



ADAPTATION OF EJECTOR REFRIGERATION SYSTEM TO GAS TURBINE POWER PLANT FOR PERFORMANCE IMPROVEMENT

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Abstract

In this paper a study of the effect of an ejector refrigeration system on the performance of a gas turbine power plant was done. A set of actual operational conditions was used to determine the compressor work, turbine net work, specific fuel consumption and the thermal efficiency of the plant. The results showed that a decrease in the ambient temperature, decreases the compressor work, increases turbine net work, decreases the specific fuel consumption and increases the efficiency of the plant. It was also observed that incorporating the refrigeration cooling system which was simulated using Matlab and Hysys software causes a drop in ambient temperature of 15°C and this leads to an increase of about 12.8% and 173.12kW in thermal efficiency and power output of the turbine respectively and also, 0.01kg/KWh drop in specific fuel consumption. Therefore, to improve the performance of an existing gas turbine power plants in high temperature regions, an ejector refrigerator system that will reduce the compressor inlet air temperature is recommended.

Keywords: Combined-power-and-cooling; Ejector-refrigerator-system; Compressor-inlet temperature; Thermal efficiency.

1. Introduction

Energy is one of the primary needs of human societies for their survival. It is needed for growing food, providing comfort and catering for a host of other application in all fields of activity such as agriculture, industry, transportation, etc. The main sources of energy are fossil fuels, solar radiation fall out, winds, tidal, and geothermal. The conversion, distribution and utilisation of energy are the domain of engineering. The demand for energy in Nigeria is increasing sharply because Nigeria is one the developing countries in the world with rising population, living standards and so there is an emphasis on developing energy to boost her economy in order to combat poverty and hardship. One of the power generating system that is used in Nigeria is the gas turbine power plant.

The gas turbine power plant is a rotor-dynamic internal combustion engine; therefore, its working cycle involves four (4) processes, such as compression, heating, expansion and discharge. It is a conventional power plant receiving fuel energy (F),

producing work and rejecting heat to a sink at low temperature. The objective is to achieve the least fuel input for a given work output as this will be economically beneficial in the operation of the power plant, thereby minimizing the fuel costs. However, the capital cost of achieving high efficiency has to be assessed and balanced against the resulting saving in fuel costs. It is important to distinguish between a closed cyclic gas power plant (or heat engine) and an open circuit power plant. In the former, fluid passes continuously round a closed circuit, through a thermodynamic cycle in which heat received from a source at a high temperature, heat is rejected to a sink at low temperature and work output is delivered, usually to drive an electric generator. Usually, a gas turbine plant operates on an "open cycle", with internal combustion. Air and fuel pass across the single control surface into the compressor and combustion chamber. The performance of the engine, that is its thermal efficiency and power output, depends on the two principal operational variables: cycle turbine entry temperature and pressure ratio.

This can be achieved by cooling the ambient air temperature using cooling systems [1].

In the Saudi Electric Company's (SEC), for example, approximately 42% of the annual energy is generated by combustion turbines, and during the summer. These turbines suffer a 24% decrease in their capacity, due to ambient temperature up to 50°C [1]. Gas turbines designed to operate at maximum efficiency at standard ambient temperatures and relative humidity may tend to reduce in performance due to adaptation problems resulting from variation in weather conditions as they are installed at different locations [2].

This work involves the modification of the Omoku gas turbine power plant using the exhaust temperature to power a refrigeration system which in turn will be used to cool the compressor inlet air which will cause an improvement on the performance of the gas turbine power plant.

