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PERFORMANCE ANALYSIS OF BENI-SUEF COMBINED CYCLE POWER PLANT (4*1200) MW.

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1. Abstract

Beni-suef combined cycle power plant (CCPP) is one of the largest and most advanced power plants in Egypt and even in the world. It is equipped with four units (modules). Each unit consists of two gas turbines (GT), two triple pressure heat recovery steam generators (HRSG), one steam turbine (ST) and three generators. The power plant has a rated net electrical output of approximately 4800 MW (1200 MW/module) with efficiency exceeds 60% at its site design conditions.

The performance of any combined cycle power plant (CCPP) is influenced by the ambient conditions of the place where it is installed like ambient temperature, atmospheric pressure, air relative humidity and cooling water inlet temperature. This study aims to investigate the effect of ambient temperature and air relative humidity on the performance of Beni-suef combined cycle power plant (CCPP) by performing an energy and exergy analysis of the combined cycle power plant components.

Energy and exergy analysis were performed at 16.1, 28.1, 37.6 ^{o}C ambient temperatures for the components of one module of Beni-suef combined cycle power plant using the in-operation data. Also, the effect of air relative humidity ratio was studied on the gas turbine performance at 16.1, 28.1, 37.6 ^{o}C ambient temperatures and different relative humidity ratios from 10 to 90%.

2. Keywords: Combined Cycle Power plant, Exergy analysis, Exergy efficiency, Exergy destruction, heat recovery steam generators, once through steam generators (BENSON), modern Combined Cycle Power plant and heat rate
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3. Introduction

Power stations are distributed over many of the Egyptian cities. Power plants generate power which is transmitted by means of overhead lines at high voltage levels up to 500 kV. The electrical power generated from the Egyptian power plants are all directed to feed a single huge grid which is commonly known as the national grid. [1]

Fossil fuel-fired power plants use either steam or combustion turbines to provide the mechanical power to electrical generators. Pressurized high temperature steam or gas expands through various stages of a turbine, transferring energy to the rotating turbine blades. The turbine is mechanically coupled to a generator, which produces electricity.[2]

The Combined Cycle power plant in is a combination of a fuel-fired turbine (Gas Turbine) with a Heat Recovery Steam Generator (HRSG) and a steam powered turbine. They combine the Rankine Cycle (steam turbine) and Brayton Cycle (gas turbine) thermodynamic cycles by using heat recovery boilers to capture the energy in the gas turbine exhaust gases for steam production to supply asteam turbine. [2]

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The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually. [3]

Because of the higher average temperature at which heat is supplied, gas-turbine cycles have a greater potential for higher thermal efficiencies. However, the gas-turbine cycles have one inherent disadvantage: The gas leaves the gas turbine at very high temperatures (usually above 500°C), which erases any potential gains in the thermal efficiency. [3]

Some Combined Cycle power plants (CCPP) use duct firing or supplementary firing to increase the power output of steam turbine regardless of gas turbine power at higher ambient temperatures. [2]

The usual method of analyzing the power plant performance is by the first law of thermodynamics. In fact, traditional first-law cycle analysis based upon component performance characteristics coupled with energy balances invariably leads to a correct final answer. However, such analysis cannot locate and quantify the sources of loss that lead to that result. This is because the first law embodies no distinction between work and heat, no provision for quantifying the quality of energy. When analyzing complex thermal systems, however, such understanding has to be supplemented by more rigorous quantitative methods. Exergy analysis provides these tools. Second law analysis cannot replace first law analysis; it is rather a supplement. The importance of this analysis results from the fact that it combines the first and second laws of thermodynamics to show the direct relationship between the exergy destruction and the use of energy [4, 5 and 6].

Aliyu, M., AlQudaihi, A. B., Said, S. A., & Habib, M. A. [7] studied the thermodynamics (energy and exergy) analysis of a combined cycle power plant using the plant design data. The plant is consisting of five units and each unit comprises of three gas turbine generators (GTG), three heat recovery steam generators (HRSG), and one steam turbine generator (STG). Exergy analysis showed that the major source of irreversibility (exergy destruction) in the steam turbine cycle (STC) of the combined cycle power plant (CCPP) is the stack followed by the HRSG, turbine, and condenser.

Javadi, M. A., Hoseinzadeh, S., Khalaji, M., & Ghasemiasl, R. [8] studied Mess (city in Iran) combined cycle power plant with a nominal capacity of 500 MW, including two gas turbine units and one steam turbine unit, is considered by a mathematical model. This study is carried out to optimize three objective functions of exergy efficiency, CO2 emission and produced power costs.

Ibrahim, T. K., Mohammed, M. K., Awad, O. I., Abdalla, A. N., Basrawi, F., Mohammed, M. N., ... & Mamat, R. [9] performed a thermodynamics analysis on each system components of a combined cycle power plant (CCPP) independently and determine the exergy destruction of the plant. It was found that in the combustion chamber (CC), there is the highest exergy destruction occurred.

Sharma, M., & Singh, O. [10] performed an exergy analysis of a dual pressure (DP) heat recovery steam generator (HRSG) having steam generation at high pressure (HP) and low pressure (LP) in the gas-steam combined cycle power plant for varying dead states. The in-operation plant data for this study are taken from a combined cycle power plant at Auraiya (U.P.), India. The study proved that with increase of dead state temperature, the exergy efficiency of all components decreases.

