International Journal of Heat and Technology
Vol. 43, No. 1, February, 2025, pp. 191-201
Journal homepage: http://iieta.org/journals/ijht

Investigation of How Cooling the Compressor Inlet in a Gas Turbine Cycle with Thermal Storage Affects the Cycle's Thermal Performance

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https://doi.org/10.18280/ijht.430120

Received: 20 December 2024
Revised: 1 February 2025
Accepted: 17 February 2025
Available online: 28 February 2025

Keywords:
exergy, economic cost, thermodynamics, ice thermal energy storage, objective function

ABSTRACT

The limiting of available energy resources prompted many researches to seek for the best investment of this resources as can as possible for best usage of this resources and converting them to electrical form with lowest possible cost. Electrical power plant working with gas turbine cycle condition closely related with ambient conditions, especially ambient temperature. As ambient temperature decrease, the thermal efficiency increase. This research aims to study the effect of cooling the air entering the compressor by a thermal ice tank that is charged (freezing and cooling the ice) at night by a compression refrigeration cycle due to the presence of an energy surplus from the plants and investing it during the day in cooling the air entering the compressor on the thermal performance of the plants and studying the economic feasibility of this addition in gas turbine power plants. The study focused in the current research on determining the optimal operational parameters that achieve the best thermal efficiency and the lowest possible operating cost through saving the value of the fuel cost added to the plants in order to obtain the same capacity. The study was conducted on the Jandar station in the Syrian city of Homs with a nominal capacity of 100 MW. The study showed an increase in the value of the thermal efficiency when cooling the air by means of a thermal ice tank by 3.5% and an exergy efficiency by 1.96%, while the operating cost decreased by 0.23% as a result of a decrease in the cost of the fuel consumed by 3.5%. The study was conduct by EES (Engineering Equation Solver) software version 9.9.

1. INTRODUCTION

Combined and gas plants are widely used to produce electricity, and they are machines that work directly with the surrounding air, so their performance is affected by several factors, including temperature, humidity, and pressure [1]. Therefore, any change in the condition of the air entering the compressor affects its efficiency. The cooling of the air entering the compressor increases the density of the air and thus improves the operating conditions of the gas station, and this in turn causes an increase in the thermal efficiency of the plant. The growing concern about the limited energy resources prompted many researchers to change the specificity of the field related to energy production and the disposal of waste resulting from the energy generation process, and prompted the scientific community to adopt a deeper study with regard to the systems used in energy conversion and the development of newer technologies that allow the investment of energy resources. at its best. The concept of exergy expresses the amount of useful energy that can be fully utilized [2]. It is a state function that describes the state of the system and was formulated mathematically based on the first and second law of thermodynamics and based on the concept of irreversibility that is associated with the generation of entropy in actual operations. The importance of the previous exergy analysis is not limited to raising the performance of thermal systems due to raising the efficiency of energy transformation processes only, but also includes the best economic investment and environmental aspects related to reducing emissions and their impact on the studied processes [3]. When studying the exergy analysis on thermal systems, this study will tell us where the greatest waste of energy is. This analysis also aims to study the economic cost of the destructive energy by linking the value of the destructive exergy to a proportionate cost function [4]. The economic exergy analysis consists of three stages, including the exergy analysis, the thermoeconomic analysis, and the exergoeconomic evaluation. Based on several studies, it was found that the ambient temperature is the most important factor that affects the performance of the combined and gas plants, as the higher the temperature of the air entering the compressor, the plant's energy production decreases and the thermal efficiency decreases [5].

At the time of the night, we notice that the power outages are almost non-existent, and the reason for this is due to the improvement in the efficiency of the gas and combined plants as a result of the decrease in ambient temperatures, but in this period the demand for electricity is very little, due to the cessation of the activities of individuals [6].

Therefore, it was necessary to find a way to exploit the surplus electricity capacity at the time of the night (peak off period) in order to benefit from it during the daytime hours (peak period) when the demand for electricity is at its peak. For this purpose, mechanical coolers and thermal energy storage (TES) were used (temporary storage of electrical energy in the form of low temperature for use when needed) [7]. This research aims to study the economic feasibility of transferring the surplus of electrical energy at late night times (when the activity of individuals is almost non-existent) to the daytime period (when the demand for electricity is at its peak) by using a cooling system equipped with a TES that contains ice water to keep the cold In it at peak off times and cooling the air entering the compressor at peak times in order to raise the thermal efficiency of gas and combined power plants. Its importance lies in striving towards investing the available energy resources and employing them so that they can be utilized to the maximum extent possible while reducing the required operating cost as much as possible.

A large number of researchers applied the equilibrium equations for exergy and energy to the cycle of gas turbines, and they reached similar results. The study differed in the areas of operating conditions and station capacity. Ibrahim et al. [1] developed a mathematical model for the components of the gas turbine, the destructive exergy is at its greatest value in the combustion chamber as a result of the irreversible processes that occur as a result of heat transfer to the working medium. Ahmed et al. [2] studied the analysis of exergy and power on Taza gas turbine in Iraq with a capacity of 70 MW under different operating conditions. the largest part of the exergy is destroyed in the combustion chamber and the least in the compressor. Salah et al. [3] studied a gas turbine plants at a load of 255 MW. The efficiency increases with the decrease of ambient temperature, while the efficiency decreases with the increase in the relative humidity value. Oyedepo et al. [8] studied and analyzed the cost and performance of a gas turbine in Nigeria for the cost analysis based on the exergy, the study showed a decrease in the cost of the destructive exergy by 29% when the turbine inlet temperature was raised by 200K. Fallah et al. [9] compared four gas turbine cycle using energy and exergy analysis. The previous four gas turbine are: Simple gas turbine cycle, gas turbine cycle with evaporative air cooler at compressor inlet, gas turbine cycle with steam injection, and Gas turbine cycle with steam injection and evaporative air cooler at inlet. it was found that the last turbine is the most advanced in terms of design. Sanaye et al. [10] carried out a thermal economic analysis for the (latent) glacial thermal energy storage system for cooling the inlet of gas turbines. For gas turbines with a net power output of 25-100 MW, due to cooling the intake air with an ITES system increased efficiency in the range 2.1-5.2%, with an increase in payback period from about 4 to 7.7 years. De Pascale et al. [11] study Analysis of Inlet Air Cooling for IGCC Power Augmentation. various systems are currently used in gas turbines and combined cycle power plants in order to reduce the gas turbine inlet air temperature and, therefore, the effect of ambient conditions on performance. Baghernejad and Anvari-Moghaddam [12] studied a gas-steam combined plant from the point of view of energy and exergy analysis, where multi-objective optimization was used by studying three objective functions, which are the exergy efficiency, the total cost rate, and the cost of electricity generation. Finally, several researchers have worked on thermal mass transfer and combustion [7, 13-15].

