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Performance of Gas Turbine Power Plant with Evaporative Air Pre-Cooler System Using Energy and Exergy Concept

[a]Enangha Obeten Ntor, [a]Adekunle Moshood Abioye*

[a]Department of Mechanical/Production Engineering, Abubakar Tafawa Balewa University, Bauchi 740102, Nigeria

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ABSTRACT

One of the main challenges in building a Solar Power Generation System at home or a Home Solar Power Plant (Home SPP) is choosing component specifications according to price. The main components of Home SPP are photovoltaic (PV) panels, inverters, and wiring systems. Given the strict price constraints, the selection of parts available on the commercial market is generally of low quality. However, low-quality components can still provide a significant advantage by optimizing the plant design. This research proves that the proper configuration can reduce electricity bills by 52.2%. This configuration does by choosing a Grid Tie Inverter (GTI) with a high working voltage and a 12 Volt PV configured in a parallel series circuit to work at 24 Volts. In addition, the 12 Volt PV panels configured in series to 24 Volts are proven to increase the conversion efficiency.

KEYWORDS

Energy
Efficiency
Exergy
Evaporative cooling
Gas turbine
Performance analysis

INTRODUCTION

Ever since man appeared on planet Earth, he has always sought forms of energy to transform his environment. The pursuit of sustainability has led to using and development of various energy sources. The rapid growth in electricity demand in certain countries is driving high investments in new power plants in the short term. Gas turbine power plants represent a prime option in this energy mix. Awareness of limited hydrocarbon resources, environmental and economic concerns, and ever-increasing demand for electricity necessitate the design of optimal gas turbine power plants from both a technical and cost perspective. Exergy analysis is based on the first and second laws of thermodynamics. It allows the characterization of the optimal analysis technique on energy systems and to identify energy levels and thermodynamically unfavorable processes in a system Radchenko et al. [1]. This method describes different energy flows and helps reduce several losses in the system. Thermodynamic analysis has been used to model energy systems, including advanced power plants, for nearly a century. The first law of thermodynamics is usually used to model a system; however, it is limited to determining the source of irreversibility in the system under consideration. As a result, an analysis based on energy balance can be misleading as it does not provide information about the internal losses in the system. For example, energy

*Corresponding Author: amadekunle@atbu.edu.ng

analysis in adiabatic systems such as adiabatic compressors, combustors, and thermal converters can lead to the hasty conclusion that no energy loss occurs in these devices. Exergy is based on both the first and second laws of thermodynamics. Ahmadi et al. [2] prove that the exergy analysis clearly shows the places of energy dissipation in a process. The results of such an analysis can lead to improved operations and technologies. In addition, the analysis can quantify the heat quality in a waste stream. In addition, the analysis can quantify the heat quality in a waste stream. In addition, Oyedepo et al. [3] define exergy as a means of assessing and comparing the reservoir of theoretically extractable work that we call energy resources. Resources are either matter or energy with properties that differ from the prevailing environmental conditions. Kamal et al. [4] examine the performance analysis and optimization of power plants with gas turbines. They found that large exergy losses occurred in the condenser. In this case, the energy could not be used elsewhere. They also suggest modifying the combustor due to the high exergy loss. Marzouk and Hanafi [5] perform a numerical simulation of the GE 7001 EA gas turbine using experimental data for compressor inlet air cooling. They also find out that irreversibility that takes place in the combustion chamber can be physical, chemical, or even nuclear exergy. Zeitoun [6] distinguishes between energy and exergy efficiency and outlines the essential features of exergy efficiency in a gas turbine power generator with two-stage inlet air cooling. They state that exergy efficiency often provides a more accurate understanding of performance than energy efficiency. Ahmadi et al. [2] presented a comparative study on the influence of different means of cooling the turbine inlet air on the thermodynamic performance of hybrid turbine power plants in humid climates. It has been reported that the highest plant efficiency and specific work is possible when closed-loop steam cooling is the least efficient. In geographic regions with significant electricity demand and the highest electricity prices during the warm months and high ambient air temperature, gas turbine inlet air cooling is a useful performance enhancement option. An exergy analysis is performed to determine the cost of exergy annihilation in each component and to evaluate the cost of a product from the system and the associated irreversibility cost ratio. It is a common misconception that evaporative cooling cannot be used in regions with high relative humidity. Such a misunderstanding prevents power plant operators from relatively inexpensively increasing the output of gas turbines on warm days or during the warmest hours of the day. Gas turbine generators (GTGs) are constant volumetric Kamal et al. [4] machines except those with inlet guide vanes (IGVs), meaning that the volumetric flow rate of the inducted air is fixed by the internal geometry and the speed of rotation of the shaft. Increasing the intake air mass flow increases the power generated by the GTG. One of the ways to increase the mass flow rate of air drawn in by the compressor is to increase the density of the combustion air by reducing its temperature, thereby increasing GTG power output. The increase in air density is achieved by evaporating water into the intake air, reducing its temperature following an adiabatic heat and mass transfer process, which correspondingly increases its density. The water vapor passes through the turbine causing a negligible increase in fuel consumption Sadighi Dizaji et. al. [7].

