International Journal of Sciences: Basic and Applied Research (IJSBAR)

ISSN 2307-4531
(Print & Online)

http://gssrr.org/index.php?journal=JournalOfBasicAndApplied

Improvement of Gas Turbine Performance Using Multi-Stage Inlet Air Cooling System

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Abstract

Gas turbine performance is extremely sensitive to the ambient air conditions, particularly in hot climates as the high ambient temperature negatively affects gas turbine performance. This work aims to determine the effect of many parameters like the temperature at the compressor inlet, relative humidity, pressure ratios, and polytropic efficiency on each component of gas turbine performance (compressor, combustion chamber, and turbine). A new combined air-cooling system is designed using chilling system before the fogging cooling system to downsize the chilling system size and to improve the gas turbine performance. A theoretical model was created using Engineering Equation Solver (EES) software to calculate exergy destruction, net power, and all efficiencies (1st law and 2nd efficiency). After that, the validation of the theoretical model was performed using the actual performance data from the gas turbine power plant 25 MW model, (GE GT-TM) made by GE, the Heliopolis power plant - Egypt as a case study. This work concerns also a comparative for costs analysis of the new cooling system to determine the payback time. The results show the comparative performance of the overall gas turbine with and without the usage of a multi-stage cooling system. It is worth mentioning that the maximum power output increase is about 14.3% at the maximum ambient temperature (313^{o}K). While the change of the 1st low efficiency and the 2nd low efficiency are so small, and it could be neglected. The present multi-stage cooling system reduces the back-period cost if it uses a chiller system. This means low initial capital investment costs and low total annual costs. In addition, the multi-stage cooling system capital cost will be cashback during the first year regarding recovered power price.

Keywords: Gas turbine; inlet air; cooling system; fogging cooling system; chiller cooling system; performance.

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1. Introduction

Gas turbines are machines with a constant volumetric flow rate and the capacity of its compressor is a function of its mass flow rate, which is usually controlled by the temperature of the input air. The rise in the air inlet air temperature is especially noticeable in the summer, and it has a significant impact on the compressor performance, especially in high summer temperature countries like Egypt. Furthermore, the output power of gas turbines is decreasing due to the reduction in both air density and mass flow rate. The influence of the ambient temperature and compression ratio are significantly affecting the performance of the combined cycle gas turbine power plant for different gas turbine configurations. Moreover, the Engineering Equation Solver (EES) software is used to perform detailed analyses of energy and exergy for the axial compressor. The energy-exergy analysis shows that, by increasing the compressor pressure ratio, the energy destruction of high-pressure compressors constantly increases and the exergy destruction of the recuperator, in contrast, decreases continuously, [1]. It will be noted that the higher values of pressure ratios can decrease the operating range of the compressor, and it will be much more vulnerable to failures that could be caused by surges, dust, and sedimentations on the blades. Hence, the values for the pressure ratio should be optimized according to the application used [2]. Besides, the main exergy parameters of the engine components were introduced, while the exergy destruction rates within the engine components are split into endogenous /exogenous and avoidable/unavoidable parts. The system has low-performance enhancement due to the unavoidable exergy destruction rate is 90 %. The relationships between the components are relatively weak since the endogenous exergy destruction is 73 %, [3]. In addition, some studies show reductions in the performance, due to each 1° ambient air temperature (Ta) growth, to be about 0.6%. On the other hand, demand for electric power, and the value of electricity are greatest at those times of high ambient air temperature, when air conditioning loads are maximized, [4]. Therefore, when the air temperature is cooled by about 15°, the air density increases by about10 percent, increasing the power output by about 10 percent, on another hand, [5]. The most famous types of inlet air cooling systems are vaporizing coolers (evaporative and fogging systems) and mechanical chilling. Those are suitable to be presented by power planet owners to increase the performance of the gas turbine output, [6]. The evaporative methods are the most common ways of cooling inlet air because of their simplicity, low installation, and O&M costs, but most effective only for dry-hot climates, [1]. Therefore, the Egyptian electricity holding company (EEHC) used the gas turbine inlet air cooling system to improve the gas turbine performance, [7]. On the other hand, fogging is slightly more effective than conventional direct evaporative cooling. However, some gas turbine manufacturers do not allow fogging due to compressor degradation and failures associated with fogging. The economic analysis shows that the cashback period for 2 years and the rate of return is 60%. Besides that, the ambient relative humidity plays an important role in the efficiency of fogging systems, [8]. While many studies examined the overall performance of the Benghazi CCPP throughout the year at different loads and different ambient conditions. The simulation results indicate that installing an absorption chiller to the existing power plant is thermodynamically possible and has the possibility of improving the overall efficiency. The plant is operating at 100% load and without absorption chillers and at ISO condition, for every 5ºK increase in ambient temperature the efficiency drop by 0.27%, although the absorption chiller improves the efficiency by 0.04%, [9]. Moreover, absorption chiller cooling in hot countries with ambient air temperatures above 40^{o}C has been studied and be concluded that absorption chillers can decrease inlet air temperatures to 10^{o}C. then the power increases about 20% for simple cycle, [10]. In addition, the analysis for the influence of various forms of inlet air heating, cooling, and supercharging on a 40 MW General Electric LM6000 gas turbine has been studied. It is concluded that the decrease by 28oC of the inlet temperature would raise the power by 30% and reduce the heat rate by 4.5%., [11]. In order, the chiller inlet air-cooling system is preferred technical than wetted evaporative media, but the wetted evaporative media is economically preferred than chiller system inlet air-cooling because the chiller system is more expensive than others inlet air cooling system by about 4-5 times. Payback periods were, 0.6, 0.4, and 2.2 years for fogging, wetted media, and refrigeration cooling, respectively, [12]. Moreover, Chiller cooling and evaporative cooling for a 264 MW gas turbine plant located at Kurimat, Upper Egypt. The results show that the annual power added by chiller cooling is 117,027 MWh, and the net cash flow is $3,787,537 while the annual power added by evaporative cooling is 86,118 MW, and the net cash flow is 4,503,548 $, [13-17]. This work aims to study the effect of ambient conditions, pressure ratio, and polytropic efficiency on the performance of gas turbine GT using the thermodynamics and economic concepts. Moreover, the design of a new inlet air cooling system is performed to analyze the influence of inlet air temperature on GT under full load which is critical in hot seasons. The newly designed system should have a combination of the advantages of conventional systems as low initial and operation cost like evaporative and fogging systems and not affected by humidity like a chiller system. On the other hand, this system avoids the drawbacks of conventional systems which can be summarized as the following: High initial and operation costs like chiller system, and the effect of relative humidity.