2. MATERIALS AND METHODS

Energy is an intuitive concept that we deal with greatly in our daily lives [16]. Although we are in contact with it a lot, there is no clear and precise definition of this quantity, and it cannot be measured directly, but it is always calculated indirectly. The energy balance equation is written for a control volume in the steady state consisting of many inputs and many outputs according to the relationship [17]:

$$\underbrace{\dot{Q}_{net}}_{\substack{\textit{energy}\\ \text{transfer}\\ \text{by heat}}} + \underbrace{\sum_{in} \dot{m}_{in} (h + \frac{1}{2} v^2 + gz)_{in}}_{\textit{energy}\ \text{transfer by mass (inlet)}} = \underbrace{\dot{W}_{net}}_{\substack{\text{energy transfer}\\ \text{by work}}} + \underbrace{\sum_{out} \dot{m}_{out} (h + \frac{1}{2} v^2 + gz)_{out}}_{\text{energy transfer by mass (outlet)}}$$

where, $\dot{Q}(KW)$ it represents the amount of heat involved in the process, $\dot{W}(KW)$ represents the amount of power, $h(\frac{kJ}{kg})$ specific enthalpy, $v(m/s)$ speed of flow, $z(m)$ height and $\dot{m}(\frac{kg}{s})$ mass flow rate. The entropy function is a function of the state and its change is related only to the initial and final state. The entropy changes for a reversible thermodynamic process and an ideal gas is given by the following relation [8]:

$$s = c_p \ln \frac{T}{T_0} - R \ln \frac{p}{p_o}$$

where, the suffix o refers to the dead state conditions. C_p heat capacity at constant pressure.

The entropy balance equation for an open system in the general case is given by the relation [17]:

$$\underbrace{\sum_i \frac{\dot{Q}_i}{T_i}}_{\substack{\textit{entropy}\ \text{transfer}\\ \text{by heat}}} + \underbrace{\sum_{in} \dot{m}_{in} S_{in} - \sum_{out} \dot{m}_{out} S_{out}}_{\textit{entropy}\ \text{transfer by mass}} + \underbrace{\dot{S}_{gen}}_{\substack{\textit{entropy}\ \text{generation due}\\ \text{to irreversibilities}}} = \frac{dS_{CV}}{dt}$$

where, $\dot{S}_{gen}(KW/K)$ represents the rate of entropy generation due to irreversible processes and CV represent control volume.

Exergy is defined as the possibility of the maximum available useful work that can be obtained from a system under specific condition, and it is a state function whose change is related to the initial final state and is not related to the path taken [18]. The physical exergy is produced as a result of the deviation of the temperature and pressure of the system from the temperature and pressure of the dead state represented by the surrounding medium [9]. The exergy flow rate is calculated for the ideal gas by the relation [19]:

$$\dot{E}x = \dot{m}(c_p (T - T_0) - T_0 c_p \ln(\frac{T}{T_0}) + R \ln(\frac{P}{P_0}))$$

And for the real gases from the relation [10]:

$$\dot{E}x = \dot{m}((h - h_0) - T_o (s - s_0))$$

Exergy is transmitted to and from the system in three ways, which are work, heat, and mass, such as energy, unlike entropy, which is transmitted only by heat and mass, where the balance equation of exergy is given in the general case that is encountered in most thermodynamic systems with the following relationship [17]:

$$\underbrace{\sum_j (1-\frac{T_o}{T_j}).\dot{Q}_j}_{exergy \text{ transfer by heat}} - \underbrace{\dot{W}_{cv}}_{\text{exergy transfer by work}} + \underbrace{\sum_{in} \dot{m}_{in}.ex_{in} - \sum_{out} \dot{m}_{out}.ex_{out}}_{exergy \text{ transfer by mass}} - \underbrace{\dot{E}x_{des}}_{exergy \text{ destroyed due to irreversibilities}} = 0$$

where, $\dot{E}x_{des}(KW)$ represents the destructive exergy. Exergy efficiency is defined as the ratio of the useful outward exergy to the value of the inward provided exergy and is calculated [20]:

$$\eta_{ex} = \frac{P}{F} = 1 - \frac{D}{F}$$

where, p represents useful exergy or production, F is provided exergy or fuel, and D is destruction exergy.

Bryton cycle: The simple gas turbine cycle consists of three basic components: The compressor, the combustion chamber, and the turbine. Figure 1 represents the scheme for this cycle [21].

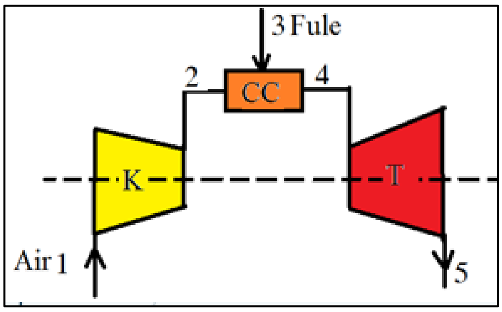

Figure 1. Gas turbine cycle scheme [21]

Table 1. Nominal data of ideal gas turbine plant

Value	Units	Nominal Data of Ideal Gas Turbine Plant
100	[MW]	Net Power
48.3	[kg/s]	Mass flow rate interning the compressor
515	[°C]	Exhaust gas temperature
491	[kg/s]	Combustion products mass flow rate
13.5	[kg/s]	Fuel mass flow rate
11		Compression ratio
0.87		Compressor isentropic efficiency
0.89		Turbine isentropic efficiency
2		Combustion chamber number

The compressor absorbs the atmospheric air at the conditions of the surrounding medium and compresses it according to a reversible adiabatic process, and its pressure and temperature rise, after which it enters the combustion chamber [21]. The fuel burns and the heat is given to the air under constant pressure, and we have a mixture of air and combustion products with a large kinetic energy that enters the turbine and it expands without heat exchange to the conditions of the surrounding medium. Figure 2 shows the scheme of the studied gas turbine plant with a cooling system for the air entering the compressor using ice thermal energy storage ITES that is cooled at off peak period and discharged at peak time.