Since the compressor absorbs approximately 2/3 of gas turbine power generation, reducing gas turbine compressor power consumption by reducing inlet air temperature for the same gas turbine fuel consumption results in a proportional increase in gas turbine power generation capacity and heating rate decrease with the attendant commercial benefits. There are several alternatives to reduce the temperature of the incoming airstream, namely evaporative cooling with wetted media, spray cooling, inlet nebulization or mist cooling, and mechanical cooling air cooling (including thermal storage and absorption refrigeration). Of all these different technologies, only water evaporation-based technologies will be analyzed in this post.

EVAPORATIVE COOLING THEOREM

One of the systems available for increasing the stale air mass flow rate at a gas turbine is to reduce the temperature of the entering air. In evaporative cooling, sensible heat is transferred from the air to the water via an isentropic process and becomes latent heat as the water evaporates. During the air-steam mixing process, the water vapor becomes part of the air mass and carries the latent heat. The air's dry bulb temperature is reduced because it gives off sensible heat. The wet-bulb temperature of the air is not affected by latent vapor heat uptake because the water vapor temperature coincides with the wet-bulb temperature of air Carmona [8].

METHODOLOGY

Operating data for a gas turbine unit was collected from ISO (International Standard Operations) conditions ($T_0 = 25\ °\mathrm{C},\ \ P_0 = 1.013\ \mathrm{bar},\ \text{and}\ \phi = 60\%$). A summary of the operating parameters of the gas turbine unit used for this study is presented in table 1 below. The analysis of the plant was divided into different control volumes, and the plant's performance was estimated using component-wise- modeling. Mass and energy conservation laws were applied to each component, and the plant's performance was determined for the simple system without air cooling technology and inlet air cooling technology. The thermodynamic modeling of the gas turbine with design data was solved using Engineering Equation Solver (EES) to get the calculated parameters for the selected gas turbine.

Plant Description

Gas turbine power plants consist of three components, including the air compressor (AC), the combustor (CC), and the gas turbine (GT). A schematic diagram of a simple gas turbine is shown in Figure 1: Ambient air is drawn into the system, compressed in the air compressor, and its temperature and pressure are further increased in the combustor. The products of combustion are expanded in the turbine to produce work. Compressor and turbine efficiencies are shown for each.

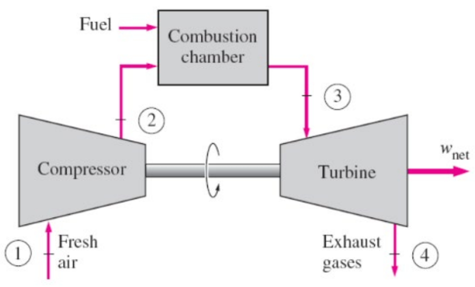

Figure 1. Open circle gas turbine power plant system

Table 1. Nominal performance at ISO conditions (25 °C and 60% RH)

Technical data	value
Total output (MW)	100 MW
Compressor stages	17
Compressor inlet pressure (bar)	1.013 bar
Compressor isentropic efficiency (%)	88%
Speed (rpm)	3000 rpm
Compressor exit temperature (°C)	662.8 °C
Pressure ratio	10.38 bar
Air flow rate (kg/s)	4.7037 kg/s
Fuel gas mass flow rate (kg/s)	14.59 kg/s
Maximum temperature (°C)	1329 °C
Turbine stage	5
Turbine isentropic efficiency (%)	89%
Fuel natural gas (LHV) kj/kg	42500 kJ/kg
Exhaust temperature (°C)	774.1 °C
Compressor inlet temperature (°C)	25 °C
Evaporative cooling effectiveness	90
Fogging cooling effectiveness	95