2. Design of the Multi-Stages Inlet Air Cooling System

The used multi-stage inlet air cooling system is consists of two stages as the following:

- The 1st stage which used an absorption chiller cooling system
- The 2nd stage which used fogging cooling system.

This combination is designed to avoid the disadvantages of each system being used as an inlet air cooling system alone. During this design, it is considered that the maximum ambient temperature is Ta = 40^{o}C and the minimum relative humidity ratio is rh = 20% where this is the worst case around the case study. The absorption chiller decreases the air temperature by about 5^{o}C using the GT gas exhaust energy. During this cooling process, condensate water is collocated into fogging water tank to reduce the fogging water consumption. Then, using the fogging system to get rh = 100% the air temperature becomes the wet-bulb temperature, as shown in Figure.1.

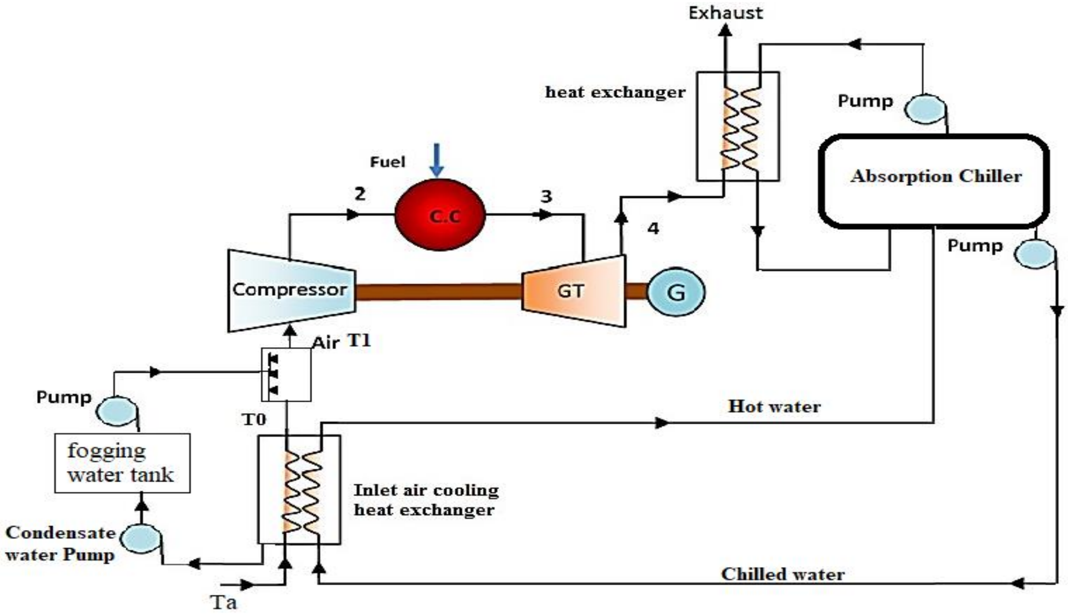

Figure 1: Schematic diagram of GT equipped with the newly designed combined cooling air system.

3. Case Study (Heliopolis power plant – Egypt)

The mobile GT unit at Heliopolis power plant has a capacity of 25 M.W which is in Cairo, EGYPT; it is selected as a real case study. This case study will concentrate on studying the effect of ambient conditions as the validity of the EES code to study the performance of GT. The GT case study model is (GE GT-TM) made by GE. For ISO conditions, the inlet air mass flow rate for a gas turbine is about 65 kg/h.

4. Simulation Model

The EES program is run with a specifically designed code to simulate all processes of design and operation for each component of GT.

4.1. Compressor Model

The isentropic efficiency studies only the start and end states of the compression process without focusing on the actual paths the compression process takes. Since the work is not a thermodynamic property and depends on the actual path, the polytropic analysis endeavors to get the actual path taken during the compression process in determining the actual work. In a polytropic process, the compression process takes place in small stages (infinitesimally small steps). Calculating the power for the polytropic process involves the summation of the power for each stage. Each stage is considered a small compressor.

The relation between polytropic and isentropic efficiency is obtained by using equation No. (1), [14].

$$\eta_p = \frac{\left[\frac{\gamma - 1}{\gamma}\right] \cdot ln(\pi s)}{ln\left[1 + \frac{1}{\eta_{is}} \cdot \left(\pi_s^{\left\{\frac{\gamma-1}{\gamma}\right\}} - 1\right)\right]} \quad \textbf{(1)}$$

Where $\boldsymbol{\pi}_s$ is the stage pressure ratio and obtained by using equation No. (2), as Nc is the number of stages and $\boldsymbol{\pi}$ is the compressor pressure ratio.

$$\pi_{sc} = (\pi)^{\frac{1}{N_c}} \quad \textbf{(2)}$$

The first low efficiency of each stage ηIs is a function in inlet air enthalpy hi, outlet air isentropic enthalpy hos, and real outlet air enthalpy of each stage and get by using equation No. (3).

$$\eta_{Is} = \frac{(h_{os} - h_i)}{(h_o - h_i)} \quad \textbf{(3)}$$

The exergy destruction is energy lost regarding the exergy difference between the inlet and outlet of each stage or compressor as shown in equation No. (4).