The current study was conducted according to the nominal capacity conditions of the Jandar station. The aim of this research is to determine the appropriate operating conditions that achieve the highest thermal efficiency and lowest operating cost. Therefore, this research did not work on other operating conditions. Table 1 shows the values of the studied plants data.

The economic analysis accompanying the exergy analysis, which is called the exergoeconomic analysis, provides a powerful tool in calculating the appropriate financial values associated with investment, operation, maintenance, and the cost of the fuel used, as these values are used in the cost balance equation. For the components that receive heat and give work, the cost balance equation states that the total cost of the exergy outgoing is equal to the cost of expenditure to obtain it, and the equation is written in the following form [16]:

$$\underbrace{\sum \dot{C}_{in} + \dot{C}_q + \dot{Z}_k}_{in} = \underbrace{\sum \dot{C}_{out} + \dot{C}_w}_{out}$$

where, subscript (In) represents paid or Fuel and subscript (out) represents output or Product, $\dot{C}_{in}, \dot{C}_{out}$ represents the average cost associated with the outgoing and entering with the exergy stream ($/sec) and is given by the relationship [16]:

$$\dot{C}(\$/\text{s}) = c(\$/\text{kJ})\dot{E}x(\text{kJ/s})$$

$\dot{C}_w, \dot{C}_q$ represents the accompanying cost rate for the output work and heat supplied and is given by the relations [16]:

$$\dot{C}_W = c_w \dot{W}$$
$$\dot{C}_q = c_q \dot{E}x_q$$

c, c_w, c_q represent the cost per unit of exergy transmitted.

$\dot{Z}_k$ represents the capital cost rate, including the costs of incorporation, construction, operation and maintenance, and is calculated from the relationship [16]:

$$\dot{Z}_k(\frac{\$}{\text{sec}}) = \frac{Z_k(\$)CRF\varphi}{H3600}$$

where, Z_k is a cost function of the device estimated in \$, H is the number of working hours of the device per year, φ is a coefficient that takes into account the costs of operation and maintenance, and CRF is the capital recovery coefficient and is given by the relationship [22]:

$$CRF = \frac{i(1+\text{i})^n}{(1+\text{i})^n - 1}$$

where, i is the interest rate and n is the operating life of the plant (years). Typical and common terminology include those used in literature [23-25].

Other operating conditions such as maintenance costs, transportation, workers' wages and other costs are very small compared to the cost of compressor operation and fuel consumption, so such costs are neglected, as indicated by many reference studies and researches.

Table 2 show Balance equation for different components [10, 16, 20, 26-30].

