

Performance Analysis of Gas Turbine Inlet Air Cooling Plant with Hybrid Indirect Evaporative Cooling and Absorption Chiller System

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Abstract

To generate electricity, Gas Turbine Power Plants are extensively used around the world but their efficiency is negatively affected on hot days by ambient air temperatures. Efficiency and output power will decrease by 2% and 12.5% respectively at ambient temperature 40°C due to lower air density and the resulting increase in compressor specific work. Ordinarily, cooling down the intake air of the gas turbine is facilitated by employing a variety of Turbine Inlet Air Cooling Systems (TIACSs), depending on the plant's immediate weather conditions. This paper studies a new hybrid cooling method, combined cooling systems (Indirect Evaporative Cooling System (IECS) + Absorption Chiller), used for a 157 MW gas power plant located in Asalouyeh in southern Iran. The Performance and Thermo-Economic analysis revealed that the annual power output can be 62,090 MW, the net income generated by the increased electricity sales can be \$ 3,851,000 for a 60% period of the year while Payback Period is found to be a year and six months. In comparison with Direct Evaporative Cooling system (DEC), the increased power output due to using Direct Evaporative Cooling system (DEC) is 12,290 MW and the net income generated by the increased electricity sales can be \$ 116,899 for a 60% period of the year while Payback Period is approximately 8 months.

Keywords: Gas Turbine; Turbine Inlet Air cooling System (TIACS); Indirect Evaporative Cooling System (IECS); Hybrid Cooling System; Thermo-Economic Analysis.

1. Introduction

Gas turbines are combustion engines often used to generate electricity. Capacity (i.e. generated power output) and efficiency of gas turbines decrease as the ambient temperatures grow. These engines in these turbines always transport a fixed volume of air while the power output depends on the total mass of the passing airflow [1]. Energy demand growth is high during warm weather conditions, and unfortunately, generated power output in gas turbines drops by about 30%. More importantly, the efficiency can dwindle by 5% in all gas turbine systems during warm seasons [2]. Turbine Inlet Cooling (TIC) is a system of air conditioning for a gas turbine to maximize the gas turbine output across all ambient conditions and prevent loss of power output and energy efficiency reduction in all gas turbine systems [2].

This is to say that more electricity can be generated using less fuel while releasing fewer pollutants into the environment [3]. Turbine Inlet Air Cooling (TIC) technologies mainly involve the following methods [4]:

1. Direct Evaporative Cooling System
2. Spraying finely atomized water (fog) and fog production (Inlet air fogging)
3. Cooling system using mechanical chillers (vapour compression cycle) or Absorption chillers [4].

In Turbine Inlet Cooling (TIC), different methods can use depending on the indigenous weather conditions [4]. Each method is known by some advantages and disadvantages. [4,3]. For example, the fogging system has limited capacity and temperature of vapour depends on the power plant's location [5]. One of the major disadvantages of the Direct Evaporative Cooling (DEC) is its low efficiency and high water consumption. In addition, unevaporated water droplets in airflow can damage the compressor blades [6], or ice crystals can form on the surface of the compressor blades due to excessive cooling [6].

This paper introduces a cooling method (hybrid cooling system) for Asalouyeh Gas Power Plant (Iran) that consists of two cooling systems (IECS + Absorption Chiller) and that can result in relatively lower temperature and humidity. In hybrid cooling systems, the secondary airflow cooled by the DEC system enters the heat exchanger while the primary airflow cools down after crossing the secondary airflow and humidity is carried outside the system as well.

In another study, Marzouk and Hanafi [7] considered the direct evaporative evaporation (DEC) cooling system in Korymat power plant in south-eastern Egypt and showed that it is possible to obtain 27 ° C by solving equations with EES software. This could be used to increase the annual power output of gas turbines (GT) up to 86,118MW through turbine inlet evaporative cooling at an annual cost lower than that of

mechanical chillers. They estimated the Payback Period would be about 0.66 years.

Najjar and Abubaker [1] showed that it is possible to obtain temperatures below 22 °C by combining IECS system with either mechanical chillers or absorption chillers. They also found that the IECS system and the absorption chiller can be combined in hot and humid conditions to increase power output by 11.9% and efficiency by 98%.

Through a Thermo-Economic analysis, Najjar and Abubaker [1] showed a combination of IECS and absorption chiller systems can be highly costly while it offers a short Payback Period (this period is 1 year in case of hot and humid weather conditions).

Ehyaei et al. [8] did Energy, Exergy and Environmental (3E) analyses for a 4900KW absorption chiller at 159 MW Bushehr power plant-Iran and showed that applying a decrease from 37°C to 15°C during the hottest months of the year can increase the gas turbine power output from 137MW to 152MW. In Iran, gas power plants and Combined Cycle Gas Turbine Plants (CHPs) are responsible for generating about 70,228 MW of electricity. As such, developing a proper cooling method can significantly help to both boost energy production and reduce pollution [9]. The present study explores the performance of gas turbine (GT) when a hybrid inlet cooling system is used. The main focus of this study are as follows:

- An introduction of Asalouyeh Gas Power Plant.
- A Thermodynamic model of Asalouyeh plant developed by EES software.
- An introduction of direct evaporative cooling system and absorption chiller.
- An introduction of Indirect Evaporative Cooling System (IECS) and its performance.
- An introduction of combined indirect evaporative cooling system and absorption chiller system (IECS + Abs-Chiller).
- EES solution.
- A Thermo-Economic analysis of the hybrid cooling system.
- Validation.

1.1 The Influence of ambient temperatures on power output and efficiency in gas turbines

Gas turbines are ordinarily designed for ISO conditions which include ambient temperature (of 15 °C) and relative humidity of 60%. However, when the ambient temperature rises to 40 °C in summer, the GT efficiency is degraded to about 2% and the power output of 12.5% in comparison with ISO conditions [7]. The impact of rising ambient temperature on power output and thermal efficiency of a 157MW gas turbine is shown in Figure 1.

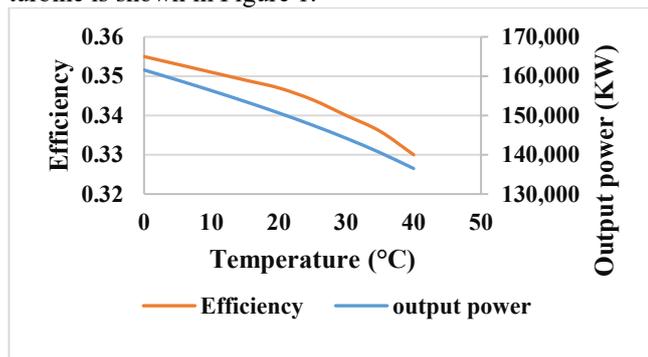


Figure 1. The impact of ambient air temperature on efficiency and power output in a gas turbine.