$$E_{desc} = m_a * T_a * (S_o - S_i) \quad \textbf{(4)}$$

The power consumption for each stage or overall compressor was obtained by using equation No. (5).

$$P_c = m_a * (h_o - h_i) \quad \textbf{(5)}$$

The reversible power was obtained by using equation No. (3.6).

$$P_{revc} = P_c - E_{desc} \quad \textbf{(6)}$$

and second low efficiency obtained by using equation No. (7).

$$\eta_{IIc} = \frac{(P_{revc})}{(P_c)} \quad \textbf{(7)}$$

The exergy destruction ratio of each stage E_{dRcs} is the ratio between exergy destruction of each stage E_{descs} to overall compressor exergy destruction E_{desc} could be obtained by using equation (8).

$$E_{dRcs}= \frac{(E_{descs})}{(E_{desc})} \quad \textbf{(8)}$$

Since axial compressors are positive displacement compressors, the volume of compressed air is constant. Moreover, the air mass flow rate depends on the density of the air as shown in equation (9).

$$M_a= \rho_{air} * V \quad \textbf{(9)}$$

Since the design operating condition is considered an ISO state, the ambient state of the design operating conditions is T_a = 288 k and rh = 0.6. Therefore, if the ambient condition changes from that design condition, it will affect the performance of the machine. The air mass flow rate ratio (m_{airR}) is the ratio between air mass flow rate at operation condition (m_{air}) at ISO condition (m_{ISO}) as the state in equation No. (10).

$$m_{airR}= \frac{m_a}{m_{ISO}} \quad \textbf{(10)}$$

The overall power consumption rat ratio (P_{cR}) is the ratio between the power consumption at operation condition (P_c) to at ISO condition (P_{cISO}) as the state in equation No. (11).

$$Pc_R = \frac{P_c}{P_{cISO}} \quad \textbf{(11)}$$

4.2. The Combustor Model

During combustor design assumed that the exit hot gas temperature constant is about T_3=1350 K, the fuel pressure about 28 bar, and the fuel temperature (T_f= 298k) with low heating value LHV =50000 kJ/kg. Since to obtain the inlet heat energy Q_{in} by using equation No. (12).

$$Q_{in}= m_f * LHV \quad \textbf{(12)}$$

While the fuel mass flow rate (m_f) be get using equation No. (13).

$$m_f = \frac{((m_{air}+m_f)*h[3])-(m_{air}*h[2])}{LHV} \quad \textbf{(13)}$$

So, the hot gas flow rate (m_g) be get using equation No. (14).

$$m_g= m_f + m_a \quad \textbf{(14)}$$

The heat energy destruction Q_{des} get by using equation No. (15).

$Q_{des}=T_0*((m_g*S_3)-(m_{air}*S_2)-(m_f*S_f))$ **(15)**

Moreover, the reversable heat energy (Q_{rev}) gets by using equation No. (16).

$Q_{rev}=Q_{in}-Q_{des}$ **(16)**

So, the 2nd low combustor η_{IIcc} efficiency gets by using equation No. (17).

$\eta_{IIcc}=\frac{Q_{crev}}{Q_{in}}$ **(17)**

4.3. The Turbine Model

Assume the pressure ratio of expansion throw the turbine like the compressor pressure ratio $\boldsymbol{\pi}$ but the number of turbine stage is N_t and the pressure ratio of each stage $\boldsymbol{\pi}_{sT}$ get by using equation No. (18).

$$\pi_{sT}=(\pi)^{\frac{1}{N_t}} \qquad \textbf{(18)}$$

The first low efficiency of each stage η_{Ist} is a function in inlet air enthalpy hi, outlet air isentropic enthalpy hos and real outlet air enthalpy of each stage and get by using equation (19).

$$\eta_{Ist}=\frac{(h_i-h_{os})}{(h_i-h_{os})} \qquad \textbf{(19)}$$

The exergy destruction is energy which lost regarding exergy difference between the inlet and outlet of each stage or compressor as shown in equation (20).

$E_{dest}=m_a*T_a*(S_i-S_o)$ **(20)**

The power generated from each stage or over all turbine obtained by using equation (21).

$P_t=m_a*(h_i-h_o)$ **(21)**

The reversible power obtained by using equation (22)

$P_{revt}=P_t-E_{dest}$ **(22)**

and second low efficiency obtained by using (23).

$$\eta_{IIt}=\frac{(P_{revt})}{(P_t)} \qquad \textbf{(23)}$$

4.4. The overall Gas Turbine Model

For the overall gas turbine performance. The out but power is the net power and obtained by using equation (24).

$$P_{net} = P_t - P_c \tag{24}$$

Moreover, the 1st low efficiency by using equation (25).

$$\eta_I = \frac{(P_{net})}{(Q_{in})} \tag{25}$$

The exergy destruction can be written as shown in equation No. (26).

$$E_{des} = T_0 * \left((m_a * S_4) - (m_g * S_1)\right) \tag{26}$$

Where s1 is the air entropy at gas turbine inlet. While s_4 is the gas entropy at gas turbine outlet. The exergy destruction ratio is the ratio between the exergy destruction at ambient condition to the exergy destruction at ISO condition (E_{desISO}) and can be written as shown in equation No. (27).

$$E_{desR} = \frac{(E_{des})}{(E_{desISO})} \tag{27}$$

The same as air flow rate ratio could be get by equations No. (28)

$$m_{aR} = \frac{m_a}{m_{aISO}} \tag{28}$$

and gas ratio as could be gotten by equation No. (29).

$$m_{gR} = \frac{m_g}{m_{gISO}} \tag{29}$$

The overall reversible power could be obtained by using equation No. (30).

$$P_{rev} = P_{net} - E_{des} \tag{30}$$

second low efficiency could be obtained by using equation No. (31).

$$\eta_{II} = \frac{(P_{rev})}{(P_{net})} \quad \textbf{(31)}$$

The 1st low efficiency ratio referring to ISO condition is the same as exergy destruction ratio too and could be obtained by using equation No. (32).

