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Modelling and Simulation of High Pressure Fogging Air Intake Cooling Unit of Omotosho Phase II Gas Turbine Power Plant

Aniekan Essienubong Ikpe[1,*], Ijeoma Camelita Iluobe[2], Desmond Iria-evbo Imonitie[1]
[1]*Department of Mechanical Engineering, University of Benin, Benin City, Nigeria.*
[2]*Department of Production Engineering, University of Benin, Edo State, Nigeria.*

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ABSTRACT

The aim of this study was to analyze the performance of Omotosho Phase II gas turbine power plant for improved performance. To obtain the required output performance of the gas turbine power plant, operation data from years of 2013 to 2016 was collected from Omotosho Phase II gas turbine power plant in Ondo State, Nigeria. ASPEN HYSYS 2016 version was used to create two models, with one representing Omotosho Phase II gas turbine power plant with fogging unit incorporated while the other represented the power plant without fogging unit. The data was fed as input variables into the models in ASPEN HYSYS 2016 version which simulated the power plant process Specific Fuel Consumption (SFC) obtained from the power plant simulation when fogging is not incorporated was 0.199 kg/kwh, whereas, SFC of the plant with fogging was 0.179 kg/kwh. Thermal efficiency of 43.93% was obtained from the result of the simulated power plant with fogging system, whereas, thermal efficiency of 39.39% obtained from the result of the simulated power plant without fogging system. Net power of 131 MW was obtained from the simulation of the power plant with fogging system while net power of 117.46 MW was obtained when the plant operates without fogging system installed. For the compressor work, 82 MW/h was obtained from the simulation of the power plant with fogging system, whereas, 112.11 MW/h was obtained from the simulation of the power plant without fogging system. Furthermore, turbine work of 213 MW/h were obtained from the simulation of the power plant operating with fogging system while turbine work of 229.57 MW/h was obtained from the power plant without fogging system. This indicates that the incorporation of fogging system into Omotosho Phase II gas turbine power plant is economically viable in terms of fuel consumption, efficiency, power requirement, and GHG emissions compared to operation of the power plant without fogging system.

1. Introduction

Nigeria has a tropical climate characterized by hot and dry weather, with an average temperature of 28 ºC which is higher than ISO 3977-2 recommended temperature for Gas Turbine (GT) air inlet temperature. According to international organization of standardization, the rated capacities of GTs are based on the standard ambient air and zero inlet and exhaust pressure drops. Hence, the air inlet conditions are air temperature of 15 °C, relative humidity of 60%, and absolute pressure of 101.325 KPa at sea level [1, 2].

As a result of the aforementioned temperature requirement under which a GT operates in Nigeria, Omotosho phase II gas turbine power plant undergoes a significant effect on both the power output,

* Corresponding author
E-mail address: aniekan.ikpe@eng.uniben.edu
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efficiency, specific fuel consumption, and net heat rate. In other words, the power and efficiency are decreased because they work at off rated capacity. The increased environmental temperature condition causes reduction in thermal efficiency of the turbine cycle, increase in fuel consumption, increase in Green House gas emissions as well as increase in cost of operation and maintenance. Such conditions in GT operation oftentimes result in downtime and equipment breakdown due to insufficient cooling of the inlet air utilized by the power plant.

The method of injecting water into the inlet duct of a GT is a well-known established process for air inlet cooling and this technique is called "FOGGING" [3]. Fog is formed when the difference between the air temperature and dew point is 2.5 ^{o}C. In other words, fog begins to form when water vapor condenses into tiny liquid droplets suspended in air [4]. Fogging system is used as a method for cooling the inlet air to the compressor via direct injection of water, in order to reduce the ambient air temperature until it reaches the wet bulb temperature and thus increasing the net power.

Different geographical regions have different climatic conditions, ambient temperature, and relative humidity ratio. The ambient temperature has a strong influence on the GT performance. Nigeria lies geographically in the tropics where the climate is seasonally damp and very humid [5] is a location that is suitable for the incorporation of fogging system in GT plants as a method of cooling the air inlet temperature of the GT for better performance and increased power output.

A typical GT engine consists of three major parts, namely compressor, combustion chamber, and turbine in addition to the generator [6, 7]. GT performance is critically limited by temperature variation, especially in hot and regions like Sub-Sahara Africa. The increase in inlet air temperature becomes pronounced especially in the hot weather, and this causes a significant decrease in GT power output. It occurs because the power output is inversely proportional to the ambient temperature and because of the high specific volume of air drawn by the compressor [8]. The efficiency and power output of GTs vary according to the ambient conditions [9]. The effects of these thermal variations greatly affect the electricity generation, fuel consumption, and plant incomes. However, cooling the air intake to the compressor has been widely used to mitigate these shortcomings [10].

The method of injecting water into the inlet duct of a GT is now a well know established tool of air inlet cooling and this technique is called inlet fogging [11]. The fine mist of water droplets is referred to as fog and is injected into the air intake by a nozzle manifold, usually mounted near the air filters by injecting less or equal amount of water to what is required for saturating the intake air at a given ambient conditions [12]. This in turn reduces the compressor inlet temperature which results in gaining back the losses in power output, efficiency, reduction in specific fuel consumption, and net heat rate [13]. The effectiveness of this technique depends on the air humidity and temperature, generally achieving maximum benefits in dry and hot climates but still delivering significant benefits in moist as well as tropical environments. Ambient temperature, humidity, and pressure are important factors that either reduces or improves the performance of a GT unit.

In a GT unit, fogging system is mainly installed to ensure that power losses due to high ambient temperature is regained. The relevance of Air In-Take Cooling (AIC) to compressor is that, it allows reduction of losses in GT power output. AIC can lead to an increase in GT output above the rated capacity by cooling the inlet air below 15 ^{o}C [14].

This research intends to show that optimum parameters can be selected for improved performance of Omotosho phase II power plant using computer aided simulation. The simulation layout involves incorporation of high pressure fogging system to reduce the intake air temperature entering the GT unit.