2. Asalouyeh Gas Power Plant

Asalouyeh Gas Power Plant is located 25 KMs south of Pars Special Economic Energy Zone (PSEEZ) in Bushehr, Iran. This plant is a gas-fired power installation with a power production strength of 942 MW in ISO conditions. It features 6×157MW gas units of V94.2: the first and second units were installed in 2007 and other units were installed in 2008. The plant is mainly powered by natural gas and by Diesel fuel as the substitute fuel. Further details and specifications are found in Table 1.

The design of Turbine Inlet Cooling System (TICS) depends on plant's local conditions. For instance, the design of a cooling system for hot and humid climates differs from that of hot and dry climates. Therefore, the climate conditions in any given region need to be studied and determine prior to any installation operations [7]. Accordingly, this study on Asalouyeh Gas Power Plant draws on Asalouyeh's climate conditions as reported by Iran's Meteorological Organization (national weather service) in 2018.

Table 1. Gas turbine design information and other specifications.

| Item | Rate |
|--|--------|
| Gag turbine output (ISO), MW | 157 |
| Air inlet temperature (ISO), °C | 15 |
| Relative humidity (ISO), % | 60 |
| Average air mass flow rate, kg/s | 504.40 |
| Ambient pressure, bar | 1.013 |
| Exhaust gases flow rate, kg/s | 513.52 |
| Heat rate, KJ/KWh | 10529 |
| Gas lower heating value, KJ/Kg | 48820 |
| Compression ratio | 10 |
| Inlet temperature to turbine, °C | 1052 |
| Fuel gas mass flow rate, kg/s | 8.802 |
| Efficiency, % | 34 |
| Compressor efficiency, % | 90 |
| Generator efficiency (η_{GE}), % | 98 |
| Turbine mechanical efficiency (η_{ME}), % | 98 |
| T13, °C | 32 |
| T17, °C | 10 |
| T18, °C | 5.5 |
| T15, °C | 32 |
| Exhaust gases temperature, °C | 538 |

The Figure 2 shows 3-hour temperature information units during a day per month over a period of one year. According to Figure 2, the lowest temperature was observed in March (27°C) while the highest temperature belonged to July (35°C). In addition, Figure 2 shows the average temperature for warm months of the year (31°C). The Figure 3 shows the relative air humidity over a 3-hour period per day per month for one year. Accordingly, the lowest relative humidity occurred in May (47%) while the highest humidity belonged to August (69%). The average relative humidity for the warm months of the year was 58%.

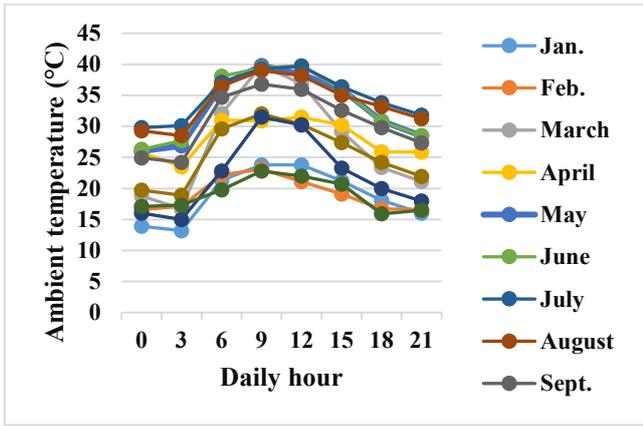


Figure 2. 3h temperature measured over a year.

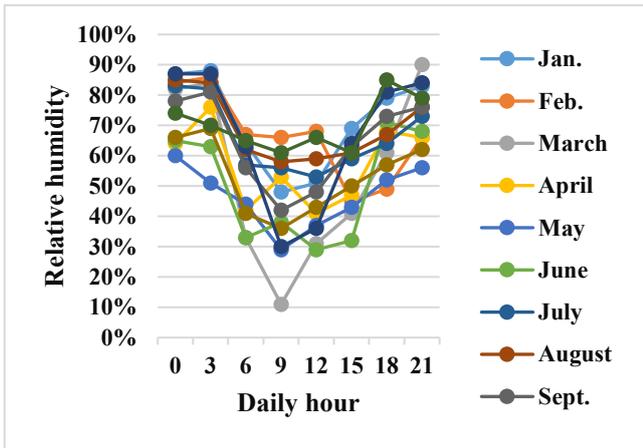


Figure 3. 3h relative humidity measured over a year.

3. Thermodynamic Model of Gas Turbine

Gas Turbines (GT) in power plants are made up of four parts: compressor, Combustion Chamber (CC), turbine and generator. Figure 4 presents a schematic diagram of a simple gas turbine. As shown in Figure 4, fresh atmospheric air is drawn into the circuit continuously and energy is added to the working fluid through the combustion process. The products of combustion expand down to the turbine, producing a shaft work output in the process. Combustion gases are eventually discharged through the exhaust into the atmosphere [7].

Efficiency in compressor and turbine is shown by η_C and η_T , respectively. The ideal process and actual process are shown in the temperature-entropy diagram in black and dashed lines, respectively [7], (Figure 5).

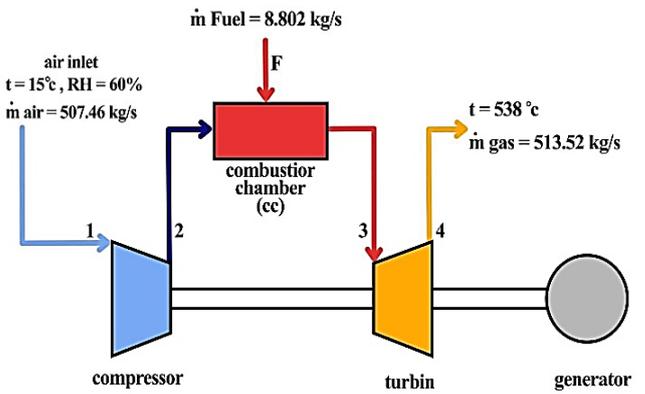


Figure 4. Components in a Gas Turbine.

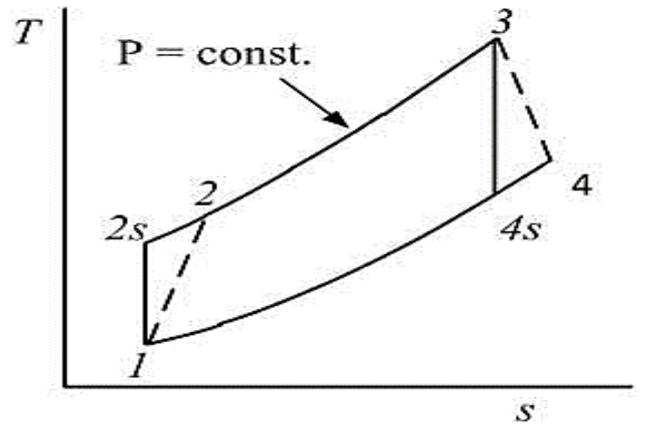


Figure 5. T-S diagram for the process in a Gas Turbine [7].