$$\eta_{IR} = \frac{\eta_{I}}{\eta_{Iiso}} \quad \textbf{(32)}$$

and 2nd low efficiency ratio referring to ISO condition could be obtained by using equation No. (33).

$$\eta_{IIR} = \frac{\eta_{II}}{\eta_{IIiso}} \quad \textbf{(33)}$$

4.5. The Inlet Air Cooling System Model

Chiller load energy Q_{ch} obtained by using equation No. (34).

$$Q_{ch} = \frac{m_{aa} * h_a - m_{a0} * h_0}{\boldsymbol{\varepsilon}} \quad \textbf{(34)}$$

The condensate water mass flow rate m_{cw} is obtained by using equation No. (35)

$$m_{cw}=(\boldsymbol{\omega}_a*m_{aa}) -(\boldsymbol{\omega}_0*m_{a0}) \quad \textbf{(35)}$$

Where $\boldsymbol{\varepsilon}$ is the effectiveness of inlet air cooling heat exchanger, m_{aa} is the air mass flow rate at ambient condition, h_a is the air enthalpy at ambient condition, m_{a0} is the air mass flow rate at the exit of inlet air cooling heat exchanger, h_0 is the air enthalpy at the exit of inlet air cooling heat exchanger and $\boldsymbol{\omega}$ is the humidity ratio.

For fogging system, the fog mass flow rate is obtained by using equation No. (36).

$$m_{fog}=(\boldsymbol{\omega}_1*m_{a1}) -(\boldsymbol{\omega}_0*m_{a0}) \quad \textbf{(36)}$$

But the required makeup water for fogging water tank is obtained by using equation No. (37).

$$m_{Rw}=m_{fog}-m_{cw} \quad \textbf{(37)}$$

is the ratio of is obtained by using equation No. (38).

$$\text{The output Ratio} =\frac{P_{net}}{P_{netISO}} \quad \textbf{(38)}$$

Where P_{net} is the output power at T_a and P_{netIso} is the output power at ISO condition

5. Results and Discussion

5.1. Validation of the simulation model

To validate the theoretical thermodynamic model, the model results were compared with the data measured regarding to the case study as mentioned previously. The operating conditions of the case study are, π=12 and rh=0.6 with different ambient temperatures T_o. By applying the EES code as a simulator for studying the GT performance and comparing its results with the real results of the case study performance test, it found that the simulator results are almost like the case study results as shown in Figure.2 with maximum error equal to 1%.

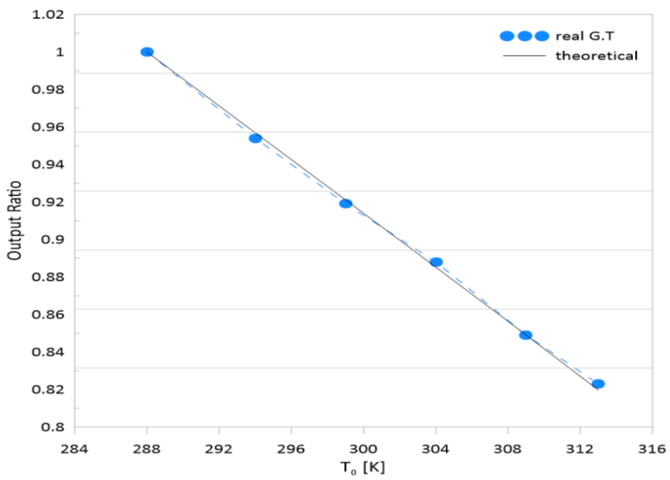

Figure 2: Comparison between the results of the present model and those of real GT performance results.

5.2. The Effect of the Inlet Air Temperature

Figure.6 shows the effect of increasing inlet air temperature $T_{0,}$ as that leads to a decrease both of net output power $P_{net,}$ and η_I, in order η_{II} increases.

5.3. The effect of pressure ratio π

Increasing the pressure ratio π leads to an increase η_I and P_{net} tell π =12, over than π =12 the P_{net} decreases but η_I still increases. In order, both η_{II} and Edes decrease as shown in Figures.6 and 7.

5.4. The effect of ηp

The increasing polytropic efficiency ηp leads to increase η_{II} and Edes decrease η_I and P_{net} as shown in Figures.3 and 4. In addition, the increases in T_0 lead to decrease exergy destruction E_{des} as shown in Figure.4.

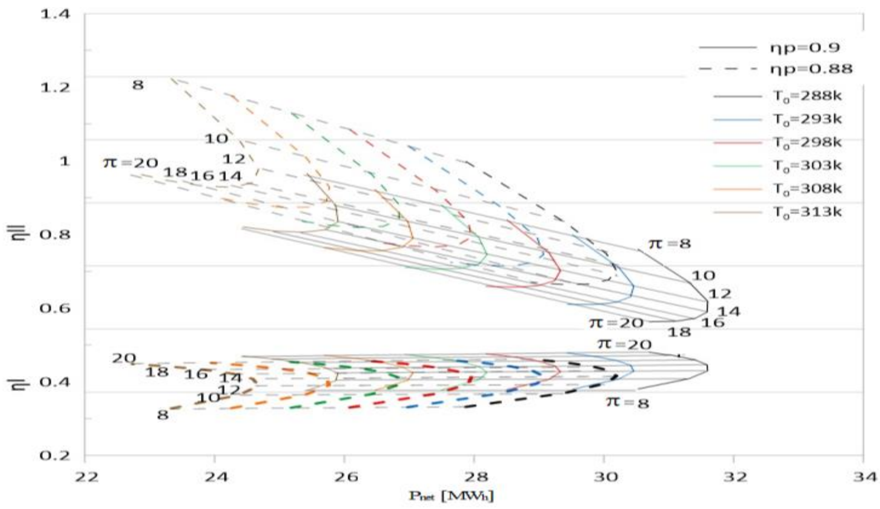

Figure 3: The effect of effective parameter on the GT P_{net} ηI and ηII.

