2022

# Performance and Efficiency of Combined Cycle Power Plants

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## ABSTRACT

In this study, the authors conducted rigorous energetic analysis for combined cycle gas turbine (CCGT) power plant in Khartoum North Station, Sudan, with similar plant data sets. The energy and performance efficiency of each component and overall plant is initiated by detailed mass and energy balances of the combined Gas and Steam power cycles. The gas turbine unit supplied exhaust temperature of over 600°C which passes through a heat recovery steam generator (HRSG) to produce steam at three different pressure levels. According to the sets of data used, the total power and net energy efficiency of the plant is predicted at 287MW and 62.57% respectively. The analysis revealed that the increase in ambient temperature leads to an increase in the demand of the steam by steam turbine plant at different levels of air humidifier water cooling flow rate, sharply decreasing the amount of steam quantity, saving water in the plant, reducing losses, and thus improving the CCGT power and efficiency. The increase in compression ratio lead to a decrease in the amount of steam at the steam turbine plant. The sharp reduction of steam quantity indicated that the ambient temperature conditions were 40.2% higher and 41.86%lower respectively. The CCGT efficiency exhibited gradual increase with an increasing steam turbine inlet temperature which reached maximum values at high compression ratio. It resulted that, the increase of inlet air humidifier water cooling mass flow rate led to a sharp decrease in the quantity of fuel required in the combustion chamber at specified compression ratio. The results reveal that the combustor fuel flow rate is higher for the lower range of compression ratio. The extensive modeling suggests a sharp increase in combustor fuel flow rate by 21.79% and 21.60% for higher and lower compressor inlet temperature respectively. The increase in the amount of steam verified at specified compression ratio resulted in gradually increasing the power of steam turbine unit thus effecting the gross power of the CCGT power plant.

## 1. INTRODUCTION

Various technologies for power supply exist and are stated in literature, such as nuclear energy, wind energy, water energy, steam turbines (ST) and gas turbines (GT). In this regard, steam is used as the main source of power for various processes such as heating, chemical reactions, power generation etc. As the cost of the fuels continuously rise, development of new methods for maximizing the electricity supply and power generation, increasing efficiency, minimizing the fuel consumption, and cost optimization of power plants are critical for researchers and development engineers. Despite the advancements made in alternative energy technologies, to date, majority of the world's power is generated in thermal power plants by utilizing fired boilers of steam turbines.

Recently, the combined cycle gas turbines (CCGT) have been considered more efficient for power generation (Harmon,1988,Sorensen,1983, Bachmann et al.,1999). CCGT normally delivers the highest efficiency for power generation (Bachmann et al.,1999, Nag,2008, Shapiro et al.,2008, Boles et al.,2008). The thermal efficiency of a CCGT can exceed 60% depending on various conditions and the design of the cycle's components (Bachmann et al.,1999).The combined cycle gas turbine (CCGT) consists of two main components, namely, the Gas turbine (Willems et al.,2003) and Steam turbine (Silvestri et al.,1989). A steam power plant mainly consist of a de-aerating feed water tank, a high-pressure steam turbine, a high-pressure steam boiler, a low-pressure steam turbine, and a water-cool condenser connected in series in a closed circuit. The function of the feed water tank is to remove air from the water before it flows into the steam boiler. The condensate from the condenser is pumped by a steam jet pump to enter the feed water tank. The feed water heater (FWH) is a component of the power plant which serves as a water preheater before it flows into the boiler to generate steam (Nag,2008, Shapiro et al.,2008, Boles et al.,2008). This preheating process is expected to reduce the irreversibilities that occur in steam generation and thus, improve the thermodynamic efficiency of the system. The Gas Turbine Power Plant consists of three components including the compressor, combustion chamber (CC), and turbine. Fresh atmospheric air is filtered and drawn continuously into the circuit, then the energy is added by the combustion of the fuel in the chamber unit. The products of combustion are expanded through the turbine (Nag,2008, Ziurys et al.,1959) and consequently produce electrical work with the rest of the exhaust gases discharging to a heat recovery steam generator (HRSG) to produce steam. Exhaust gas from the gas turbine unit is a main energy agent for transferring thermal power for other utilizing equipment on the plant and is used in a heat recovery steam generator (HRSG) to produce steam in the steam plant. The produced steam is then used in steam turbines for electricity production. The gas turbine and the steam turbine produce approximately 65% and 35% respectively of the total produced power in the combined cycle plant (Nag,2008). An advantage of the gas turbine technology (Mast et al.,2002) is the different types of gas turbines which are available with different operating and design parameters such as back pressure and the exhaust gas temperature. The exhaust temperature, ranging from 700 to 950K (Nag,2008), is a very important parameter in utilizing the HRSG design. The exhaust temperature has great influence on the design of the HRSG and thus effects the thermal efficiency of the steam plant. In combined cycle power plants, steam pressure and temperature of the HRSG are increased to produce more power (Bachmann et al.,1999, Nag, 2008). Triple-pressure HRSGs are designed and developed to increase energy recovery, thus increasing the efficiency of the steam plant. Probable emissions of $NO_x$, $SO_x$ and CO could be treated in the process of heat recovery.