Gas turbine power plant in Nigeria has in the past decades had an average thermal efficiency at the range of 27-30%. This low efficiency is tied to many factors which include operation mode, poor maintenance procedures, age of the plant, and high ambient air temperature and relative humidity [2]. As ambient air temperature rises, less air can be compressed by the compressor since the swallowing capacity of the compressor is known, and then the gas turbine output is reduced at a given turbine entry temperature. The compressor work also increases because the specific volume of the air increases in proportion to the intake air temperature. Generally, the gas turbine power plant is designed with a constant volume flow in the compressor and so when the ambient air temperature increases, its specific mass is reduced, so that the mass flow rate entering the gas turbine is accordingly decreased, this would in turn decrease the power output of the gas turbine. Thermodynamic analysis was used to prove that the thermal efficiency and specific power output decreased with an increase in humidity and ambient air temperature. This research intends to mitigate this problem by adopting the ejector refrigeration system which uses the waste heat from the gas turbine in cooling the compressor inlet air temperature and in turn increases the mass flow rate and reduces the compressor work and thereby increases the output power and efficiency of the plant.

Nasser et al [3], explains that the gas turbine performance is critically limited by the predominating ambient temperature, mainly in hot and dry regions. It occurs because the power output is inversely proportional to the ambient temperature. The temperature drop provides and augments in the air density and consequently elevates air mass flow rate; this behaviour increases the power output and efficiency at about 0.7% per degree Celsius for heavy duty gas turbine [4].

Mohanty et al. [5] studied a similar system for a 100MW gas turbine in Bankok, taking the inlet temperature down to 15°C. They achieved instantaneous power output increases of between 8 and 13%, with an overall increase of 11%.

Investigation was carried out on the auxiliary power consumption for different turbine inlet cooling systems in Oman at two locations named Marmul and Fahud considering evaporative cooler, fogging, absorption chillers using both water-lithium bromide and aqua-ammonia, and mechanical compression systems. The water-lithium bromide chiller provides 40% and 55% more energy than fogging at Fahud and Marmul, respectively. But applying aqua-ammonia water and mechanical compression, a further increase in annual energy production of 39% and 46% is predicted in comparison with water-lithium bromide absorption chiller at Fahud and Marmul, respectively [5].

In the works of Alhazmy et al. [6], they compared two different techniques of air coolers, cooling coils and water spraying system, and the results were analyzed for a specific set of operational and design conditions. The chiller coils are more expensive than spray coolers. However, the latter is extremely affected by ambient temperature and relative humidity. This method is capable of reducing the ambient air temperature by 3-15°C, producing power augmentation by 1-7%, and increasing the efficiency by 3%. Chiller cooling offers full control over the air intake conditions, yet it has high parasitic power consumption and this power is removed from the gas turbine output. Their result also showed that although the cooling coil during cold and humid conditions ($T=50^{\circ}\text{C}$ and $\phi=80\%$), the demand operational power reduces the net power by 6.1% and 37.6% respectively.

The results of the influence of air cooling intake on the gas turbine performance by comparing two different cooling systems, evaporative and cooling coil showed that the chiller and evaporative cooling system present similar improvement in the power output about 1.0-1.5MW, but the cooling coil of the mechanical chiller consumes more energy to run the vapour-compression refrigeration unit and the overall plant performance decrease, [7].

Al-Ibrahim et al. [1] reported that refrigerative cooling uses mechanical or electrical vapour compression refrigeration equipment. Equipment and O&M costs for mechanical chillers are cheaper than absorption systems, but capital costs are higher and parasitic power requirement can be 30% of the power gain.

Al-Ibrahim et al. [1] performed a review of inlet air cooling methods that can be used for improving the power output of Saudi Arabian industry's gas turbine during summer days. Their results at the end of the review shows that the evaporative cooling system requires a large amount of water and this puts a

limitation on its use in the desert climates. The absorption chiller is an expensive system and its cost of investment is too high. Air humidification is normally carried out by spraying water in the ambient air flow upstream of the compressor inlet. This method requires a good quality water to avoid corrosion and erosion of the compressor blades. Also, the droplet drift may increase the amount of water for the humidification process.

Ali et al. [8] studied a chiller and evaporative cooling systems thermally and economically for a 264MW gas turbine plant located at Korymat, Sonther Egypt. The results obtained shows that the annual power gained by chiller cooling is 117,027mwh and the net cash flow is \$3,787,537 while the annual power gained by the evaporative cooling system is 86,118Mw and the net cash flow is \$4,503,548.

