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Original Article

Comparative Analysis of Environmental Sustainability and Performance Indices of Different Gas Turbine Inlet Cooling Systems

Enefiok Okon Usungurua[1*], **Imoh Ime Ekanem**[2], **Aniekan Essienubong Ikpe**[2]

[1]Department of Mechanical Engineering, Federal University of Technology, Ikot Abasi, Nigeria
[2]Department of Mechanical Engineering, Akwa Ibom State Polytechnics, Ikot Osurua, Ikot Ekpene, Nigeria

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***Corresponding author:**
Enefiok Okon Usungurua,
Email: enefiokusungurua@gmail.com

Abstract

Background: A thermodynamic analysis was performed on the inlet air cooling system of a natural gas-fueled turbine at the Afam power plant in Nigeria. The objective was to identify optimal conditions for environmental sustainability, performance output, fossil fuel economy, and cost-effectiveness.

Methods: Three cooling methods were evaluated: spray cooler with wetted media, fogging system, and mechanical chiller. Energy, exergy, and exergy-economic models for both the existing and modeled turbine units were compared. The developed program source code indicated that both sustainability and performance indices improved when the turbine inlet cooling system adhered to design specifications.

Results: The refrigeration method achieved the lowest inlet temperature at 15 °C and a sustainability index (SI) of 2.243. The SI decreased with increasing ambient temperatures due to higher total exergetic destruction. Regarding performance indices—net turbine output, energy efficiency, and exergy efficiency—the spray cooler had the highest values at 98.845 WM, 29.05%, and 28.49% respectively. This was followed by the mechanical chiller at 100.692 WM, 29.25%, and 29.03%, and the fogging system at 98.695 WM, 29.03%, and 28.45%. The base system recorded the lowest values at 95.70 WM, 28.69%, and 27.58%, attributed to increased compressor work. Both the environmental effect factor (EEF) and waste exergy ratio (WER) decreased with rising ambient temperature due to total exergetic destruction.

conclusion: The refrigeration cooling method is recommended due to its lower inlet conditions, minimal exergy destruction, and high environmental sustainability, performance efficiency, and cost-effectiveness. Inlet-air cooling remains an efficient technique for improved performance.

Keywords: Gas turbine, Environmental sustainability, Performance output, Cooling systems, Techno-economy

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Introduction

Gas turbine engines are essential components in power plants, aircraft, and heavy-duty vehicles. A standard gas turbine engine comprises three main components: the compressor, combustion chamber, and turbine, along with the generator.[1,2] While gas turbines have been globally used for power generation for many decades, their performance efficiency can be inconsistent due to varying climatic conditions. Specifically, the dry seasons often negatively affect the operation of gas turbines due to the adverse impact of ambient temperature on the air compression process. This results in reduced power output, which is problematic given the high power demands during these periods.[3] Perez-Blanco et al suggest that the effect of ambient air temperature on gas turbine output is significant, with output drops ranging from 0.58% to 0.89% per degree rise in ambient temperature.[4] Furthermore, there is a correlation between an increase in ambient air temperature and a substantial rise in the gas turbine heat transfer rate, along with an increase in system operating costs.[5] Therefore, it is expected that during extended dry weather conditions, as seen in Nigeria, power depletion becomes inevitable. To address this issue of power shortages during such harsh weather conditions, and in countries with high ambient air temperatures, compressor inlet air cooling has proven to be an effective solution."

Among the frequently used inlet air cooling methods, including spray cooler and wetted media, wet compression, mechanical chiller system, absorption chiller system, and fogging system technology, the latter has gained widespread application and acceptance over the years due to its effectiveness in power augmentation and cost efficiency.[5,6] Chaker and his colleagues defined fogging as the cooling of the compressor inlet air by injecting demineralized water at high pressure through atomizing nozzles.[5] As an evaporative system, this technology ensures a reduction in the gas inlet air temperature to reasonably low levels and increases the relative humidity to between 95% and 100%.[7] Gas turbine power augmentation through the use of an inlet air cooling system has several methods, as mentioned in the literature. However, the choice of a method for a particular system may be influenced by the location, available resources, and environmental properties.[8]

It is common practice to directly apply the fog inlet air cooling system to cool the compressor air inlet temperature. According to sustainability and performance indices of gas turbine systems, a decrease in inlet air temperature enhances the overall system efficiency and reduces heat flow rates.[3,5] This scenario also enhances air density under isobaric conditions. As a result, a higher mass flow rate leads to improved gas turbine power output. Increasing turbine capacity implies lower heat rates, indicating a rise in power output without a corresponding increase in fuel input.[5] Generally, the fog system technique boosts air density and mass flow rate by introducing water to dry air. It also elevates the specific heat ratio due to changes in the gas-phase composition, transitioning the compressor process from adiabatic to a quasi-isothermal process.[6]

In addition to the reduced power generation due to high ambient temperatures, other significant considerations include environmental sustainability, ecological efficiency, and performance indices of power generation systems. Environmental degradation is directly linked to high levels of exergy destruction, especially when not proportionally matched with the power output from the system in question. To develop a techno-economically and environmentally friendly power augmentation system, sustainability, ecological efficiency, and performance indices must be considered when selecting a turbine inlet air cooling method. Numerous studies in the literature have extensively examined turbine air cooling systems, focusing intently on their application, droplet size evaluation, and optimization.[9-11]

Athari et al. introduced and examined a gas turbine cycle incorporating fog cooling and steam injection, integrated with biomass gasification, through energy, exergy, and exergoeconomic analyses.[11] However, the study did not consider the sustainability and environmental impact of steam injection cooling.[12] Another study analyzed the inlet fogging process in a gas turbine system with varying water injection ratios, focusing on modeling the evaporation of the injected water droplets. Under carefully assumed conditions, a model was developed to calculate the time required for complete or partial evaporation of the droplet phase. Turbine inlet air cooling systems are generally categorized into refrigerative or evaporative systems. Among these, spray cooler and wetted media are the simplest, least expensive, and most widely applied techniques, especially for base load turbines.[12] This method involves passing the inlet air through a film of water carried by a medium, such as a honeycomb, which evaporates the water. It has been reported to be effective between 85 to 95%.[13] These techniques operate more efficiently during hot and dry weather but are less effective in high humidity conditions. Another technology, the fogging system, involves producing and introducing fog—tiny droplets of de-mineralized water—into the compressor's inlet air. The fog increases the water's exposed surface area, enhancing water evaporation into the air.[14,15] This technology is similar to wetted media in terms of cost, simplicity, and widespread application, boasting an efficiency of 95 to 100%. Droplet size can be adjusted (reduced) to improve the evaporation rate, but this increases the cost. Additionally, wet compression is an evaporative cooling technique where more fog is added to the inlet air than can be evaporated under the prevailing ambient conditions. Excess water, which remains un-evaporated, is carried by the airstream into the compressor, where further evaporation occurs. This cooling reduces compressor work and increases the turbine's output power. This method usually complements other cooling technologies.[16]