2. Research Methodology

Omotosho generation station (phase II) is a GT plant owned by Niger Delta Power Holding Company (NDPHC). The power plant is located at Omotosho, Okitipupa Local Government area of Ondo State. It has four GE frame machines with the installed capacity of 125 MW for each machine bringing its total installed capacity to 500 MW. The operating parameters for gas turbine unit of Omotosho power plant was collected from the daily turbine control log sheet. Summary of operating parameters of the gas turbine used for the simulation is presented in *Table 1*. The thermodynamic analysis of the plant and its performances were carried out without cooling and with cooling of the inlet air entering the compressor unit.

Table 1. *Operating parameters from 125 MW Omotosho power plant phase II.*

S/N	Operating Parameters	Values	Unit
1	*Mass flow rate of air through compressor (ma).*	*361*	*kg/s*
2	*Temperature of inlet air to compressor (T1).*	*302*	*ºK*
3	*Pressure of inlet air to compressor (P1).*	*101.3*	*Kpa*
4	*Outlet temperature of air from compressor (T2).*	*611*	*ºK*
5	*Outlet pressure of air from compressor (P2).*	*1030*	*kpa*
6	*Fuel gas (natural gas) mass flow rate (mf).*	*6.5*	*kg/S*
7	*Fuel – air ratio at full load (on mass basis).*	*56:1*	
8	*Inlet pressure of fuel gas.*	*24*	*Bar*
9	*Inlet temperature of gas turbine (T3).*	*1405*	*ºK*
10	*Maximum exhaust temperature of T. outlet.*	*851*	*ºK*
11	*Combustion efficiency ∩ce.*	*90*	*%*
12	*Pressure drop in the combustion chamber.*	*10*	*%*
13	*Installed capacity.*	*125*	*MW*
14	*Isentropic eff. of compressor.*	*89.20*	*%*
15	*Isentropic eff. of Turbine.*	*89.80*	*%*
16	*Specific heat capacity of air 1 Cpa.*	*1.005*	*KJ/kg K*
17	*Specific capacity of gas Cpg.*	*1.15*	*KJ/kg K*
18	*Lower Heating Value (LHV).*	*45880*	*KJ/kg K*

The compressor inlet temperature is equal to ambient temperature once the base-case neglects the cooling effect and simulates the cycle under ISO conditions and without pressure drop at inlet and exhaust ducts [15]. From the Bryton cycle P-v diagram in *Fig. 1*, the inlet pressure is given by *Eq. (1)*.

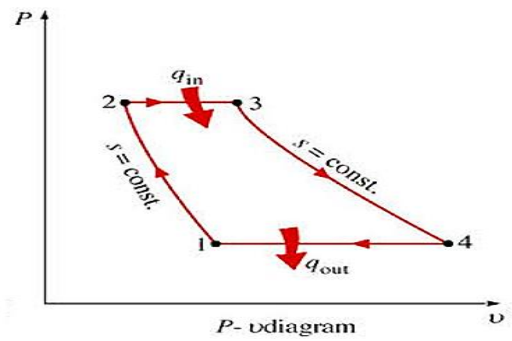

Fig. 1. *P-v diagram for ideal Bryton open cycle.*

$$\text{All } P_1 = P_3, \tag{1}$$

where, P_1 is the atmospheric pressure, P_3 is the inlet pressure of the compressor.

The air and combustion products are assumed to behave as ideal gases [15]. The gas turbine process is based on Brayton cycle while the gas turbine plant essentially consists of compressor, combustion chamber, and turbine. Air enters the compressor, is compressed and heated after that, it goes to the combustion chamber, fuel is burned at constant pressure then raises the temperature of air to the firing temperature. The resulting high temperature gases then enter the turbine where they expand to generate the useful work. A part of the work developed by the gases passing through the turbine is used to run the compressor and the remaining is used to generate the electrical energy. When the heat is given to the air by mixing and burning the fuel in the air and the gases coming out of the turbine are exhausted to the atmosphere, the cycle is known as an open cycle power plant. Using the polytrophic relation of the ideal gas and knowing the isentropic efficiency of the compressor, the discharge temperature (T_2) can be determined using *Eq. (2)*:

$$T_2 = \frac{T_1}{\eta_c}\left[r_p^{\frac{\gamma-1}{\gamma}} - 1\right] + T_1. \tag{2}$$

Where T_1 is the ambient temperature, η_c is the isentropic efficiency of the compressor and T_2 is the Compressor Temperature Discharge (CTD). The compressor work (W_C) can be estimated using the first law of thermodynamics that is given by *Eq. (3)*:

$$W_c = m_a * C_{pa}(T_2 - T_1). \tag{3}$$

Where m_a is the mass flow rate of air and C_{pa} is the specific heat of dry air at constant pressure, determined as a function of the average temperature across the compressor. The heat delivered by the combustion discharge pressure is given by *Eq. (4)*:

$$Q_{in} = m_a + m_f \times C_{pg}(T_3 - T_2). \tag{4}$$

By knowing the Fuel Gas Heat Value (FHV), the natural gas mass flow rate is given by *Eq. (5)*:

$$\dot{m}_f = \frac{Q_{in}/FHV}{\eta_{combustor}}. \tag{5}$$

Where $\eta_{Combustor}$ is the combustion chamber efficiency. The turbine discharge temperature can be expressed by *Eq. (6)*:

$$T_6 = T_5 - \eta_t . T_4\left[1 - \left(\frac{1}{(P_5/P_6)}\right)^{\frac{\gamma-1}{\gamma}}\right]. \tag{6}$$

Where η_t is the turbine isentropic efficiency and P_6 is the ambient pressure. Hence, the turbine power is given by *Eq. (7)*:

$$\dot{W}_t = C_{Pg} \times (T_5 - T_6). \tag{7}$$

Where $\dot{m}_T$ is the total mass flow rate composed of fuel and air mass flow rate in *Eq. (8)*.