According to the diagram in Figure 5, the compression ratio in the compressor can be expressed by the following equation [7]:

$$R_p = \frac{P_2}{P_1} \quad (1)$$

Isentropic efficiency in compressor is approximately 80% to 90%.

$$\eta_C = \frac{T_{2s} - T_1}{T_2 - T_1} \quad (2)$$

The exhaust air temperature in the compressor can be expressed by the following equations:

$$T_2 = \frac{T_1}{\eta_C} \cdot \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right] + T_1 \quad (3)$$

or

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma_a - 1}{\gamma_a}}$$

The compressor work can be expressed in the following relationship:

$$W_C = \dot{m}_a \cdot C_{Pa,avg} \cdot (T_2 - T_1) \quad (4)$$

Where \dot{m}_a is the inlet to the compressor and $C_{Pa.avg}$ is the specific heat of dry air at constant pressure. The energy balance in the combustion chamber can be expressed in the following equation:

$$\dot{m}_a \cdot C_{Pa.avg} \cdot T_2 + \dot{m}_f \cdot LHV = (\dot{m}_a + \dot{m}_f) \cdot C_{Pg.avg} \cdot T_3 \quad (5)$$

Where \dot{m}_T represents the total mass flow rate obtained by combining airflow mass and fuel mass.

$$\dot{m}_T = \dot{m}_a + \dot{m}_f \quad (6)$$

The temperature of the turbine exhaust gas is calculated by the following equation:

$$T_4 = T_3 \cdot \left[1 - \eta_T \cdot \left(1 - \frac{1}{R_p^{\frac{\gamma_a - 1}{\gamma_a}}} \right) \right] \quad (7)$$

or
 $T_4 = T_3 \cdot (1 - \eta_T) \cdot R_{P.g}$

or
 $T_4 = T_3 - \eta_T \cdot T_3 \cdot \left[1 - \left(\frac{p_3}{p_4} \right)^{\frac{\gamma_g - 1}{\gamma_g}} \right]$

and also:

$$R_{P.g} = \left(1 - \frac{1}{R_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right)$$

Where η_T is the isentropic efficiency of the turbine and P_4 is the pressure of the turbine exhaust gas.

The work in the turbine is calculated by the following equation:

$$W_T = C_{Pg.avg} \cdot (T_3 - T_4) \quad (8)$$

or
 $W_T = C_{Pg.avg} \cdot T_3 \cdot \eta_m \cdot \frac{R_{P.g}}{\eta_m}$

Where $C_{Pg.avg}$ represents the (average) specific heat of the exhaust gas passing through the turbine. The air-fuel ratio γ is calculated from the following equation:

$$\gamma = \frac{\dot{m}_a}{\dot{m}_f} \quad (9)$$

The net output work is obtained from the following equation:

$$W_{net} = W_T - W_C \quad (10)$$

Here, the isentropic turbine efficiency is about 93%.

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4S}} \quad (11)$$

The specific fuel consumption can be expressed in the following equation:

$$SFC = \frac{360^\circ}{W_{net}} \quad (12)$$

HR heat rate is the next parameter for the gas turbine and is expressed in the following equation:

$$HR = SFC \cdot LHV \quad (13)$$

or
 $HR = \frac{3600}{\eta_{th}}$

The heat added to the combustion chamber is calculated in the following equation:

$$Q_{add} = C_{Pg.avg} \cdot (T_3 - T_2) \quad (14)$$

The thermal efficiency of a gas turbine is expressed in the following equation:

$$\eta_{th} = \frac{3600}{SFC \cdot LHV} \quad (15)$$

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

The turbine's power output is expressed in the following equation:

$$P(MW) = \eta_{GE} \cdot \eta_{ME} \cdot \dot{m}_a \cdot W_{net} \quad (16)$$

or
 $P(MW) = (\dot{m}_a + \dot{m}_f) \cdot W_{net}$

Where η_{GE} is generator efficiency (about 98%) and η_{ME} is a turbine's mechanical efficiency (about 98%).

Further, the difference between the power output values in presence of cooling system and in absence of cooling system is calculated by the following equation:

$$\Delta P = P_{with \text{ cooling system}} - P_{without \text{ cooling system}} \quad (17)$$

The Table 2 shows the characteristics of some of the variables used in this paper. The Table 3 shows the input parameters of the system which are essential for calculation by EES software.

Table 2. Specification of some variables.

| Variable Title | Variable type | Measurement unit |
|--------------------------|---------------|------------------|
| Pressure (P) | independent | bar |
| Temperature (T) | independent | K |
| Relative humidity | independent | % |
| Humid ratio (ω) | dependent | - |
| Entropy (s) | dependent | KJ/Kg-k |
| Dewpoint (dp) | dependent | K |
| Enthalpy (h) | dependent | KJ/Kg-k |

Table 3. Input parameters.

| Input parameters | ISO condition | During %60 of a year (average) |
|-----------------------|---------------|--------------------------------|
| T (°C) | 15 | 31 |
| P (bar) | 1/013 | 1/013 |
| Relative humidity (%) | 60 | 58 |

4. Direct Evaporative Cooling System and Absorption Chiller

4.1 Direct evaporative cooling system (media)

The direct evaporative cooling system (media) is very suitable for hot and dry weather because it cools ambient air temperature through latent evaporative heat: Temperature of Inlet Air is increased from the temperature of Dry Bulb Temperature of Inlet Air to Wet Bulb Temperature of Inlet Air. Through the procedure used in this cooling method, the sensible heat is converted to latent heat. Thus, the ambient air is cooled down as water evaporates from the wet media surface into the system [10]. This procedure is shown in Figure 6 along with a psychrometric diagram.

After the cooling process in the media, inlet air temperature is calculated by the following equation [7, 11]:

$$T_{a2} = T_{a1} - \epsilon(T_{a1} - T_{wb}) \quad (18)$$

Where T_{a1} is dry Bulb Temperature of Inlet Air, T_{wb} is the Wet Bulb Temperature of Inlet Air and ϵ is the cooler effectiveness which is assumed to be 90% in this paper.

The Cooling Load of the direct evaporative cooling (DEC) system is calculated by the following equation [11]:

$$\dot{Q}_{CL} = \dot{m}_a \cdot C_{p,avg} \cdot (T_{a1} - T_{a2}) \quad (19)$$

Where \dot{m}_a is the Air Flow Rate and $C_{p,avg}$ is the specific heat of the dry air with a constant pressure. In addition, the water vapour mass is calculated from the following equation [7]:

$$R = \frac{V(\omega_2 - \omega_1)\rho_a}{\rho_w} \quad (20)$$

Where v is water volume (m^3/s) and ρ_{air} is dry air density and ρ_w is water density.