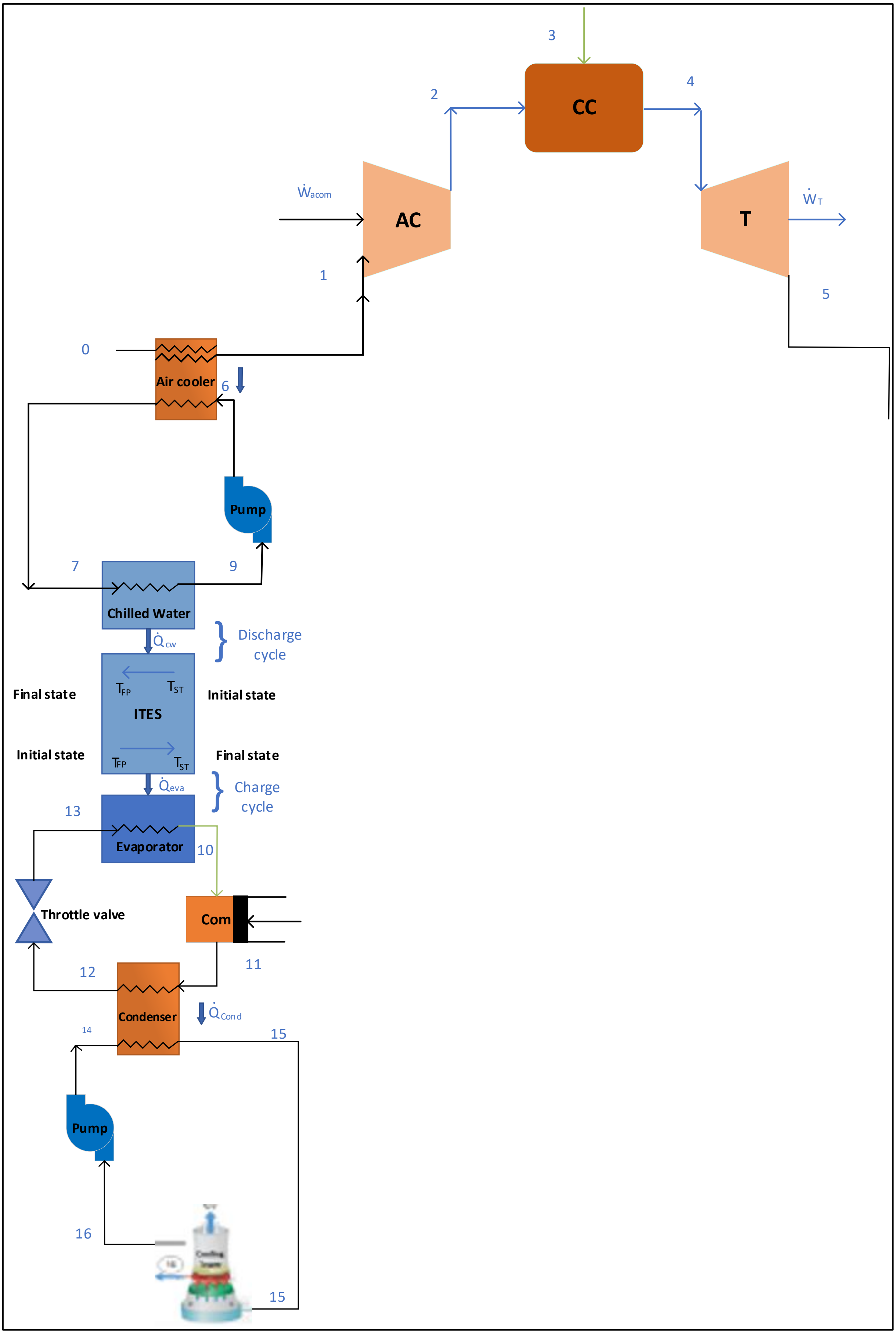

Figure 2. Gas turbine cycle with cooling system

Table 2. Balance equations for different components of plants

Mass	$\dot{m}_1 = \dot{m}_2 = \dot{m}_{air}$	Compressor
Energy	$\dot{W}_{acom} = \dot{m}_{air}\,\mathrm{C}_{p,air}(\mathrm{T}_2 - \mathrm{T}_1)$	
Entropy	$\dot{S}_{gen,acom} = \dot{m}_2 s_2 - \dot{m}_1 s_1$	
Exergy	$\underbrace{\lvert\dot{W}_{acom}\rvert}_{F} - \underbrace{(\dot{E}X_2 - \dot{E}X_1)}_{P} - \underbrace{\dot{E}X_{des,acom}}_{D} = 0$	
Mass	$\dot{m}_2 + \dot{m}_3 = \dot{m}_4$ $\dot{m}_3 = \dot{m}_{fuel}$ $\dot{m}_4 = \dot{m}_{gas}$	Combustion chamber
Energy	$\dot{Q}_{in} = \dot{m}_{air}((1/\mathrm{AF}+1)\mathrm{C}_{p,gas}T_4 - C_{p,air}T_2) = \frac{1}{AF}LHV$	
Entropy	$\dot{S}_{gen,cc} = \dot{m}_4 s_4 - \dot{m}_3 s_3 - \dot{m}_2 s_2$	
Exergy	$\underbrace{(\dot{E}X^{ch}{}_2 + \dot{E}X_2) + (\dot{E}X_3 + \dot{E}X^{ch}{}_3)}_{F} - \underbrace{(\dot{E}X_4 + \dot{E}X^{ch}{}_4)}_{P} - \underbrace{\dot{E}X_{des,cc}}_{D} = 0$	
Mass	$\dot{m}_4 = \dot{m}_5 = \dot{m}_{gas}$	Turbine
Energy	$\dot{W}_T = \dot{m}_{gas}C_{p,gas}(\mathrm{T}_4 - \mathrm{T}_5)$	
Entropy	$\dot{S}_{gen,T} = \dot{m}_5 s_5 - \dot{m}_4 s_4$	
Exergy	$\underbrace{(\dot{E}X_4 - \dot{E}X_5)}_{F} - \underbrace{\dot{W}_T}_{P} - \underbrace{\dot{E}X_{des,T}}_{D} = 0$	
Mass	$m_1 + m_3 = m_5$	Entire gas cycle plant
Energy	$\dot{W}_{net} = \dot{W}_T - \lvert\dot{W}_{acom}\rvert = m_{air}((1+1/\mathrm{AF})\mathrm{c}_{p,gas}(\mathrm{T}_4 - \mathrm{T}_5) - \mathrm{c}_{p,air}(\mathrm{T}_2 - \mathrm{T}_1))$	
Entropy	$\dot{S}_{gen} = \dot{m}_5 s_5 - \dot{m}_1 s_1 - \dot{m}_3 s_3$	
Exergy	$\underbrace{(\dot{\mathrm{E}}\mathrm{X}_1 + \dot{\mathrm{E}}\mathrm{X}_3 + \dot{\mathrm{E}}\mathrm{X}_3{}^{ch})}_{F} - \underbrace{\dot{W}_{net}}_{P} - \underbrace{\dot{\mathrm{E}}\mathrm{X}_5}_{L} - \underbrace{\dot{E}X_{des,\mathrm{plant}}}_{D} = 0$	
Mass	$\dot{m}_0 = \dot{m}_1 = \dot{m}_{air}$ $\dot{m}_6 = \dot{m}_7 = \dot{m}_{cw}$	Air cooler
Energy	$\dot{m}_0 h_0 + \dot{m}_6 h_6 = \dot{m}_7 h_7 + \dot{m}_1 h_1$	
Entropy	$\dot{S}_{gen,\mathrm{AC}} = (\dot{\mathrm{m}}_1\mathrm{s}_1 + \dot{\mathrm{m}}_7\mathrm{s}_7) - (\dot{\mathrm{m}}_0\mathrm{s}_0 + \dot{\mathrm{m}}_6\mathrm{s}_6)$	
Exergy	$\underbrace{(\dot{\mathrm{E}}\mathrm{X}_6 - \dot{\mathrm{E}}\mathrm{X}_7)}_{F} - \underbrace{(\dot{\mathrm{E}}\mathrm{X}_1 - \dot{\mathrm{E}}\mathrm{X}_0)}_{P} - \underbrace{\dot{\mathrm{E}}\mathrm{X}_{des,\mathrm{AC}}}_{D} = 0$	
Mass	$\dot{m}_9 = \dot{m}_6 = \dot{m}_{cw}$ $\dot{m}_{14} = \dot{m}_{16} = \dot{m}_{CT}$	pump
Energy	$\dot{W}_{pump1} = \dot{m}_{cw}(\mathrm{h}_6 - \mathrm{h}_9)$ $\dot{W}_{pump2} = \dot{m}_{CT}(\mathrm{h}_{14} - \mathrm{h}_{16})$	
Entropy	$\dot{S}_{gen,pump1} = \dot{m}_6 s_6 - \dot{m}_9 s_9$ $\dot{S}_{gen,pump2} = \dot{m}_{14} s_{14} - \dot{m}_{16} s_{16}$	
Exergy	$\underbrace{\lvert\dot{W}_{pump1}\rvert}_{F} - \underbrace{(\dot{E}X_6 - \dot{E}X_9)}_{P} - \underbrace{\dot{E}X_{des,pump1}}_{D} = 0$ $\lvert\dot{W}_{pump2}\rvert - (\dot{E}X_{14} - \dot{E}X_{16}) - \dot{E}X_{des,pump2} = 0$	
Mass	$\dot{m}_7 = \dot{m}_9 = \dot{m}_{cw}$	Discharged cycle
Energy	$Q_{cw} = Q_{ST}$	
Entropy	$\dot{S}_{gen,ST,dis} = \dot{m}_9 s_9 - \dot{m}_7 s_7 + \frac{\Delta s_{ST,dis}}{t_{dis}}$	
Exergy	$\underbrace{\left\lvert\frac{\Delta Ex_{ST,dis}}{t_{dis}}\right\rvert}_{F} - \underbrace{(\dot{E}X_9 - \dot{E}X_7)}_{P} - \underbrace{\dot{E}X_{des,ST,dis}}_{D} = 0$	