Popli et al. [9] applied different turbine inlet cooling systems in order to enhance the gas turbine performance for two Brazilian sites called Campos and Goiania. The results showed that the best turbine cooling system for both sites is the absorption chiller in terms of energy cost and generation, while the evaporative cooler has provided only a limited enhancement for the performance with the lowest cost.

Robert et al. [10] carried out a study on the evaporative cooling on the performance of gas turbine plant operating in Bayelsa State, Nigeria where a set of actual operational conditions such as ambient temperature and relative humidity were employed to determine compressor work output, actual turbine work output, thermal efficiency and specific fuel consumption of the plant. The results obtained showed that a decrease in the ambient temperature, causes decreases in the compressor work, increase turbine network.

The current research involves the modification of the an existing gas turbine power plant with an ejector refrigeration system powered by its exhaust heat (combined cycle), which is used to cool the compressor inlet air. The combined cycle is simulated to ascertain the performance improvement of the gas turbine power plant. Results for power output and gas turbine thermal efficiency are obtained and compared with the results of the gas turbine without a cooling system.

The influence of performance parameters of the ejector refrigeration system on the compressor intake air temperature drop of the Omoku power plant is also investigated

2. Materials and Methods

2.1 Data collection

The data for this study were obtained from an operational unit of a 25MW GE Nuovo Pignone, Industrial gas turbine plant located at Omoku, Rivers State, Nigeria. The values of the input parameters used

for this study were obtained from the log sheet for an interval of three (6) months, (October, 2017-March, 2018). Data were also collected daily during the operation of the plant for a period of one (1) month.

The Matrix laboratory (Matlab) software which is a technical computation software will be used to calculate the modelled equations of the gas; turbine and the combined system while Aspen Hysys software (a process simulation software) is also used to simulate the gas turbine plant data without cooling and with cooling of the inlet air.

2.2 Description of the gas turbine cycle

The gas turbine is considered as a heat engine because it receives heat at high temperature from a heat source, and produces power, and after that, it discharges the heat at a lower temperature to a heat sink. Figure 1 shows the schematic diagram of a simple gas turbine cycle.

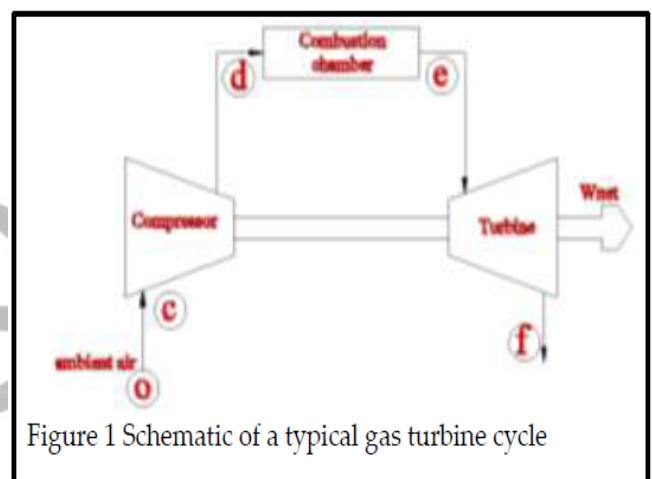


Figure 1 Schematic of a typical gas turbine cycle

The governing equations used to achieve a better understanding of this work is derived using thermodynamics relations which are then subsequently used for the simulation of the gas turbine plant.