The blow down rate can be determined from the blow down ratio which is [7]:

$$E = \frac{\text{The blow down rate}}{\text{evaporation rate}} = \frac{B}{R} \quad (21)$$

This ratio is calculated according to the E curve of water hardness. In this study, water hardness is assumed to be 150 PPM while the E ratio is assumed to be 4.

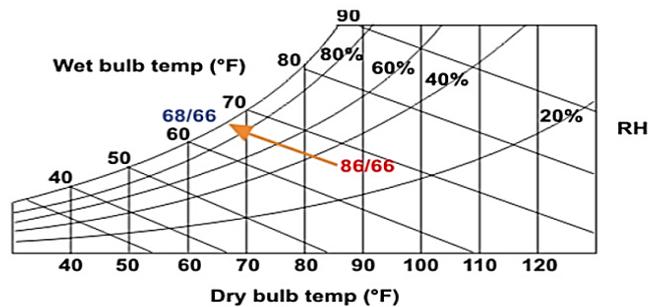
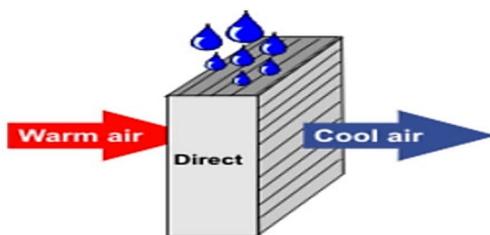


Figure 6. A Schematic representation of a Heat Exchanger in Direct Evaporative Cooling (DEC) system and its Psychrometric Diagram [10].

$$B = 4 \cdot R \quad (22)$$

Total water consumption Q_w is the sum of evaporated water and blow down rate [7]:

$$Q_w = B + R \quad (m^3/s) \quad (23)$$

4.2 Absorption Chiller

The absorption chiller is another way for the Gas Turbine Inlet Cooling (TIC). In this cooling method, the absorption chiller uses the exhaust heat from the Gas Turbine (GT). The atmosphere is cooled when the cooled water passes through the heat exchanger. The Figure 7 shows the structure of the heat exchanger in a chiller system and a psychrometric diagram [11].

The cooling load that is excluded from the airflow at the ambient conditions of the power plant can be calculated using the first law of thermodynamics [11].

$$\dot{Q}_{CL} = \dot{m}_a \cdot [(h_1 - h_2) - h_{fg} \cdot (\omega_1 - \omega_2)] \quad (24)$$

Where \dot{m}_a represents the airflow rate, h_{fg} is the latent heat of evaporation, and ω_1 and ω_2 show the inlet and outlet (the humidity ratio) of the evaporator system. One advantage of the absorption chiller cooling system is that it is not limited to ambient air temperatures [11].

5. Indirect evaporative cooling system (IECS)

An Indirect Evaporative Cooling System (IECS) is a relatively new method for Gas Turbine Inlet Cooling (GTIC) [1]. In fact, this method is a direct evaporative cooling system that uses evaporation for cooling in a way that no moisture is released into the gas turbine inlet (compressor) [12], thus protecting compressor blades and Combustion Chamber (CC) against possible damages due to moisture [13]. The Figure 8 shows an overall schematic representation of the indirect evaporative cooling system along with its psychrometric diagram. 8 shown.

In fact, an indirect evaporative cooling system is made up of a Direct Evaporative Cooling system (DEC) and an air-to-air heat exchanger.

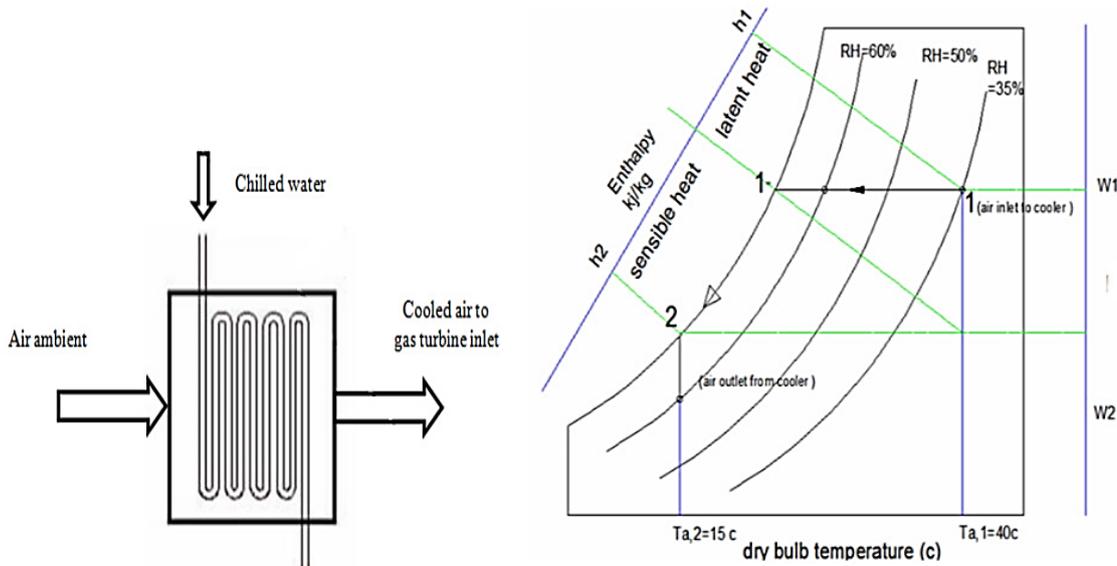


Figure 7. A Schematic Representation of Cooling Coil in a Chiller Cooling System and its Psychrometric Diagram [10].

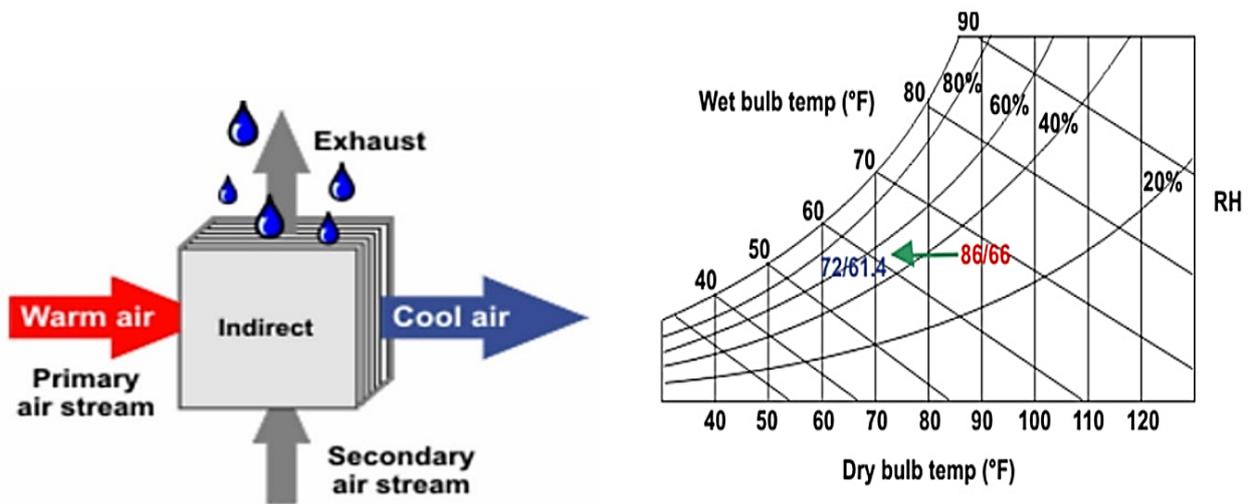


Figure 8. Indirect evaporative cooling system (IECS) with its psychrometric diagram.