Reddy et. al. (2008) analyzed a dual pressure combined cycle and investigated the effect of steam injection on its performance. Manfrida et. al. (1998) studied a semi-closed gas turbine combined power plant in terms of energy balances and efficiency, for the purpose of identifying the critical plant devices, considering several operating conditions. Aljundi (2009) used energy analysis to identify and quantify the components having largest energy losses in a steam power plant. Besides, the effects of ambient air temperature variation on the power plant are also presented. Murugesan et. al. (2009) applied energy analysis on lignite fired power plant and compared the energy losses of the individual components of the plant. Kahveci et. al. (2006) performed energy analysis of a combined cycle power plant located in Lüleburgaz. Hepbaşlı et. al. (2007) performed energy analysis of a power plant located in Eskişehir. Some modifications were also suggested, and exergy efficiency of the modified power plant was increased. Kishore et. al. (2018) performed energy analysis of a combined cycle to develop the design of a combined cycle with higher efficiency and output work.

In this work, similar data sets of a combined cycle gas turbine (CCGT) power plant, located in Khartoum North Station, Sudan, investigate the performance and efficiency of the plant. The CCGT produces 287 of megawatts of power for one unit of gas and steam turbines. The energy efficiency, power consumptions and other design parameters of each component and overall plant, are studied by analyzing of the energy equations. Furthermore, a parametric study and analysis is carried out by changing some critical parameters of the system. The real data sets are simulated EES software. Heat losses in the pipes and devices are assumed to be zero and all components are considered to operate at steady state. The obtained data from parametric analysis is to be considered for further research and for the thermoeconomic optimization analysis of the power plant.

## 2. PLANT DESCRIPTION

A brief schematic diagram of the power plant is given in Fig.1. Siemens SGT5-2000E model gas turbine-generator (Mast et al.,2002) with 187 MW electrical power (in ISO conditions), heat recovery steam generator and steam turbine-generator with100 MW electrical power is used. Table 1 shows the technical specifications of this power plant. Total power consumption and net electric efficiency of the plant is predicted at287 MW and 62.57% respectively. Ambient air with 15ºC enters the cycle through a filter and is then compressed with an average pressure ratio of 15 in the compressor. An evaporative cooling system (EV) (Chavan et al.,2014, Sanjay et al.,2014, Prasad et al.,2012) is installed in the path of the compressor's inlet air supplying pressurized water by atomizing it into fine droplets, which gradually decreases the ambient temperatures and increases the diffusivity and mass flow rate to the compressor.