$$\dot{m}_T = \dot{m}_a + \dot{m}_f. \tag{8}$$

Cpg is the specific heat capacity of dry gas of combustion product assumed to be 1.15 kJ/kgK and determined as function of the average temperature across the turbine [16]. The maximum exhaust temperature T_4 from the turbine outlet is given by *Eq. (9)*:

$$T_4 = T_3\left[1 - \eta_T\left[1 - \left(\frac{P_3}{P_4}\right)\frac{1-\gamma_g}{\gamma_g}\right]\right]. \tag{9}$$

The net power obtained from the gas turbine is given by *Eq. (10)* [17]:

$$W_{net} = W_t - W_c\,. \tag{10}$$

The Specific Fuel Consumption (SFC) can be determine using *Eq. (11)*:

$$Sfc = \frac{3600.mf}{W_{net}}. \tag{11}$$

The Heat Rate (*HR*) can be calculated using *Eq. (12)*:

$$HR = Sfc\ x\ LHV. \tag{12}$$

The thermal efficiency of the gas turbine is given by *Eq. (13)* [18]:

$$\eta_{th} = \frac{work\ output}{Heat\ Supplied} = \frac{W_{net}}{Q_{in}}. \tag{13}$$

From *Eq. (13)*, Thermal Discharge Index (TDI) can be obtained. This is the number of thermal energy units discharged to the environment for every unit of electrical energy generated by a power plant. According to El Wakill [19], TDI is strongly dependent on the thermal efficiency of the plant, the lower it is the more efficient in power plant; thus a low value of TDI is desirable. Equation for the TDI of a gas turbine unit is given by *Eq. (14)*:

$$TDI = \frac{P_{th}(1-\eta_{th})}{P_{th}\eta_{th}} = \frac{1}{\eta_{th}} - 1. \tag{14}$$

Where P_{th} is the thermal exergy input. The work ratio is given by *Eq. (15)*:

$$Work\ Ratio = \frac{Net\ Work\ output}{Gross\ Work\ output}. \tag{15}$$

The specific heat of real gas varies with temperature and also with pressure at extreme high pressure levels. However, in the present model, it is assumed that specific heat of gas varies only with temperature in the form of polynomials given by *Eq. (16)*:

$$C_P(T) = a + bT + cT^2 + dT^3, \tag{16}$$

where a, b, c, and d are the coefficient of polynomials, as taken from the work of [20].

A factor called humidity accounts the increase in specific humidity of ambient air across the air-humidifier and it is calculated using *Eq. (17)*:

$$f_h = 1 + 0.05\emptyset_{h,e}, \tag{17}$$

where $\emptyset_{h,e}$ is the relative humidity at the humidifier outlet. Thus, enthalpy of the gas is given by *Eq. (18)* [21]:

$$h = \int_{T_a}^{T} C_P(T)dT. \tag{18}$$

2.1. Thermodynamic Analysis of Gas Turbine Fogging System

High pressure fogging includes a high pressure reciprocating pump providing demineralized water to fogging nozzle where the fog is mixed with air to ensure proper saturation located after the air filter elements. The fog then provides cooling when it evaporates in the inlet duct of gas turbine. As air is compressed through the compressor stages of the gas turbine engine, temperature and pressure of the incoming air increase while the volume decreases. The air can attain 100% relative humidity at the compressor inlet and thereby gives the lowest temperature possible without refrigeration (wet bulb temperature) [22, 23]. The gas turbine power plant was modeled base on the following assumptions:

– All component has adiabatic boundaries.
– The air and the combustion products assume ideal characteristics.
– Kinetic and potential components of energy are neglected.
– The ambient conditions of temperature and pressure are at 25.69 ^{o}C and 101.3 KPa.

A simple fogging cooling system with various phases involved in the process is shown in *Fig. 2*. Effectively with water, the following recommendation have to be adopted to avoid damage to compressor blade [24]:

– The quality of the water used in the fogging system must be controlled.
– The pH should be between 6 and 8.
– The total dissolved solid content should be less than 5 ppm.
– Sodium and calcium content should be less than 0.1 ppm and silica content less than 0.1 ppm.
– Chloride and sulphate content less than 0.5 ppm.

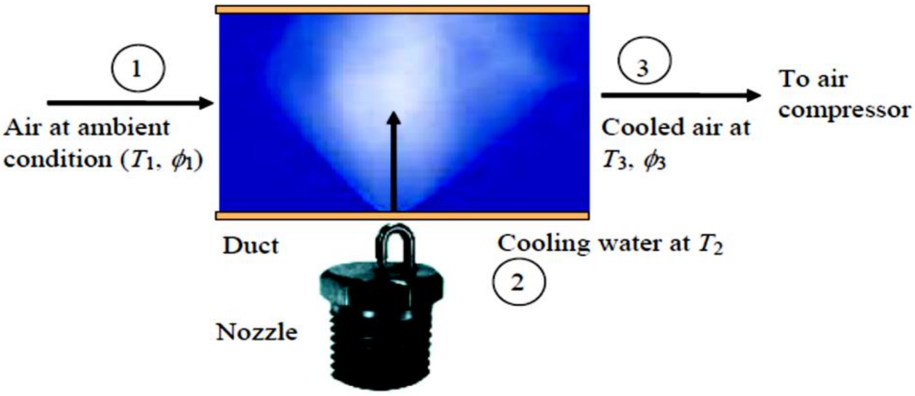

Fig. 2. *Fogging cooling system.*

As shown in *Fig. 2*, phase 1 is the original ambient condition and phase 3 is air after it undergoes cooling. Pressurized liquid water is sprayed into the air at phase 2. The temperature of air after fog cooling can be obtain from an energy balance on the dry air, water spray, and air-borne water vapor before and after the system. Assuming adiabatic mixing, the energy gained by the sprayed water is balance by the energy lost by the atmospheric air, after cooling. Mass flow rate of the water is given by *Eq. (19)*:

$$\dot{m}_w = (h_{v3} - h_{w2}) = \dot{m}(h_{a1} - h_{a3}) + w_1\dot{m}_a(h_{v1} - h_{v3}). \tag{19}$$