Turbine inlet air cooling systems are broadly categorized into refrigerative or evaporative systems. Among these, spray cooler and wetted media stand out as the simplest, least expensive, and most widely applied techniques, particularly for base load turbines.[12] This method involves passing the inlet air through a film of water carried by a medium, such as a honeycomb, leading to the evaporation of the water. The effectiveness of this technique has been reported to range between 85 and 95%.[13] These systems operate more efficiently during hot and dry weather but are less effective in high humidity conditions. Another technology, known as fogging system technology, involves the production and introduction of fog—tiny droplets of de-mineralized water—into the compressor's inlet air. The fog increases the water's exposed surface area, thereby enhancing water evaporation into the air.[14,15] This technology is comparable to wetted media in terms of cost, simplicity, and widespread application, boasting an efficiency of 95% to 100%. However, adjusting (reducing) the droplet size to improve the evaporation rate can increase the cost. Furthermore, wet compression is an evaporative cooling technique in which more fog is added to the inlet air than can be evaporated under the prevailing ambient conditions. Excess water, which remains un-evaporated, is carried by the airstream into the compressor, where further evaporation occurs. This cooling process reduces compressor work, thereby increasing the turbine's output power. This method typically complements other cooling technologies.[16]

Another method includes the mechanical chiller system and absorption chiller.[12] While evaporative cooling methods can only cool the inlet air to the wet-bulb temperature of the ambient air and not below,[17,18] refrigerative chiller technology can lower the inlet ambient air temperature to as low as 5.6 °C.[19] Mechanical chillers can be powered by an electric motor, steam turbine, or an engine, and they utilize cooling coils. These systems are more capital-intensive than evaporative systems. For absorption technology, the system reduces the compressor inlet air temperature to approximately 10 °C and employs Lithium Bromide (Li-Br) and water; with Li-Br serving as the absorption medium and water as the refrigerant. The required steam can be generated by the combustion turbine exhaust gases.[20] Absorption cooling systems can be designed as either single or double effect. The coefficient of performance (COP) for the single effect system is reported to range between 0.7 and 0.9, whereas the double effect system has a COP of 1.15.[20] Compared to mechanical chillers, absorption cooling systems have lower parasitic loads, higher capital costs, and lower operation and maintenance costs.

Marzouk and Hanafi[7] compared chiller cooling and evaporative cooling for a 264 MW turbine. They concluded that the evaporative cooler is more economical due to its low maintenance cost, low electricity consumption, and low capital cost. Perez-Blanco et al[4] investigated the performance enhancement of a turbine with an evaporative cooler. Their results showed that for each 5 °C decrease in inlet air temperature, the net output power and thermal efficiency increased by 5-10% and 2-5%, respectively. Indeed, gas turbine inlet air cooling has limitations and varying degrees of success related to economy, power requirement, and maintenance cost.

Young and Wilcock developed a self-consistent approach for modeling air-cooled gas turbines, which are particularly suitable for thermodynamic cycle calculations.[21] They emphasized the importance of representing losses in terms of irreversible entropy creation rather than total pressure loss. A set of models for the components of the cooling loss was presented, and sample calculations were used to illustrate the division and magnitude of the losses in the system. The results provided a firm foundation to enhance more accurate thermodynamic models of the air-cooled gas turbine.

Aliehyaei et al conducted comprehensive thermodynamic modeling and optimization of a fog inlet air cooling system for combined cycle power plants (CCPP) using a genetic algorithm.[22] The results showed that using an inlet air cooling system for the CCPP and optimizing it led to an increase in the average output power, first-law efficiency, and second-law efficiency by 17.24, 3.60%, and 3.50%, respectively, during three warm months of the year.

Kim and Perez-Blanco studied the potential of regenerative gas turbine systems with high fogging compression. The results showed that, compared to a dry cycle with no recuperation and a pressure ratio of 25, efficiency can increase from 45% to 51.5%, and specific work can increase from 410 to 680 kJ/kg when compression cooling and recuperation are implemented.[23]

Ahmadi Boyaghchi and Molaie employed advanced exergy analysis to examine the sensitivity of various components' exergy destruction and the performance of a combined cycle power plant in response to turbine inlet temperature and compression ratio parameters. They found that an increase in turbine inlet temperature led to an increase in avoidable endogenous exergy destruction in certain components: the air compressor by 13%, the combustion chamber by 130%, the high-pressure superheater by 343%, the low-pressure evaporator by 54%, and the air preheater by 74%.[24]

Marzouk and Hanafi conducted a thermal and economic study on chiller and evaporative cooling systems for a 264 MW gas turbine plant located in Korymat, southern Egypt.[7] For chiller cooling, the total yearly gas turbine output power due to the chiller system was 117 027 MWh, with an annual cost of $7 624 548.90, a net cash flow from the plant of $3 787 587.00, and a payback period of 3.3 years. The average gas turbine thermal efficiency was recorded at 36.46%. In contrast, with evaporative cooling, the total yearly gas turbine output power was 86,118 MWh, the annual cost was $1 524 779.70, the net cash flow stood at $4 503 548.50, and the payback period was 0.66 years. The average gas turbine thermal efficiency was slightly higher at 37.205%

Ondryas et al conducted a study in a co-generation plant, employing absorption chillers (powered by steam from gas turbine exhaust), vapour compression chillers (electricity-powered), and thermal storage systems to cool the intake air during elevated ambient temperatures. Their findings highlighted the high energy efficiency and cost-effectiveness of air chilling under such conditions.[25]

Chen et al. conducted an off-design performance analysis of a combined cooling, heating, and power (CCHP) system. This system consisted of a small-scale gas turbine, an exhaust-fired double-effect absorption chiller, and a heat exchanger. Their findings revealed that the system achieved energy savings when the gas turbine's power output was above 30% of its full load. Additionally, carbon dioxide emissions were reduced by 66.7% to 70.5% compared to a conventional separation system as the gas turbine's power output increased from approximately 30% to 100%. The study concluded that the performance of the CCHP system was mainly influenced by the combustor of the small-scale gas turbine.[26] Also, Kurzke introduced a performance modeling methodology that defines efficiency for both cooled single-stage and multistage turbines.[27]

Given the rising energy demands, increasing source costs, and environmental concerns related to waste, there is a pressing need for sustainable and renewable energy sources. Sustainability is crucial for addressing current ecological, economic, and developmental challenges.[7] This study evaluates and compares the environmental sustainability and performance metrics of various gas turbine inlet cooling systems. We modeled the basic inlet turbine system to assess thermo-sustainability, techno-economy, and eco-friendliness, establishing baseline performance efficiencies for cooling methods such as spray cooler, wetted media, fogging system technology, wet compression, mechanical chiller system, and absorption chiller after thorough analysis.

Materials and Methods

A schematic diagram of a basic gas turbine with an inlet fogging system is depicted in Figure 1, while the corresponding T-s diagram is shown in Figure 2. The fogging arrangement reduces the compressor's inlet temperature. The system comprises an air compressor (1-2, AC), a combustion chamber (2-3, CC), and the turbine (3-4, GT). Exergy from the ambient conditions at state 1 (T_0 and P_0) enters the GT system via the compressor. Mechanical work, in the form of exergy, is supplied to the compressor by the turbine shaft. Some exergy is lost in the compressor due to irreversibility, while part of the exergy in the AC is consumed during compression. The compressed air, carrying exergy, enters the CC at a higher pressure. In the combustion chamber, fuel and air mix under high turbulence, resulting in combustion and the flow of a high-pressure, high-temperature exergy stream into the turbine. Irreversibilities in the turbine lead to exergy destruction, part of which drives the compressor, with the remaining available as the net work produced by the turbine. The energy, exergy, and exergoeconomic models of the overall system are briefly presented below.[28,29]

Gas Turbine With Fogged Cooling System

The air and combustion products were assumed to behave as ideal gases. The gas turbine process was based on the Brayton cycle, with the gas turbine plant essentially consisting of a compressor, combustion chamber, and turbine.[30] Figure 1 depicts the gas turbine with the fogging system. Fresh water was drawn into the demineralization plant, where certain mineral deposits were removed, maintaining the water's pH at a maximum of 7.5. The demineralized water was then stored and pumped to the position nozzles. These nozzles atomized the demineralized water into tiny water droplets or fog. Each droplet, according to the nozzle design, was recommended to be less than 50 microns in size. Critical parameters of the fogger included the droplet mean size, their distribution pattern, and their extent of penetration into the air duct from their point of production. A T-s diagram of an open-type gas turbine system is presented in Figure 3.