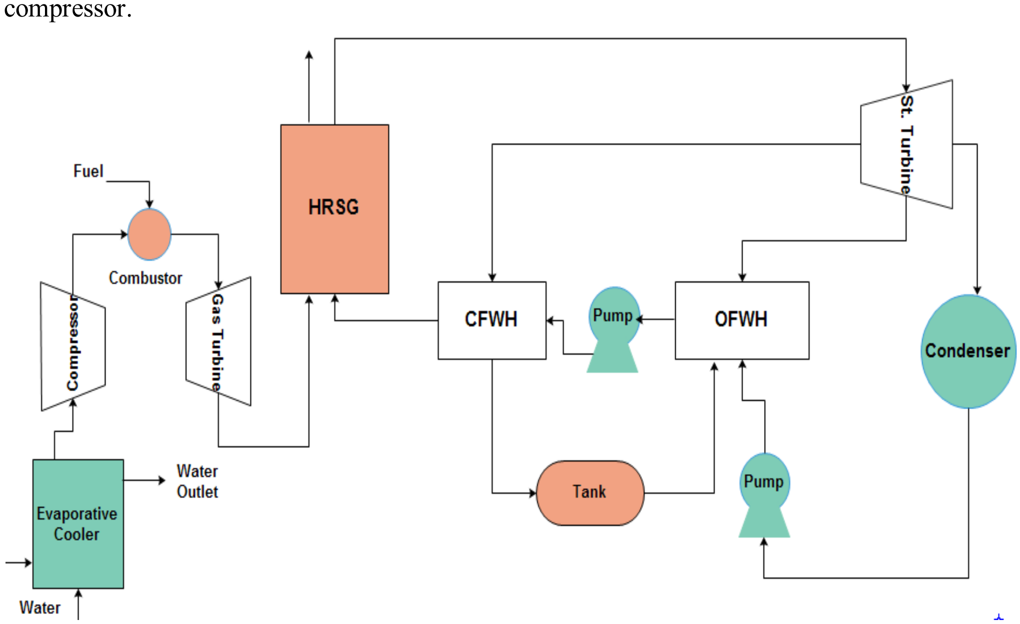


**Figure 1**: Brief diagram of a combined cycle power plant.

Compressed air enters to the combustion chamber where combustion takes place and diesel is used as fuel source with a 48100 kJ/kg LHV. The combustion gasses leave the gas turbine at nearly 600°C and enter the duct burner. After the duct burner, combustion gasses pass through heat exchangers inside of the heat recovery steam generator (HRSG) to produce steam at three different pressure levels. Steam at three different pressure levels from the HRSG are led to high-intermediate- and low-pressure steam turbines that are connected to each other via a common shaft. The Rankine cycle represent the heat cycle of a regenerative feed water heating, commonly used for increasing the thermal efficiency of steam plant. The feed water heater (FWH) is a component of the power plant which serve as a water preheater before it flows into the boiler to generate steam. This preheating process is expected to reduce the irreversibilities that occur in steam generation and thus, improve the thermodynamic efficiency of the system. Produced steam rotates the steam turbines while passing through the narrow openings between the casing and the blades. This rotational mechanical energy is converted to electrical energy on the steam turbine generator, connected on the same shaft. The expanded steam, leaving the low-pressure steam turbine, is condensed in a condenser and then fed back to integral deaerator for utilizing the waste heat capacity of flue gases. A natural draft cooling tower (Tichenor et al., 1970) ejects heat to the environment using performing-air and recirculated water flow as cooling agents.

Table (1): Main operating conditions of the combined cycle gas turbine (CCGT) power plant.

| Operating Conditions | Value | Units |
|---|---|---|
| Gas turbine output | 187 | MW |
| Steam turbine output | 100 | MW |
| Air mass flow rate | 600 | kg/s |
| Average pressure ratio | 15 | - |
| Evaporative cooling water flow | 10 | kg/s |
| Fuel mass flow rate | 13 | kg/s |
| Inlet gas turbine pressure | 9 | bar |
| Total steam flow rate | 70 | kg/s |
| Efficiency of gas turbine plant | 34.50 | % |
| Steam turbine inlet temperature | 500 | °C |
| Exhaust gas temperature | 600 | °C |
| Efficiency of steam turbine plant | 42.53 | % |
| Turbine inlet temperature | 1200 | °C |
| Gas turbine compressor, combustor and turbine efficiency | 90 | % |
| Feed water heater pressure | 3 | bar |
| Efficiency of the combined cycle plant | 62.57 | % |
| Cooling tower circulated water | 1180 | kg/s |

## 3. THERMODYNAMIC ANALYSIS

The intake pressure at the compressor inlet was modeled with the following equation (Nag,2008):

$$P_1 = P_{ATM} - \Delta P_{intake} \tag{1}$$

Where the intake pressure drop ($\Delta P_{intake}$ ) was taken to be 0.005 bar, and the intake temperature was modeled as the ambient temperature. The process on the temperature-entropy diagram is represented in Fig.1.The compressor compression ratio ( $r_P$ ) can be defined as (Nag,2008):

$$r_P = \frac{P_{jc}}{P_{ic}} \tag{2}$$

where $P_i$ and $P_j$ are compressor inlet and outlet air pressure, respectively. Accordingly, the isentropic outlet temperature leaving the compressor is modeled by the equation (Nag,2008, Shapiro et al.,2008, Boles et al.,2008):

$$\frac{T_{ic}}{T_{jcs}} = \left(\frac{P_{jc}}{P_{ic}}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \tag{3}$$

The specific heat ratio for air $\gamma_a$ was taken as 1.4 and was predicted at $\gamma_g = 1.3$ for the gas.The isentropic efficiency of the compressor and turbine was taken to be in the range of 85% to 90%. The isentropic compressor efficiency is expressed by the equation (Nag,2008, Shapiro et al.,2008, Boles et al.,2008):

$$\eta_c = \frac{T_{jcs} - T_{ic}}{T_{jc} - T_{ic}} \tag{4}$$

Where, $T_{iC}$ and $T_{jC}$ are the compressor inlet and outlet air temperatures respectively and $T_{jCs}$ is the compressor isentropic outlet temperature. The specific work required to run the compressor work ($W_C$) is modeled with the following equation (Nag,2008):

$$\dot{W}_c = \dot{m}_a C_{P_a}\left(T_{jc} - T_{ic}\right) = \dot{m}_a C_{P_a} T_{ic}\left[\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right] \tag{5}$$

From the energy balance in the combustion chamber (Nag,2008, Sheng et al., 2013):

$$\dot{m}_a C_{P_a} T_x + \dot{m}_f LHV + \dot{m}_f C_{P_f} T_f = \left(\dot{m}_a + \dot{m}_f\right) C_{Pg} TIT \tag{6}$$

Where $\dot{m}_f$ is the fuel mass flow rate in (kg/s), $\dot{m}_a$ is the air mass flow rate (kg/s), LHV is the caloric value (The fuel used has a value of 48100 kJ/kg) , TIT is the turbine inlet temperature, $C_{P_f}$ is the specific heat of fuel, and $T_f$ is the temperature of the fuel. Combustor efficiency was set at 95%, and a pressure drop of $\Delta P_{C,C} = 0.4785\ bar$ in the combustor. Accordingly, the efficiency of the combustor was modeled as (Nag,2008, Sheng et al., 2013):

$$\eta_{C,C} = \frac{\dot{m}_g C_{Pg} TIT - \dot{m}_a C_{P_a} T_x}{\dot{m}_f LHV_g} \tag{7}$$