Where $\dot{m}_w$ is the mass flow rate of water, h_{w2} is the enthalpy, $\dot{m}_a$ is the mass flow rate of dry air, h_{a1}-h_{a3} is the enthalpy change of dry air, W_1 is the specific humidity of inlet air of water per kg of dry air, h_{v1}-h_{v3} is the enthalpy change of air-borne water vapor after cooling. The specific humidity W_1 is calculated using *Eq. (20)*:

$$w_1 = \frac{0.622P_{V1}}{P_1 - P_{V1}}. \tag{20}$$

Where P_{V1} is the partial pressure of water vapor and P_1 the total atmospheric pressure from conservation of mass, the amount of water spray is equal to the mass of water vapor at point 3 minus the water vapor originally in the air at point 1, i.e.

$$\dot{m}_w = (w_3 - w_1)\,\dot{m}_a. \tag{21}$$

Where $\dot{m}_a$ is the mas flow rate of air, $\dot{m}_w$ is the mass flow rate of water, W_3 is the humidity ratio of air after cooling which can also be calculated from *Eq. (20)*, if P_{V1} is replaced by P_{V3}. Partial pressure of water vapor at point 1 and point 3 can be calculated from relative humidity $(\emptyset_1, \emptyset_3)$.

$$P_{v1} = \emptyset_1 P_{sat1,} \tag{22}$$

$$P_{v1} = \emptyset_3 P_{sat3}.$$

Where P_{sat1} and P_{sat3} are the saturation pressures of water vapor at the corresponding temperature (T_1 and T_3). Neglecting pressure losses in the process, then P_1 equals to P_3. The inlet air temperature after cooling process T_3 can be calculated using *Eq. (23)*:

$$T_3 = T_{db1} - (T_{db1} - T_{wb2})_{\varepsilon}. \tag{23}$$

Where Tdb_1 is the dry-bulb temperature, Twb_2 is the wet-bulb temperature, ε is the cooling effectiveness. The cooling load associated with the evaporative cooling system is given by *Eq. (23)*:

$$\dot{Q}_{cl} = \dot{m}_a cp_a (T_1 - T_3). \tag{24}$$

Where cp_a is the specific heat of dry air at constant pressure assumed to be 1.005 kJ/kgK. The enthalpy of air at inlet and exit ha_1 and ha_3 is calculated using *Eq. (25)*:

$$h_{a1} = cp_{a1} t_{a1} + (2500 + 1.88 t_{a1}) w_{a1}, \tag{25}$$

$$h_{a3} = cp_{a3} t_{a3} + (2500 + 1.88 t_{a3}) w_{a3}.$$

The working fluid passing through the compressor is assumed to be an ideal mixture of air and water vapor. The total temperature of the fluid leaving the compressor having an isentropic efficiency η_c can be calculated using *Eq. (26)* [24]:

$$T_{4S} = T_3 + \frac{T_3}{\eta_c}\left[(\gamma p)\frac{\gamma-1}{\gamma} - 1\right]. \tag{26}$$

Similarly, the total temperature leaving the turbine having isentropic efficiency of turbine η_T is given by *Eq. (27):*

$$T_{6S} = T_5 - \eta_T\,(T_5 - T_6). \tag{27}$$

The power produced by the turbine due to expansion of hot gasses is obtain using *Eqs. (28) - (29)*:

$$\dot{w}_t = \dot{m}_t c_{pg} (T_5 - T_6) + w(h_5 - h_6), \tag{28}$$

$$m_t = \dot{m}_a + \dot{m}_w + \dot{m}_f = \dot{m}_a (1 + w + f). \tag{29}$$

Where f is given by *Eq. (30)*:

$$f = \frac{\dot{m}_f}{\dot{m}_a}. \tag{30}$$

2.2. Assumptions for Simulation of Gas Turbine with Fogging System

- Air contain 23.3% Oxygen and 76.7% Nitrogen by mass.
- The combustion of the process was assumed to be a conversion reaction in HYSYS.
- The conversion is 100% in the reactor.
- In the compressor, the isentropic efficiency was 87.80%, while turbines isentropic efficiency was 89.40%.
- The component of the natural gas is Methane.
- The natural gas in the feed comes directly at the pressure of 22.8 bars and temperature of 55 ^{o}C.
- Assumed mechanical loss of 97%.
- Assumed that there are no losses on the conversion energy.
- The pressure drop across the combustion chamber was assumed to be 2%.
- The pressure, ambient air temperature and mass flow rate of air are constant.

2.3. Modelling Procedures of Gas Turbine with Fogging Units

Aspen HYSYS was used to model the gas turbine units with fogging and without fogging system. The first step in creating the model was the selection of a standard set of components and a thermodynamic basis to model the physical properties of these components. When the component list was created, HYSYS created a new component list called Component List-1. The next step was the selection of a 'Fluid Package' for it. The 'Fluid Package' is the thermodynamic system associated with the chosen list of components. After completing the aforementioned procedure, the process simulation environment was initiated to begin the simulation process. The pump, mixer, separator, compressor, conversion reactor, and turbine icons from the model palette were clicked and placed on the flow sheet. Moreover, performance sensitivity analysis was carried out to determine the effects of pumping rate of fogging unit on the power plant output, inlet temperature, and the gas turbine efficiency. The schematic diagram of the simulated plant model with fogging system is presented in *Fig. 3*.