Plant Energy Model without Inlet-air Cooling

Compressor model:

The compressor used in a gas turbine power plant is of the axial flow type. The thermodynamic losses in an axial flow compressor have been modeled, and the temperature and air pressure in the compressor section is given by the expression Betelmal and Farhat [9].

$$P_2 = P_1 r_p \tag{1}$$

Compressor discharge temperature is given as;

$$T_2 = T_1(1 + R_{c,p}) \tag{2}$$

Where

$$R_{c,p} = \left[\frac{r_p^{\left(\frac{\gamma-1}{\gamma}\right)} - 1}{\eta_c}\right] T_2 = T_1(1 + R_{c,p}) \tag{3}$$

The work required to drive the compressor per unit mass is given as

$$w_c = m_a c_{pa}(T_2 - T_1) \tag{4}$$

The specific heat of air at constant pressure and the specific heat of the gas at constant pressure was assumed to function in a temperature range of 200k < T < 800k Deng et al. [10].

$$C_{pa}\,(T) = 1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.4903T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right) \tag{5}$$

$$C_{pa}(T) = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.22442T^3}{10^{10}}\right) \tag{6}$$

Combustion chamber model:

The energy balance of a combustor with a steady flow is shown in Equation 7. Also, the pressure drops associated with the combustor are assumed to be 3% for this analysis.

$$m_a\, c_{pa} T_2 + m_f\, LHV + m_f c_{pf} T_f = m_f c_{pf} T_3 \tag{7}$$

The air-fuel ratio is expressed as Carmona (2015)

$$f = \frac{c_{pg} T_3 - c_{pa} T_1 \left(1 + R_{pg}\right)}{LHV + c_{pf} T_f - c_{pg} T_3} \tag{8}$$

The mass of fuel burned in the combustion chamber is given as $m_f = f \times m_a$

The mass flow of combustion products is given as

$$m_{g\,=}\, m_a + m_f \tag{9}$$

Heat supply to the combustion chamber

$$Q_{add} = m_g LHV \tag{10}$$

Gas turbine model

The turbine exit temperature is expressed as follows

$$T_4 = T_3\left(1 - R_{p,g}\right) \tag{11}$$

Where:

$$R_{p,g} = \left(1 - \left(\frac{p_4}{p_3}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right) \tag{12}$$

Work done by the turbine per unit mass

$$W_T = m_g c_{pg} (T_3 - T_4) \eta_T \tag{13}$$

The network is specified as

$$W_{net} = W_T - W_C \tag{14}$$

The overall thermal efficiency is given as

$$\eta_{th} = \frac{W_{net}}{Q_{add}} \tag{15}$$

The specific fuel consumption (SFC)

$$SFC = \frac{3600 \times m_f}{W_{net}} \tag{16}$$

The heat rate (HR)

$$HR = \frac{SFC}{\eta_{th}} \tag{17}$$

Plant Energy Model with Inlet Air Cooling

Evaporative Cooling:

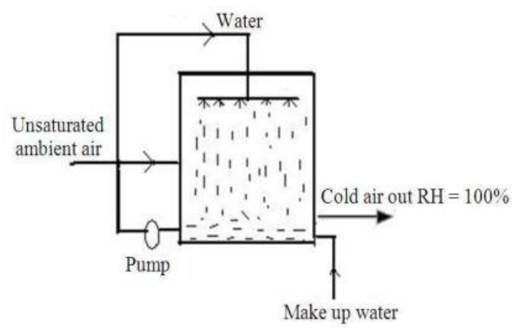

Figure 2. Schematic diagram of evaporative cooling

In evaporative cooling, sensible heat from the air is transferred to water because of latent heat as the water evaporates. The water vapor became part of the air and carried the latent with the dry bulb temperature decrease because it gives up the sensible heat. The air wet bulb temperature is not affected by the absorption of latent heat in the vapor because the water vapor enters the air at the wet bulb temperature M.M. Alhazmy and Najjar [11].