The air fuel ratio at the combustor was modeled according to the following equation:

$$AFR = \frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \tag{8}$$

Where the total mass flow rate is given by:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{9}$$

The discharge gas of the turbine was predicted according to the equation (Nag,2008, Shapiro et al.,2008, Boles et al.,2008):

$$\frac{T_{iT}}{T_{jTs}} = \left(\frac{P_{iT}}{P_{jT}}\right)^{\frac{\gamma_g - 1}{\gamma_g}} \tag{10}$$

where $P_{iT}$ and $P_{jT}$ are turbine inlet and outlet gas pressure, respectively. The actual outlet temperature leaving the turbine at the isentropic conditions was modeled according to (Nag,2008, Shapiro et al.,2008, Boles et al.,2008):

$$\eta_t = \frac{T_{iT} - T_{jT}}{T_{iT} - T_{jTs}} \tag{11}$$

where $T_{iT}, T_{jT}$ and $T_{jT\,s}$ are turbine inlet, outlet and isentropic outlet gas temperature, respectively. The shaft work produced from the turbine is determined by the equation (Nag,2008):

$$\dot{W}_{GT} = \dot{m}_g C_{Pg}(T_{jT} - TIT) = \dot{m}_g C_{Pg} TIT \eta_t \left[1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}}\right] \tag{12}$$

The network of the gas turbine unit was expressed by the equation (Nag,2008):

$$\dot{W}_{GT,Net} = \dot{W}_{GT} - \dot{W}_c = \dot{m}_g C_{Pg} TIT \eta_t \left[ 1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right] - \dot{m}_a C_{P_a} T_1 \left[ \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right] \quad (13)$$

The output power from the gas turbine is expressed with the equation (Nag,2008):

$$P_{GT} = \left[ \dot{W}_{GT} - \dot{W}_c \right] * \eta_{Mech} \eta_{Gen} \quad (14)$$

The mechanical ($\eta_{Mech}$) and generator ($\eta_{Gen}$) efficiencies were taken to be 92% and 95% respectively.
The heat supplied was expressed with the equation (Nag,2008):

$$\dot{Q}_{add} = \dot{m}_g C_{Pg} TIT - \dot{m}_a C_{P_a} T_{jc} \quad (15)$$

The heat supplied to gas turbine (per kg. air) to the combustor was modeled according to the equation (Nag,2008):

$$\dot{Q}_{add} = \frac{\dot{m}_f LHV_g \eta_{C,C}}{\dot{m}_{air}} = \frac{LHV_g * \eta_{C,C}}{AFR} \quad (16)$$

The gas turbine efficiency was determined by the equation (Nag,2008):

$$\eta_{over,GT} = \frac{\dot{W}_{GT,Net}}{\dot{Q}_{add,GT}} \quad (17)$$

Accordingly, the heat rate (HR) of gas turbine plant which is defined as the consumed heat to generate unit energy of electricity was determined by the equation (Nag,2008):

$$HR = \frac{3600 * m_f * LHV_g}{\eta_{over,GT}} \quad (18)$$

The specific fuel consumption (SFC) is determined by the equation (Nag,2008):

$$SFC = \frac{3600 * \dot{m}_f}{\dot{W}_{RGT,Net}} \quad (19)$$

For the individual components of the regenerative steam power plant, mass and energy balances are expressed under a steady-state condition as follows:

$$\sum_i \dot{m}_{steam} = \sum_e \dot{m}_{steam} \quad (20)$$

$$\dot{Q}_{in} + \sum_i \dot{m}_{steam} h = \dot{W}_{steam,output} + \sum_e \dot{m}_{steam} h \quad (21)$$

The mass flow of steam, steam turbine work and heat input are indicated as $\dot{m}_{steam}$ ,h, $\dot{W}_{steam\,,output}$ , respectively.The output power is determined by the work produced by the steam turbines (Mrakovčić et al.,2017):

$$\dot{W}_{steam,output} = \eta_{over,ST} \left[ \sum_j \dot{m}_{steam} h_i - \sum_i \dot{m}_{steam} h_e + \left( \dot{m}_{i,steam} - \dot{m}_{1,steam} - \ldots - \dot{m}_{n,steam} \right) \left( h_n - h_e \right) \right] \quad (22)$$

Eq. (22) is dependent on the number of steam turbines installed in the steam power plant. The subscripts 1, i, e …n refer to the number of steam extraction in the steam turbines. The heat input generted on the heat recovery steam generator (HRSG) combining the mass and energy balances of the gas turbine (equation (13)) and steam cycles (equation (22)) to develop the total net power of the plant as follows:

$$\dot{W}_{Net,GT,ST} = \dot{W}_{GT,Net} + \eta_{over,ST}\left[\sum_j \dot{m}_{steam} h_i - \sum_i \dot{m}_{steam} h_e + \left(\dot{m}_{i,steam} - \dot{m}_{1,steam} - \ldots - \dot{m}_{n,steam}\right)\left(h_n - h_e\right)\right] \quad (23)$$

