient temperature, which simply means that the electric power of the GT plant increases during the hot hours of the day when electricity demand is high (10:00 AM to 6:00 PM). The increase in the output power of the GT plant reaches a maximum of 15.46% with a little change in the plant thermal efficiency. The practical illustrative application indicates that a maximum decrease in the thermal efficiency change of only 0.223% occurs at 13:00 PM when the air temperature is 45.2oC, and RH is 34%.

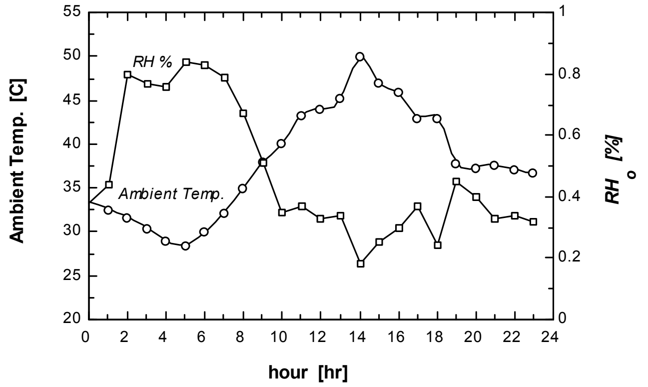

Fig. 4. Ambient temperature and RH variations on August 18th at Yanbu Industrial City, KSA.

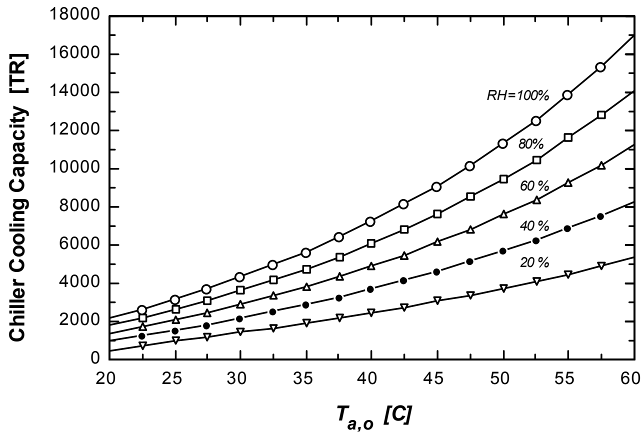

Fig. 5. Dependence of chiller cooling capacity on the climatic conditions.

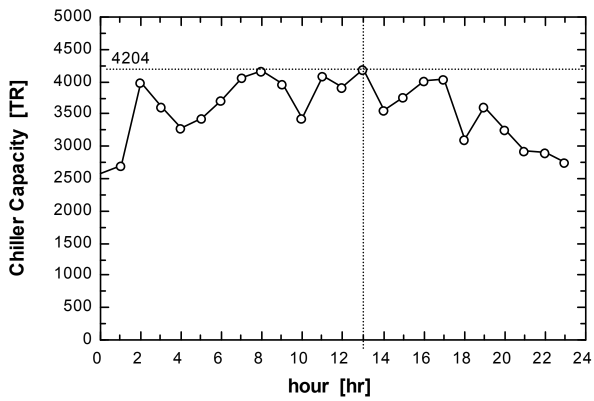

Fig. 6. Chiller capacity with the variation of the climatic conditions (temperature and RH).

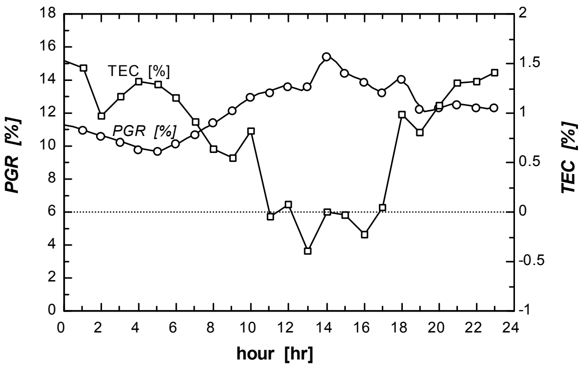

Fig. 7. Variation of gas turbine PGR and TEC during August 18th operation.