The inlet air temperature to the compressor in typical gas turbine is equal to ambient temperature. Air has been considered as ideal gas in all gas turbine cycle, also using the polytrophic relation for ideal gas.

$$\frac{T_d}{T_c} = (r_p)^{\left(\frac{k-1}{k}\right)} \quad (1)$$

Where

r_p = compression ratio in compressor,

$$k = \frac{C_p}{C_v} \quad (2)$$

k = specific heat ratio:

Where

C_p and C_v are specific heat at constant pressure and volume, respectively, also pressure ratio:

$$r_p = \frac{P_d}{P_c} \quad (3)$$

From equation (1) T_d can be calculated as:

$$T_d = T_c(r_p)^{\frac{k-1}{k}} \quad (4)$$

Considering the first law of thermodynamic on the compressor, the workdone is estimated as follows:

$$\dot{W}_{comp} = \dot{m}_a C_{pa}(T_d - T_c) \quad (5)$$

Where

\dot{m}_a = air mass flow rate and

$$C_{pa}(T) = 1.04841 - \frac{3.8371T}{10^4} + \frac{9.4537T^2}{10^7} - \frac{5.49031T^3}{10^{10}} + \frac{7.9298T^4}{10^{14}} \quad (6)$$

Where C_{pa} is the specific heat capacity of air at constant pressure, [7].

The heat delivered by the combustion chamber is determined from energy balance:

$$\dot{Q} = \dot{m}_a C_{pg}(T_e - T_d) \quad (7)$$

Where

$$C_{pg}(T) = 0.991615 + \frac{6.99703T}{10^5} + \frac{2.7129T^2}{10^7} - \frac{1.22442T^3}{10^{10}} \quad (8)$$

C_{pg} = flue gas specific heat at the combustion as function of the average value [7]. Also fuel gas consumption in combustor chamber is defined as:

$$\dot{m}_f = \frac{\dot{Q}_{in}}{LHV \times \eta_{cc}} \quad (9)$$

Where

η_{cc} = combustion chamber efficiency

LHV = fuel gas lower heat value.

The discharge temperature of the exhausted gas that living the turbine is defined as:

$$T_f = T_e(r_p)^{\frac{k-1}{k}} \quad (10)$$

Where

P_f = turbine exhaust gas pressure in the last step of typically gas turbine cycle. Therefore, the total power produced from the turbine is equal to:

$$\dot{W}_t = \dot{m}_T C_{t,av}(T_e - T_f) \quad (11)$$

$$\dot{m}_T = (\dot{m}_a + \dot{m}_f) \quad (12)$$

Where

\dot{m}_T = total mass flow rate

\dot{m}_a = air mass flow rate

Also $C_{t,av}$ = fuel gas specific heat in average temperature through the turbine [7].

The net power obtained from the gas turbine cycle can be calculate by using (5) and (11):

$$\dot{W}_T = \dot{W}_t - \dot{W}_c \quad (13)$$

Lastly the thermal efficiency of gas turbine can be calculated as follow:

$$\eta_{th} = \frac{\dot{W}_T}{\dot{Q}_{in}} \quad (14)$$

In the absence of a totalizer the nearest estimation of total gas consumed by the gas turbine unit is calculated by the specific consumption.

$$SFC = \frac{3600 * f}{P_{net}} \quad (15)$$

where SFC is the turbine specific fuel consumption

2.3 Combined Gas Turbine and Refrigeration cycle

The gas turbine power plant consists of compressor, combustion chamber and turbine, but in this work the ejector refrigeration system inlet air cooling technique is proposed for analysis. Fig. 2 shows a schematic diagram of the system herein studied, which consists of a standard gas turbine power plant and an intake air cooler.

The working principle of the system is generalized as follows:

- Low-grade heat (Q_g) is delivered from the exhaust of the gas turbine to the generator for vaporization of the refrigerant.
- The high-pressure vapour out from the generator, i.e. the primary flow, enters into the ejector nozzle and draws low-pressure vapour from the evaporator, i.e. the secondary flow.
- The two flows undergo mixing and pressure recovery inside the ejector. The mixed flow is then fed into the condenser, where condensation takes place by rejecting heat (Q_c) to the heat sink.
- The liquid from the condenser is divided into two parts. One 5 goes through the expansion device to the evaporator, where it evaporates and hence produces a cooling effect (Q_e).
- The remaining liquid is pumped back to the generator by the pump, and completes the cycle. It is also recognized that the ERS can be considered as two loops, a power loop (an organic rankine cycle), and a refrigerating loop (a vapour compression refrigeration cycle), [13].