The heat interaction between the ambient air and the saturated air is presented as

$$CP_a(T_a - T_1) = (\omega_1 - \omega_a)h_{fg} \tag{18}$$

Where ω = the specific humidity and is calculated at a certain temperature

$$\omega = \frac{0.662 \times pv}{p - pv} \tag{19}$$

Where $pv = \emptyset \times p_{sat.}$ is the partial pressure of vapor and $\emptyset$ = the relative humidity and p_{sat} = the saturation pressure of air corresponding to the desired temperature. And ω_a and ω_1 are humidity ratios before and after the saturation, respectively. In general, ω is related to the water vapor pressure at saturation, as presented in equation 20 by Cengel and Boles [12].

$$T_1 = T_a - (T_a - T_{wb}) \times \eta \tag{20}$$

Where:
T_a = Ambient temperature
T_1 = cooler outlet temperature
T_{wb} = wet bulb temperature
η = evaporative cooler efficiency
It is believed that the working fluid flowing through the compressor is an ideal mixture of air and water vapor. The total enthalpy of atmospheric air is given in Equation 21 below Cengel and Boles [12].

$$h = h_a + \omega \times h_v = C_P \times T + \omega \times h_g \tag{21}$$

Where: $h_{a,}$ and h_v are the enthalpy of dry air and enthalpy of water vapor.
The enthalpy of water vapor can be evaluated approximately as in equation 22.

$$h_v = (2500.9 + 1.88 \times T) \tag{22}$$

The total temperature of the fluid leaving the isentropic efficiency compressor can be found using the ideal gas relationship obtained in equation 23 Ehyaei et al. [13].

$$T_{02} = T_1 + \frac{T_{2-}T_1}{\eta_c} = T_1 + \frac{T_1}{\eta_c} + \left[r_p^{\frac{y_{a+1}}{y_a}} - 1 \right] \tag{23}$$

Where:
r_p = compression ratio
y_a = specific ratio

Similarly, the total temperature leaving the turbine having isentropic efficiency is given in equation 24

$$T_{04} = T_{03} - \eta_t (T_3 - T_4) \tag{24}$$

The compressor is calculated from the mass flow and the enthalpy change over the compressor Egware et al. [14].

$$W_c = m_a(1+\omega) \times \left(C_{pa}T_{02} - T_1\right) + \omega(h_{02} - h_1) \tag{25}$$

Similarly, the turbine work is obtained as

$$W_T = \left(m_a + (1+\omega)m_f\right) \times \left(C_{pg}T_3 - C_{pg}T_{04}\right) + \omega(h_3 - h_{04}) \tag{26}$$

The energy of exhaust gas E_G is obtained by Majed M. Alhazmy et al. [15].

$$E_G = \left(m_a + (1+\omega) + m_f\right) \times \left(C_{pg}T_{04}\right) - \left(C_{pg}T_{01}\right) + \omega(h_{04} - h_{01}) \tag{27}$$

The net power obtained from the gas turbine is given as

$$W_{net} = W_T - W_C \tag{28}$$

The thermal efficiency of the gas turbine is given as

$$\eta_{th} = \frac{W_{net}}{m_f \times LHV} \tag{29}$$

The specific fuel consumption of the gas turbine is computed by

$$SFC = \frac{3600 \times m_f}{W_{net}} \frac{kg}{kwh} \tag{30}$$

EXERGY ANALYSIS

The exergy flow rate of the air stream leaving the compressor

$$Ex_2 = m_a\left[c_{pa}\left(T_2 - T_{ref}\right) - \left(c_{pa} ln\left(\frac{T_2}{T_{ref}}\right) - Rln\left(\frac{P_2}{P_{ref}}\right)\right)\right] \tag{31}$$

Exergy destruction rate or irreversibility in compressor output current

$$I_c = W_c - Ex_2 \tag{32}$$

The exergy balance in the combustion chamber is expressed as

$$Ex_f + Ex_2 = Ex_3 + I_{cc} \tag{33}$$

Exergy rate in fuel is expressed by Ku et al. [16].

$$Ex_f = m_f \left[c_p^{-h}(T_3 - T_{ref}) - T_{ref} c_p^{-s} ln\left(\frac{T_3}{T_{ref}}\right) + RT_{ref} ln\left(\frac{P_3}{P_{ref}}\right) + \sum_j x_j e_{jo} RT_1 \sum_j ln\gamma_j \, x_j \right] \tag{34}$$

Exergy destruction rate in the combustion chamber exit stream

$$I_{cc} = m_g T_{ref} \left[c_p^{-s} ln\left(\frac{T_3}{T_2}\right) - Rln\left(\frac{P_3}{P_2}\right) \right] \tag{35}$$