5.1 Indirect Evaporative Cooling System (IECS) Performance

In an IECS, an airflow (called the primary airflow) is considerably cooled by a heat exchanger (no moisture added). The secondary airflow gets cooled by a Direct Evaporation Cooling (DEC) and then is directed to a heat exchanger in order to remove thermal energy from the primary air flow [12]. This air-to-air heat exchanger is used to cool the primary airflow by a Secondary airflow that has been already cooled. In this way, the primary airflow that exits the heat exchanger is a cooled flow free of moisture that can be used as the turbine inlet air. The Figure 9 shows a schematic representation of an IECS system performance [12].

In an IECS system, the heat exchanger can have two structures: tubular or plate. In an IECS system, both wet bulb and dry bulb are cooled. As mentioned above, no moisture is applied to the compressor for Turbine Inlet Cooling (TIC) in

an indirect evaporative cooling. However, IECS is costlier than a direct evaporation system (DEC). IECS efficiency is 40% up to 80% [12].

The temperature of the secondary airflow that is emitted from a DEC system is obtained from the following equation (Figure 9) that can be solved using EES software [12].

$$T_{S,2} = T_{S,1} - \eta_{DEC} \cdot (T_{S,1} - T'_{S,1}) \quad (25)$$

Where $T_{S,1}$ shows the Dry Bulb Temperature of Inlet Air in the DEC system, $T'_{S,1}$ is the Wet Bulb Temperature of outlet Air from the DEC system and $T_{S,2}$ is the Dry Bulb Temperature of outlet Air from DEC system.

The primary air temperature at the heat exchanger outlet is calculated from the following equation [12].

$$T_{P,2} = T_{P,1} - \eta_{HX}(T_{P,1} - T_{S,2}) \quad (26)$$

That is solved using EES software. Here, $T_{P,1}$ shows the Dry Bulb Temperature of Inlet Air at heat exchanger, $T_{P,2}$ represents Dry Bulb Temperature of outlet Air at the heat exchanger (compressor feed flow), $T_{S,2}$ shows the Dry Bulb Temperature of Inlet Air at the heat exchanger (DEC system outlet) and η_{HX} is the heat exchanger effectiveness.

6. Combining the indirect evaporative cooling system and lithium bromide/water system

6.1 The types of hybrid cooling systems are [1]:

- A combination of the IECS and Vapour - compression system (mechanical chiller).
- A combination of the IECS and absorption chiller system.

This paper is primarily focused on a hybrid cooling system (a combination of IECS and absorption chiller) and how it affects the efficiency and the power output of the gas turbine at Asalouyeh Gas Power Plant while offering an economic analysis of this installation.

Environmental conditions have a smaller impact on a combined IECS and absorption chiller system. However, the best environmental condition for such a system is warm and humid climates [1].

6.2 Hypotheses for a combined IECS and absorption chiller system

- The thermal energy required in a generator in the absorption chiller is supplied by the turbine outlet gases after heat is effectively reduced using a Heat Recovery Steam Generator (HRSG).
- Evaporator effectiveness is 0.9.
- The evaporator outlet water temperature at the absorption chiller is around 6°C.

6.3 Performance of the indirect evaporative cooling system combined with an absorption chiller

The secondary cooled airflow emitted by the Direct Evaporative Cooling (DEC) system enters the heat exchanger (HX). At the same time, an atmospheric airflow (as a primary airflow) enters the heat exchanger. A collision between the primary airflow and the secondary airflow leads to removal of heat and moisture from former and the latter, respectively. As a result, the outlet flow of this (IECS) is cooled and less humid than the one coming from a Direct Evaporative Cooling (DEC) system. At this point, the temperature and humidity of this airflow are reduced after contacting the evaporator extracted from the absorption chiller system. This airflow can then be used to supply a gas turbine compressor. Figure 10 is a schematic representation of combined IECS and absorption chiller (i.e. hybrid cooling system).

6.4 Psychrometric Diagram

Point P1 is the entry to the psychrometric diagram (Figure 11) with mean values for temperature and humidity determined. The red arrow marks the Direct Evaporation Cooling (DEC) system with a media system for cooling the inlet airflow. As moisture leaves the media surface, the cooling effect occurs along the constant enthalpy line. The relative humidity is 95%, at S2 and humidity is emitted from S2 to S3 on the grey dashed line. The dark blue arrow represents the cooling action in primary airflow from P1 to P2. This is in fact the onset of the cooling operation in the heat exchanger (HX) with a constant humidity ratio and relative humidity of 58%. The light blue arrow illustrates the decrease in temperature that happens along with the decrease in humidity ratio. At P3, this cooled airflow enters the gas turbine compressor.

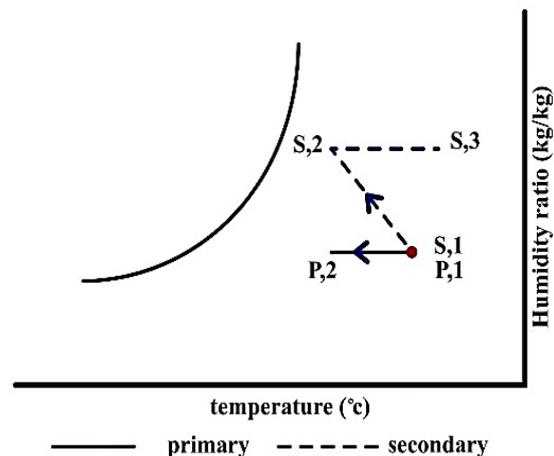
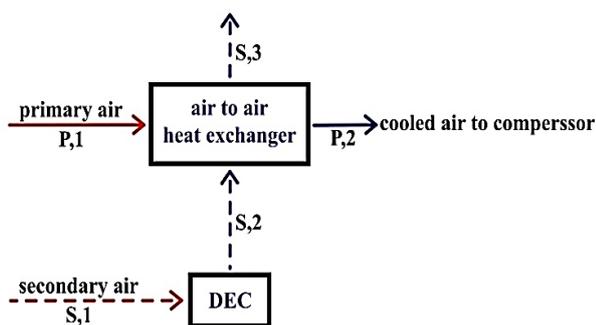


Figure 9. Schematic representation of the performance in an indirect evaporative cooling system (IECS) with a psychrometric diagram.