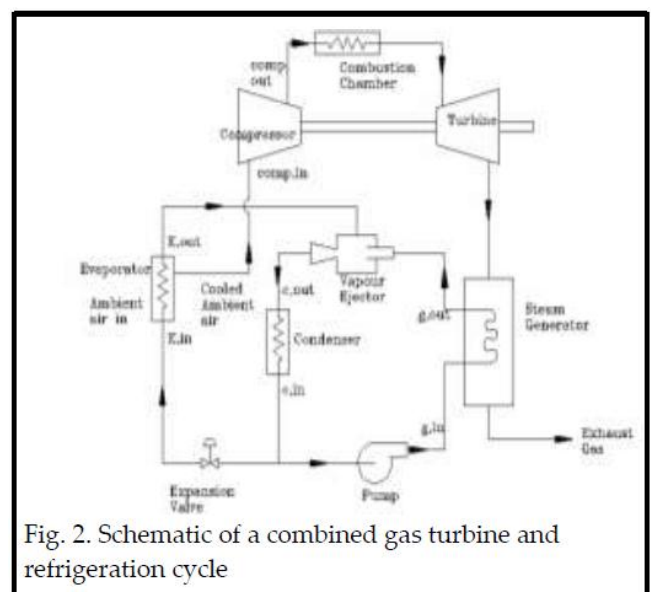


Fig. 2. Schematic of a combined gas turbine and refrigeration cycle

3. Results and Discussion

The parameters pertinent for this study are those of the turbine compressor inlet temperature starting with the refrigeration system, temperature at the end of the compressor process, specific fuel consumption, net

turbine work output, and the thermal efficiencies of the turbine plant ascertained in the period under examination.

Table 1. Summary of the overall average of the working parameters

MONTHS [017/018]	T3 [K]	T4 [K]	T5 [K]	T6 [K]	P3 [BAR]	P4 [BAR]	RP [-]
OCTOBER	299.00	551.46	1299.12	743.00	1.00	8.52	8.52
NOVEMBER	300.00	553.49	1299.99	743.30	1.00	8.53	8.53
DECEMBER	300.80	555.15	1300.68	743.50	1.00	8.54	8.54
JANUARY	01.00	555.70	1301.20	743.60	1.00	8.55	8.55
FEBRUARY	301.50	556.81	1301.90	743.80	1.00	8.56	8.56

3.1 Effect of pressure ratio variation on compressor work

Figure 3 clearly indicates the variations of the pressure ratio with the workdone in the compressor. It shows that the compressor work and the pressure ratio are proportional (i.e. the increase in the pressure ratio causes an increase in the compressor work).

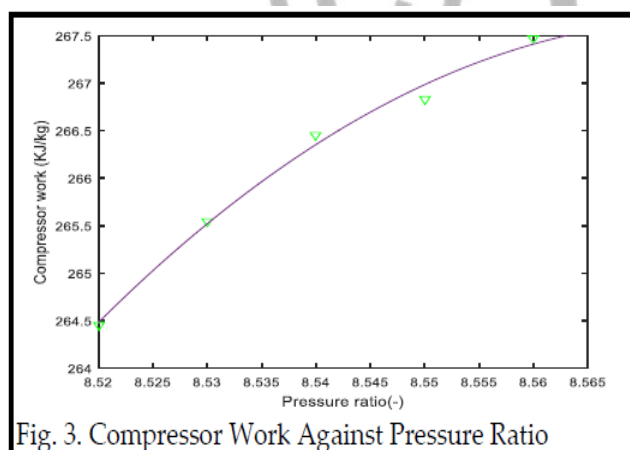


Fig. 3. Compressor Work Against Pressure Ratio

3.2 Effect of cooled air temperature variation on compressor work

Figure 4a clearly indicates the variation of the ambient air temperature with the workdone in the compressor. It shows that the compressor work and the ambient air temperatures are proportional (i.e. the increase in the ambient temperature causes an increase in the compressor work). But when the refrigeration cooling system is used to cool the ambient air temperature, the mass flow rate of the air increases and this in turn decreases the compressor work as shown Figure 4b.