The overall exergetic efficiency of the system expresses the ratio of the usable work output to the maximum achievable work output Marzouk and Hanafi [5].

$$\xi_{all} = \frac{W_{net}}{Ex_{fuel}} \tag{36}$$

The rate of exergy destroyed in a component explains the component's efficiency defects and can be represented by the component exergy destruction ratios

$$y_{D,k} = \frac{I_k}{Ex_{ln}} \tag{37}$$

Evaporative Exergy Analysis

In general, the expression for the exergy destruction, Ahmed et al. [17].

$$I = T_o \left[(S_{out} - S_{in}) - \sum_{i-1}^{n} \frac{Q_i}{T_i} \right] \geq 0 \tag{38}$$

and the exergy balance for each component of the coupled GT and refrigeration cycle is expressed by Ku et al. [18].

$$E_{in} + E^Q = E_{out} + W + I \tag{39}$$

In the final expressions, different magnitudes of exergy destruction terms due to irreversibility are given for each component in the gas turbine and the proposed air cooling system.

Air Compressor

$$I_{comp,air} = m_a(1 + \omega_1)T_o \left[c_{pa} \ ln\left(\frac{T_2}{T_1}\right) - R_a \ ln\left(\frac{P_2}{P_1}\right) \right] \tag{40}$$

$$W_{eff,comp} = W_{comp} + I_{comp} \tag{41}$$

Combustion Chamber

$$I_{comb\ comber} = m_a T_0 (1 + f + \omega_1)\left[c_{pg}\ ln\left(\frac{T_3}{T_0}\right) - R_g\ ln\left(\frac{P_3}{P_0}\right)\right] - (1 + \omega_1)\left[c_{pa}\ ln\left(\frac{T_2}{T_0}\right) - R_a ln\left(\frac{P_2}{P_0}\right)\right] + T_0 \Delta S_0 \quad (42)$$

Gkoutzamanis et al. [19].

$T_0 \Delta S_0$ = rate of exergy loss in combustion or reaction = $m_a \times f \times NCV(\varphi - 1)$

$$Q_{eff,comb} = Q_{comb} + I_{comb} \quad (43)$$

Where:

$$R_g = \frac{C_{pg}(\gamma_g - 1)}{\gamma_g}$$

NVC = fuel net calorific value
φ = typical values of some industrial fuel

The effective heat to the combustion chamber is expressed as in equation 43

Gas Turbine

According Sadighi Dizaji et al. [7] gas turbine model with evaporative is given as in equation 44

$$I_{gas\ turbine} = m_a (1 + f + \omega_1) T_0 \left[c_{pg}\ ln\left(\frac{T_4}{T_3}\right) - R_g\ ln\left(\frac{P_4}{P_3}\right)\right] \quad (44)$$

$$W_{eff,t} = W_t - I_t \quad (45)$$

RESULTS AND DISCUSSION

The gas turbine cycle's power output and thermal efficiency were calculated for different ambient air temperatures and relative humidity of 60%. Thermodynamic properties such as specific heat, humidity ratio of air, and enthalpy of steam and combustion gases were evaluated as a function of temperature, pressure, and relative humidity using Engineering Equation Software (EES). Figures 3, 4, and 5 show the change in power output, thermal efficiency, and specific fuel consumption for turbines operating at 60% relative humidity without an inlet air cooling system. Thus, a 10-degree increase in ambient temperature resulted in a 0.10% drop in rated gas turbine efficiency, and a 3-degree increase in ambient temperature resulted in a 4000 kW drop in rated gas turbine power. Furthermore, an increase in ambient temperature by 10 degrees leads to an increase in specific fuel consumption by 0.00701 [kg/kWh].

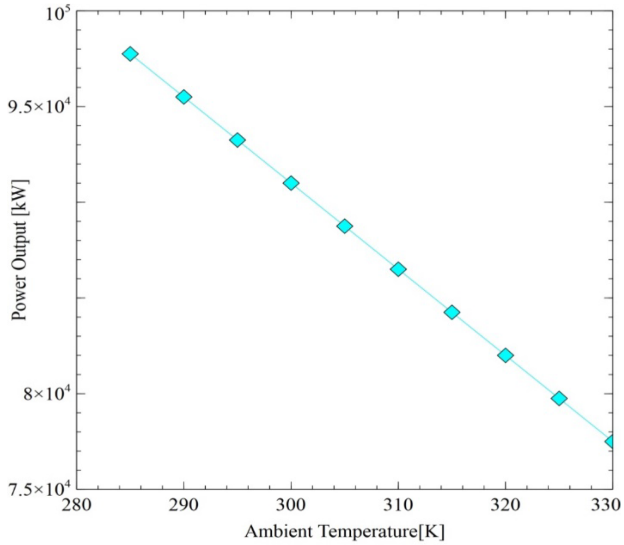

Figure 3. Power output vs ambient temperature for gas turbine without inlet air cooling system