Boyaghchi, F. A., & Molaie, H. [11] analyzed and optimized a real Combined Cycle power plant (CCPP) based on advanced exergy method. Turbine inlet temperature (TIT), compression ratio (rc) and Duct Buring (DB) are selected as the main variables, while the total exergy destruction rate and CO2 emission are selected as the objective functions for the optimization. They concluded that the growth of compression ratio (rc) has a negative impact on exergy destruction rate of the combined cycle power plant (CCPP).

Kaviri, A. G., Jaafar, M. N. M., Lazim, T. M., & Barzegaravval, H. [12] studied Neka combined cycle power plant (CCPP) which has a dual pressure heat recovery steam generator (HRSG) and duct firing. The applied exergy analysis showed that, for the given heat recovery steam generator (HRSG), the exergy destruction for high pressure evaporator section (HP-EVP) was more than other components exergy destruction. Their results also showed that, increasing heat recovery steam generator (HRSG) inlet gas temperature more than 650 ^{o}C has less improvement on thermal efficiency and exergy efficiency of the bottoming cycle.

Law, B. [13] simulated the performance of a combined cycle cogeneration configuration based on energy and exergy analyses point of view. The operating conditions investigated included gas turbine pressure ratio, gas turbine inlet temperature, process heat load and ambient conditions.

Ameri, M., & Ahmadi, P. [14] studied the thermal and exergy analyses of a 400 MW combined cycle power plant. This power plant has two gas turbines, two compressors, two heat recovery steam generators (HRSGs) with duct burning, two deaerators, one steam turbine and a cooling system. Their results showed that, the exergy losses have a minimum point at 19^{o}C which is the design temperature. Moreover, the results revealed that, the duct burning increases the exergy loss of heat recovery steam generator (HRSG).

Cihan, A., Hacıhafızogˇlu, O., & Kahveci, K. [15] performed energy and exergy analysis of the combined-cycle power plant of 1000 MW capacity and located near the city of Lu¨ leburgaz in TURKEY by calculating the exergy of the natural gas as well and exergy losses of each device in the power plant using the in operation data.

Hassan A.A. [1] performed energy and exergy analysis of El-Kurimat steam power plant in Egypt which is a 2*627 MW capacity. The analysis was used to determine the proportions of exergy losses among the plant components. The effect of the load on the thermal efficiency, exergy efficiency, and exergy destruction of different components of the plant were presented for constant and sliding pressure modes.

4. Aim of the present work

In this work, the effect of ambient temperature and air relative humidity on the performance of Beni-suef combined cycle power plant (CCPP) are investigated based on energy and exergy analysis. Energy and exergy analysis were studied at 16.1, 28.1, 37.6 ^{o}C ambient temperature for almost all components of one unit of Beni-suef combined cycle power plant (CCPP). The study was used to calculate the proportions of exergy losses among the plant components. Also, the effect of air relative humidity was studied on gas turbine (GT)at 16.1, 28.1, 37.6 ^{o}C ambient temperatures and different relative humidity ratios from 10 to 90% which helps in predicting the performance of the machine at different locations.

5. Beni-suef combined cycle power plant (CCPP) description

Beni-Suef combined cycle power plant (CCPP) is equipped with four units (Modules). Each unit consists of two gas turbines (GT), two triple pressure heat recovery steam generators (HRSG), one steam turbine (ST) and three generators. The power plant has a rated net electrical output of approximately 4800 MW at site design conditions. The power plant is mainly designed for base load operation. The gas turbines (GTs) are fired with natural gas. The fuel gas preheating system heats the natural gas up to 215°C during combined cycle operation. The exhaust gas from each gas turbine (GT) is fed via bypass stack with diverter damper to an unfired heat recovery steam generator (HRSG).

Each heat recovery steam generator (HRSG) has a vertical casing design with triple pressure and reheat system. HP evaporators are once through steam generators (BENSON). The IP and LP stages are drum types and are designed as natural circulation evaporators. Each steam turbine (ST) is fed with steam from two heat recovery steam generators (HRSGs) via common steam headers. The steam turbine (ST) is a double-casing condensing turbine with one combined HP/IP turbine and one double flow LP turbine. The turbine exhaust steam is condensed in a condenser which is aligned to the LP turbine. The condenser is arranged on one side of the LP turbine (single side-exhaust) and is cooled by recirculating water cooling from the cooling tower. Four cooling towers - one for each unit - are installed at the power plant as shown in the schematic diagram Fig.1.