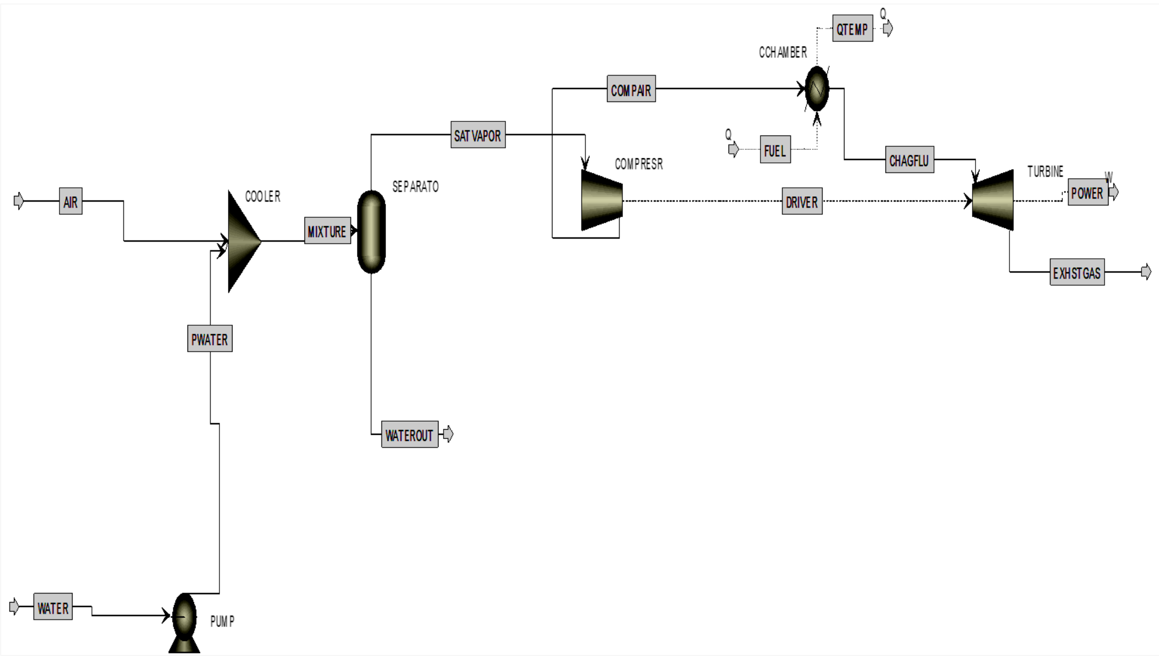

Fig. 3. *Flow chart of the simulated plant model with fogging system.*

The combined units of mixer and separator are used to simulate a fogging unit. When the incoming air enters the fogging unit, the high pressure pump then converts the demineralized water to fog through the nozzle operating at high pressure sprayed to the hot air in the mixer. The hot air mixes with the fog to ensure proper saturation of air before it leaves the mixer; saturated air then enters the separator where water and vapor are separated. The water that is not vaporized by the incoming hot air leaves the bottom of the fogging unit as shown in *Fig. 3*. Schematic diagram of the simulated plant model without fogging system is presented in *Fig. 4*. Given in the model of *Fig. 4*, air flows at ambient temperature into the compressor. In the model (*Fig. 3*) where fogging system is incorporated, air at ambient temperature passes through cooling systems before entering the compressor.

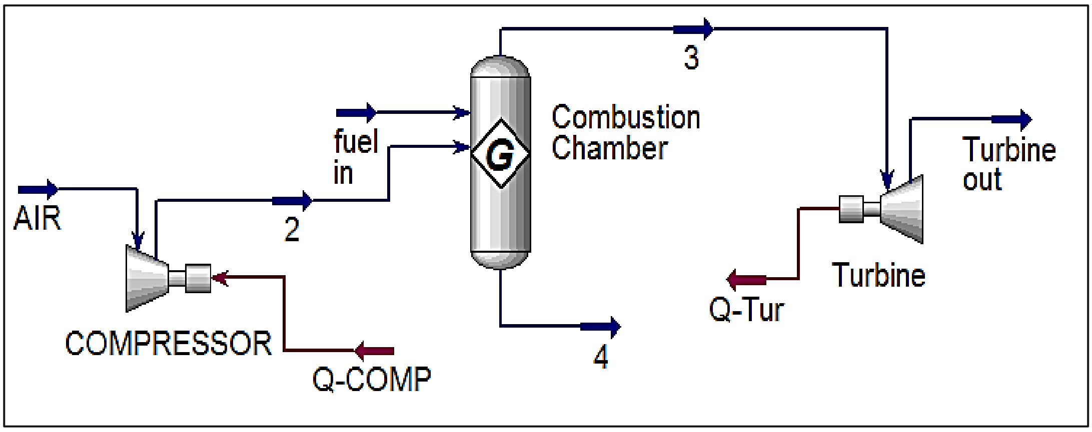

Fig. 4. *Schematic diagram of the simulated plant model without fogging system.*

3. Results and Discussion

Simulated performance of the power plant without fogging system and simulated performance of the power plant with fogging system incorporated are presented in *Table 2*. The average inlet temperature was taken as 294 K. *Fig. 5* is a graphical representation of output results with fogging and without fogging system.

Table 2. *Simulated performance of the power plant without fogging.*

Parameters	Without Fogging	With Fogging	Unit
Compressor Work	*112.11*	*82*	*MW/h*
Turbine Work	*229.57*	*213*	*MW/h*
Net Power	*117.46*	*131*	*MW/h*
Thermal efficiency	*39.39*	*43.93*	*%*
Net heat rate HR	*9140.06*	*8195.35*	*kJ/kWh*
Specific Fuel Consumption	*0.199*	*0.179*	*kg/kwh*