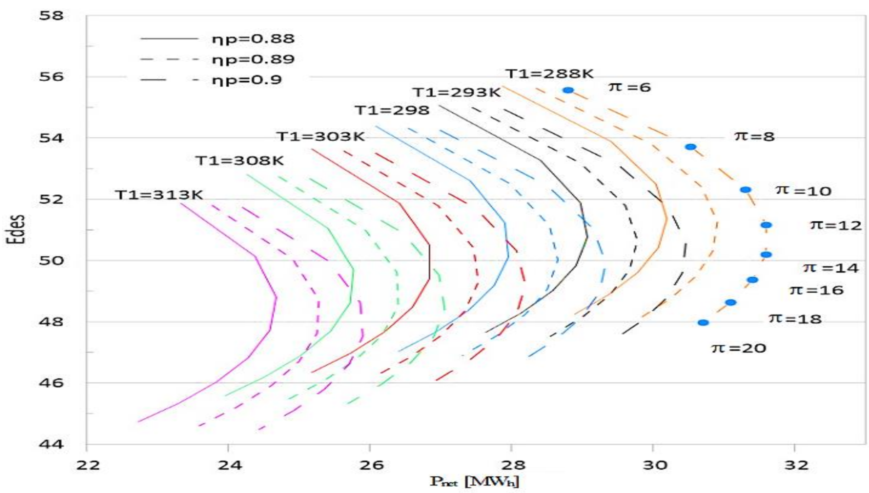

Figure 4: effective parameter on the GT P_{net} and Edes.

So, the maximum power (P_{net}) was 30 MWh in January 2020 without using any inlet air cooling system. While, the minimum power was 26.65 MWh in June 2021, as shown in Figure.5.

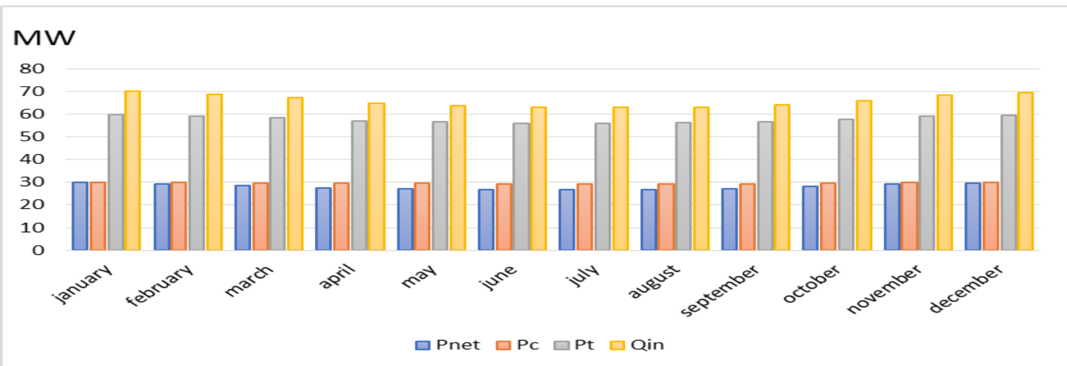

Figure 5: Expected performance of the entire GT (2021) without inlet air cooling system.

In addition, the effect of ambient condition effect may be neglected on η_I, η_{Ic}, η_{IIc}, η_{IIcc}, η_{It}, and η_{IIt}. But during the hot months, they decrease, while, η_{II} increases, as shown in Figure.6.

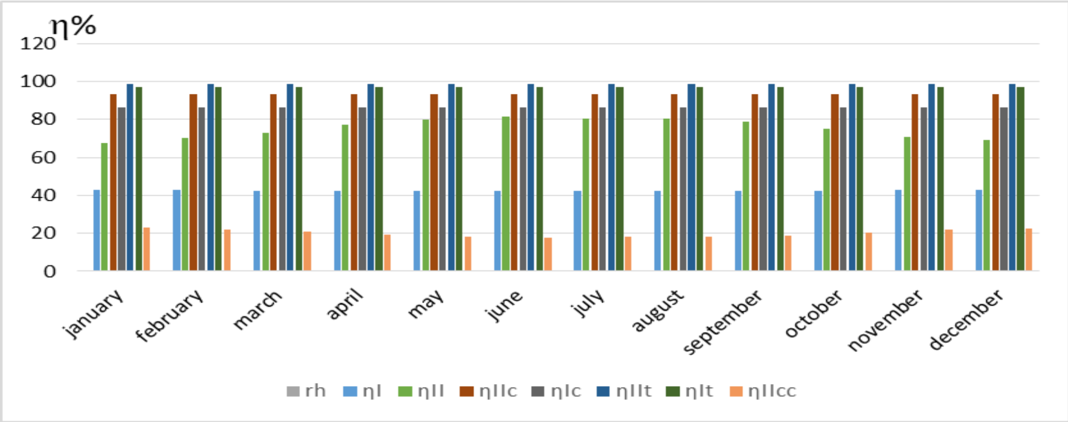

Figure 6: Expected η_I, η_I of the GT and each of its components within one year (2021) without inlet air cooling system.

5.5. *Inlet air cooling system effect*

This section discusses the effect of using a multi-stage cooling system on the performance of each GT component. So, every parameter ratio is the ratio between this parameter without using the cooling system to with cooling multi-stage inlet air cooling system.

5.5.1. The Effect on Inlet Air Temperature

There are many types of GT inlet air cooling systems. The effect of rh is clear when using an inlet air fogging system (Tf) and using an inlet air multi-stage cooling system (Tcomb) too. However, there is no effect of rh using chiller cooling system (Tch). In addition, for using Tf or T_{comb} systems, there is a relation between T_a and T_1 as by increasing T_a the T_1 increases, but by using Tch system T_1 is constant even if T_a changed, as shown in Figure.7.