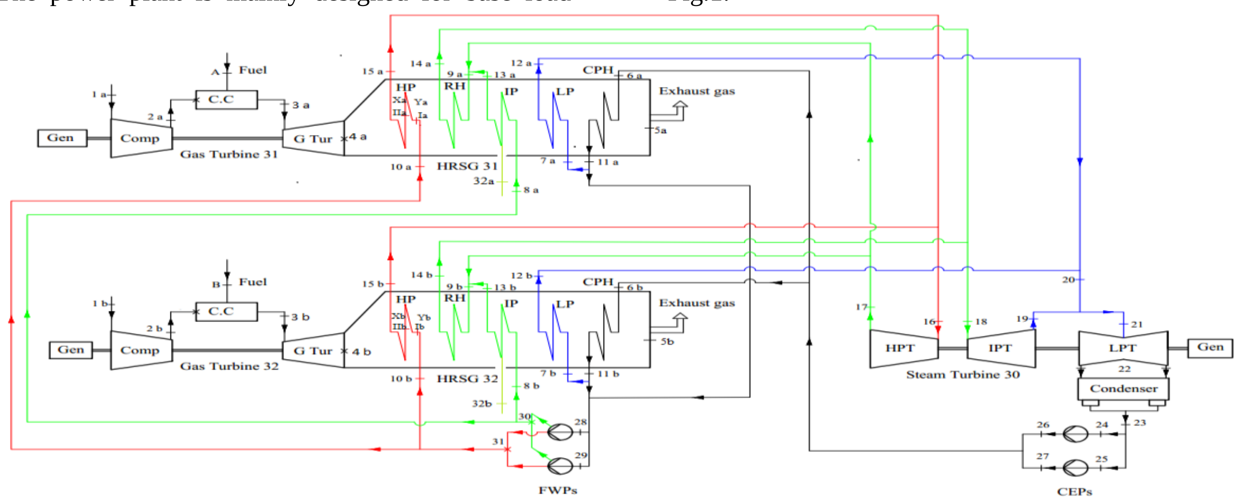

Fig.1 Beni-Suef Combined Cycle power plant (CCPP) one-unit schematic diagram.

5. Methodology

As we all know that the first law of thermodynamics deals with the amount of energy but the second law identifies the quality of energy. To fulfill the first law of thermodynamics, energy balance has to be done however for the second law of thermodynamics, exergy balance has to be performed.

The following assumptions are made in calculating the performance of Beni-suef combined cycle power plant (CCPP):

- All components associated with the cycle are steady flow devices.
- All processes in the cycle can be analyzed as steady flow processes.
- Kinetic and potential energy changes are neglected.
- The amount of cooling air extracted from gas turbine (GT) compressor outlet air to cool down the turbine blades is neglected.
- Gas turbine (GT) inlet temperature equals 1370 ^{o}C and exhaust gases are at atmospheric pressure.
- Pressure drop across gas turbine (GT) combustion chambers is neglected.

- Exergy destroyed in fuel gas preheaters values are small compared with other values.

Mass balance for any control volume can be expressed as

$\sum \dot{m}_i = \sum \dot{m}_e$ (1)

energy balances for any control volume at steady state with neglected potential and kinetic energy changes can be expressed as

$\dot{Q} - \dot{W} = \sum \dot{m}_e * h_e - \sum \dot{m}_i * h_i$ (2)

We can use the following formula for exergy balance for any control volume at steady state with neglected potential and kinetic energy changes

$X_{heat} - \dot{W} = \sum \dot{m}_e \psi_e - \sum \dot{m}_i \psi_i + X_{dest}$ (3)

Where the net exergy transfer by heat, X_{heat}at temperature T_H is given by

$\dot{X}_{heat} = \sum \left(1 - \frac{T_o}{T_H}\right) * \dot{Q}_{in}$ (4)

Where $\dot{Q}_{in}$ is the heat input to the cycle, which is heat input to the GT combustion chamber.

$\dot{Q}_{in} = \Sigma \dot{m}_e h_e - \Sigma \dot{m}_i h_i$ (5)

And the specific exergy is given by

$\psi = (h - h_o) - T_o (s - s_o)$ (6)

$\psi = C_p(T - T_o) - T_o(C_p * \ln\frac{T}{T_o} - R \ln\frac{P}{P_o})$ (7)

the exergy destroyed in processes which is equivalent to process irreversibility$\dot{I}$.

$\dot{X}_{dest} = \dot{I} = \dot{W}_{rev} - \dot{W}_u = T_O \dot{S}_{gen} = T_O (\dot{S}_{out} - \dot{S}_{in}) = T_O(\sum \dot{m}_e s_e + \frac{\dot{Q}_{out}}{T_L} - \sum \dot{m}_i s_i - \frac{\dot{Q}_{in}}{T_H})$ (8)

Where $\dot{S}_{gen}$ is the entropy generated during the process and T_H .and T_L is the temperatures of the system boundary where heat is transferred into and out of the system respectively.

For a cycle that involves heat transfer only with a source at T_H and a sink at T_L the exergy destruction can be expressed as

$\dot{X}_{dest} = T_o \left(\frac{\dot{Q}_{out}}{T_L} - \frac{\dot{Q}_{in}}{T_H}\right)$ (9)

The reversible work for a single stream steady flow device is expressed as

$\dot{W}_{rev} = \dot{m} (\psi_1 - \psi_2) + \Sigma(1 - T_o/T_H) \dot{Q}$ (10)

Which reduces for adiabatic device to

$\dot{W}_{rev} = \dot{m} (\psi_1 - \psi_2)$ (11)

Then the exergy change of a stream undergoes a process from state 1 to state 2 with neglected potential and kinetic energy changes is given by:

$\Delta\psi = (\psi_1 - \psi_2) = (h_2 - h_1) - T_o (S_2 - S_1)$
(12)

Then the total exergy rate associated with the flow stream becomes:

$\dot{X} = \dot{m}.\psi = \dot{m}((h - h_o) - T_o (S - S_o))$
(13)

The second law efficiency, also referred to as exergy efficiency, is a measure of the performance of a device relative to the performance under reversible conditions for the same end states and is given by:

$\eta_{II} = \frac{Exergy\ recovered}{Exergy\ supplied} = \frac{Exergy\ destroyed}{Exergy\ supplied}$
(14)