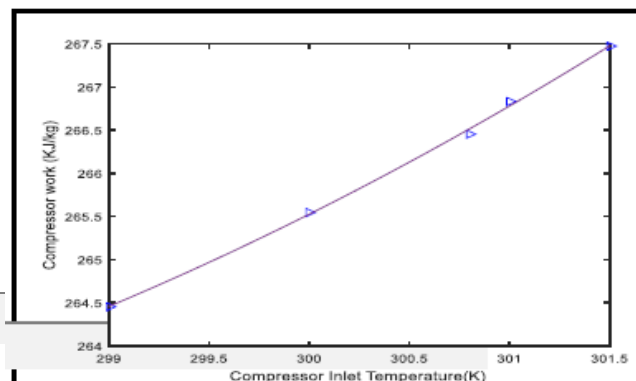


Fig. 4a. Compressor Work Against Compressor Inlet Temperature before Cooling

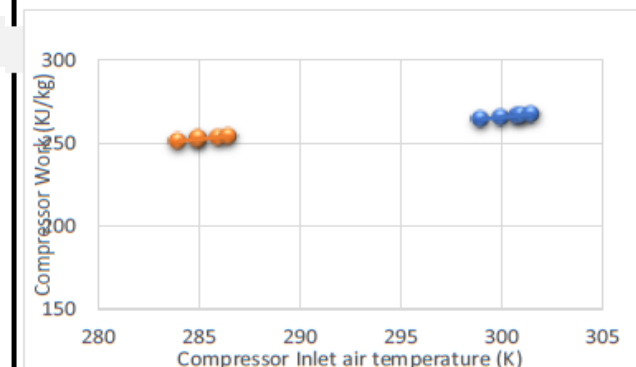


Fig. 4b. Compressor Work Against Compressor Inlet Temperature after Cooling

3.3 Effect of cooled air temperature variation on the thermal efficiency of the gas turbine plant.

Figure 5a also, illustrates the impact of compressor inlet temperatures on the efficiencies of a gas turbine plant. But as can be seen in Figure 5b, the effect of compressor inlet temperature is such that a fall in the ambient temperature of 15°C does results to thermal efficiency rise of about 12.8%. This reveals that a fall in the compressor inlet temperature gives rise to the efficiency of the plant. Plant efficiency is also dependent on the pressure ratio, when the turbine runs at a high pressure, the plant efficiency increases greatly.

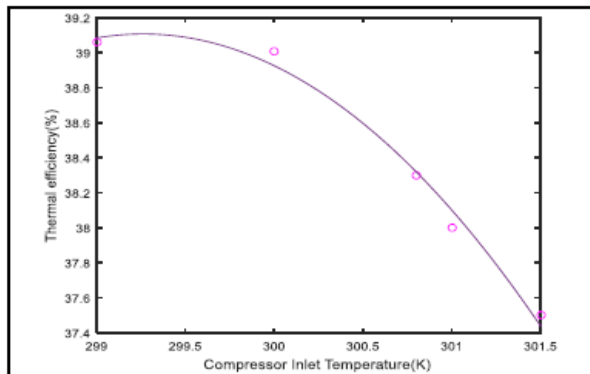


Fig. 5a. Turbine Thermal Efficiency Against Compressor Inlet Temperature before Cooling