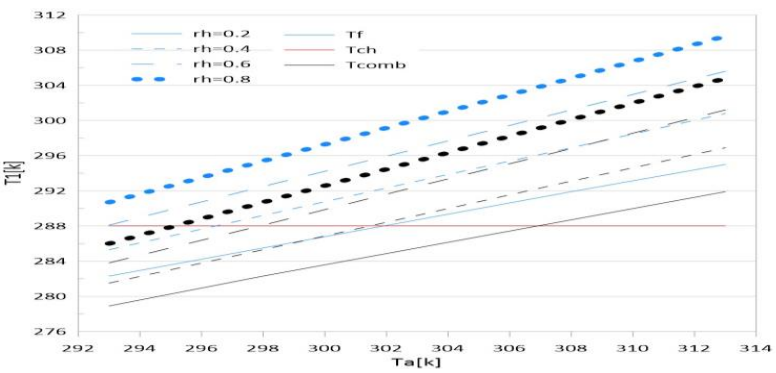

Figure 7: The effect of different inlet air cooling system at variable T_a and relative humidity rh on inlet air temperature T_1

So, regarding using Tcomb system the maximum decrease of the inlet air gas turbine is about 14.7^{o} as shown in Figure.8.

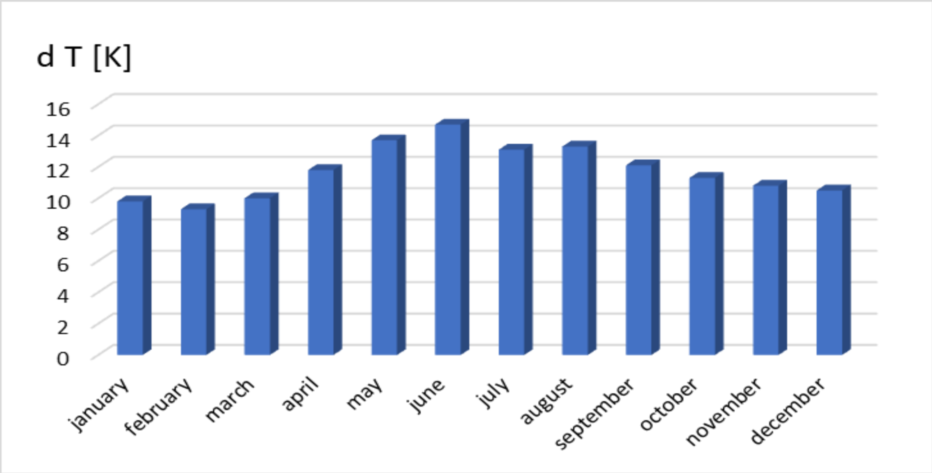

Figure 8: The effect of using the multi-stage inlet air cooling system.

5.5.2. The effect on the overall GT performance.

The results indicated the effect of using a multi-stage inlet air cooling system on the overall gas turbine performance as shown in Figure 9, Figure 10, as the curves shown clarify that at low Ta the curves are closer than at higher Ta, that indicated to the effect of rh on the used inlet air cooling system effective is clearer at high Ta. In addition, Figure.9, indicated to the increase of rh leads to a decrease P_{netR}. Moreover, the used inlet air cooling system leads to an increase of recovered power ratio (P_{netR}) with increasing Ta. Also, in Figure.10, an increase of Ta leads to an increase η_{IR} and, decrease η_{IIR} as shown in Figure 13, this effect on the 1st and 2nd low efficiency of the gas turbine is very small and could be ignored.

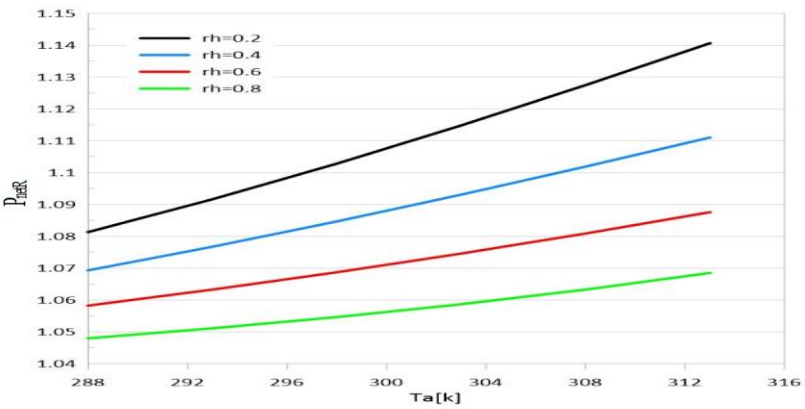

Figure 9: The effect of multi-stage cooling system on the P_{netR}

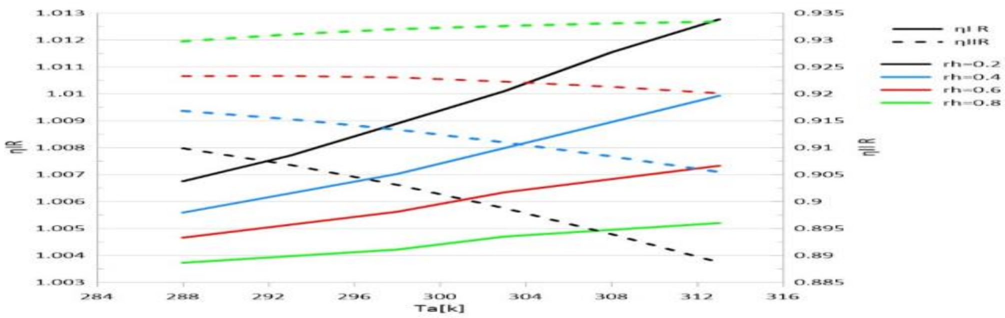

Figure 10: The effect of the multi-stage cooling system on the η_{IIR} and η_{IR}

The maximum expected additional power capacity of GT when using a multi-stage inlet air cooling system for one year (2020) is about 2.7 MW_h in June as shown in Figure.11, and Figure.12, where PnetR= (Pnet with cooling)/ (Pnet without cooling)