The exergy destruction rate and the exergy efficiency for the whole system and for each component in the power plant can be expressed as:

- Compressor:

Exergy destruction rate:

$\dot{X}_{des} = \dot{X}_{in} - \dot{X}_{out} + \dot{W}_{comp}$ (15)

Exergy efficiency:

$\eta_{II} = \frac{\dot{X}_{out} - \dot{X}_{in}}{\dot{W}_{comp}} = 1 - \frac{\dot{X}_{des}}{\dot{W}_{comp}}$ (16)

- Combustion Chambers:

Exergy destruction rate:

$\dot{X}_{des} = \dot{X}_{in\ fuel} + \dot{X}_{comp\ out\ air} - \dot{X}_{CC.out} + \dot{Q}_{in}$
(17)

$\dot{X}_{in\ fuel} = \dot{m}_{fuel}((h_{fuel} - h_{o\ fuel}) - T_o (s_{fuel} - s_{o\ fuel})) + \dot{m}_{fuel}(\xi * LHV)$ (18)

$\xi = x_{fuel}/LHV$ (19)

$\xi = 1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x}$ (20)

For a fuel gas C_xH_y

Exergy efficiency:

$\eta_{II} = \frac{\dot{X}_{out} - \dot{X}_{in}}{\dot{Q}_{in}} = 1 - \frac{\dot{X}_{des}}{\dot{Q}_{in}}$ (21)

- Turbine:

Exergy destruction rate:

$\dot{X}_{des} = \sum_{in,T} \dot{X} - \sum_{out,T} \dot{X} - \dot{W}_T = T_o(\sum \dot{S}_{out,T} - \sum \dot{S}_{in,T})$ (22)

Exergy efficiency:

$\eta_{II} = \frac{\dot{W}_T}{\dot{X}_{in} - \dot{X}_{out}} = 1 - \frac{\dot{X}_{des}}{\dot{X}_{in} - \dot{X}_{out}}$ (23)

- HRSG

Exergy destruction rate:

$\dot{X}_{des} = \left(\sum_{in,exh.gas} \dot{X} - \sum_{out,exh.gas} \dot{X}\right) - \left(\sum_{out,Steam} \dot{X} - \sum_{in,water} \dot{X}\right)$ (24)

Exergy efficiency:

$$\eta_{\mathrm{II}} = \frac{(\sum_{out,S}\dot{X} - \sum_{in,S}\dot{X})}{(\sum_{in,P}\dot{X} - \sum_{out,P}\dot{X})} = 1 - \frac{\dot{X}_{des}}{(\sum_{in,P}\dot{X} - \sum_{out,P}\dot{X})}$$
(25)

- Pumps

Exergy destruction rate:

$$\dot{X}_{des} = \sum_{in,P}\dot{X} - \sum_{out,P}\dot{X} + \dot{W}_P \quad (26)$$

Exergy efficiency:

$$\eta_{\mathrm{II}} = \frac{(\sum_{out}\dot{X} - \sum_{in}\dot{X})}{\dot{W}_P} = 1 - \frac{\dot{X}_{des}}{\dot{W}_P} \quad (27)$$

- Condenser

Exergy destruction rate:

$$\dot{X}_{des} = T_o(\dot{S}_{steam\ out} - \dot{S}_{steam\ in}) + \dot{Q}_{cooling\ water}$$
(28)

$$\eta_{\mathrm{II}} = 1 - \frac{\dot{X}_{des}}{\dot{Q}_{cooling\ water}} \quad (29)$$

- The whole CCPP Exergy efficiency:

$$\eta_{II,CCPP} = \frac{\dot{W}_{net,out}}{\dot{Q}_{in}(1-\frac{T_O}{T_H})} \quad (30)$$

$$\dot{W}_{net,out} = \sum \dot{W}_{turbines} - House\ load\ power$$
(31)

Where the amount of total heat added to the gas turbines (GTs) at combustion chambers is

$$\dot{Q}_{in} = \Sigma\dot{m}_e h_e - \Sigma\dot{m}_i h_i \quad (32)$$

For determining the effect of air relative humidity on gas turbine (GT) performance at temperatures of 16.1, 28.1, 37.6 ^{o}C, the following formulas was used

$$Cp_{air} = Cp_{dry\ air} + \omega * Cp_{vapor} \quad (33)$$

Where ω is the air specific humidity which may be taken from psychometric chart .

Substituting in the above equation with air relative humidity ratios from 10 to 90% for each ambient temperature of 16.1, 28.1, 37.6 ^{o}C.

6. Results and discussion

6.1 effect of ambient temperature on the performance of Beni-suef combined cycle power plant (CCPP):

Beni-suef combined cycle power plant (CCPP) was analyzed using the above relations at 16.1, 28.1, 37.6 ^{o}C ambient temperatures and 101.325 kPa for atmospheric pressure. Gas turbines (GTs) combustion chambers (CCs) have the largest proportion of the combined cycle power plant (CCPP) exergy destruction percentage >86 % and condensate extraction pumps have the lowest proportion of the combined cycle power plant (CCPP) exergy destruction percentage about 0.005 % in all ambient conditions. Fig. 2 indicates the proportions of exergy destruction of the combined cycle power plant (CCPP) at different ambient temperatures.