ISO Operating Data for Gas Turbine Plant

The operating data for this study was sourced from the

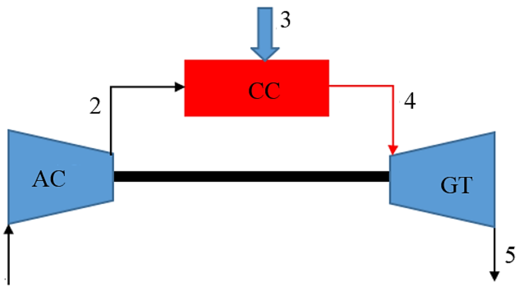

Figure 2. Simple Open Type Gas Turbine Configuration

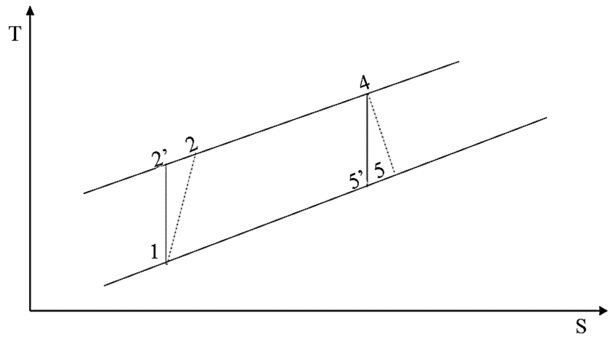

Figure 3. T-s Diagram of a Simple Open Type Gas Turbine Configuration

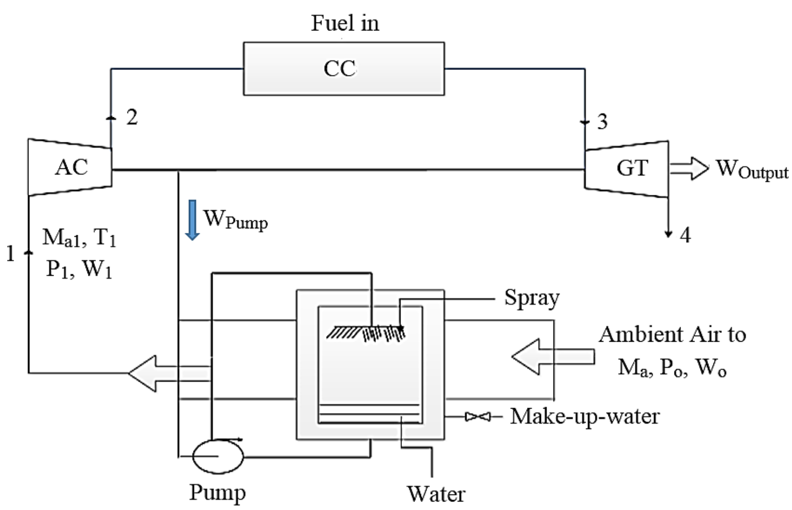

Figure 1. Fogged Gas Turbine System

Afam gas turbine plant in Nigeria. The design data for the base load is detailed in Table 1. The plant uses natural gas as fuel with the following composition: 88% methane, 5% ethane, 2% propane, 0.6% butane, and 4.4% carbon dioxide.

Thermodynamic Assumptions

The following assumptions were made in the analysis of the gas turbine system:

- The ambient pressure was maintained at 1.013 bar, and the temperature was set at 250 °C.
- The gas turbine plant was operated under steady-state conditions.
- A negligible pressure drop was considered in the combustion chamber.
- The air entering the compressor after fogging was completely saturated adiabatically, resulting in 100% relative humidity
- All components were assumed to have adiabatic boundaries.
- Both air and combustion gases were treated as ideal gases with constant specific heats at specified temperatures.

Energy Analysis of the Gas Turbine Unit

This thermodynamic technique is based on the second law of thermodynamics and serves as a means to evaluate and compare gas turbine units and their processes.

Gas Turbine Energy Balance

Energy balances for the schematic diagrams presented in Figures 1 and 2 are provided. The only difference between these two systems lies in the alteration of the compressor inlet temperature. The pressure values for both the base turbine and the fogged system remained consistent with ambient values. In the model analysis, the impact of fogging was incorporated by increasing the fluid mass, which was in turn a function of the specific humidity of the fogged air.

Table 1. Gas Turbine Plant Parameters

Component	Parameter	Unit	Value
Air compressor	Inlet tempt. T_{in}	K	298
	Outlet tempt. T_{out}	K	644.7
	Inlet pressure P_{in}	Bar	1.013
	Outlet pressure P_{out}	Bar	10.47
	Air flow $\dot{m}_a$	kg/s	427
	Isent. eff. η_{comp}	%	80
Combustion chamber	Inlet tempt. T_{in}	K	644.7
	Outlet tempt. T_{out}	K	1324
	Inlet pressure P_{in}	Bar	10.47
	Outlet pressure P_{out}	Bar	10.47
	Inlet gas flow $\dot{m}_f$	kg/s	7.039
	Inlet air flow $\dot{m}_a$	kg/s	427
	Outlet mass flow $\dot{m}_{out}$	kg/s	434.04
Gas turbine	Inlet tempt. T_{in}	K	1324
	Outlet tempt. T_{out}	K	834
	Inlet pressure P_{in}	Bar	10.47
	Outlet pressure P_{out}	Bar	1.013
	Inlet mass flow $\dot{m}_{in}$	kg/s	434.04
	Outlet mass flow $\dot{m}_{out}$	kg/s	434.04
	Isent. eff. η_{comp}	%	85

The Air Compressor

The work input to the compressor raises the enthalpy and temperature of the air from ambient conditions to values that vary based on the compression rate. The energy balance in the compressor, as shown in Figures 1 and 2, is expressed as follows[31]:

$$C_{pair}T_1 + W_{ac} = C_{pair}T_{2a} \tag{1}$$

The actual compressor work was

$$W_{ac} = C_{pair}\left(T_{2a} - T_1\right) \tag{2}$$

While the ideal specific compressor work was obtained as:

$$W_{AC'} = C_{pair}\left(T_2 - T_1\right) \tag{3}$$

By introducing isentropic relation in the compressor, the exit temperature of the compressor was obtained as[25]:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\propto - 1}{\propto}} \tag{4}$$

$$\therefore T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\propto - 1}{\propto}} \tag{5}$$

In the real case, the actual temperature T_{2a} after compression was obtained thus using the relationship with isentropic efficiency:

$$\eta_{comp.} = \frac{Ideal\ work}{Actual\ work} \tag{6}$$

It follows that Equation (6) can be expressed as:

$$\eta_{comp.} = \frac{T_2 - T_1}{T_{2a} - T_1} \tag{7}$$

$$\therefore T_2 = \eta_{comp.}\left(T_{2a} - T_1\right) + T_1 \tag{8}$$

Substituting T_2 Equations (8) into (5) gives:

$$\eta_{comp.}\left(T_{2a} - T_1\right) + T_1 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\propto - 1}{\propto}} \tag{9}$$