The net electrical output of the steam turbine considering the pump efficiency is calculated as follows:

$$\dot{W}_{Net,steam} = \sum_i \dot{W}_{steam,output} - \sum_j \frac{\dot{m}_{steam}\left(h_i - h_e\right)}{\eta_{Pump}} = \sum_i \dot{W}_{steam,output} - \sum_i \dot{W}_{Pump} \quad (24)$$

## 3. RESULTS AND DISCUSSIONS

A computational model is developed by using Engineering Equation Solver for evaluating energy and efficiency analysis of combine cycle plant. The main operating data for evaluations are mentioned in Table (1) except the parameters, whose effects are discussed in particular plots, and has been varied to predict the performances and efficiencies of the plant. The simulation results verified main design parameters and find its influences on other operating conditions. The results explained different ways to assess the design, reliability, operation and maintenance procedures of the combined cycle gas turbine (CCGT) power plant.

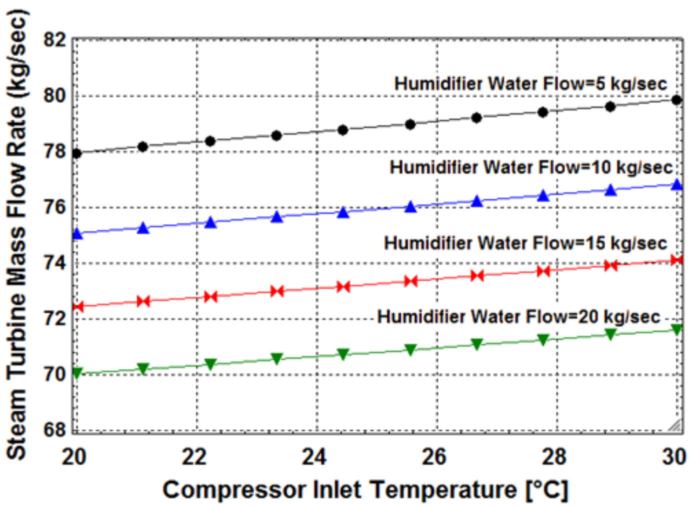


**Figure 2**: ST mass flow rate versus compressor temperature for different cooling air humidifier water flow rate.

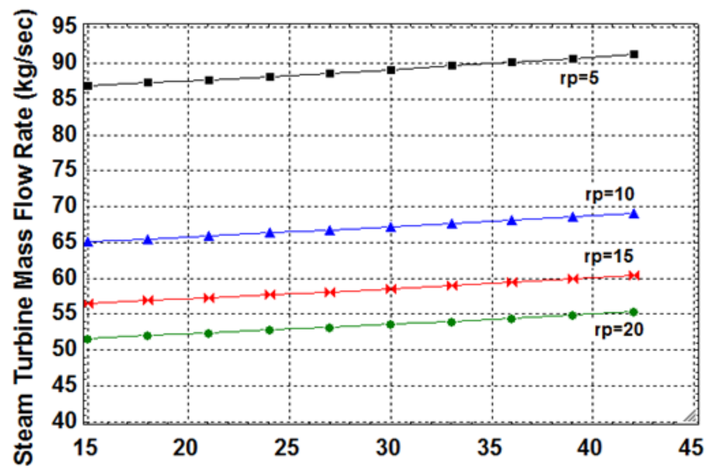


**Figure 3**: Compressor inlet temperature versus ST mass flow rate at different compression ratios.

The combined cycle gas turbine (CCGT) power plant was connected to the country load in April 2020, using diesel as a fuel. The gas turbine loads are performed at different operating conditions. It is observed that air plays a major role on supplying power of GT and thereafter transfer its thermal energy to ST plant to provide the final power load of the CCGT as stated on Fig.1. An evaporative water-cooling unit (EV) was installed to decrease the ambient temperature in a purpose to enhance the thermal efficiency (Sanjay et al., 2014) of the GT unit. The simulation results verified the quantity of the cooling agent used at different levels. As observed in Fig.2, the air humidifier cooling water of the evaporative cooling unit (EV) is verified at 5,10,15,20 kg/s for various ambient conditions ranged between 20-30$^{o}$Cand the amount of steam consumed on ST plant. It resulted in the increase of ambient temperature leading to an increase in the demand of the steam by ST plant at different levels of air humidifier water cooling flow rates. This is due to the fact that, that as the temperature increases the air will expand gradually results in increasing the amount of gas at HRSG and thus demanding large amount of steam by ST unit. This resulted in a drop of steam flow rate by 10% and 10.3% for higher and lower ambient temperature conditions respectively. The results delivered that the increase in the amount of cooling agent will sharply decrease the amount of steam quantity, which saves water, reducing its losses, and thus improving the CCGT power and efficiency. Fig.3 investigated the influence of the compression ratio at 5,10,15,20 on the amount of steam used at different compressor inlet temperatures verified between 15-45$^{o}$C.The increase of the compression ratio led to a decrease in the amount of steam at ST plant (see also Fig.5), this is due to decrease of gas amount at HRSG.As verified in Fig.3, the sharp reduction of steam quantity indicated 40.2 % and 41.86% for higher and lower ambient temperature condition respectively.