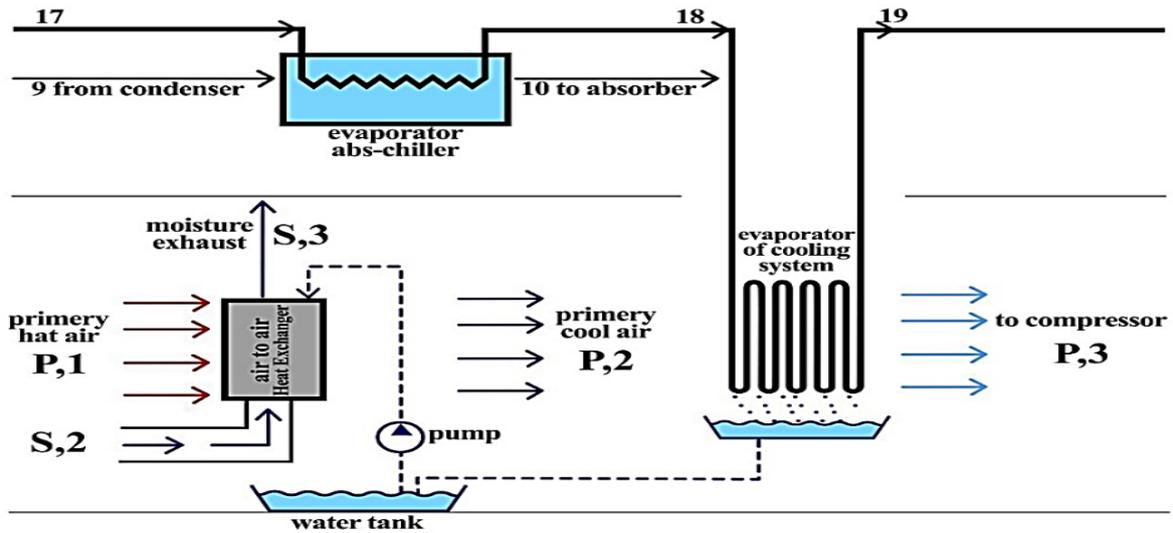


Figure 10. A combined IECS and absorption chiller system.

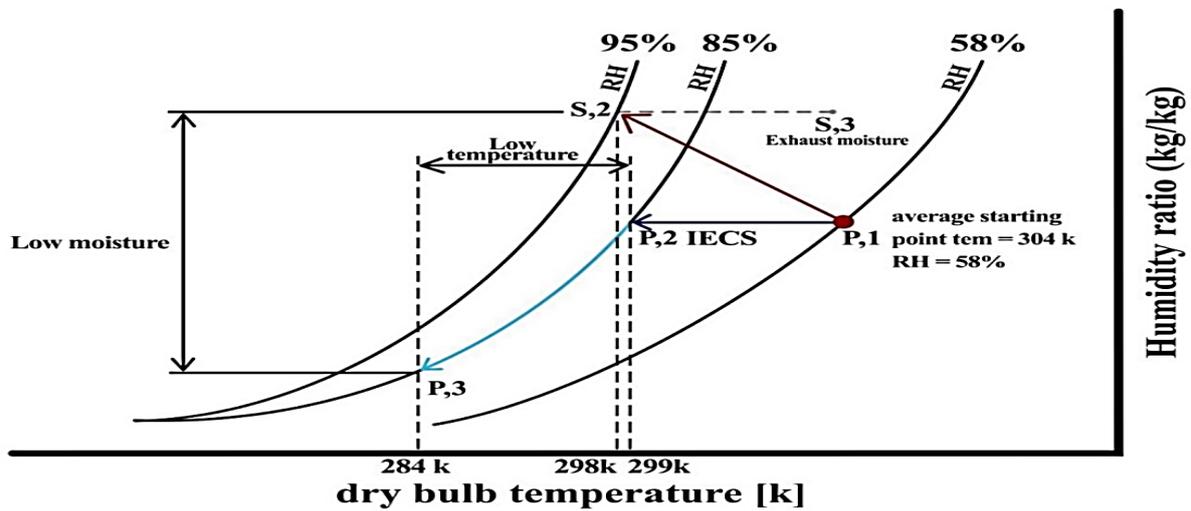


Figure 11. Psychrometric chart of cooling the inlet in a Hybrid system.

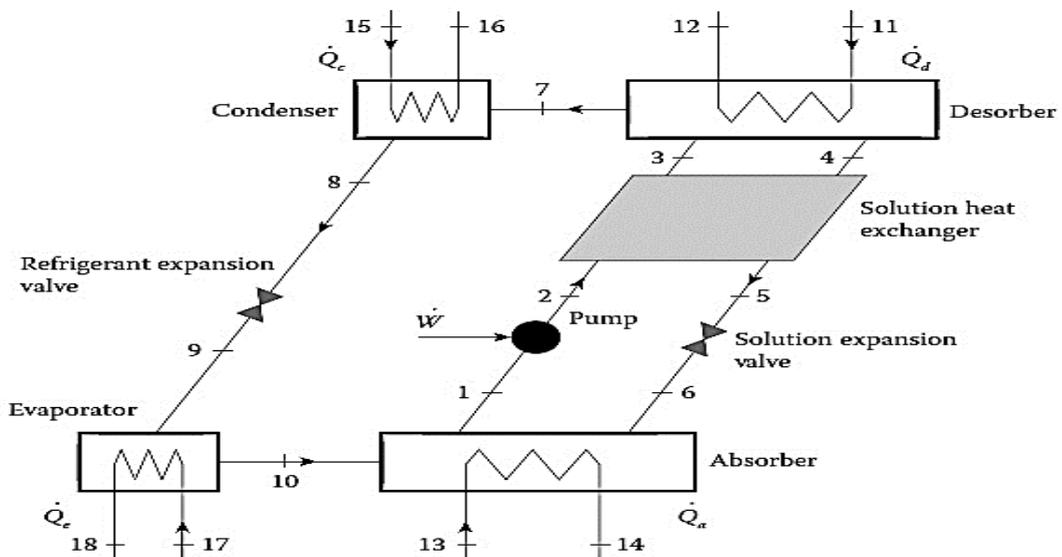


Figure 12. A schematic representation of a simple cycle in an absorption chiller [17].