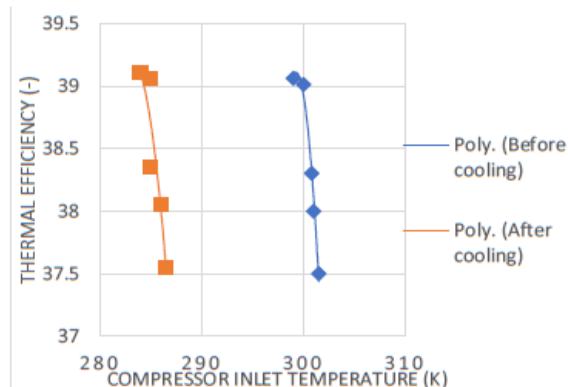


Fig. 5b. Turbine Thermal Efficiency Against Compressor Inlet Temperature after Cooling

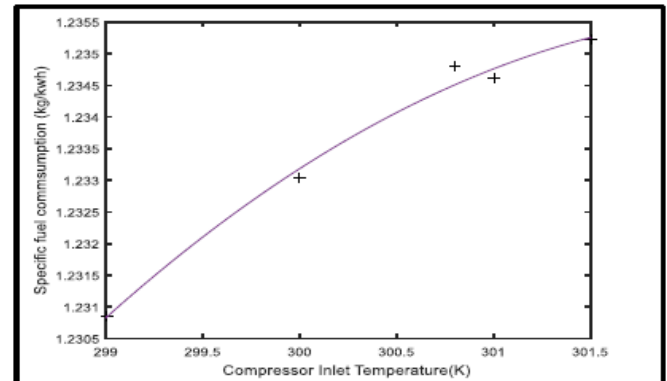


Fig. 6a Specific Fuel Consumption Against Compressor Inlet Temperature before Cooling

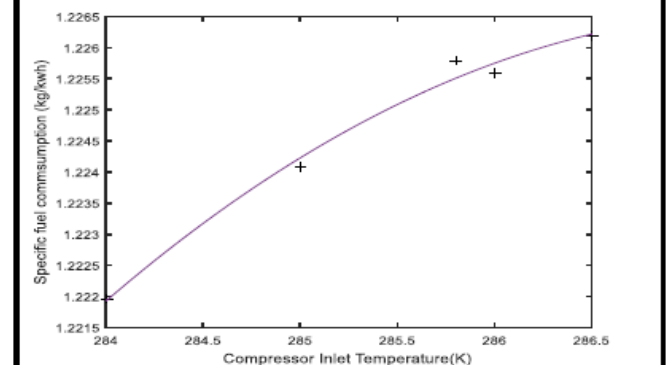


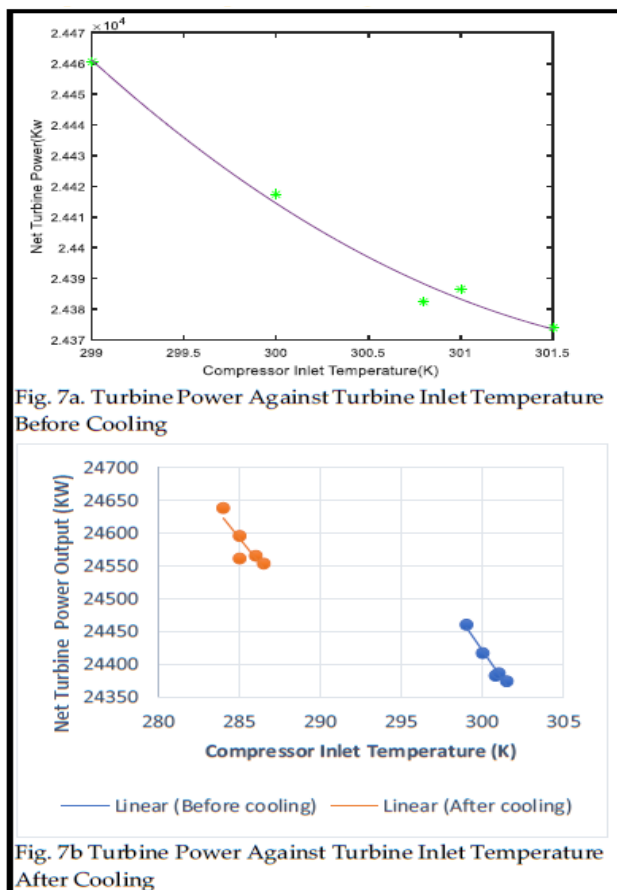
Fig. 6b. Specific Fuel Consumption Against Compressor Inlet Temperature after Cooling