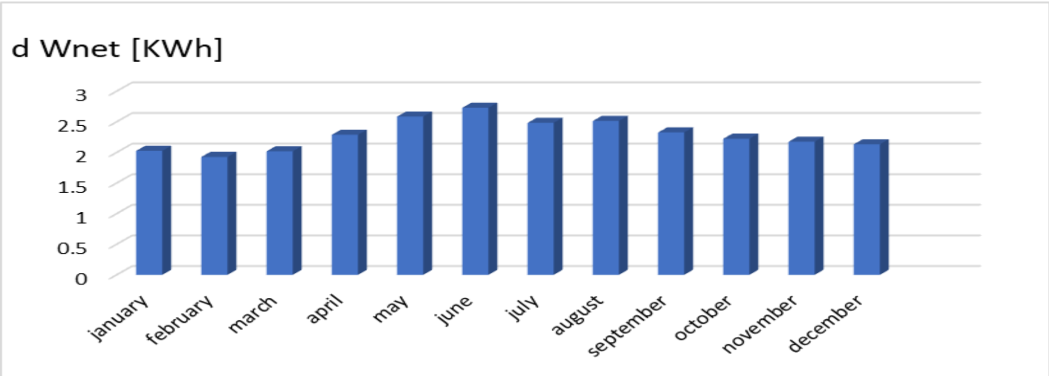

Figure 11: The expected additional power capacity of GT when using a multi-stage inlet air cooling system for one year (2021).

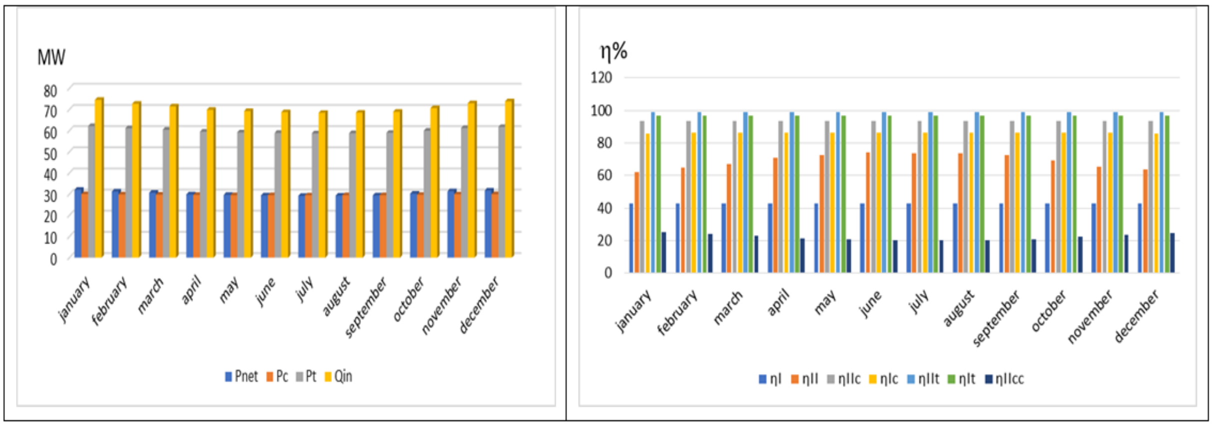

Figure 12: Expected performance of the GT and each of its components within one year (2021) with multi-stage inlet air cooling system.

6. Multi-stage cooling system cost analysis

Cost analysis is one of the most important indicators which helps in choosing the optimum design for the inlet air cooling system.

6.1. The Capital Cost

The multi-stage cooling system total capital cost and price including pipes, control, and measurement system, pumping station with accessories, installation, pipe installation, and mechanical works, tanks, Absorption chiller cooling system, and fogging cooling system on average 310,000$. With the straight-line method of depreciation, the annual depreciation is 12,400$/year with an average lifetime of 25 years.

6.2. The Operation Cost

The operation cost included the pumping station electrical power consummation, and the electricity price of the Gas Turbine cycle power plant is 0.097 $/kW, So the recovered power price is about 1,774,001 $/year. Also, maintenance costs include spare parts, and workers each year in about 10,000 $/year. Therefore, the multi-stage cooling system operation total cost is 443,891.61 $/year.

6.3. The final cost analysis

The net gained = economically recovered – (operation total cost + the annual depreciation).

The net gained = 1,774,001 – (443,891.61 + 12,400) = 1,317,709 $/year

The above equation results mean that, the system will cashback during the first year.

7. Conclusions

The analyses of the gas turbine performance through the effect of varying ambient conditions and the use of different cooling systems to improve the gas turbine performance is performed. The following are the conclusions of the most important issues from this study: -

1. The effect of increasing inlet air temperature T_o, as that leads to a decrease both of net output power Pnet, and η_i, in order ηii increases.
2. The increases in T_o leads to decrease the exergy destruction E_{des}.
3. Increasing the pressure ratio π leads to an increase $\eta_{I,}$ while P_{net} increases up to a certain pressure ratio, over than this value the P_{net} decreases again. In order, both ηii and E_{des} decrease.
4. Increasing the polytropic efficiency η_p leads to improve η_{II} and E_{des}. In order, has a bad effect on η_I and P_{net}.
5. Using the inlet air cooling system leads to It is worth mentioning that the maximum power output P_{net} increase is about 14.3% at the maximum ambient temperature (313^{o}K), with very small effect on 1st and 2nd law efficiencies of the overall gas turbine and could be ignored.

6. The multi-stage cooling system has the advantages of fogging cooling and absorption cooling and avoids their disadvantages.