6.5 A Theoretical Analysis of the Absorption Chiller System (Figure 12):

Evaporator:

$$\dot{Q}_{evap} = \dot{m}_{ref} \cdot (h_{10} - h_9) \quad (27)$$

$$\dot{Q}_{load} = \dot{Q}_{evap} \quad (28)$$

$$\dot{m}_{ref} = \frac{\dot{Q}_{load}}{h_{10} - h_9} \quad (29)$$

Generator:

Equations for energy balance in generator:

$$\dot{M}_{weak} = \frac{x_4 - x_7}{x_4 - x_3} \cdot \dot{m}_{ref} \quad (30)$$

$$\dot{Q}_G = \dot{M}_{weak} \cdot (h_4 - h_3) + \dot{m}_{ref} \cdot (h_7 - h_4) \quad (31)$$

Condenser:

$$\dot{Q}_{cond} = \dot{m}_{ref} \cdot (h_7 - h_8) \quad (32)$$

Absorbent:

$$\dot{Q}_{abs} = \dot{M}_{weak} \cdot (h_6 - h_1) + \dot{m}_{ref} \cdot (h_{10} - h_6) \quad (33)$$

And pump:

$$\dot{W}_{pump} = \dot{M}_{weak} \cdot (P_2 - P_1) \cdot v_1 \quad (34)$$

7. EES solution

The gas turbine model was solved using EES software. At the beginning of modeling the desired Gas turbine, the power production of the gas turbine was simulated in ISO conditions to evaluate the validation of solving equations. In standard state, the output power of software result was closed to ISO condition of gas turbine specifications. Average annual temperature and relative humidity of Assaluyeh entered the software, the result of simulation shows the output power of the gas turbine without the cooling system. The relative humidity and temperature gained from the simulation of the hybrid cooling system were replaced and the result was the output power of the gas turbine using the hybrid cooling system.

8. A Thermo-Economic Analysis of Hybrid Cooling System (IECS + Abs-Chiller)

In the case of using a hybrid cooling system to cool gas turbine inlet (TIC), the length of Recovery Period can be calculated. First, total investment should be estimated. For an IECS system, the total investment cost is \$33KW⁻¹ for

expanding power output while the total investment cost for an absorption chiller system is roughly \$477KW⁻¹ [1].

8.1 Extra fuel consumption cost

In case of base conditions, fuel consumption at a power plant is 8.802 kg/s and in case of power increase using a cooling system, fuel consumption is 8.907 kg / s [14, 15]. 1m³ of natural gas is 1000 Rials in Iran for power plant consumption purposes.

$$\begin{aligned} \text{fuel consumption in base condition} \\ &= 8.802 \text{ kgs}^{-1} \times 3600 \text{ sec} \\ &= 31687.2 \text{ kgh}^{-1} \end{aligned} \quad (35)$$

$$\begin{aligned} \text{fuel consumption with cooling system} \\ &= 8.907 \text{ kgs}^{-1} \times 3600 \text{ sec} \\ &= 32065.2 \text{ kgh}^{-1} \end{aligned} \quad (36)$$

$$\begin{aligned} \Delta_{gas \text{ consumption}} &= 32065.2 - 31687.2 \\ &= 378 \text{ kgh}^{-1} \end{aligned} \quad (37)$$

Where $\Delta_{gas \text{ consumption}}$ is the gas consumption per hour in an absorption chiller system. When converted to cubic meters, this amount becomes 527.932 m³h.

If 60% of the cooling system remains in operation during a year (or 5256 hours), the gas consumption cost will be about 2,774,810,592 Rials or USD23,123.42.

8.2 Operation maintenance costs

For an IECS+Absorption Chiller system, the Operation Maintenance costs are:

2% of the entire investment cost in an absorption chillers system [16] and 4% of the entire investment cost in a direct evaporation system [7].

8.3 Water Consumption

The amount of evaporated water in a direct evaporative cooling system as well as the total water consumption is 0.1592 m³/s or 573.3 m³/h. The price of water consumed in plants is 4000 Rials for one cubic meter. Thus, the cost of water consumed during the operating period of the cooling system is \$ 100,433.3 [7].

The annual income generated from the increased electricity sales can be calculated from the following equation. The unit price of increased electricity sales (C_{els}) is estimated to be between 0.07 and 0.15.

$$\begin{aligned} \text{Annual saving power augmented} \\ &= (P_{with \text{ cooling}} - P_{without \text{ cooling}}) \\ &\quad \cdot C_{els} \end{aligned} \quad (38)$$

The Payback Period is calculated from the following equation:

Payback Period for IECS + Absorption chiller (year)

$$= \frac{\text{Capital cost of IECS} + \text{Capital cost of Absorption Chiller}}{\text{annual saving power augmented} - (\text{annual increased fuel consumption} + \text{annual O\&M cost} + \text{annual water cost})} \quad (39)$$

Table 4. Validation table.

| Parameters | Najjar and Abubaker [1] (for IECS+Absorption Chiller) | Marzouk and Hanafi [7] (for evaporative cooling system) | Current manuscript (for IECS+Absorption Chiller) | Current manuscript (for evaporative cooling system) |
|--|---|---|--|---|
| payback period | 1/079 (A Year) | 0/66 (8 months) | 1/59 (A Year and six months) | 0/692 (approximately 9 months) |
| Average power increasing | 2/5% | 6/84% | 6/72% | 1/60% |
| Average efficiency increasing (η) | 2% | 1/5% | 2/4% | 2/94% |

9. Validation

In order to validate the modeling results of the intended gas turbine performance analysis, the results of some published papers have been considered. For this purpose, the results of Najjar and Abubaker [1] and Marzouk and Hanafi [7] have been used to compare and validate the current manuscript results.

Najjar and Abubaker [1] showed that using the IECS + absorption Chiller system during hot and humid climate the average power increasing was about %2.5 and Payback period achieved a year, on the other side the average efficiency of gas turbine was thrived %2.

In comparison with the current manuscript with using IECS + Absorption chiller system during hot and humid climate the average power increasing is about %6.72 and payback period is estimated about a year and six months, average efficiency increasing calculated approximately %2.4.

Marzouk and Hanafi [7] in their research revealed obviously through the evaporative cooling system during hot and humid climate in Egypt the average power and efficiency increasing was respectively %6.84 and %1.5. The payback period was evaluated for 8 months.

In opposite in this manuscript average power and efficiency increasing through evaporative cooling system during hot and humid climate are respectively %1.60 and %2.94, the payback Period is calculated about 9 months. The Table 4 shows the some prominent parameters that compared.

10. Conclusion

If a hybrid cooling system is used for 60% of a year (i.e. 5256 hours), the increased power output due to the hybrid cooling system is 62090 MW approximately. As seen in the linear diagram in Figure 13, the greatest increased output in the gas turbine occurred in July (10.6%) while the increased output in January is (3.57%). The average increased output over a year is %6.72. As shown in the linear diagram in Figure 14, the highest increase in thermal efficiency of gas turbine occurred in July (4.49%) while the lowest efficiency was observed in February (1.15%). In addition, the average increase in thermal efficiency is 2.4%.