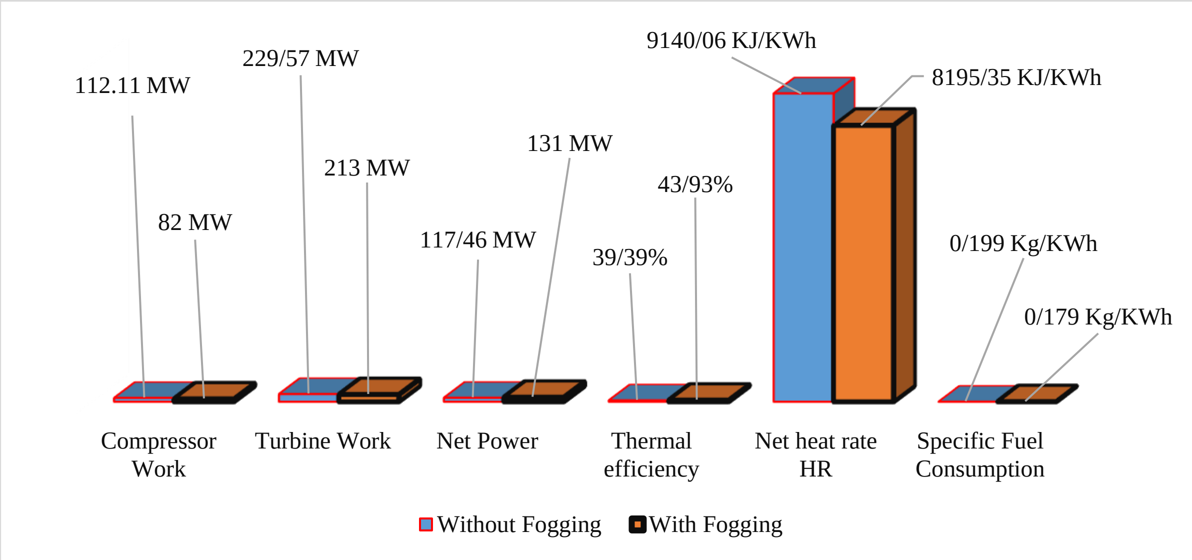

Fig. 5. *Representation of output results with fogging and without fogging unit*.

SFC is an engineering term that is used to describe the fuel consumed by an engine for each unit of energy produced. In gas turbine engines, it is the ratio of the mass of fuel consumed to the output power of an engine. It is measured in kilogram per mega joules or kilogram per megawatt and it is best when the value is minimum. As shown in *Fig. 5*, the SFC obtained from the power plant simulation when fogging is not incorporated was 0.199 kg/kwh, whereas SFC of the plant with fogging was 0.179 kg/kwh. Comparing the two values of SFC indicated that operation of the plant with fogging provides a better fuel economy/efficiency and cost effective operation. Like the case of Omotosho power plant, the fuel used in most power plants all over the world as well as the gas turbine units is natural gas [25], because it is a clean energy resource that burns with less emissions.

Thermal efficiency is the ratio of work done (output) by a system to the heat input or heat supplied to it. As shown in *Fig. 5*, thermal efficiency of 43.93% was obtained from the result of the simulated power plant with fogging system, whereas, thermal efficiency of 39.39% was obtained from the result of the simulated power plant operated without fogging system. Compared to thermal efficiency of 35.82% obtained from fogging system in a study carried out by Orhorhoro et al. [3], the aforementioned thermal efficiency obtained in this study is higher and better for Omotosho plant operation. From a thermodynamic point of view, a system with higher thermal efficiency is likely to produce more work output and is more reliable than a system with low thermal efficiency. This implies that operating the power plant with fogging system which produces higher thermal efficiency will produce more work output. This is necessary in power plant operations in terms of boosting profits, as industrial operators pay no attention to non-value added activities that may influence production output [26].

Net power is a property that relates to the amount of energy transmitted per unit time by a system. As shown in *Fig. 5*, the net power obtained (131 MW) from the simulation of the power plant is higher when the fogging system is incorporated than the net power obtained (117.46 MW) when the system operates without fogging system. This implies that work requirement will reduce and the desired output will increase when the system operates with fogging system incorporated in it.

For the compressor work and turbine work, 82 MW and 213 MW were obtained from the simulation of the power plant operating with fogging system, as 112.11 MW and 229.57 were obtained from the simulation of the plant operating without fogging. This is the power required for operation of the turbine and the compressor. From the aforementioned values, the power requirement for these two systems are comparably lower when the power plant operates with fogging system than when it operates without fogging system. Temperature is vital to the operation of compressors and turbine systems in a power plant, but to achieve a single unit of temperature for effective operation of the plant goes at a cost. The simulated temperature requirement for both compressor and turbine system in Omotosho power plant is presented in *Fig. 6*.

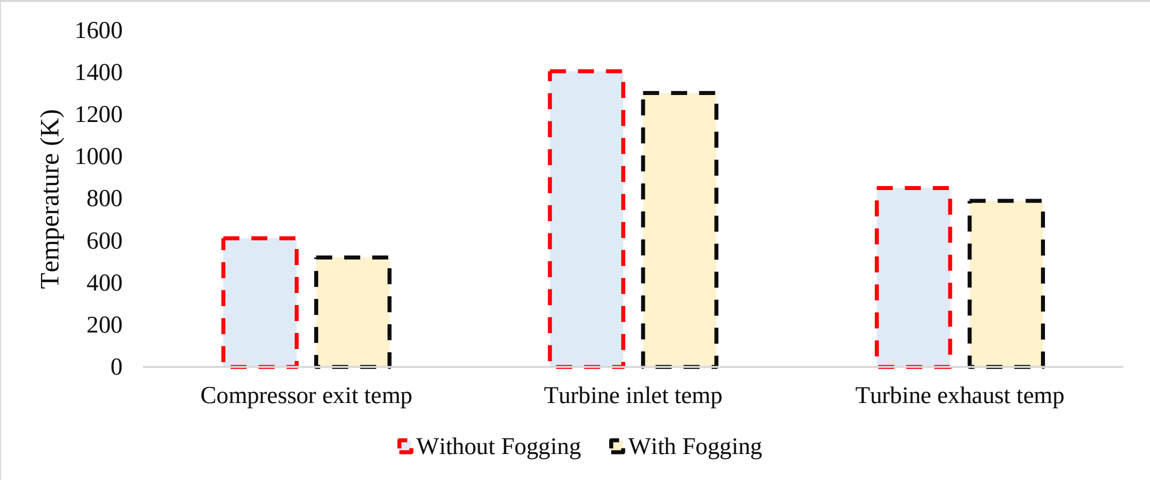

Fig. 6. *Temperature requirement of the power plant with and without fogging unit.*

It can be observed in *Fig. 6* for both compressors exit temperature and the gas turbine inlet and exhaust temperature that operate with fogging system require low temperature compared to when fogging system is not incorporated. The low temperature requirement offers great advantage in terms of the cost needed to attain the temperature of the said equipment. Another point is that, materials with high conductivity and thermal properties may be required when the temperature requirement is high and this is where the importance of fogging system comes into play.