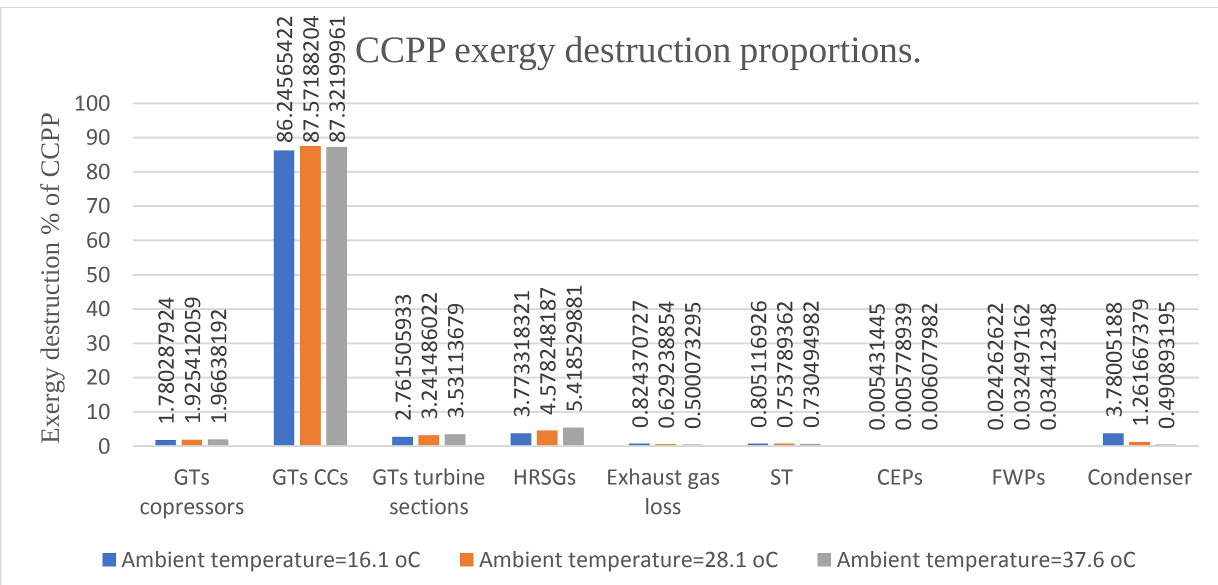

Fig. 2 The proportions of the combined cycle power plant (CCPP) components exergy destruction.

Starting from gas turbine (GT) compressor, Fig. 3 shows that with the increase of ambient temperature by 1 ^{o}C the specific exergy destruction in compressor is increased by 0.16871473 kJ/kg.

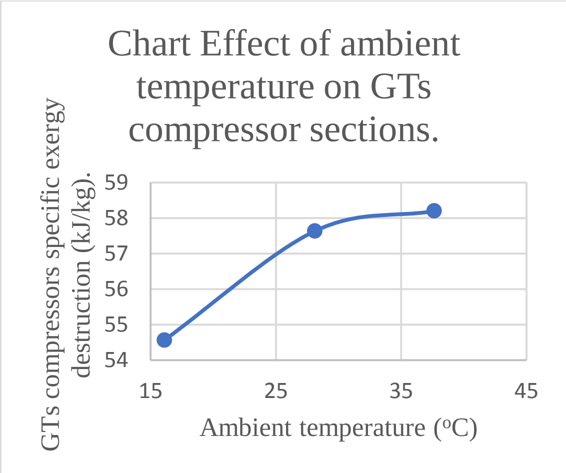

Fig. 3 The effect of ambient temperature on compressor exergy destruction.

For gas turbine (GT) combustion chambers section, the exergy destruction is decreased with the increase of ambient temperature because the air and fuel flow rates decrease as show in Fig. 4.

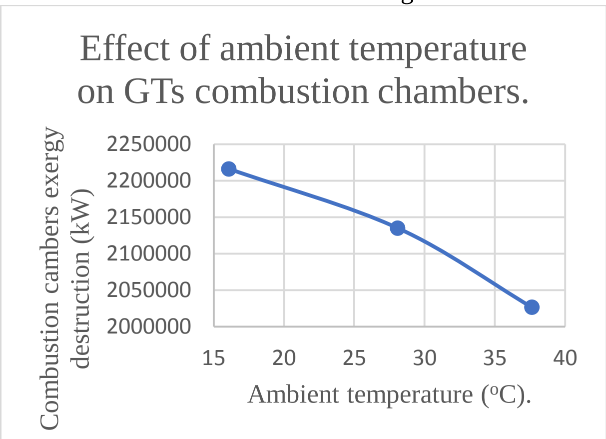

Fig. 4 The effect of ambient temperature on GT combustion chambers exergy destructions.

For gas turbine (GT) turbine sections the exergy destruction is significantly increased with the increase of ambient temperature as shown in Fig. 5.

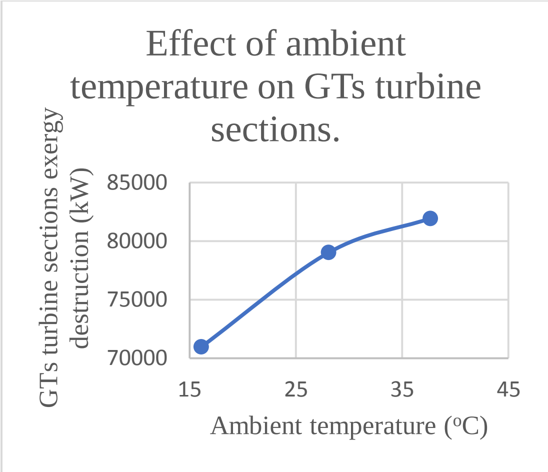

Fig. 5 The effect of ambient temperature on GTs turbine section exergy destruction.