Nomenclature			email	Air mass ratio
Edes	Exergy destruction	(KW_h)	**Acronyms**	
h_i	Inlet air enthalpy	(KJ)	CCPP	Combined Cycle Power Plant
h_{0s}	Outlet air isentropic enthalpy	(KJ)	EES	Engineering Equation Solver
h_0	Real outlet air enthalpy	(KJ)	GT	Gas Turbine
m_a	Air mass flow rate at ISO condition	(kg/hr)	ISO	International Organization for Standardization
m_{ISO}	Air mass flow rate at ISO condition	(kg/hr)	**Abbreviations**	
T0	Inlet Air Temperature	(K)	$\boldsymbol{\pi}$	Pressure ratio
Ta	Ambient Temperature	(K)	ps	Stage pressure ratio
S	Entropy	(KJ/Kg. K)	η	Efficiency %
Pc	Compressor Consumption Power	(KW_h)	η_{Is}	isentropic efficiency
P_{revc}	Reversible power	(KW_h)	η_p	and polytropic efficiency
P_{cISO}	Compressor Consumption Power at ISO condition	(KW_h)	Ɣ	Specific heat ratio (Cp/CV)
P_{cR}	Compressor Consumption Power ratio			
P_t	Turbine power generated	(KW_h)		
P_{tISO}	Turbine power generated at ISO condition			
P_{revt}	Reversible power	(KW_h)		
P_{net}	Gas turbine net output power	(KW_h)		
ρ	Density	(kg/m^3)		
V	Compressor volume	(m^3)		
m_g	Hot gas flow rate	(kg/hr)		
LHV	Low heating value	(kJ/kg)		
Q	Heat energy	(KW_h)		
Q_{ch}	Chiller load energy	(KW_h)		

References

[1] Mohtaram Soheil, Chen Wen, T. Zargar, Ji Lin, "Energy-exergy analysis of compressor pressure ratio effects on thermodynamic performance of ammonia water combined cycle", Energy Conversion and Management 134 (2017).

[2] R. K. Dinindu, "Effect of cooling charge air on the gas turbine performance and feasibility of using absorption refrigeration in the “Kelanitissa” power station, Sri Lanka", Master of Science Thesis, STOCKHOLM, (2014).

[3] N.N. Farah, Espindola Juan, M. S. Osama, and S. Amano Ryoichi, "Energy, Exergy, and Emission Analysis on Industrial Air Compressors”, Journal of Energy Resources Technology, (2021).

[4] M. M. El-Awad, M. A. Siraj and H. A. Saeed, "Feasibility of Gas Turbine's Inlet-Air Cooling by Air

Washing and Evaporative Cooling in the Dry and Hot Climate ", UofKEJ Vol. 2 Issue 1 pp. 10-15 (February2012).

[5] K. Ibrahim Thamir, M. M. Rahman, and N. Abdalla Ahmed, " Optimum Gas Turbine Configuration for Improving the performance of Combined Cycle Power Plant", Available online at www.sciencedirect.com, Procedia Engineering 15, 4216 – 4223, (2011).

[6] S.K. Abhishek, A. Singhania, A. Kumar, R. Ranendra, and M.B. Kumar, " Improvement of Gas Turbine Power Plant Performance: A Review", International Journal of Innovative Research in Engineering & Management (IJIREM) ISSN: 2350-0557, Volume-4, Issue-3, (May-2017).

[7] Siemens Power and Gas Division, Bid Ref. No. EG1020 (2015/05) PG ES SGEA PROP, II_0004010000_EG1020 _ EKH_ Gas Compressor_R2.doc, Rev:2, Vol / Section: II / 0.4.1, page: 7 of 9, EEHC, Cairo Egypt, (2015).

[8] S.H. Yousef, Najjar, and M. Ahmad Abubaker, "Indirect evaporative combined inlet air cooling with gas turbines for green power technology", international journal of refrigeration 59, (2015) 235–250.

[9] Masheiti Salah, Abdusamad Jamal, Shamekh Awad, and Bodalal Awad, " Improvement of overall performance of Benghazi North combined power plant by retrofitting the Inlet Cooling with a single effect absorption chiller", 3rd International Conference on Renewable Energies for Developing Countries (REDEC), (2016).

[10] B. Mohanty, and G. Paloso, “Enhancing gas turbine performance by intake air cooling using an absorption chiller”. Heat Recovery Systems and CHP 15 41-50, (1995).

[11] D.A. Kolp, W.M. Flye, and H.A. Guidotti, “Advantages of air conditioning and supercharging an LM6000 gas turbine inlet, engineering for gas turbine and power”, Transactions of the ASME 117, 513–527, (1995).

[12] Kareem Emara, Ahmed Emara, El Sayed Abdel Razek, “Turbocharger Selection and Matching Criteria in A Heavy-Duty Diesel Engine”, International Journal of Scientific & Engineering Research Volume 7, Issue 12, December-2016 ISSN 2229-5518.

[13] K.N. Abdalla, and Z.A.M. Adam, “Enhancing gas turbine output through inlet air cooling”. Sudan Engineering Society Journal 7-14 52, (2006).

[14] Abdelghany Tarek, "Effect of Climatic Conditions on Gas Turbines Performance", Master of Science Thesis, Helwan University, Cairo, (2021).

[15] S.S. Ali and Safari Mahbod, " Study and Comparison of Inlet Air Cooling Technique of Gas Turbines and Their Effects on Increase of the Efficiency and Outlet Power", International Journal of Materials, Mechanics and Manufacturing, Vol. 2, No. 4, (November 2014).

[16] Saleh S. Baakeem, Jamel Orfi, Hany Al-Ansary,” Performance improvement of gas turbine power plants by utilizing turbine inlet air-cooling (TIAC) technologies in Riyadh, Saudi Arabia”, Applied Thermal Engineering, Volume 138, 25 June 2018, Pages 417-432.

[17] Muhammad Reshaeel, Adeel Javed, Ahmad Jamil, Majid Ali, Mariam Mahmood, Adeel Waqas, “Multiparametric optimization of a reheated organic Rankine cycle for waste heat recovery-based repowering of a degraded combined cycle gas turbine power plant”, Energy Conversion and Management, Volume 254, 15 February 2022, 115237.