Mass	$\dot{m}_{13} = \dot{m}_{10} = \dot{m}_r$	Charge cycle
Energy	$Q_{eva} = Q_{ST}$	
Entropy	$\dot{S}_{gen,ST,\mathrm{ch}} = \dot{m}_{10}s_{10} - \dot{m}_{13}s_{13} + \frac{\Delta s_{ST,\mathrm{ch}}}{t_{ch}}$	
Exergy	$\underbrace{(\dot{EX}_{13} - \dot{EX}_{10})}_{F} - \underbrace{\frac{\Delta Ex_{ST,\mathrm{ch}}}{t_{ch}}}_{P} - \underbrace{\dot{EX}_{des,ST,\mathrm{ch}}}_{D} = 0$	
Mass	$\dot{m}_{11} = \dot{m}_{10} = \dot{m}_r$	Compression cycle compressor
Energy	$\dot{W}_{com} = \dot{m}_r(\mathrm{h}_{11} - \mathrm{h}_{10})$	
Entropy	$\dot{S}_{gen,\mathrm{com}} = \dot{m}_{11}s_{11} - \dot{m}_{10}s_{10}$	
Exergy	$\underbrace{\lvert\dot{W}_{com}\rvert}_{F} - \underbrace{(\dot{EX}_{11} - \dot{EX}_{10})}_{P} - \underbrace{\dot{EX}_{des,\mathrm{com}}}_{D} = 0$	
Mass	$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_r$ $\dot{m}_{14} = \dot{m}_{15} = \dot{m}_{cT}$	condenser
Energy	$\dot{Q}_{cond} = \dot{m}_r(\mathrm{h}_{11} - \mathrm{h}_{12}) = \dot{\mathrm{m}}_{CT}(\mathrm{h}_{15} - \mathrm{h}_{14})$	
Entropy	$\dot{S}_{gen,\mathrm{cond}} = (\dot{\mathrm{m}}_{12}\mathrm{s}_{12} + \dot{\mathrm{m}}_{15}\mathrm{s}_{15}) - (\dot{\mathrm{m}}_{11}\mathrm{s}_{11} + \dot{\mathrm{m}}_{14}\mathrm{s}_{14})$	
Exergy	$\underbrace{(\dot{Ex}_{11} - \dot{Ex}_{12})}_{F} - \underbrace{(\dot{\mathrm{E}}\mathrm{x}_{15} - \dot{Ex}_{14})}_{P} - \underbrace{\dot{\mathrm{E}}\mathrm{x}_{des,cond}}_{D} = 0$	
Mass	$\dot{m}_{13} = \dot{m}_{12} = \dot{m}_r$	Throttle valve
Energy	$h_{12} = h_{13}$	
Entropy	$S_{gen,valve} = \dot{m}_{13}s_{13} - \dot{m}_{12}s_{12}$	
Exergy	$\underbrace{\dot{EX}_{12}}_{F} - \underbrace{\dot{EX}_{13}}_{P} - \underbrace{\dot{EX}_{des,valve}}_{D} = 0$	

3. RESULTS AND DISCUSSION

In this paper, the economic feasibility of installing an air cooling system on the gas power plant operating with natural gas was studied. For the purposes of comparison, we consider plant work on constant capacity, which is 100 MW, and the study was done at an average temperature throughout the year, which is 296K.

Since the study aims to search for the best operating conditions that achieve the highest thermal efficiency and the lowest operating cost during all periods of the year, we relied on the average environmental conditions during the year. If we had conducted the study during all months of the year, month by month, the results would have been long and we would have seen a significant improvement during some months of the year and a lesser or even undesirable improvement in other months. Therefore, since the requirement in this research was to determine the amount of average benefit during the year, we relied on the average conditions of the year.

The study of other cooling techniques such as evaporative cooling, absorption cycle cooling, and compression cooling cycle cooling have been studied extensively in scientific articles and research. However, the cooling technique studied in the current research depends on utilizing the surplus capacity of the station at night when the load applied to the network is at its lowest consumption value and utilizing it to improve the thermal performance of the station at peak times and at the lowest costs.

The study was conducted using the EES program and the environment of the examples in it is limited to Single objective optimization, which makes it difficult to conduct these examples multi-objective optimization, and focusing on the environmental aspects will make the research long, and therefore this aspect was cancelled from this research, in addition to the fact that the environmental aspect is studied in many previous researches.

In order to determine the values of the optimal variables that contribute to the improvement of the thermal system, we have chosen five objective functions, each of which we will study in two cases:

1. First case without ITES cooling system.
2. Second case with ITES cooling system.

The optimization will be done using the genetic algorithm in the EES program, then a comparison will be made between the two cases to determine whether there is an economic feasibility of this design or not. Table 3 shows these five-object function:

Table 3. Objective functions

Objective Function	Function No.
Minimization of operational cost rate	1
Maximization of thermal efficiency	2
Maximization of exergy efficiency	3
Minimization of electricity cost	4
Minimization of fuel cost	5

The Table 4 shows the decision variables used in this paper, which are the compression ratio r_p, the temperature of the air entering the compressor T_1, the temperature at the turbine inlet T_4, the temperature of the refrigerant solution at the air cooler inlet T_6, the temperature of the frozen water in the thermal ice tank T_{st}, and the condensation and evaporation pressures in the compression refrigeration cycle P_{10}, P_{11}.