The overall gas turbines (GTs) exergy destruction is decreased with the increase of ambient temperature as it is seen in Fig. 6.

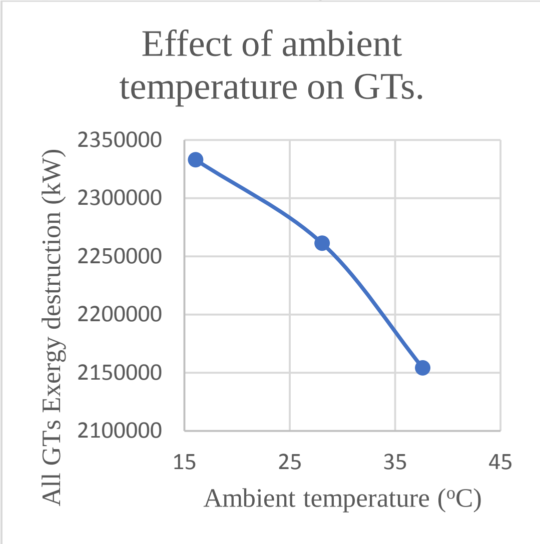

Fig. 6 The effect of ambient temperature on overall gas turbine (GT) exergy destruction.

As for heat recovery steam generators (HRSGs) part, the increase of ambient temperature leads to an increase in heat recovery steam generators (HRDGs) exergy destruction as it is illustrated in Fig. 7.

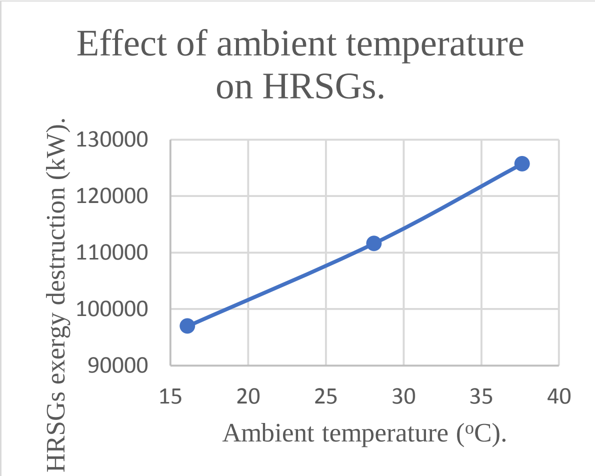

Fig.7 The effect of ambient temperature on heat recovery steam generators (HRSGs) exergy destruction.

The HP evaporator contributes by about 20 % of the overall HRSG exergy destruction. Also the exergy destruction in HP evaporator is significantly increased with the increase of ambient temperature as shown in Fig. 8.

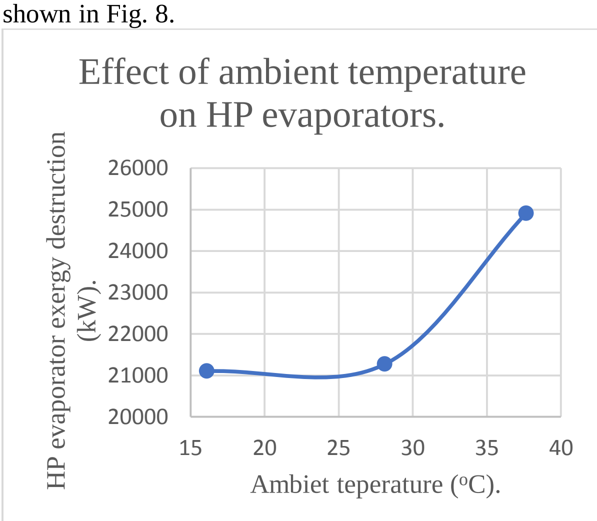

Fig. 8 The effect of ambient temperature on HP evaporator exergy destruction.

Regarding to the exhaust gas exergy lost to the atmosphere, the exhaust gases exergy lost is decreased with the increase of ambient temperature as shown in Fig. 9. The exhaust gases temperature to the atmosphere approximately remains constant in all operating conditions however the increase of ambient temperature decreases the gaseous mass flow rate.

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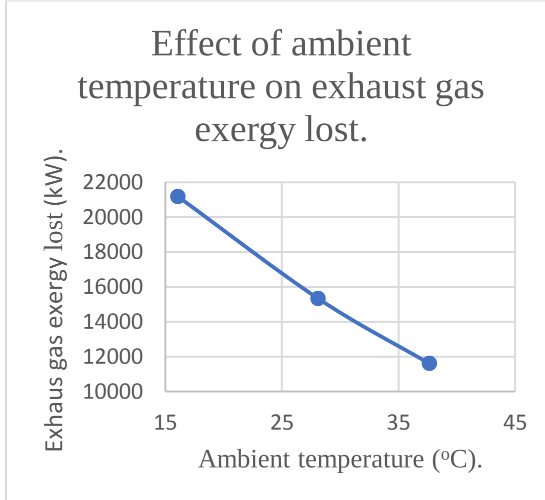

Fig. 9 The effect of ambient temperature on exhaust gases exergy lost.