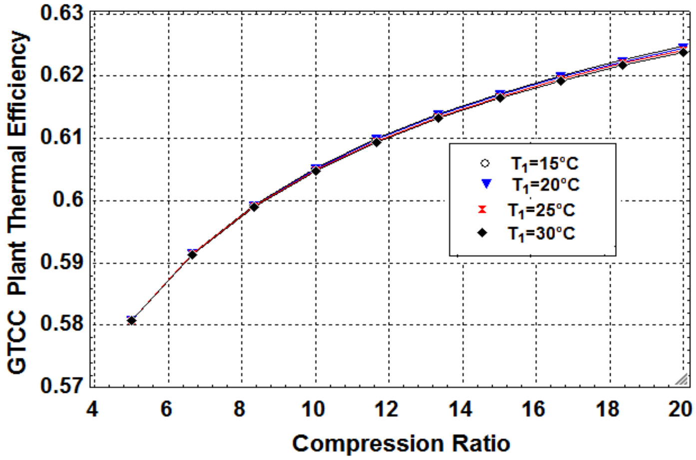


**Figure 4**: Variation of compression ratio and thermal efficiency of GTCC at different ambient temperatures.

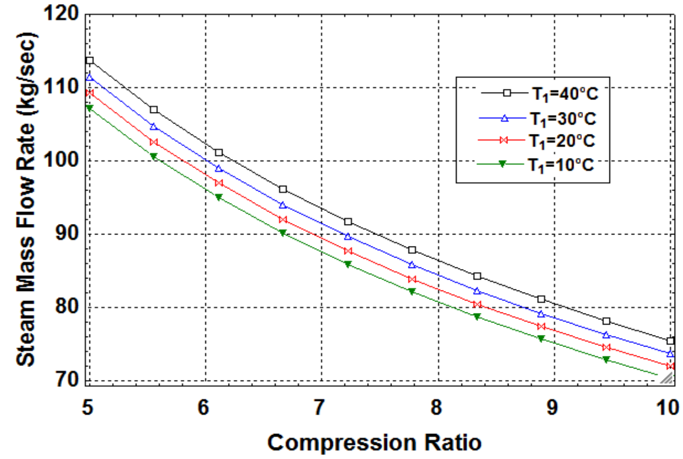


**Figure 5**: Variation of compression ratio and ST mass flow rate of GTCC at different ambient temperatures.

As observed in Fig.4, for combined cycle gas turbine (CCGT) power plant, the increase of the compression ratio led to increase in the thermal efficiency of the plant at different ambient temperatures verified at 15,20,25,30 °C. The combined cycle provided a higher efficiency reaching 62.5% at a pressure ratio of 20. A maximum efficiency has been verified at lower ambient temperature 15 °C. Consequently, the efficiency will reach a maximum value at optimum compression ratio through which the maximum real work occur, thereafter the work will decrease gradually and the increase of the compression ratio will decline the thermal efficiency of the plant.Fig.6 represents the variation of steam turbine inlet temperature with thermal efficiency of CCGT at different compression ratios.