Table 4. Decision variables

Reason	Rang	Variables
r_p	2-16	Domain approved for a wide range of applications
T_1	$274k - 283k$	Increase thermal efficiency
T_4	$900k - 1500k$	Restricted by the ability of materials to withstand high temperatures
T_6	$273k - 276k$	To provide sufficient air cooling within the mentioned area
T_{st}	$263k - 270k$	Within the feasible range of cooling and tank size
P_{10}	$0.8bar - 2bar$	So that the temperature of the Freon in the evaporator does not become higher than the temperature in the ice tank
P_{11}	$8bar - 30bar$	Suitable for a wide range of applications

3.1 First case: The plant without ITES

Before starting to determine the values of the optimal variables according to the five functions, we will study determining the values of the variables that correspond to the lowest operating cost value for the plant without an ITES cooling system (the first case - Objective function 1) by observing the behavior of the curves that show the change in the operating cost value and the fuel cost with the change in the compression ratio in the compressor at different inlet turbine temperature values And deduce the value of the compression ratio and inlet turbine temperature that gives the lowest value of the operating cost (objective function 1) and the lowest cost of fuel (objective function 4) and compare the results with the results of the optimization used using the genetic algorithm in order to verify the effectiveness of this algorithm in determining the optimal values for the required objective function with an acceptable degree of accuracy. Figure 3 and Figure 4 show these curves. We note that the lowest fuel cost occurs at high compression ratios and high inlet turbine temperatures, and that the lowest operating cost occurs at a compression ratio of 8.16 and at a temperature of 1500k at the turbine inlet. This is consistent with the values obtained using the genetic algorithm. Table 5 shows the values of these parameters.

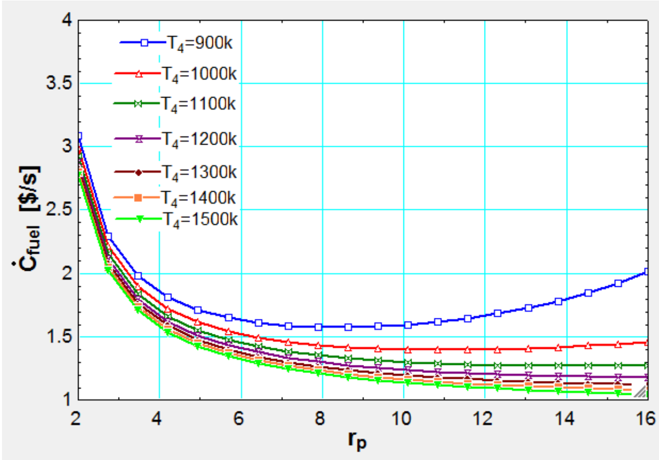

Figure 3. Variation of fuel cost with compression ratio at different temperatures

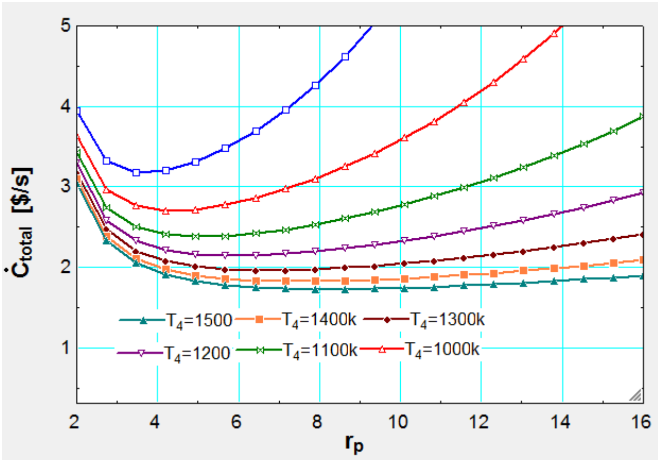

Figure 4. Change of total cost with compression ratio at different inlet turbine temperatures

Table 5. Decision variable for first case

	Objective Function				
variable	1	2	3	4	5
rp	8.16	16	16	2	16
T4	1500	1500	1500	1500	1500