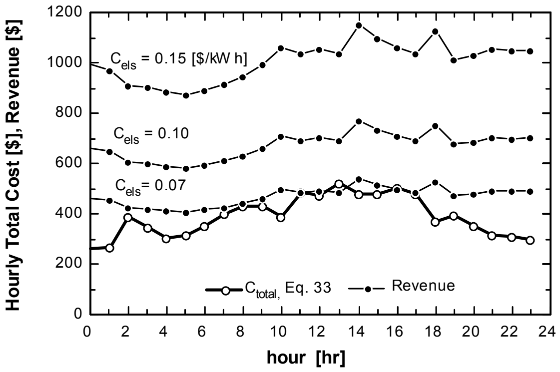

Fig. 8. Variation of hourly total cost and excess revenue at different unit cost of selling electricity.

Based on the daily variation of the ambient conditions on August 18th, assuming different values for selling the electricity (Cels), Eq. 39 gives the hourly revenues needed to payback the investment after a specified operation period (selected by 3 years). The different terms in both Eqs. 32 and 39 are calculated and presented in Fig. 8. Firstly, the effect of the climate changes is quite obvious on both the GT net power output as shown in Fig. 7 and on the total expenses as presented in Fig. 8. The variations in Ctotal are due to the changes in $\dot{Q}_{ev}$ in Eq. 32 that depends on T0, T1, ω0, and ω1. The revenues from selling additional electricity are also presented in the same figure, which shows clearly the potential of adding the cooling system. A profitability of the system, being the difference between the total cost and the revenues, is realized when the selling rate of the excess electricity generation is higher than the base rate of 0.07 $/kWh.

Fig. 8 shows that selling the electricity to the consumers at the same price ($C_{els} = C_{el}$ = 0.07 $/kWh) makes the cooling system barely non-profitable during the morning, night time, and hot hours of the day. This result encourages the utilities to consider adding a time-of-use tariff during the high demand periods, which is customary case in many courtiers. Should this become the case also in KSA, installing an air cooling system becomes economically feasible and profitable. Economics calculations for one year of 7240 operation hours with different electricity rates (C_{els}) and fixed electricity rate (C_{el} = 0.07 $/kWh) are summarized in Table 3.

Table 1. Specifications of the GT plant.

Parameters		Range
Ambient air	Ambient air temperature, T_o	28-50 ºC
	Ambient air relative humidity, RH_o	18%-84%
Gas turbine	Pressure ratio, P_2/P_1	10
	Turbine inlet temperature T_3	1273.15 K
	Volumetric air flow rate	250 m^3s^{-1}at NPT
	Fuel net calorific value, NCV	46000 kJ kg^{-1}
	Turbine efficiency, η_t	0.88
	Air compressor efficiency, η_c	0.82
	Combustion efficiency, η_{comb}	0.85
Generator	Electrical efficiency	95%
	Mechanical efficiency	90%
Water chiller	Refrigerant	R22
	Evaporating temperature, T_e	$T_{chws} - TD_e$ ºC
	Superheat	10K
	Condensing temperature, T_c	$T_o + TD_c$ K
	Condenser design temperature difference TD_c	10 K
	Evaporator design temperature difference TD_e	6 K
	Subcooling	3K
	Chilled water supply temperature, T_{chws}	5 ºC
	Chiller evaporator effectiveness, $\varepsilon_{eff,er}$	85%
	Chiller compressor energy use efficiency, η_{eu}	85%
	α_{ch}	172 *$/kW*
Cooling coil	Cooling coil effectiveness, $\varepsilon_{eff,cc}$	85%
	Contact factor, CF	50%
Economic analysis	Interest rate i	10%
	Period of repayment (payback period), n	3 years
	The maintenance cost, α_m	10% of C_{ch}^c
	Electricity rate, C_{el}	0.07 $/kWh
	Cost of selling excess electricity, C_{els}	0.07-0.15 $/kWh
	Hours of operation per year, t_{op}	7240

Table 2. *The ambient conditions and the cooling coil outlet temperature and humidity during August 18th operation.*

Hour	$T_o{}^oC$	RH	$T_1{}^oC$	RH_1
0	33.4	0.38	19.2	0.64
1	32.6	0.44	18.8	0.70
2	31.7	0.8	18.35	0.99
3	30.5	0.77	17.75	0.98
4	29.0	0.76	17.0	0.99
5	28.5	0.84	16.75	0.97
6	30.0	0.83	17.5	0.99
7	32.2	0.79	18.6	0.96
8	35.1	0.67	20.05	0.99
9	38.0	0.51	21.5	0.84
10	40.2	0.35	22.6	0.64
11	43.3	0.37	24.15	0.69
12	44.0	0.33	24.5	0.64
13	45.2	0.34	25.1	0.66
14	50.0	0.18	27.5	0.43
15	47.0	0.25	26.0	0.53
16	45.9	0.30	25.45	0.61
17	43.0	0.37	24.0	0.69
18	43.0	0.24	24.0	0.50
19	37.9	0.45	21.45	0.76