From Equation (9) the actual temperature after compression was obtained as:

$$T_{2a}=T_1\left[1+\frac{1}{\eta_{comp.}}\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right)\right] \tag{10}$$

Substituting Equation (10) into (2) the actual specific compressor work is given. Thus,

$$W_{ac}=C_{pair}\left\{T_1\left[1+\frac{1}{\eta_{comp.}}\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right)\right]-T_1\right\} \tag{11}$$

$$\therefore W_{ac}=C_{pair}\frac{T_1}{\eta_{comp.}}\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right) \tag{12}$$

Consequently for a given mass flow rate, $\dot{m}_a$ the actual compressor work can be calculated as:

$$W_{ac}=\frac{\dot{m}_a C_{pair}}{\eta_{comp.}}T_1\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right) \tag{13}$$

Equation (13) is further simplified to give:

$$W_{ac}=\dot{m}_a C_{pair}\left(T_{2a}-T_1\right)+\dot{m}_v\left(l_{g2}-l_{g1}\right) \tag{14}$$

The mass of vapour, $\dot{m}_v$ was related to the specific humidity (W)[32],

$$\mathrm{W}=\frac{\dot{m}_v}{\dot{m}_a} \tag{15}$$

$$\therefore \dot{m}_v=\mathrm{W}\dot{m}_a \tag{16}$$

Substituting equation (16) into (14) the compressor work for the fogged system was obtained as:

$$W_{ac}=\dot{m}_a C_{pair}\left(T_{2a}-T_1\right)+\mathrm{W}\dot{m}_a\left(l_{g2}-l_{g1}\right) \tag{17}$$

$$W_{ac}=\dot{m}_a\left[C_{pair}\left(T_{2a}-T_1\right)+\mathrm{W}\left(l_{g2}-l_{g1}\right)\right] \tag{18}$$

$$\therefore W_{ac}=\dot{m}_a\left\{\frac{C_{pair}}{\eta_{comp.}}T_1\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right)+\mathrm{W}\left(l_{g2}-l_{g1}\right)\right\} \tag{19}$$

Where l_{g2} and l_{g1} are the enthalpies of saturated water vapour at compressor exit and inlet respectively.

The combustion chamber

The energy balance for the combustion chamber for the base case was expressed as[32];

$$\dot{m}_f C_v \eta_{comb.}+\dot{m}_a C_{pair}T_{2a}=\left(\dot{m}_a+\dot{m}_f\right)_g T_3 \tag{20}$$

Rearranging equation (20) and substituting $Q_{in}=\dot{m}_f C_v \eta_{comb.}$ into it the following was obtained:

$$Q_{in}=\left(\dot{m}_a+\dot{m}_f\right)_g C_p T_3-\dot{m}_a C_{pair}\left\{T_1\left[1+\frac{1}{\eta_{comb.}}\left(\left[\frac{P_2}{P_1}\right]^{\frac{\propto-1}{\propto}}-1\right)\right]\right\} \tag{21}$$

Hence, the energy balance for the combustion chamber for the fogged system was expressed as:

$$\dot{m}_f C_v \eta_{comb.}+\dot{m}_a C_{pair}T_{2a}+\dot{m}_v h_{v2}=\left(\dot{m}_a+\dot{m}_f\right)_g T_3+\dot{m}_v h_{v3} \tag{22}$$

$$\therefore Q_{in}=\left(\dot{m}_a+\dot{m}_f\right)_g C_p T_3-\dot{m}_a C_{pair}T_{2a}+\dot{m}_v\left(h_{v3}-h_{v2}\right) \tag{23}$$

The enthalpies of water vapour h_{v2} and h_{v3} expressed in Equation (23) were estimated as[33]:

$$h_{v1i}=2501.3+1.8723T_i \tag{24}$$

The term f expressed in Equation (25) is the ratio of air mass and fuel approximated thus[8]:

$$f=\frac{\dot{m}_f}{\dot{m}_a}=\frac{\left(T_3-298\right)_g C_p-C_{pair}\left(T_2-298\right)+W\left(h_{v3}-h_{v2}\right)}{C_v\eta_{comb.}-\left(T_3-298\right)_g C_p} \tag{25}$$

The Turbine Unit

Energy balance for the turbine was presented in line with the schematic diagram and T-s diagram of Figures 1 and 2 respectively[33]:

$$C_{pg}m_t T_3=W_T+C_{pg}m_t T_{4a} \tag{26}$$

Where $m_t=\dot{m}_a+\dot{m}_v+m_f=\left(1+w+f\right)$ (27)

Applying the isentropic expansion in the turbine, the exit temperature was related to the isentropic efficiency as shown[33]:

$$T_{4a}=T_3\left(1-{}_{T}\left[1-\frac{1}{\left(r_p\right)^{\frac{\propto-1}{\propto}}}\right]\right) \tag{28}$$

The turbine work can further be presented according to[8] as:

$$W_T=\dot{m}_a\left(1+w+f\right)C_{pg}\eta_T T_3\left(1-\frac{1}{\left(r_p\right)^{\frac{\propto-1}{\propto}}}\right) \tag{29}$$

Exergy Analysis of the Gas Turbine Units

Exergy balances were presented for the two cases to allow for the determination of the exergetic sustainability of the entire system. At state points, the exergy was determined through established relationships followed by the exergy balances for the gas turbine components system.

Air Compressor

In general, exergy balance for a steady process was given by[28]:

$$\sum Ex_{in} + E_Q = \sum Ex_{out} + Ex_w + Ex_D \tag{30}$$

Where $\sum Ex_{in} =$ summation of all exergy streams entering a component, $\sum Ex_{out} =$ summation of all exergy streams entering a component, Ex = exergy due to heat transfer, [Ex_w = exergy associated with work and Ex_D = exergy destruction due to irreversibility caused by entropy generation.

From the equation (30), compressor exergy balance using the T-s diagram of Figure 2 was expressed as:

$$Ex_1 + Ex_w = Ex_2 + Ex_D \tag{31}$$

Where E_{x1}, Ex_w, and Ex_D are the compressor work, exergy streams into and out of the compressor and exergy destruction, respectively. The exergy at state points 1 and 2 were written with respect to the combustion turbine base case configuration and that of the fogged system as shown:

$$ex_1 = (h_1 - h_0) - T_0(s_1 - s_0) \tag{32}$$

Expressing the terms to reflect temperature and pressure components gave;

$$ex_1 = C_{pa}(T_1 - T_0) - T_0\left(C_{pair} ln\left(\frac{T_1}{T_0}\right) - Rln\left(\frac{P_1}{P_0}\right)\right) \tag{33}$$

However, the exergy at point 1 for the base case was zero since the temperature at point 1 was ambient, and exergy at point 2 was expressed as:

$$ex_2 = C_{pa}(T_2 - T_0) - T_0\left(C_{pair} ln\left(\frac{T_2}{T_0}\right) - Rln\left(\frac{P_2}{P_0}\right)\right) \tag{34}$$