The increase in ambient temperatures increases the exergy destruction in the high pressure turbine (HPT) and decreases it in the intermediate pressure turbine (IPT). While a decrease in overall steam turbine (ST) exergy destruction and in the low pressure turbine (LPT) sections is resulted by increasing the ambient temperatures as show in Fig.10.

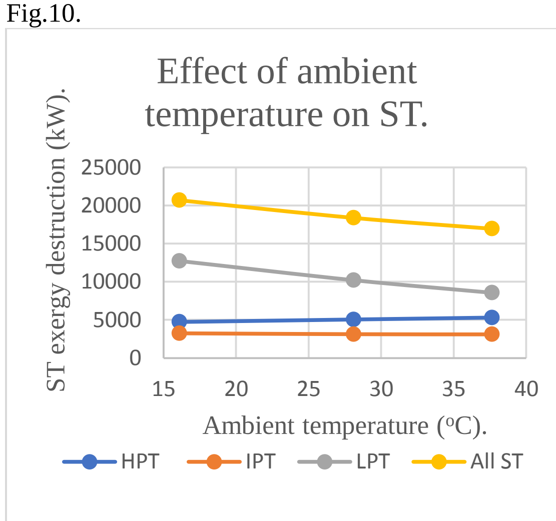

Fig. 10 The effect of ambient temperature on steam turbine (ST) exergy destruction.

With the increase in ambient temperature the feed water pumps (FWPs) exergy destruction is significantly increased but it seems to be constant for condensate extraction pumps (CEPs) as shown in Fig. 11.

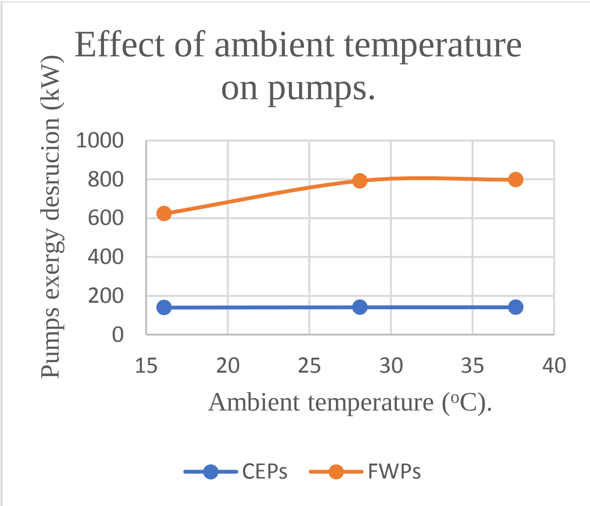

Fig. 11 The effect of ambient temperature on pumps exergy destruction.

The condenser exergy destruction is decreased with the increase in the ambient temperatures and with the decrease of the steam mass flow rates as shown in Fig. 12.

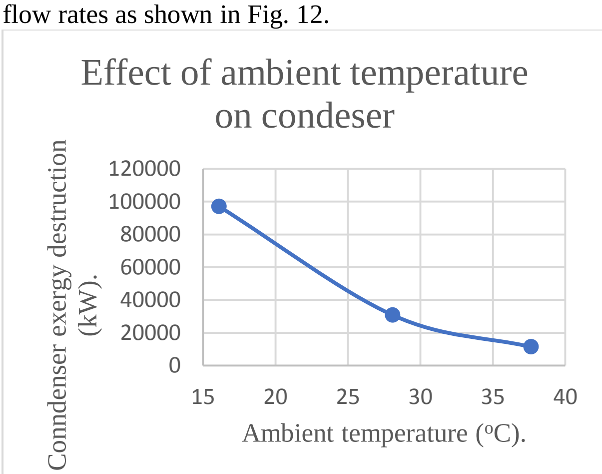

Fig. 12 The effect of ambient temperature on condenser exergy destruction.

The overall combined cycle power plant (CCPP) exergy destruction is decreased with the increase of ambient temperature as the fuel and air mass flow rates decrease which enter the combustion process which have a high proportion of exergy destruction >86 % of the overall combined cycle power plant (CCPP) exergy destruction as shown in Fig. 13.

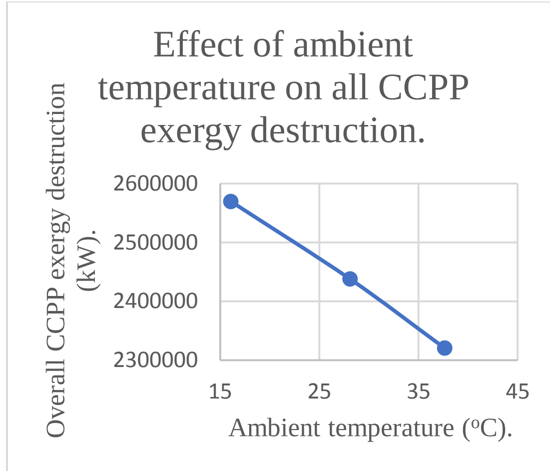

Fig. 13 The effect of ambient temperature on overall combined cycle power plant (CCPP) exergy destruction.

With the increase in ambient temperature, gas turbines (GTs) gross power output decrease as air and fuel mass flow rates decrease resulting a decrease in exhaust gases total flow rate and also a decreases in the generated steam from heat recovery steam generators (HRSGs) which also leads to a decrease in the steam turbine (ST) gross power output. This is shown in Fig. 14.