3.4 Effect of cooled air temperature variation on specific fuel consumption

Figure 6a and Figure 6b shows that the ambient temperature does have a great impact on the specific fuel consumption of the gas turbine plant. In the two graphs, it may be seen that in spite of the fact that the specific fuel consumption is in direct proportion with increase in ambient temperature, it could be made less with decrease in compressor inlet temperature. Comparison of both figures (Figure 6a and Figure 6b) proves that a temperature reduction of 15°C produces a drop in the specific fuel consumption by 0.01kg/KWh. The implication of this is that since the specific fuel consumption compares the ratio of fuel used by an engine to the power produced, decrease in the ambient air temperature causes the consumption of fuel to be reduced by 0.81%.

3.5 Effect of cooled air temperature variation on the power output of the plant

The effect of ambient temperature on the power output of the plant is illustrated with Figure 7a and Figure 7b. The two figures showed that the power output (network) gradually increase with lower turbine inlet temperature. However, the benefit of the application of the refrigeration cooling system is such that a decrease in the ambient temperature could lead to a gain of 178.21KW power output. The simple explanation to this is, when the air temperature is lowered, its density increases which leads to lower volume of air required for same mass handled. The result of this is a decline in the compressor specific work. Therefore, if the peak temperature of the turbine and its pressure ratio remain constant, then the only variable parameter due to the cooling becomes the compressor. This means that lower compressor work produces higher net work.



r_p = Compression ratio [bar]
 k = Specific heat ratio [-]
 C_p = Specific heat capacity at constant pressure [kJ/kg K]
 C_v = Specific heat capacity at constant volume [kJ/kg K]
 P_e = Compressor inlet pressure [bar]
 P_d = Compressor discharge pressure [bar]
 W_{comp} = Compressor work [KJ]
 M_a = Mass flowrate of air [kg/s]
 C_{p_a} = Specific heat capacity of air at constant Pressure [KJ/kg K]
 C_{p_g} = Specific heat capacity of gas at constant pressure [KJ/kg K]
 Q = Heat delivered by the combustion chamber [KJ]
 T_f = Turbine exhaust temperature [K]
 M_f = Mass flowrate of fuel [kg/g]
 LHV = Fuel gas lower heating value [K]
 T_e = Combustion chamber outlet temperature [K]
 W_t = Turbine power [Kw]
 m_T = Total mass flowrate [kg/s]
 η_{th} = Thermal efficiency [%]
 W_T = Work net of the system [KJ]
 SFC = Specific fuel consumption [kg/KWh]
 COP = Coefficient of performance

4. Conclusion

An ejector refrigeration system has been adapted to a gas turbine power plant inspired by the Omoku gas turbine power plant facility. The parameters considered in this paper are the ambient temperatures, work done by compressor, combustion chamber inlet temperature, turbine network and the thermal efficiency of the plant. The study of the performance parameters showed that:

- As the ambient temperature decreases, the density of the air taken into the compressor increases and this causes an increase in the air mass flow rate. Also, less work is required from the turbine to drive the compressor. These all contributes to the increase in efficiency and the power output of the turbine.
- The gas turbine plant performance improves with small but continual rise in specific fuel consumption and air-fuel ratio. This suggests that as the compressor inlet air temperature decreases, fuel consumed by the turbine reduces.
- Since it has been proved that decrease in compressor inlet temperature reduces NO_x emission, refrigeration cooling system can be a means of reducing NO_x emission from the gas turbine power plant.

Nomenclature

T_c = Compressor Inlet Temperature [K]
 T_d = Compressor Inlet Temperature [K]

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