The exergy at point 1 for the fogged case was obtained per *kg* of air flow as thus[34]:

$$ex_{1if} = (C_{pa} + wC_{pv})T_0\left[\frac{T_1}{T_0} - 1 - ln\left(\frac{T_1}{T_0}\right)\right] + (1 + 1.607w)R_a T_0 ln\left(\frac{P_1}{P_0}\right) \tag{35}$$

Incorporating the mass flow, it becomes obvious that the mass flow at point 1 includes air and water vapour. Thus, equation (35) was transformed as presented:

$$ex_{1if} = (\dot{m}_a + \dot{m}_v)$$
$$\left\{(C_{pa} + wC_{pv})T_0\left[\frac{T_1}{T_0} - 1 - ln\left(\frac{T_1}{T_0}\right)\right] + (1 + 1.607w)R_a T_0 ln\left(\frac{P_1}{P_0}\right)\right\} \tag{36}$$

Or

$$ex_{1if} = \dot{m}_a(1 + w)$$
$$\left\{(C_{pa} + wC_{pv})T_0\left[\frac{T_1}{T_0} - 1 - ln\left(\frac{T_1}{T_0}\right)\right] + (1 + 1.607w)R_a T_0 ln\left(\frac{P_1}{P_0}\right)\right\} \tag{37}$$

Similarly, the exergy at point 2 was expressed as:

$$ex_{2if} = \dot{m}_a(1 + w)$$
$$\left\{(C_{pa} + wC_{pv})T_0\left[\frac{T_2}{T_0} - 1 - ln\left(\frac{T_2}{T_0}\right)\right] + (1 + 1.607w)R_a T_0 ln\left(\frac{P_2}{P_0}\right)\right\} \tag{38}$$

$$\text{Exergy efficiency} = \frac{Ex_2 - Ex_1}{Ex_w} \tag{39}$$

The Combustion Chamber

For the combustion chamber, the exergy balance was expressed as:

$$Ex_2 + Ex_{fuel} = Ex_3 + Ex_D \tag{40}$$

The value of exergy at point 2 was already stated in the Equations (36) and (38) for the fogged and base cases respectively.

The exergy of fuel was composed of the physical and chemical parts as presented:

$$Ex_{fuel} = Ex_{fuel(chem.)} + Ex_{fuel(phy.)} \tag{41}$$

The chemical component of fuel exergy was obtained for the base and fogged cases with the following relationships[28,35]:

For the fogged case,

$$Ex_{fuel(chem.)} = \sum\nolimits_{i=1}^{n} x_i Ex_{ex1} + RT_0 \sum x_i ln(x_i) RT_0$$
$$\left[(1 + 1.609w) ln\{(1 + 1.607w_0)/(1 + 1.607w)\} + 1.607wln(w/w_0)\right] \tag{42}$$

And for the base case[28]:

$$Ex_{fuel(chem.)} = \sum\nolimits_{i=1}^{n} x_i Ex_{chi} + RT_0 \sum\nolimits_{i=1}^{n} x_i ln(x_i) \tag{43}$$

The physical exergy component of fuel was calculated thus with the relationship[28]:

$$Ex_{fuel(phy.)} = C_{Pfuel}(T_{fuel} - T_0) - T_0$$
$$\left\{C_{Pfuel} ln\left[\frac{T_{fuel}}{T_0}\right] - R_{fuel} ln\left[\frac{P_{fuel}}{P_0}\right]\right\} \tag{44}$$

However, since the fuel was admitted into the combustion chamber at a temperature T_0, the expression in equation (44) was reduced to:

$$Ex_{fuel(phy.)} = T_0 R_{fuel} ln\left[\frac{P_{fuel}}{P_0}\right] \tag{45}$$

The exergy of gas stream after combustion at state point 3 was obtained as:

$$Ex_3 = C_{pg}\left(T_3 - T_0\right) - T_0\left\{C_{pg} ln\left(\frac{T_3}{T_0}\right) - R_{pg} ln\left(\frac{P_3}{P_0}\right)\right\} \tag{46}$$

Exergy efficiency in the combustion chamber was written utilizing terms in Equation (45) as follows:

$$\psi_{CC} = \frac{exergy\ of\ product}{exergy\ of\ fuel} \tag{47}$$

Or $\psi_{CC} - \dfrac{Ex_3}{Ex_2 + Ex_{fuel}}$ (48)

The exergy at point 3 for the fogged system was obtained per kilogram of fluid as:

$$Ex_3 = \left(C_{pa} + wC_{pv}\right)T_0\left[\frac{T_3}{T_0} - 1 - ln\left(\frac{T_3}{T_0}\right)\right] + \left(1 + 1.607w\right)R_a T_0 ln\left(\frac{P_3}{P_0}\right) \tag{49}$$

And for a given mass, Equation (49) takes the form:

$$Ex_3 = \dot{m}_a\left(1 + w + f\right)$$
$$\left\{\left(C_{pa} + wC_{pv}\right)T_0\left[\frac{T_3}{T_0} - 1 - ln\left(\frac{T_3}{T_0}\right)\right] + \left(1 + 1.607w\right)R_a T_0 ln\left(\frac{P_3}{P_0}\right)\right\} \tag{50}$$

The Gas Turbine

For the turbine, considering the T-s diagram of Figure 3 and the general expression of equation (32) the exergy balance in the gas turbine was given as:

$$Ex_3 = Ex_4 + Ex_W + Ex_D \tag{51}$$

The exergy Ex_3 at state point 3 was already calculated with equations (49) and (50). The exergy at point 4 was computed with the relationship in equation (52) using the temperature and pressure values of this point as well as its mass flow rate as follows:

$$Ex_4 = C_{pg}\left(T_4 - T_0\right) - T_0\left\{C_{pg} ln\left(\frac{T_4}{T_0}\right) - R_{pg} ln\left(\frac{P_4}{P_0}\right)\right\} \tag{52}$$

The expression for the exergy of the fogged system was presented per kg of gas thus:

$$Ex_4 = \left(C_{pa} + wC_{pv}\right)T_0\left[\frac{T_4}{T_0} - 1 - ln\left(\frac{T_4}{T_0}\right)\right] + \left(1 + 1.607w\right)R_a T_0 ln\left(\frac{P_4}{P_0}\right) \tag{53}$$