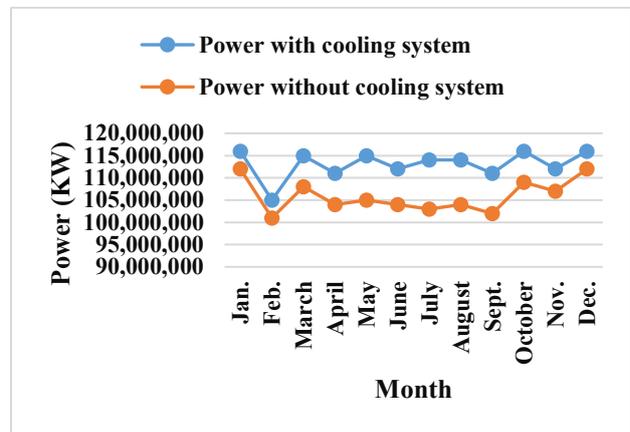


Figure 13. Power output with/without a hybrid cooling system.

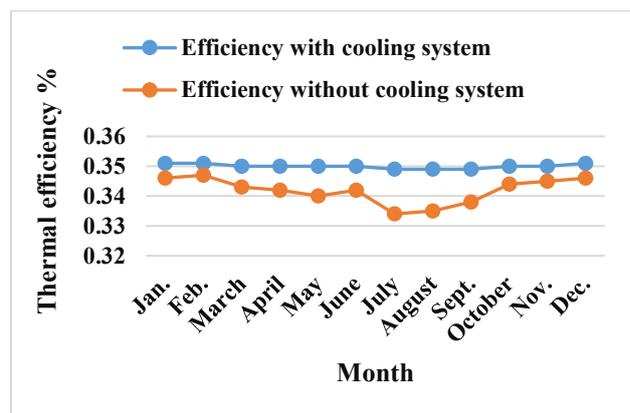


Figure 14. Thermal efficiency with/without a hybrid cooling system.

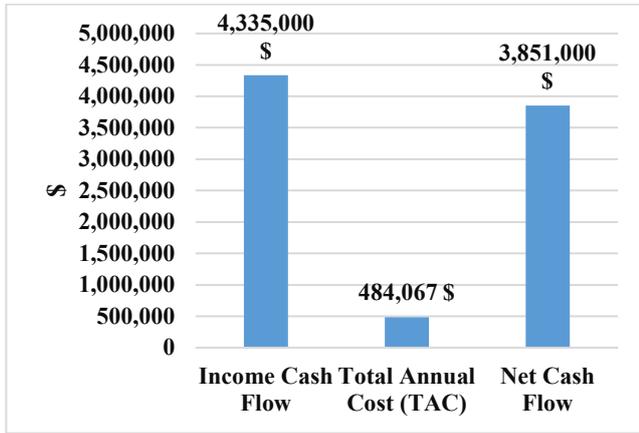


Figure 15. Annual income, Cost, and net Income.

This study of the Asalouyeh Gas Power Plant (25 km south of Pars Special Economic Zone, Bushehr, Iran) showed that combining IECS and absorption chiller system with temperature and mean relative humidity of (inlet) 31°C and 58% resulted in (outlet) 11°C and 85%, respectively. The average increase in power output over a year is 6.75%. Besides, Thermo-Economic analysis showed that using a Hybrid Turbine Inlet Cooling system (TIC) for 60% of the year would increase income from selling the increased electricity generated by the use of such a cooling system. According to Figure 15, this increase is \$4,335,000 while the total annual cost is \$484,067. Thus, total annual net income is \$ 3,851,000 and the Recovery Period is a year and six months, a significantly shorter period when compared to the Payback Period in case of combined IECS and vapour compressor chillers. Also, fuel consumption (natural gas) for generating this increased amount of power output due to the cooling system is much less than the fuel consumption for generating the same amount of power output without the cooling system.

In comparison with Direct Evaporative Cooling system (DEC), the increased power output due to using the DEC system is 12,290 MWh and the net income generated by the increased electricity sales can be \$ 116,899 for a 60% period of the year while Payback Period estimated at 8 months.

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Nomenclature

| | |
|-----------|---|
| avg | Average |
| B | Blow Down Rate, (m^3/s) |
| CC | Combustion Chamber |
| C_{els} | Cost of selling excess electricity, (estimated between 0.07 – 0.15), (\$/kwh) %60 a year = 5256, (hours) |
| CHP | Combine heat and Power |
| CL | Cooling Load |
| DEC | Direct Evaporative Cooling |
| E | Blow Down Ratio |
| EES | Engineering Equation Solver |
| GTIC | Gas Turbine Inlet Cooling |
| GT | Gas Turbine |
| h_{fg} | latent heat of vaporization of water, (KJ/Kg) |
| h_1 | Ambient Air Enthalpy At Cooler Inlet, (KJ/kg-k) |
| h_2 | Ambient Air Enthalpy At Cooler Outlet, (KJ/kg-k) |
| HX | Heat Exchanger |
| Hybrid | combination of two different things |

| | |
|-------------|--|
| IECS | Indirect Evaporative Cooling System |
| ISO | International Organization for Standardization |
| LHV | Lower Fuel Heat Value, (KJ/kg) |
| \dot{m}_a | Air Mass Flow Rate At Inlet Of Cooler, (kg/s) |
| \dot{m}_f | Fuel Mass Flow Rate Without Cooling, (kg/s) |
| \dot{m}_T | Total mass flow rate, (kg/s) |
| P,1 | primary air of Heat Exchanger |
| P,2 | primary cooled air from Heat Exchanger |
| P,3 | final cooled air stream by Evaporator of cooling system |
| Δp | Difference Between Gas Turbine Output (without and with cooling), (MW) |
| Q_{add} | Heat Added To Combustion Chamber, (KJ/Kg-k) |
| \dot{Q} | Heat Transfer rate, (KW) |
| Q_w | total water consumption, (m^3/s) |
| R | Evaporative Cooler Vaporization rate, (m^3/s) |
| RH | Relative Humidity, (%) |
| Rial | National currency of Iran |
| SFC | Specific Fuel Consumption, (kg/kwh) |
| S,1 | secondary air of DEC system |
| S,2 | secondary cooled air from DEC system |
| S,3 | exhaust moisture from Heat Exchanger |
| TAC | Total Annual Cost, (\$) |
| TIACS | Turbine Inlet Air cooling System |
| TIC | Turbine Inlet Cooling |
| T_{a1} | Ambient Air Temperature At Inlet, (°C) |
| T_{a2} | Ambient Air Temperature At Outlet, (°C) |
| T_{wb} | Wet Bulb Temperature Of Inlet Air, (°C) |
| V | Air 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) |
| X | mass fraction LiBr (%) |

Greek symbols

| | |
|---------------|---------------------------------------|
| ε | Evaporative Cooler Efficiency |
| η_C | Compressor isentropic efficiency, (%) |
| η_{GE} | Generator Efficiency, (%) |
| η_{ME} | Turbine Mechanical Efficiency, (%) |
| η_{th} | Gas turbine thermal efficiency, (%) |
| ρ_a | Air Density, (kg/m^3) |
| ρ_w | Water Density, (kg/m^3) |

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