3.1. Green House Gas Emission Saving

Eq. (31) was used in calculating greenhouse gas emission of the fuel (natural gas) for both plants.

$$\text{Emission GHG fuel} = fuel\ consumption \times emission\ factor\ of\ CO_2. \tag{31}$$

Where emission GHG fuel= emission of a given GHG by type of fuel (kg/GHG); fuel consumption= amount of fuel combusted in kg/kWh; emission factor GHG fuel= emission factor of a given GHG by type of fuel (kWh) for CO_2. Thus, calculating emission of GHG with fogging specific fuel consumption= 0.179 kg/kWh; emission factor of CO_2 natural gas= 34260.008 kWh; GHG emission with fogging $0.179 \times 34260.008 = 6{,}132.54$ kg/h.

Calculating emission of GHG without fogging specific fuel consumption= 0.199 kg/kWh; emission factor of CO_2 natural gas = 34260.008 kWh; GHG emission without fogging= $0.199 \times 34260.008 =$ 6,817.74 kg/h; emissions saved with fogging= 6,817.74 - 6,132.54 = 685.2 kg/h.

The total emissions generated when the power plant operates with fogging system, without fogging system, and the total emission saved with fogging system are presented in *Fig. 7*.

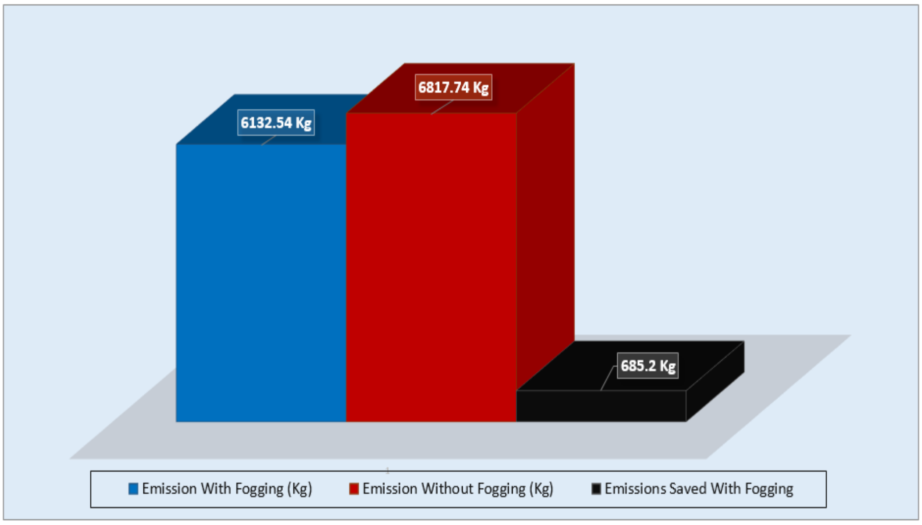

Fig. 7. *Emission savings from power plant operating with and without fogging unit.*

A greenhouse gas is a type of gas that absorbs and emits radiant energy within the thermal infrared range. Increasing greenhouse gas emissions cause the greenhouse effect such as global warming, climate change, increase in earth's temperature, etc. Since the beginning of industrial revolution, the human activities have been a major emitter of greenhouse gases. A comparative chart showing the GHG emission from Omotosho power plant operating with fogging system and without fogging system is presented in *Fig. 7*. From the emission analysis presented in this section, it can be observed that operating the power plant with fogging system emitted over 6,132.54 kg/h of greenhouse gases, whereas, operating the power plant without fogging system emitted over 6,817.74 kg/h.

Comparing the two calculated values when fogging system is incorporated in the power plant operation and when the power plant operates without fogging system, it can be observed that fogging system saved about 685.2 kg per hourly operation of the power plant. Assuming the power plant operates every day for 365 days, a total of 6002353 kg of greenhouse gas will be saved when fogging system is incorporated into the power plant operation. This is a green technology that can go a long way in minimizing the rate of global emissions from power plants which is now designed in many forms for industrial purposes. For example, Renewable Energy Power Plant (REPPs) which also uses gas turbines are designed not only for investing purpose but also for maximizing the resource usage (sun, water, and wind) and minimizing the raw materials such as aluminum, iron, and silicon among others [27].

3.2. Sensitivity Analysis

The effect of pumping rate of fogging unit on the mass flow rate ranging from 3.5 kJ-4.5 kJ on the power output, temperature, and efficiency were investigated. Results of the simulated sensitivity analysis is shown in *Figs. (8) - (10)* as follows:

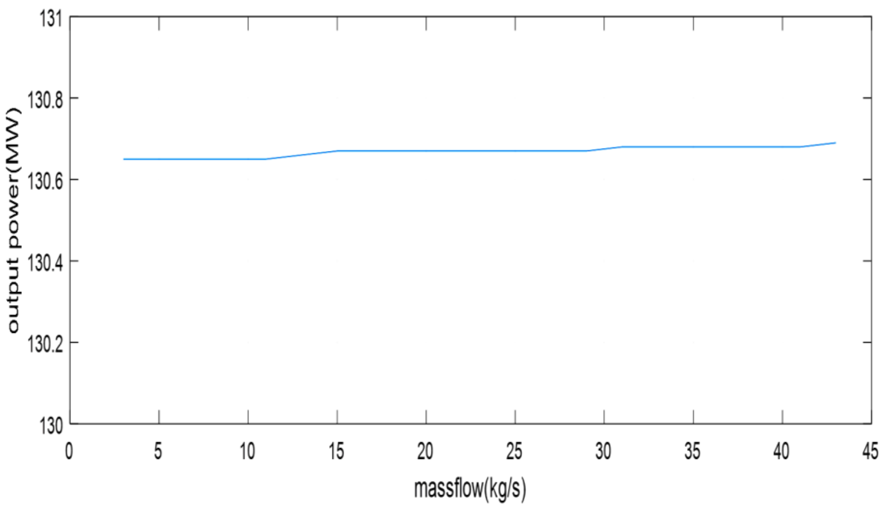

Fig. 8. *Effect of pumping rate of fogging unit on the power output.*

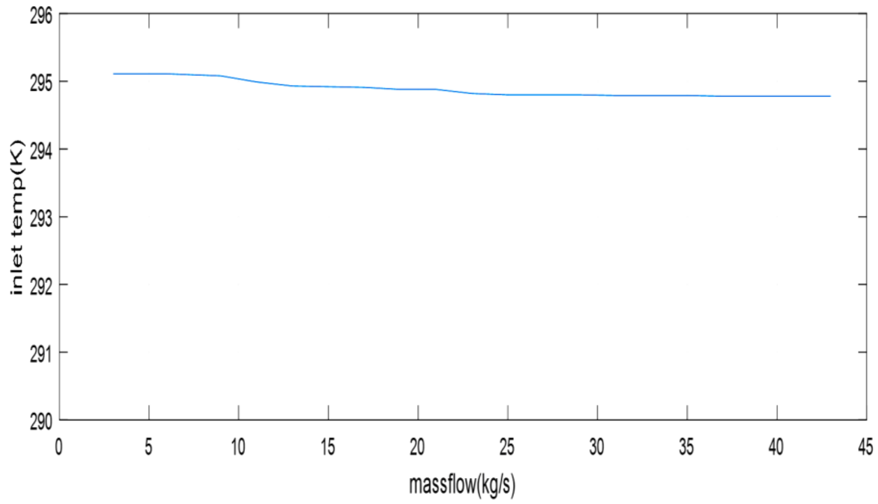

Fig. 9. *Effect of pumping rate of fogging unit on the inlet temperature.*

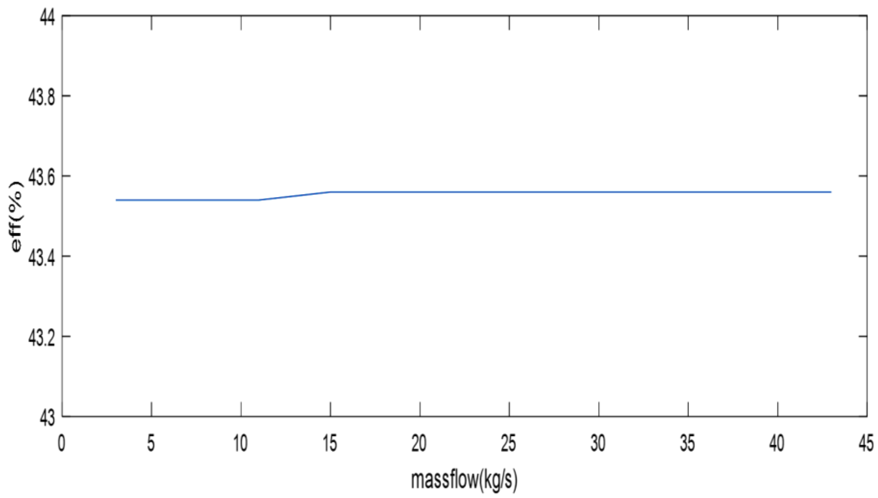

Fig. 10. *Effect of pumping rate of fogging unit on the gas turbine efficiency.*

The variation of indicated power output to the mass flow rate is presented in *Fig. 8*. It is observed that as the mass flow rate of the water entering into the fogging unit increases from 3 kg/s to 43 kg/s which is the range of values chosen at random for this analysis during simulation; there is a corresponding increase in the indicated power output from 130.65 MW to 130.69 MW which shows no significant effect on the output indicated power. Thus, this indicates that an external power supply for energizing the fogging unit will be of high consideration if a significant amount of power needs to be produced from the gas turbine. The effect of pumping rate of fogging unit on the inlet temperature is shown in *Fig. 9*. It is observed that higher cooling rate is achieved if the mass flow of spray fluid increases. However, a less temperature change is achieved as it only reduces from 295.11 to 294.78 ºC. From *Fig. 10*, it is observed that there is no significant change on the gas turbine efficiency as the mass flow rate increases. Consequently, in order to achieve further improvement of the plant efficiency, the fogging system needs to get extra power from an external device or system such as solar or other forms of renewable energy system.

4. Conclusions

The models developed for high pressure fogging system were successfully employed in the analysis of Omotosho power plant cooling system as follows:

- With reduced SFC achieved through the incorporation of fogging system, cost effective operation is obtained, such that the cost required to achieve higher temperature input and the cost required to purchase more fuel are reduced.
- With the incorporation of fogging system in the power plant, it was observed that GHG emission was minimized and this led to reduction in the rate of environmental pollution.
- From the performance sensitivity analysis, there was no significant effects of the pumping rate of fogging unit on the efficiency of the power plant.

The thermodynamic effects of incorporating high pressure fogging system in Omotosho power plant indicated that the performance of some operating parameters such as power output and thermal efficiency improved while there was 944.71 kj/kwh reduction in the net heat rate and 0.02 kg/Kwh reduction in specific fuel consumption. Increased thermal efficiency of 4.5% and increased net power of 13.54 MW were also achieved in the power plant through the incorporation of fogging system. This study has successfully shown that reducing the temperature of intake air entering a gas turbine unit increases the mass flow rate and enhances its net output, and above all, reduces the specific fuel consumption and heat rate.

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