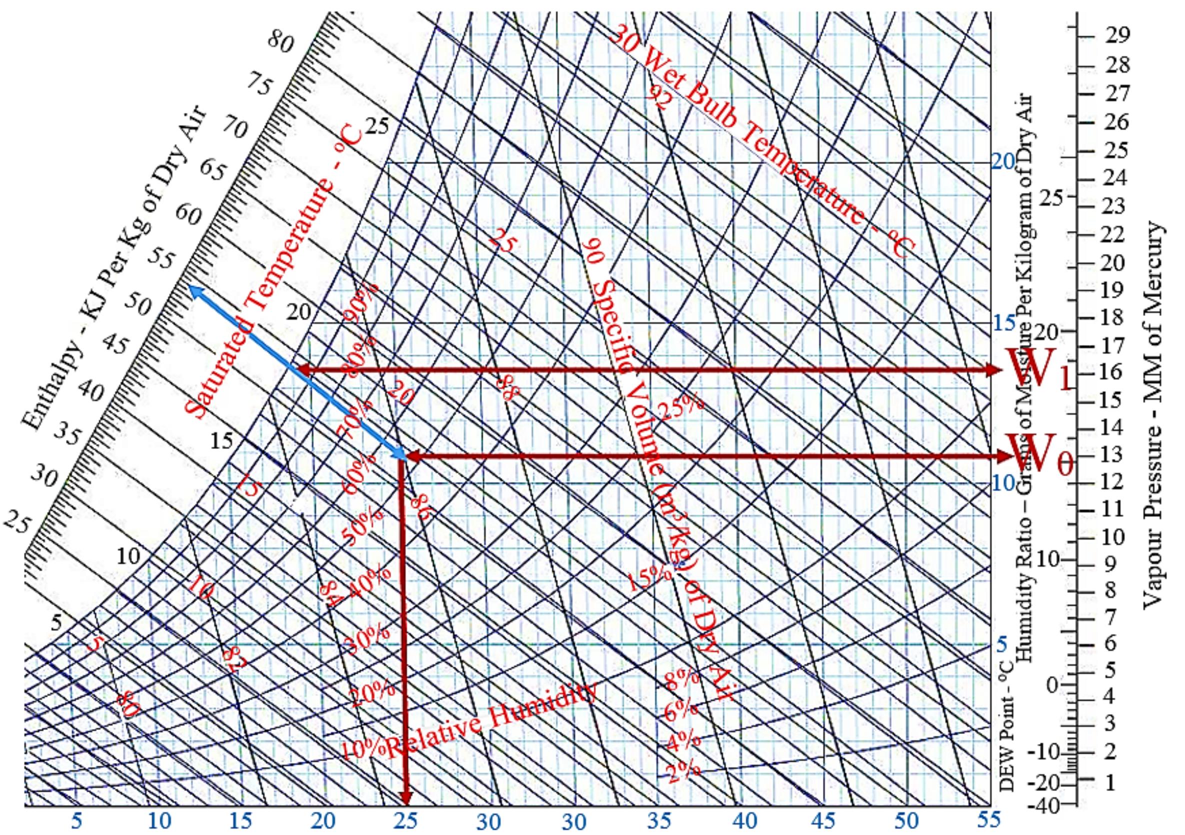

Figure 4. Psychrometric Chart of Turbine Inlet Fogging System

For the mass stream Equation (53) was written as:

$$Ex_4 = \dot{m}_a(1+w+f)$$
$$\left\{(C_{pg}+wC_{pv})T_0\left[\frac{T_4}{T_0}-1-ln\left(\frac{T_4}{T_0}\right)\right]+(1+1.607w)R_aT_0ln\left(\frac{P_4}{P_0}\right)\right\} \quad (54)$$

The exergy of turbine work was same as the turbine work output. This work term was already presented in Equation (29). The exergetic efficiency for the gas turbine was expressed as[28,29]:

$$\psi_{GT} = \frac{Ex_w}{Ex_3 - Ex_4} \quad (55)$$

Exergetic Sustainability

Environmental sustainability indicators[36] are exergy based indices which comparatively assesses the performance of energy conversion systems based on exergy efficiency, useful system output and the environmental impact of such systems resulting from thermodynamic irreversibility. They consist of four basic indicators as follows:

Waste Exergy Ratio

The waste exergy ratio (WER) quantifies the degree of cumulative thermodynamic irreversibility in a plant with respect to the available external exergy input to the system. For thermal power plants and boilers, the available external exergy input is the chemical exergy of the fuel used. The WER was obtained mathematically as the overall exergy waste (or destruction) for the system divided by the total exergy input. This was expressed as follows[36]:

$$\dot{e}_{WR} = \frac{\dot{D}_{Total}}{\left[1-\frac{T_0}{T_{CC}}\right] * \dot{m}_{fuel\ oil} * LHV_{fuel\ oil}} \quad (56)$$

Environmental Effect Factor

The environmental effect factor (EEF) quantifies the degree of cumulative thermodynamic irreversibility in a plant relative to the plant's net exergy efficiency. It also provides a comparative measure of how a plant's useful output is adversely affected by high thermodynamic irreversibility, leading to environmental concerns. The EEF was calculated as the ratio of the WER to the exergy efficiency. This is expressed as follows[36]:

$$EEF = \frac{\dot{e}_{WR}}{\psi} = \frac{\dot{D}_{Total}}{\sum_i x_i Ex_{x_i}^{CH} + RT_0 \sum x_i ln x_i} \quad (57)$$

Exergy Efficiency

The exergy efficiency represents the ratio of the exergy in the product to the total exergy of the fuel, serving as a general performance index of a plant based on the concept of availability. This is expressed as follows[37, 32]:

$$\psi = \frac{exergy\ in\ product}{\sum_i x_i Ex_{x_i}^{CH} + RT_0 \sum x_i ln x_i} \quad (58)$$

Sustainability Index

The sustainability index (SI) is a non-dimensional term that quantifies the extent to which the total useful output of an energy conversion system surpasses its total internal thermodynamic irreversibility, as shown in equation (59). Conversely, the reciprocal of the EEF is termed the exergetic SI. This index serves as a basis for comparing the environmental degradation to the exergetic output of each system follows.[36]

$$\mathrm{SI} = \frac{Exergy\ of\ product}{Grand\ exergy\ destruction} \quad (59)$$

Results and Discussion

The results obtained from the comparative analysis of environmental sustainability and performance indices of different gas turbine inlet cooling systems in this study are presented and discussed in this section as follows:

Results

The results of the system parameters, sustainability analysis and performance indices are presented as follows:

System Operating Data

For gas turbine plant parameters without inlet cooling system:

The simulation involved investigating the effects of various turbine inlet cooling systems on environmental sustainability indicators, net power output, exergetic and energetic efficiencies, as well as the impact of various operating parameters on the system. To model these conditions dynamically, we utilized the basic operating data from an existing power plant. Models were developed for each component of the system, and a program source code was created using engineering equation solver (EES) to assist in the simulation. This program was designed for evaluating similar plant configurations using any combination of operating data. The summarized operating data is presented in Table 1.

The data presented in Table 1 were used to perform all subsequent analyses. This presentation was made possible by utilizing all performance criteria of the gas turbine plant under consideration. These criteria include the isentropic efficiencies of the compressor and turbine, as well as the compression and expansion indices for air and burnt gases, respectively. However, the mass flow rates for air and fuel remained as shown in Table 1.

For fogged inlet cooling system, the data is presented thus in Table 2.

Sustainability Analysis for Different Inlet Cooling System

Regarding the sustainability analysis of the gas turbine with different inlet cooling systems, the details are presented as follows: For the fogged inlet cooling system, the results are shown in Figure 4.

Figure 5 presents the sustainability analysis of the gas turbine inlet cooling system with mechanical refractive chiller as follows:

Performance Indices for the Gas Turbine Plant at Different Parametric Conditions

The performance indices for the gas turbine plant at different parametric conditions are presented under this section:

Gas Turbine Plant at Base Condition

A compendium of the performance indices for the plant at base condition is shown in Table 3.

Table 2. Operating Data for the Fogged System

Parameter	Unit	Value
Inlet temp. (T_1)	K	292
Temp. after compression (T_2)	K	644.7
Temp. combustion (T_4)	K	1324
Temp. combustion (T_5)	K	832.4
Mass of air inlet comp. ($\dot{m}_a$)	kg/s	427
Mass of fuel ($\dot{m}_f$)	kg/s	7.039
Specific humidity (w)	kg/kg of dry air	0.0136
Relative humidity	%	55

Gas Turbine Plant With Fogging

Using the compressor inlet temperature dropping to 292 K, a simulation of the system to show its effect on sustainability indicators was performed with the data compiled as shown in Table 4.