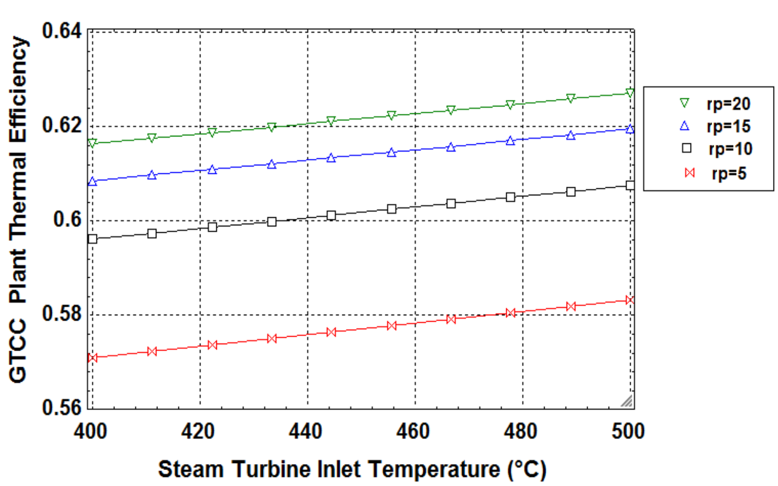


**Figure 6**: Influence of ST inlet temperature on thermal efficiency of GTCC at different compression ratios.

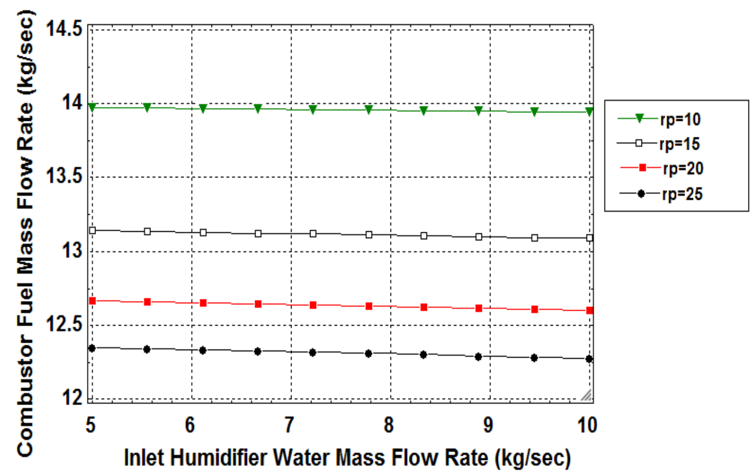


**Figure 7**: Influence of cooling air humidifier water flow on the combustor fuel mass flow rate at different compression ratios.

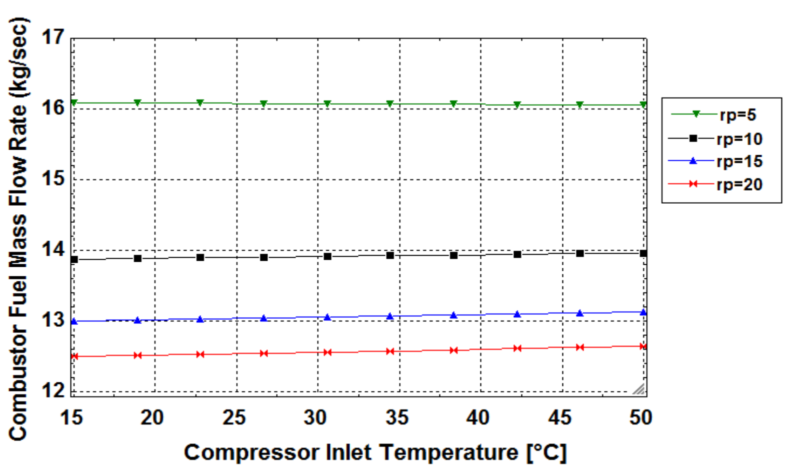


**Figure 8**: Variation of compressor inlet temperature with combustor fuel mass flow rate at different compression ratios.

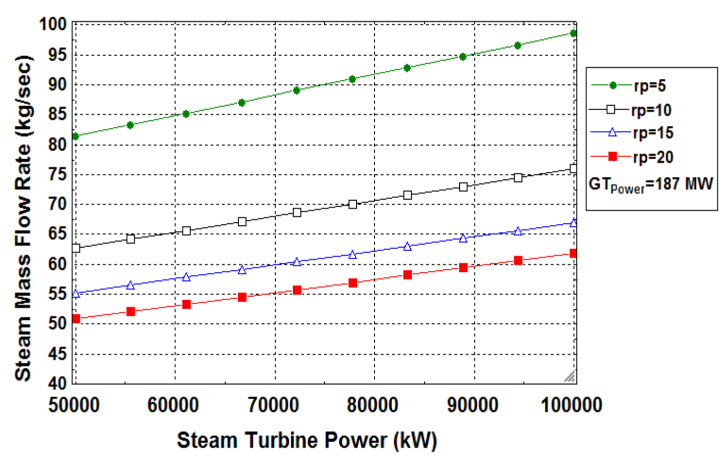


**Figure 9**: Variation of steam turbine power with ST mass flow rate at different compression ratios

The CCGT efficiency exhibited gradual increase on increasing the steam turbine inlet temperature. This is referring to the fact that as ST temperature increases, the net power of ST cycle increases, thus enhancing the efficiency of ST unit and the overall efficiency. It has been found that the thermal efficiency of CCGT is higher for high compression ratio. For higher and lower steam turbine inlet temperature ranged between 400-500$^{o}$C and compression ratio verified at 5,10,15,20, the percentage reduction on CCGT efficiency reached values of 6.35% and 6.56% respectively. The influence of humidifier water cooling mass flow rate on the combustor fuel consumption at different compression ratio was investigated in Fig.7. The increase of the inlet humidifier water cooling mass flow rate led to a sharp decrease in the quantity of fuel required in the combustion chamber at specified compression ratio. As observed in Fig.7, the increase of the compression ratio resulted in decreasing the combustor fuel mass flow rate due to the change in the mass of air and the compressor work. The result shows the increase in the amount of water at evaporative cooling unit (EV), which will cause a reduction in the compressor inlet temperature, support gathering of air molecules, causing a dramatic increase of air density and reduction of compressor work leading to a higher output power and efficiency of GT unit. As results of this, the demand of fuel at combustor will decrease. For higher and lower humidifier water cooling mass flow rate varied between 5-10kg/s and compression ratios verified at 10,15,20,25, the percentage reduction on fuel required in the combustion chamber reached values of 10.11% and 10.85% respectively. Consequently, Fig.8 represents the variations between compressor inlet temperature with combustor fuel mass flow rate for different compression ratios. The amount of fuel required by the combustion chamber at specified compression ratio increases on increasing the compression inlet temperature. Results show that the combustor fuel rate is higher for the lower range of compression ratio. It results in a sharp increase of combustor fuel flow rate by 21.79% and 21.60% for higher and lower compressor inlet temperature respectively. In Fig.9, the effect of steam turbine power varied between 50-100 MW on the steam turbine mass flow rate at specified compression ratio verified at 5,10,15,20. Accordingly, the increase of steam mass flow rate resulted in increasing the power of steam turbine unit, thus demanding a higher amount of the working fluid. The steam mass flow rate is higher for low compression ratio. For higher and lower steam turbine power varied between 50-100 MW and compression ratio verified at 5,10,15,20, the percentage reduction on steam mass flow rate effects the produced power of the unit reached values of 37.37 % and 39.75% respectively.