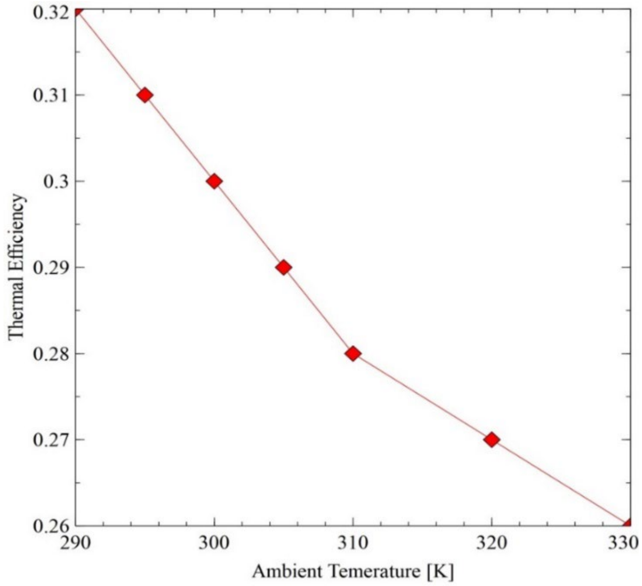

Figure 4. Efficiency vs ambient temperature for gas turbine without inlet air cooling system

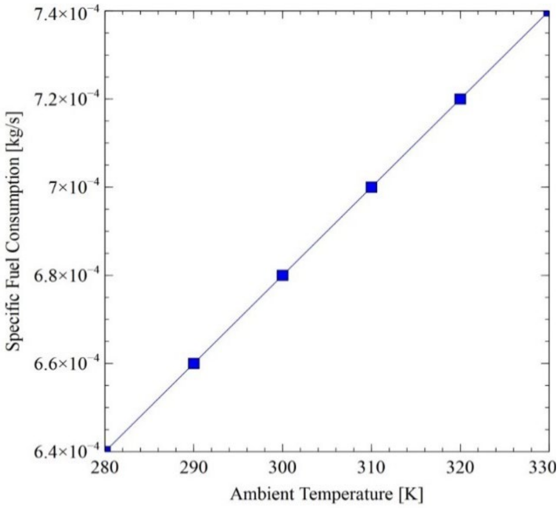

Figure 5. Specific fuel consumption vs ambient temperature for gas turbine without inlet air cooling system

The gas turbine power output is presented in Figure 6 for evaporative cooling inlet conditions. Note that the power output obtained is lower at operating conditions without inlet air cooling technology than when the evaporative cooling techniques are employed. Figure 7 presents the gas turbine thermal efficiency with inlet air cooling technology as occurred for the power output results at an air intake temperature of the air cooling process enhances the gas turbine thermal efficiency from 0.29% to 0.38% respectively.

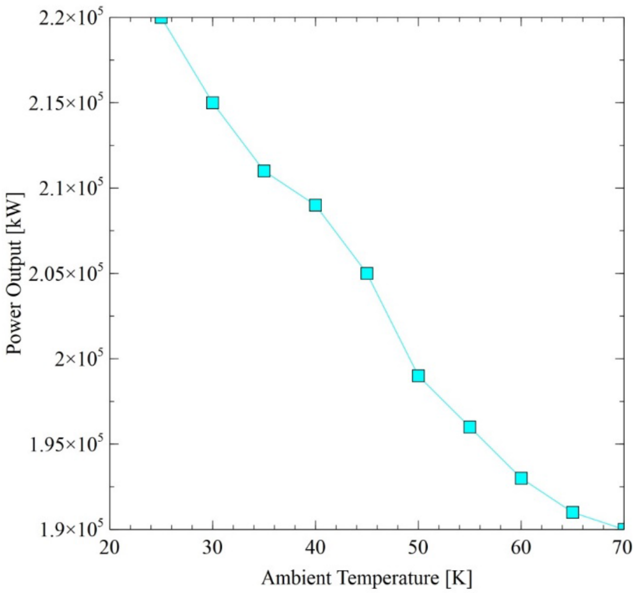

Figure 6. Effect of ambient intake temperature on the gas turbine power output using evaporative cooling