Gas Turbine With Mechanical Refractive Cooling

Using the compressor inlet temperature dropping to 288 K, a simulation of the system to show its effect on sustainability indicators was performed with the data compiled as indicated in Table 5.

Gas Turbine With Spray Cooler

Using the compressor inlet temperature dropping to 298 K, a simulation of the system to show its effect on sustainability indicators was performed with the data compiled as presented in Table 6.

Comparative Sustainability Analysis of Gas Turbine Inlet Cooling System

Comparative sustainability analysis of the system was conducted based on the parameters set by the compressor inlet conditions. Figure 6 presents a relative sustainability analysis of the system with different turbine inlet cooling technologies.

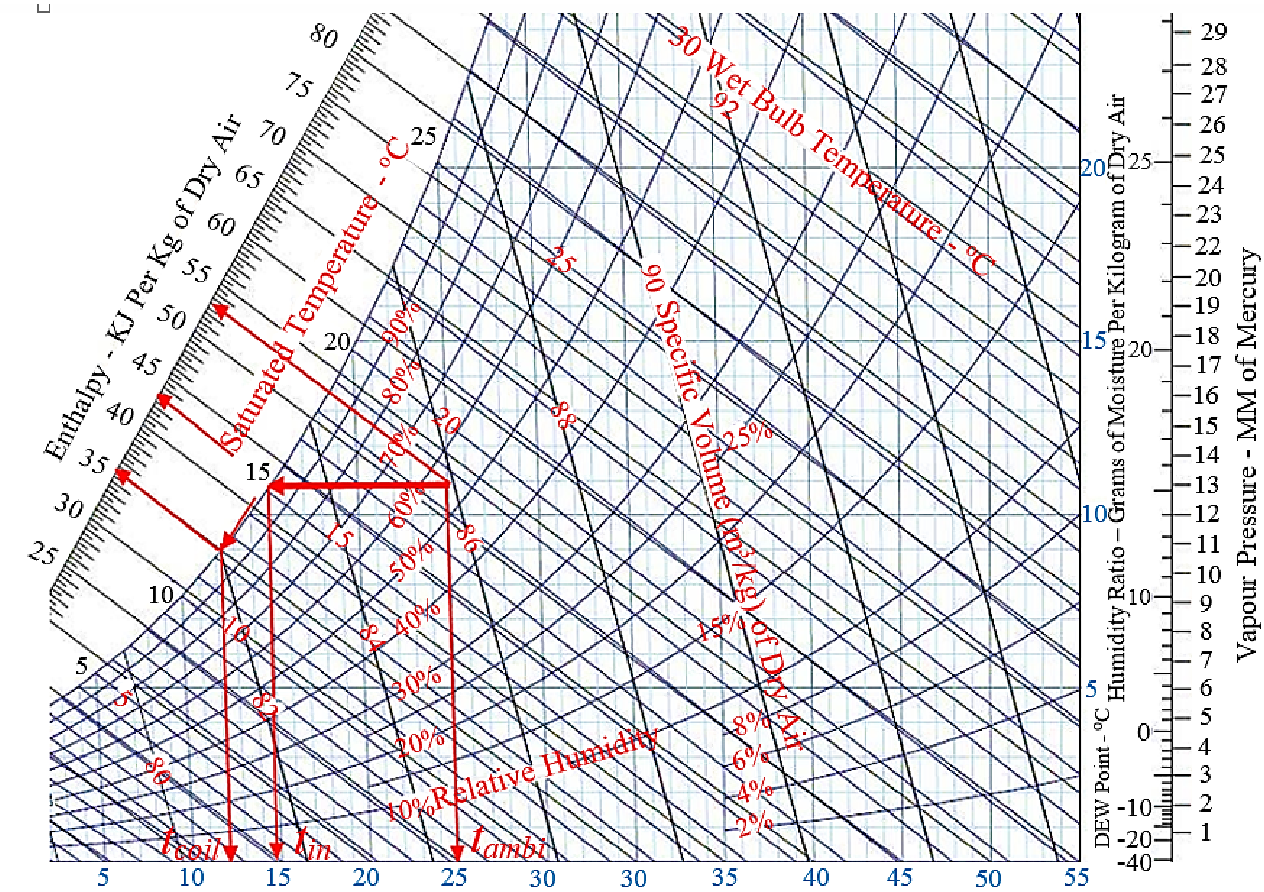

Figure 5. Psychometric Chart of Mechanical Refrigerative Cooling

Table 3. Performance Indices for the Gas Turbine Plant at Base Condition

Parameter	Value	Unit
Compressor work	148.775	MW
Turbine work	244.474	MW
Net turbine output	95.7	MW
Energy efficiency	28.69	%
Exergy efficiency	27.58	%
WER	0.3425	Dimensionless
Environmental effect factor	0.486	Dimensionless
SI	2.057	Dimensionless

Table 4. Performance Indices for the Gas Turbine Plant With Fogging

Parameter	Value	Unit
Compressor Work	145.779	MW
Turbine Work	244.474	MW
Net Turbine Output	98.695	MW
Energy Efficiency	29.03	%
Exergy Efficiency	28.45	%
WER	0.3256	Dimensionless
Environmental Effect Factor	0.462	Dimensionless
SI	2.165	Dimensionless

Table 5. Performance Indices for the Gas Turbine Plant with Mechanical Refrigerative Cooling

Parameter	Value	Unit
Compressor Work	143.782	MW
Turbine Work	244.474	MW
Net Turbine Output	100.692	MW
Energy Efficiency	29.25	%
Exergy Efficiency	29.03	%
WER	0.3142	Dimensionless
Environmental Effect Factor	0.4458	Dimensionless
SI	2.243	Dimensionless

Table 6. Performance Indices for the Gas Turbine Plant With Spray Cooler

Parameter	Value	Unit
Compressor Work	145.629	MW
Turbine Work	244.474	MW
Net Turbine Output	98.845	MW
Energy Efficiency	29.05	%
Exergy Efficiency	28.49	%
WER	0.3247	Dimensionless
Environmental Effect Factor	0.4608	Dimensionless
SI	2.17	Dimensionless

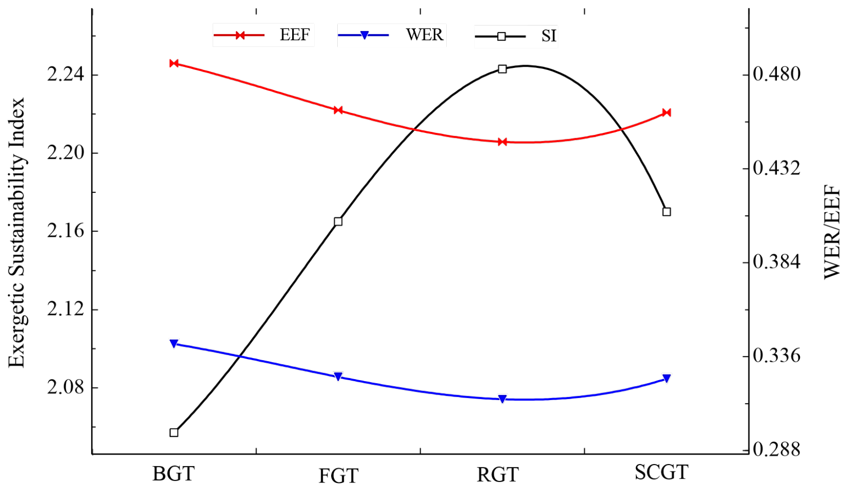

Figure 6. Comparative Sustainability Analysis of Different Turbine Inlet Cooling System

Discussion

From the performance evaluation presented in Table 3, the system generated 95.7 MW of power, which is 4.3 MW less than the rated 100 MW of the plant. The SI, which relates the ratio of useful output to the total exergetic destruction, was calculated as 2.057. This indicates that the turbine output exceeded the total destruction by 2.1 times, aligning with the trend of 2.003 SI reported by Ahmadi and Dincer. The compressor work was found to be 148.775 MW, noticeably higher due to the elevated compression ratio. In contrast, the turbine output was 244.474 MW. A study by Alhazmy et al[8] reported a similar pattern with 137.233 MW for compressor work and 229.218 MW for turbine output in an additionally cooled inlet turbine system.