Table 3. *Annual net profits out of retrofitting a cooling system to a GT, Hitachi MS700 GT at Yanbu for different product tariff and 3 years payback period.*

Electricity selling rate C_{els} ($/kWh)	Annuity for chiller and maintenance ($/y)	Annual operating cost ($/y)	Annual net profit for the first 3 years ($/y)	Annual net profit for the fourth year ($/y)
0.07	1,154,780	1,835,038	-1,013,600	+141180
0.1	1,154,780	1,835,038	-166,821	+ 987,962
0.15	1,154,780	1,835,038	1,244,978	+ 2,399,758

The results in Table 3 show that there is always a net positive profit starting after the payback period for different energy selling prices. During the first 3 years of the cooling system life, there is a net profit when there is an increase in selling rate of the excess electricity generation to 0.15 S/kWh, which nearly doubles the base tariff.

6. Conclusions

There are various methods to improve the performance of gas turbine power plants operating under hot ambient temperatures far from the ISO standards. One proven approach is to reduce the compressor intake temperature by installing an external cooling system. In this paper, a simulation model that consists of thermal analysis of a GT coupled to refrigeration cooler and economics evaluation is developed. The performed analysis is based on coupling the thermodynamics parameters of the GT and cooler unit with the other variables such as the interest rate, life time, increased revenue, and profitability in a single cost function. The augmentation of the GT plant performance is characterized using the PGR and the TEC.

The developed model is applied to a GT power plant located at the city of Yanbu (20o 05” N latitude and 38o E longitude), KSA, where the maximum DBT has reached 50oC on August 18th, 20xx. The recorded climate conditions on that day are selected for sizing out the chiller and cooling coil capacities. The performance analysis of the GT, for a pressure ratio of 10, rate of air intake of 250 m3/s, and 1000oC maximum cycle temperature shows that the intake air temperature decreases from 12 to 22 K, while the PGR increases to maximum of 15.46%. The average increase in the plant power output power is 12.25%, with slight change in plant thermal efficiency.

In this study, the profitability resulting from cooling the intake air is calculated for electricity rates between 0.07 and 0.15 $/kWh and a payback period of 3 years. Cash flow analysis of the GT power plant in the city of Yanbu shows a potential for increasing the output power of the plant and increased revenues. The profitability is a result of adding the cooling system increase as the electricity rate increase during the peak demand periods, beyond the current base rate of 0.07 $/kWh.

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Nomenclatures

A_{cc} cooling coil heat transfer area, m^2
C_{cc}^{c} capital cost of cooling coil ($)
C_{ch}^{c} capital cost of chiller ($)
C_{el} unit cost of electricity, $/kWh
c_p specific heat of gases, kJ/kg K
CF contact factor
E energy kWh
EES engineering equation solver
h_v specific enthalpy of water vapor in the air, kJ/kg
i_r interest rate on capital
k specific heat ratio
$\dot{m}$ mass flow rate, kg s^{-1}
$\dot{m}_a$ air mass flow rate, kg/s
$\dot{m}_{cw}$ chilled water mass flow rate, kg/s
$\dot{m}_r$ refrigerant mass flow rate, kg/s
$\dot{m}_w$ condensate water rate, kg/s
NCV net calorific value, kJ kg^{-1}
P pressure, kPa
PGR power gain ratio
P_o atmospheric pressure, kPa
PR pressure ratio = P_2/P_1
$\dot{Q}_h$ heat rate, kW
$\dot{Q}_{e,r}$ chiller evaporator cooling capacity, kW
$\dot{Q}_{cc}$ cooling coil thermal capacity, kW
T temperature, K
TEC thermal efficiency change factor
U overall heat transfer coefficient, kW/m^2K
x quality
$\dot{W}$ power, kW
Greek symbols
η efficiency
ε_{eff} effectiveness, according to subscripts
ω specific humidity (also, humidity ratio), according to subscripts, $kg/kg_{dry\ air}$
Subscripts
a dry air
cc cooling coil
ch chiller
comb combustion
comp compressor
el electricity
f fuel
g gas
o ambient
t turbine
v vapor

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