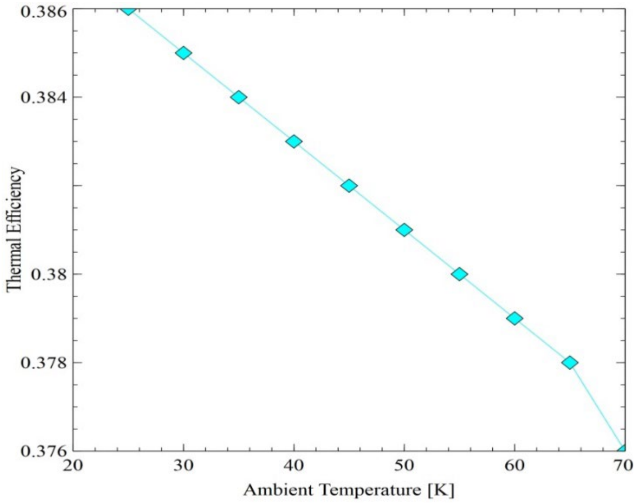

Figure 7. Effect of ambient temperature intake temperature on the gas turbine thermal efficiency employing evaporative cooling system

Table 2 shows the results of the plant's energy performance output with an evaporative air cooling system. From the table, it can be seen that the plant performed maximally when retrofitted with an evaporative cooling system with a percentage difference in thermal efficiency of 9% and with a network output increase of 96145KJ/Kg. At the same time, the heat rate reduces at a difference of 137kj/kg and specific fuel consumption has a difference of 0.000137kg/s.

Table 2. Energy performance output of the plant

Plant analysis	Without cooling	Evaporative cooling
Thermal efficiency	29.85%	38.84%
Net Work output	123647kJ/kg	136011kJ/kg
Heat rate	9511kJ/kg	9374kJ/kg
SFC	0.0005014kg/s	0.0004877kg/s

Table 3 explains the results of the second law based on the components, and this shows that when all components are retrofitted with an evaporative cooling system, the plant's exergy efficiency and exergy loss increase by 2.05% and the plant's exergy loss decreases enormously, when retrofitted with an evaporative cooling system in compressor, combustor and gas turbine by 3371.88%, 15548.49%, and 7122.50%.

Table 3. Exergy destruction rate and component efficiency

Component	Compressor	Combustion chamber	Turbine
Without inlet cooling technology			
Exergy destruction rate (kW)	11841.38kW	66368.76kW	26352.57kW
Component efficiency (%)	70.20%	30.07%	60.35%
Evaporative cooling technology			

Component	Compressor	Combustion chamber	Turbine
Exergy destruction rate (kW)	351.18kW	426.85kW	369.99kW
Component (%)	72.21%	33.08%	62.54%

CONCLUSION

Operating parameters, including compression ratio, turbine inlet temperature, ambient temperature, and relative humidity, significantly affect the performance of a gas turbine power plant. The results are summarized as follows. As ambient temperature increases, thermal efficiency and grid performance decrease while specific fuel consumption increases. The evaporative cooling system improved the performance of a gas turbine power plant because the thermal efficiency and grid performance are higher than the energy analysis without cooling. The air cooling system helps to reduce the environmental impact as the specific fuel consumption is lower than the energy analysis without cooling.

REFERENCES

[1] A. Radchenko, E. Trushliakov, K. Kosowski, D. Mikielewicz, and M. Radchenko, "Innovative turbine intake air cooling systems and their rational designing," *Energies*, vol. 13, no. 23, 2020, doi: 10.3390/en13236201.

[2] P. Ahmadi, I. Dincer, and M. A. Rosen, "Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants," *energy*, vol. 36, no. 10, pp. 5886–5898, 2011, doi: 10.1016/j.energy.2011.08.034.

[3] S. O. Oyedepo, R. O. Fagbenle, S. S. Adefila, and M. M. Alam, "Exergoenvironomic modelling and performance assessment of selected gas turbine power plants," *World J. Eng.*, vol. 13, no. 2, pp. 149–162, 2016, doi: 10.1108/WJE-04-2016-020.