The inlet temperature for the fogging system was determined based on the ambient temperature and relative humidity. The temperature and relative humidity at the fogging arrangement were 25 °C and 55%, respectively. Assuming complete adiabatic saturation, the ambient air temperature and specific humidity were calculated as 18.5 °C and 0.0136 kg/kg of dry air, as indicated on the psychrometric chart (Figure 4). The mechanical chiller system is represented in the psychrometric chart shown in Figure 5. Given the ambient conditions of 25 °C and 55% relative humidity, the cooling process at constant specific humidity resulted in an air temperature reduction to approximately 15 °C, compared to the reported 12.3 °C.[13]

Under these conditions, the coil surface temperature was around 12 °C, leading to an established coil bypass factor of 0.23. Moreover, with a mass flow rate of 427 kg/s for air entering the gas turbine, and a specific volume of air at 25 °C dry bulb temperature and 55% relative humidity being 0.86 m^3/kg, the recirculated air by the refrigerative system was 443 kg/s. This yielded a mechanical cooling requirement of 18.18 kJ/kg, equivalent to approximately 8328.4 kW. These results align with the findings presented by Ameri et al. for similar configurations.[17]

In terms of sustainability analysis, the spray cooler and wetted media methods emerge as the simplest and most cost-effective cooling techniques, frequently applied to base load turbines.[12] The wetted media approach involves directing the inlet air through a water film spread over a medium, such as a honeycomb. As the air passes through this medium, it evaporates the water, drawing latent heat of vaporization from the surrounding air and thereby reducing the temperature. This method boasts an effectiveness ranging between 85 and 95%.[13] In practical terms, this means the inlet air temperature (dry-bulb) can be reduced to 85 to 95% of the difference between its wet-bulb temperature (WBT) and dry-bulb temperature (DBT).

Given the ambient conditions of 25 °C and 55% relative humidity, the wet bulb temperature was determined to be 18 °C, resulting in a WBT-DBT difference of 7 °C. Applying an efficiency of 90% to this difference, the turbine inlet temperature was calculated to be 18.7 °C (Table 6).

The basic gas turbine plant exhibited a certain degree of sustainability, as its value exceeded unity. However, Figure 6 reveals significantly enhanced sustainability values when turbine inlet cooling methods are employed. The level of sustainability is directly influenced by the degree of turbine inlet temperature reduction achievable within the design constraints. For example, the refrigerative cooling method, which achieved a relatively lower inlet temperature of 15 °C, demonstrated an SI of 2.243. This was followed by the spray cooling technique at 2.170 and fogging at 2.165. The lowest SI was recorded under the base conditions of the turbine system at 2.057. These findings align with those presented in the study conducted by Aydin.[36]

To assess performance indices, we compared net turbine output, energy efficiency, and exergy efficiency. The results were consistent with those reported in references: Cataldi et al[6] and Athari et al,[11] particularly for base conditions, with average values of 90.44 MW, 25.94%, and 26.01% for turbine output, energy efficiency, and exergy efficiency, respectively. The performance indices for the selected parameters similarly recorded the lowest values under base conditions. Specifically, the turbine output, energy efficiency, and exergy efficiency were 95.70 MW, 28.69%, and 27.58% at base conditions. For the fogged system, the values were 98.695 MW, 29.03%, and 28.45%; for mechanical chillers, they were 100.692 MW, 29.25%, and 29.03%; and the spray cooler recorded 98.845 MW, 29.05%, and 28.49%. Therefore, the mechanical chiller exhibited the highest efficiency, followed by the spray cooler and then the fogging system, with the base condition showing the least efficiency.

Conclusion

Turbine inlet air cooling remains highly efficient for enhancing gas turbine performance during adverse conditions, particularly in hot and humid summer periods when electrical power demands are at their peak. In striving to boost power generation with an emphasis on ecological efficiency, environmental friendliness, sustainability, and cost-effectiveness, turbine inlet air cooling has proven to be a suitable method. Numerous turbine inlet air cooling systems have been extensively analyzed in the literature based on application, performance, and sustainability indices. This study, therefore, examined the spray cooler and wetted media system, fogging system technology, and mechanical chiller system in terms of sustainability and performance indices under both base conditions and parametric variations.

The following conclusions were drawn from the study:

- Generally, the simple gas turbine plant exhibited a degree of sustainability, with its value exceeding unity. However, the results indicated significantly improved sustainability values through the use of turbine inlet cooling systems. SI varied directly with the efficiency of turbine inlet temperature reduction based on design requirements. Specifically, the refrigerative method achieved a 2.243 SI, spray cooling had a 2.17 SI, and fogging registered a 2.165 SI. The base conditions recorded the lowest SI at 2.057.
- An analysis of ambient temperature effects on the system revealed a general increase in exergetic destruction throughout the system, predominantly in the combustion chamber. This increase was caused by a significant temperature difference between the compressed air and natural gas, resulting in a large temperature gradient and subsequent exergetic destruction.
- Both energetic and exergetic efficiencies of the plant declined with rising ambient temperatures due to increased compressor work requirements.
- The SI decreased with increasing ambient temperatures, attributed to higher total exergetic destruction within the plant at elevated ambient temperatures. Both the EEF and WER exhibited an increasing trend in line with ambient temperature, as they are functions of total exergetic destruction.

Performance indices inversely correlated with increasing ambient temperature. The improvement in net turbine output, energy efficiency, and exergy efficiency between different cooling systems was contingent on reduced compressor work.

As a recommendation, given the existence of various cost-effective turbine inlet cooling methods, a comparative cost evaluation incorporating specific exergy costing for both the cooling equipment and the turbine as a unified system would provide insights into determining the most exergetically sustainable, efficient, and cost-effective cooling method.

Authors' Contribution

Conceptualization: Enefiok Okon Usungurua, Aniekan Essienubong Ikpe.
Data curation: Enefiok Okon Usungurua, Imoh Ime Ekanem.
Formal analysis: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Funding acquisition: Enefiok Okon Usungurua.
Investigation: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Methodology: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Project administration: Enefiok Okon Usungurua.
Resources: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Software: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Supervision: Enefiok Okon Usungurua.
Validation: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Visualization: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Writing–original draft: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.
Writing–review & editing: Enefiok Okon Usungurua, Imoh Ime Ekanem, Aniekan Essienubong Ikpe.

Competing Interests

None declares.

Ethical Approval

The authors declare that this research do not require any ethical committee approval or legal authorization. All authors have read and agreed to the publication of this research work.

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