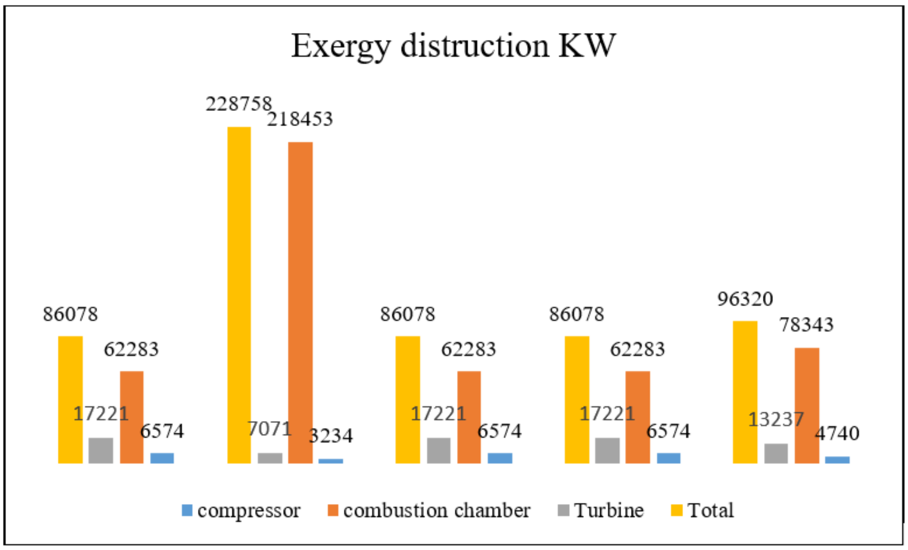

Figure 5. Exergy destruction rate

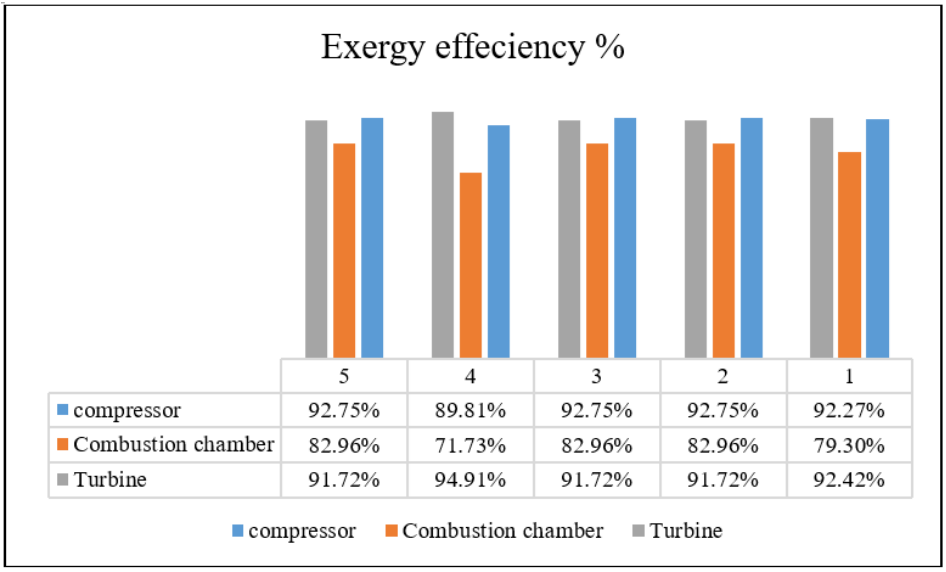

Figure 6. Exergy efficiency

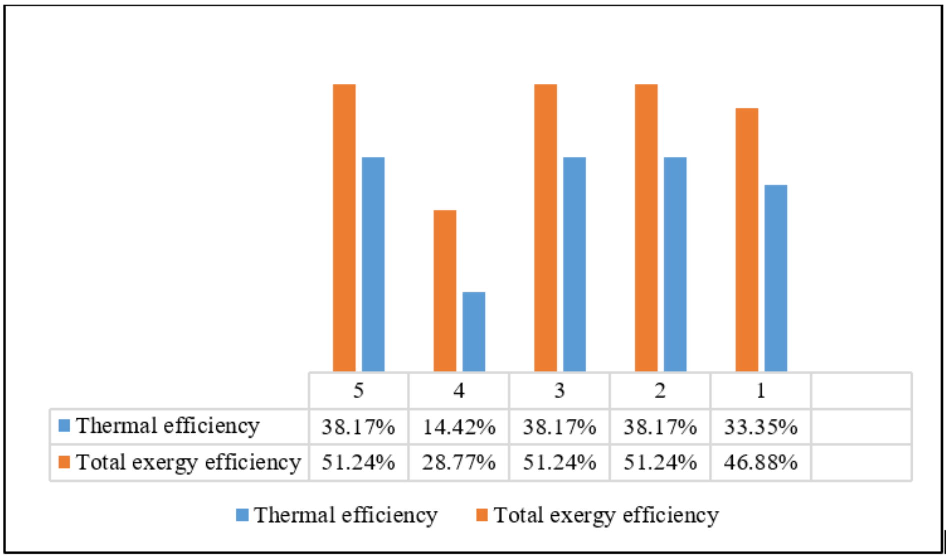

Figure 7. Comparison between thermal and exergy efficiency

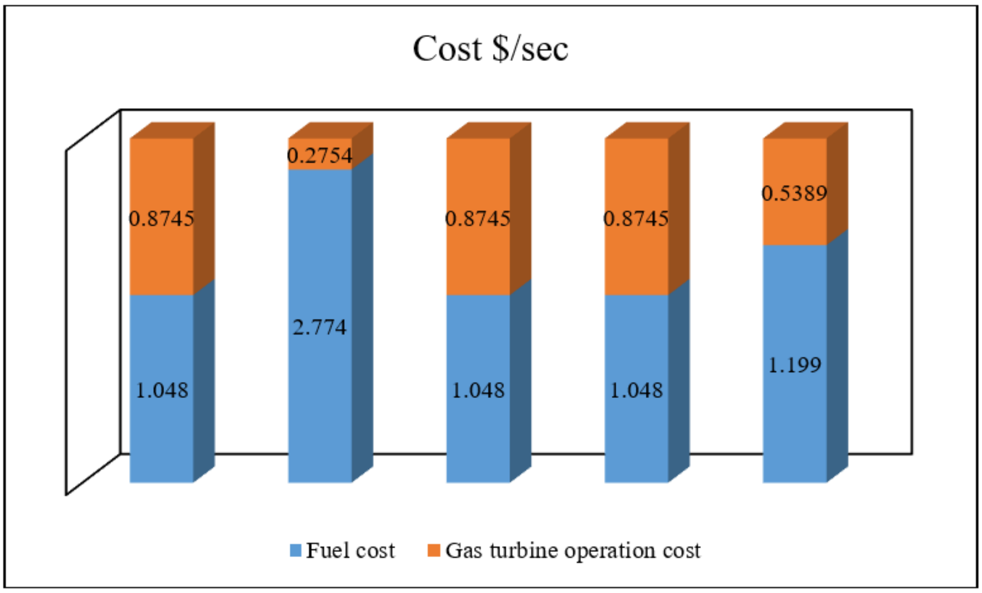

Figure 8. Cost rate

We notice that the value of the highest exergy efficiency (objective 3) corresponds to the highest thermal efficiency (objective 2) and corresponds to the lowest fuel consumption value (objective 5). This is due to the low value of the destructive exergy in the combustion chamber, as is clear in the Figure 5. The Figure 6 shows the exergy efficiency of the five objective functions above. As we said, the exergy efficiency is a reflection of the value of the destroyed exergy. The higher the exergy efficiency, the lower the exergy destruction rate. The Figure 7 also shows the values of both the thermal and total exergy efficiency of the plants corresponding to the five objective functions. The Figure 8 shows the distribution of operating costs between fuel costs and gas turbine operating costs.

3.2 The second case: The station with ITES

Table 6 shows the results of the objective functions resulting from the optimization process for the aforementioned parameters according to the previous five functions.

Figure 9 shows the value of the thermal efficiency and the exergy efficiency in the previous five functions. We note that the worst thermal performance value occurs in the fourth objective function for the same reason mentioned above.

Figure 10 shows the total operating cost in detail according to its three components, which are the operating cost of the gas turbine, the operating cost of ITES, and the cost of fuel used. It is clear from Figure 10 that the cost of electricity is a result of the large consumption of it by the compressor. When the compressor capacity decreases, the value of the resulting electricity cost will decrease, and thus the cost of operating the gas turbine will decrease, but this will be accompanied by a large flow of air in order to maintain the same capacity, and this is accompanied by a large increase in fuel consumption and thus High costs of fuel used and this mainly explains the observed behavior in the fourth objective function. Table 6 shows a summary of operating results according to the lowest operating cost function (function 1).