[4] S. N. Omar Kamal, D. A. Salim, M. S. Mohd Fouzi, D. T. Hong Khai, and M. K. Yusri Yusof, "Feasibility Study of Turbine Inlet Air Cooling using Mechanical Chillers in Malaysia Climate," *Energy Procedia*, vol. 138, pp. 558–563, 2017, doi: 10.1016/j.egypro.2017.10.159.

[5] A. Marzouk and A. Hanafi, "Thermo-Economic Analysis of Inlet Air Cooling In Gas Turbine Plants," *J. Power Technol.*, vol. 93, no. 2, pp. 90–99, 2013.

[6] O. Zeitoun, "Two-stage evaporative inlet air gas turbine cooling," *Energies*, vol. 14, no. 5, 2021, doi: 10.3390/en14051382.

[7] H. Sadighi Dizaji, E. J. Hu, L. Chen, and S. Pourhedayat, "Using novel integrated Maisotsenko cooler and absorption chiller for cooling of gas turbine inlet air," *Energy Convers. Manag.*, vol. 195, pp. 1067–1078, 2019, doi: 10.1016/j.enconman.2019.05.064.

[8] J. Carmona, "Gas turbine evaporative cooling evaluation for Lagos - Nigeria," *Appl. Therm. Eng.*, vol. 89, pp. 262–269, 2015, doi: 10.1016/j.applthermaleng.2015.06.018.

[9] E. H. Betelmal and S. A. Farhat, "Energy and Exergy Analysis of a Simple Gas Turbine Cycle with Wet Compression," *Mech. Eng. Res.*, vol. 8, no. 1, p. 30, 2018, doi: 10.5539/mer.v8n1p30.

[10] C. Deng *et al.*, "Air cooling techniques and corresponding impacts on combined cycle power plant (CCPP) performance: A review," *Int. J. Refrig.*, vol. 120, pp. 161–177, 2020, doi: 10.1016/j.ijrefrig.2020.08.008.

[11] M. M. Alhazmy and Y. S. H. Najjar, "Augmentation of gas turbine performance using air coolers," *Appl. Therm. Eng.*, vol. 24, no. 2–3, pp. 415–429, 2004, doi:

10.1016/j.applthermaleng.2003.09.006.
[12] Y. a. Çengel, *Thermodynamics: An Engineering Approach*, 5th editio. 2006.
[13] M. A. Ehyaei, M. Tahani, P. Ahmadi, and M. Esfandiari, "Optimization of fog inlet air cooling system for combined cycle power plants using genetic algorithm," *Appl. Therm. Eng.*, vol. 76, pp. 449–461, 2015, doi: 10.1016/j.applthermaleng.2014.11.032.
[14] H. O. Egware, U. P. Onochie, and H. Itoje, "Effect of incorporating fogging inlet air cooling system: a case study of Ihovbor Thermal Power Plant, Benin City," *Int. J. Ambient Energy*, vol. 43, no. 1, pp. 2173–2179, 2022, doi: 10.1080/01430750.2020.1722231.
[15] M. M. Alhazmy, R. K. Jassim, and G. M. Zaki, "Performance enhancement of gas turbines by inlet air-cooling in hot and humid climates," *Int. J. Energy Res.*, vol. 30, no. 10, pp. 777–797, 2006, doi: 10.1002/er.1184.
[16] A. K. Mohapatra and Sanjay, "Comparative analysis of inlet air cooling techniques integrated to cooled gas turbine plant," *J. Energy Inst.*, vol. 88, no. 3, pp. 344–358, 2015, doi: 10.1016/j.joei.2014.07.006.
[17] A. H. Ahmed, A. M. Ahmed, and W. H. Al Doori, "Energy–exergy analysis of 70-MW gas turbine unit under the variable operation condition," *Heat Transf.*, vol. 49, no. 2, pp. 743–754, 2020, doi: 10.1002/htj.21636.
[18] A. K. Mohapatra and Sanjay, "Thermodynamic assessment of impact of inlet air cooling techniques on gas turbine and combined cycle performance," *energy*, vol. 68, pp. 191–203, 2014, doi: 10.1016/j.energy.2014.02.066.
[19] *et al.*, "Thermal Energy Storage for Gas Turbine Power Augmentation," *GPPS Proc. GPPS Zurich19*, 2019, doi: 10.33737/gpps19-tc-072.