## 4. CONCLUSIONS

This work has discussed the performance and efficiency of the combined cycle gas turbine (CCGT) power plant with plant data sets similar to that of Khartoum North Station, Sudan. The work has investigated various results and different design and operating parameters such as compression ratio, steam flow rate, air cooling agent, steam turbine temperature, ambient temperature, and amount of fuel consumption on the efficiency and performance of CCGT. The thermodynamic performance in terms of first and second law efficiency improved significantly by combining gas turbine power cycle with steam turbine vapor power system. An improvement of thermal efficiency of 62.5% is found as compared to single cycle of gas and vapor power system. The gas turbine system in a (CCGT) plant was considered as a main unit to supply the fuel to the steam turbine plant via a heat recovery steam generator (HRSG) to produce steam at three different pressure levels. The work revealed that the compression ratio, inlet air temperature, steam flow rate, humidifier cooling amount, fuel flow rate at combustor had a major effect on the thermal efficiency and power output of the combined cycle gas turbine (CCGT) power plants.

## NOMENCLEATURE

***Symbols&***

| | | | | | |
|---|---|---|---|---|---|
| T | Temperature | (K) | $TIT$ | Turbine Inlet Temperature | ($^{o}$C) |
| S | Entropy | (kJ/kg.K) | $\Delta P_{C,C}$ | Combustor Pressure Drop | (bar) |
| P | Pressure | (kPa) | $\eta_{C,C}$ | Combustor Efficiency | - |
| $r_P$ | Compression Ratio | - | $h$ | Enthalpy | (kJ/kg) |
| $\gamma$ | Specific Heat Ratio | - | $\dot{W}_{GT}$ | Turbine Shaft Work | (MW) |
| $\eta_C$ | Isentropic Compressor Efficiency | - | $\eta_T$ | Turbine Efficiency | - |
| $T_S$ | Compressor Isentropic Temperature | ($^{o}$C) | $P_{GT}$ | Gas Turbine Power | (MW) |
| $\dot{W}_C$ | Specific Compressor Work | (MW) | $\dot{Q}_{add}$ | Heat Supplied | (kW) |
| $\dot{m}_a$ | Air Mass | (kg.air) | $C_{P_a}$ | Heat Capacity of Air | (kJ/kg.K) |
| $\dot{m}_f$ | Fuel Mass | (kg.fuel) | $C_{P_f}$ | Heat Capacity of Fuel | (kJ/kg.K) |

| | | | | | |
|---|---|---|---|---|---|
| $\dot{m}_g$ | Gas Mass | (kg.gas) | $C_{Pg}$ | Heat Capacity of Flue Gas | (kJ/kg.K) |

***Subscripts***

| | |
|---|---|
| CCGT | Combined Cycle Gas Turbine |
| HRSG | Heat Recovery Steam Generator |
| rp | Pressure Ratio |
| FWH | Feed Water Heater |
| GT | Gas Turbine |
| ST | Steam Turbine |
| EV | Evaporative Cooling |
| Mech | Mechanical |
| Gen | Generator |
| AFR | Air Fuel Ratio |
| EES | Engineering Equation Solver |
| CC | Combustion Chamber |
| ATM | Atmospheric |
| LHV | Lower Heating Value |
| TIT | Turbine Inlet Temperature |
| HR | Heat Rate |
| SFC | Specific Fuel Consumption |

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## ACKNOWLEDGMENTS

The authors greatly acknowledge the technical support of Chemical Engineering Department and the Energy Research Centre of the University of Khartoum, Faculty of Engineering for providing laboratory and facilitating the field of works. The Financial, technical and academic supervisions of the World Academy of Science (TWAS) and the Islamic Development Bank (IsDB),grant number “506798/2019” under the scheme of the Postdoctoral Fellowship program, is greatly appreciated.
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