Table 6. Decision variable for second case

	Objective Function				
variable	1	2	3	4	5
T1	274.5	274.5	274	283	274
T4	1500	1500	1500	1500	1500
T6	273	273	273	276	273.2
T-st	270	270	263.2	268.3	263.8
rp	8.86	16	16	2	16
P10	1.96	1.96	2	1.057	0.9995
P11	12.64	12.64	8	17.22	17.42

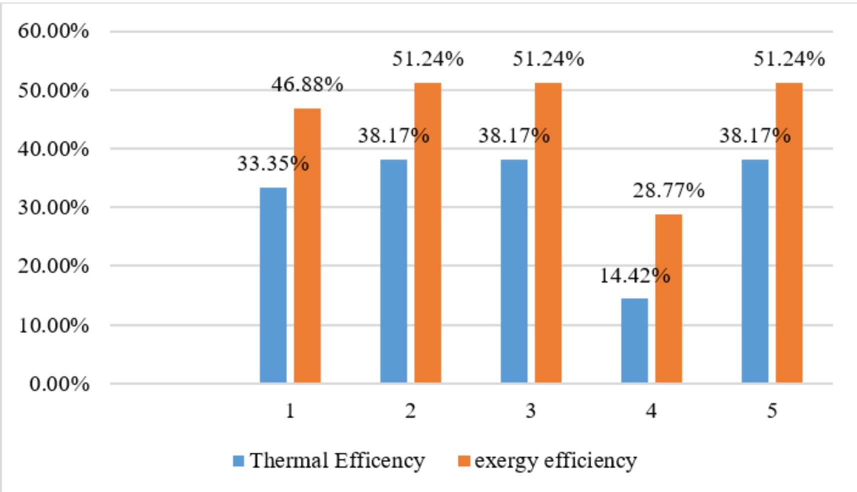

Figure 9. The exergy and thermal efficiency and the thermal yield

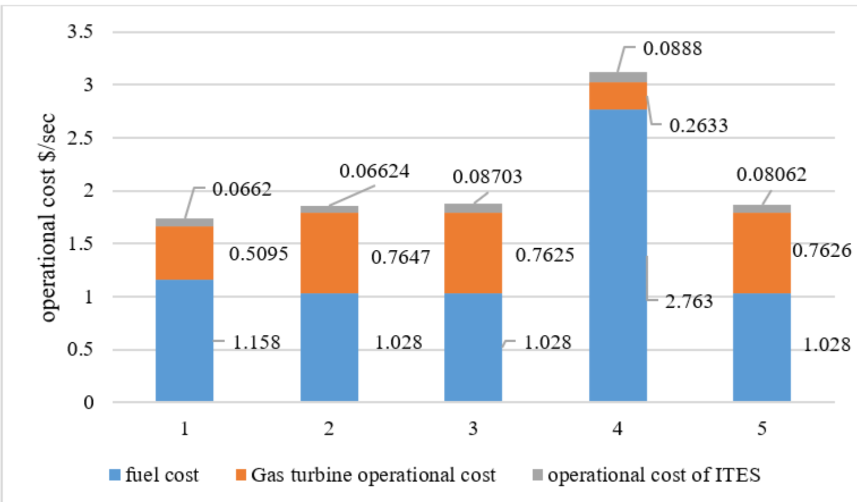

Figure 10. Cost distribution

Table 7. Summary of operational cost results according to objective function 1

Objective Function 1			
Output	with	without	difference%
$\dot{C}_{total}$	1.734	1.738	0.23015
$\dot{C}_{Fuel}$	1.158	1.201	3.58035
$\dot{W}_{acom}$	56216	61281	8.265205
Gas turbine operational cost rate	0.5095	0.5389	5.45
η_{ex}	47.80%	46.88%	1.962457
η_{Th}	34.53%	33.35%	3.538231

For the sake of the first objective function (Table 7) corresponding to the lowest operating cost, we note that the operational cost with the previous cooling system has decreased by 0.23% compared to the operating cost without the presence of this system. This decrease in the operational cost comes from the decrease in the fuel cost in the cycle equipped with a cooling system by 3.5% and the decrease in the cost required to operate the compressor, and thus the operation of the gas turbine, by 5.4%, as the capacity of the compressor in the cycle equipped with a cooling system decreased by 8.26%.

Compared to the cycle that does not contain the cooling system, this confirms the importance of this system in raising the thermal efficiency of the origin plant by 3.53% and decreasing the operational cost, by a very slight amount, and thus confirms the existence of an economic feasibility for this system.

4. CONCLUSIONS

- The operating parameters of the plants generally affect the operating cost, as they determine the amount of fuel consumed and, consequently, the cost of the fuel, the price of the compressor to be used, and the amount of compressor work. The most important parameters that affect the performance and operating cost of the station are the compression ratio and the temperature at the turbine inlet. At the compression ratio corresponding to the lowest cost (rp=8.16), fuel consumption decreases by 21% when the temperature at the turbine inlet increases from 900K to 1500K, while the total operating cost decreases by 62%.
- The lowest fuel cost corresponds to the highest exergy efficiency and the highest thermal efficiency to some extent, because most of the destructive exergy occurs in the combustion chamber. Therefore, the less fuel enters the combustion chamber while maintaining a constant air-to-fuel ratio, the less destructive exergy will be in it and the higher the thermal performance of the facility.
- By comparing the gas turbine plants without a cooling system and the plants with an ITES cooling system, we notice that the thermal efficiency of the turbine with the presence of the previous cooling system has increased and the operating cost has decreased compared to the turbine cycle without a cooling system. If we take the first objective function corresponding to the lowest operating cost, we notice that the operating cost with the presence of the thermal tank has decreased by 0.23% and the thermal efficiency has increased by 3.53%. This confirms the economic feasibility of equipping power stations operating with a gas turbine cycle with such systems, especially when the station operates in a desert climate (hot during the day and cold at night).

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NOMENCLATURE

$\dot{Q}$	amount of heat involved $[KW]$
$\dot{W}$	amount of power $[KW]$
h	specific enthalpy $\left[\frac{kj}{kg}\right]$
v	speed of flow $[m/s]$
z	height $[m]$
$\dot{m}$	mass flow rate $[kg/s]$
$\dot{S}_{gen}$	rate of entropy generation $[KW/K]$
$\dot{Ex}_{des}$	destructive exergy $[KW]$
CRF	capital recovery coefficient $[-]$