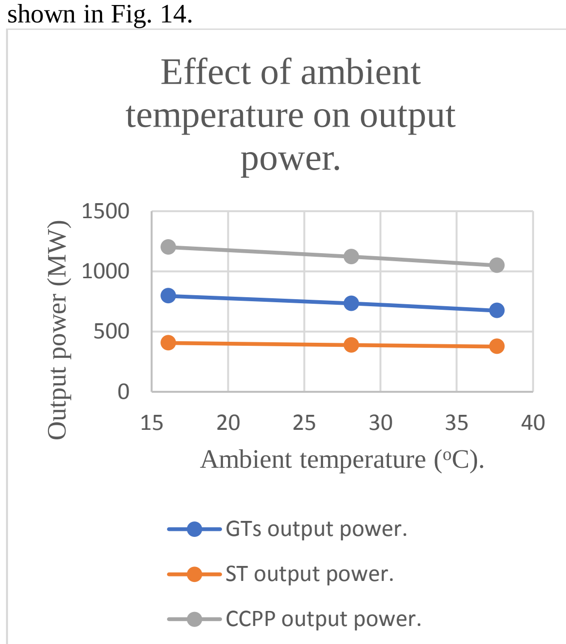

Fig. 14 The effect of ambient temperature on output gross power.

With the increase in ambient temperature, gas turbines (GTs) and the combined cycle power plant (CCPP) efficiencies are decreased as shown in Fig.15.

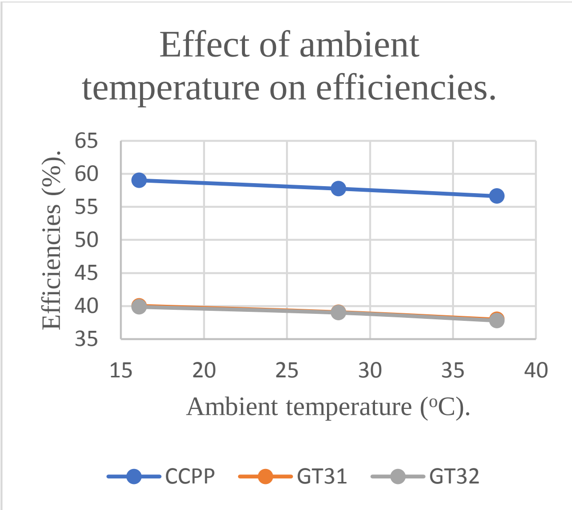

Fig.15 The effect of ambient temperature on efficiencies.

With the increase of ambient temperature, the combined cycle power plant (CCPP) net heat rate is increased as shown in Fig. 16.

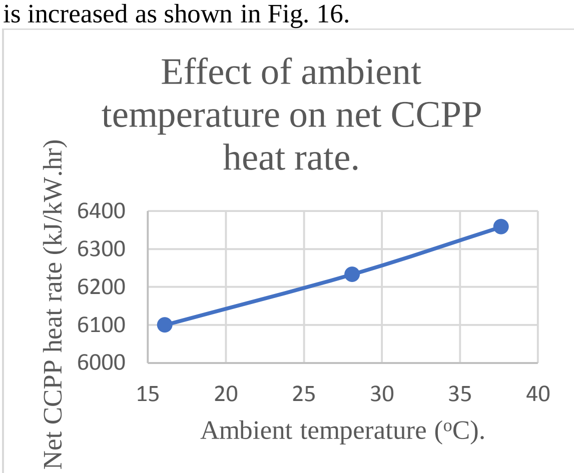

Fig. 16 The effect of ambient temperature on combined cycle power plant (CCPP) net heat rate.

6.2 Effect of air relative humidity on the performance of Gas Turbine (GT):

The effect of air relative humidity was studies at ambient temperatures of 16.1, 28.1, 37.6 °C at different air relative humidity ratios from 10% to 90% on gas turbine (GT) compressor performance.

Fig. 17 indicates that with the increase of air relative humidity, compressor specific work increases. It is obvious that at humidity ratios over 50 % the compressor specific work at 37.6 °C is higher than that at 28.1 °C and at 16.1 °C at the same air relative humidity ratio.

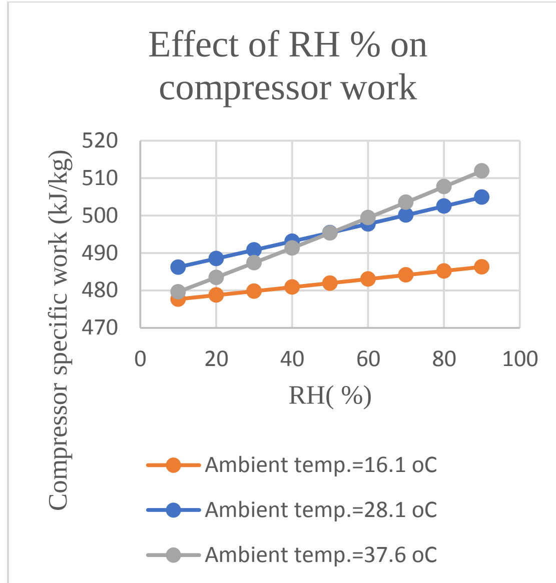

Fig. 17 The effect of air relative humidity on compressor specific work.

With the increase of air relative humidity, compressor power is increased. At the same air relative humidity ratios, the compressor power at 37.6 °C is higher than that at 28.1 °C and at 16.1 °C as shown in Fig. 18.

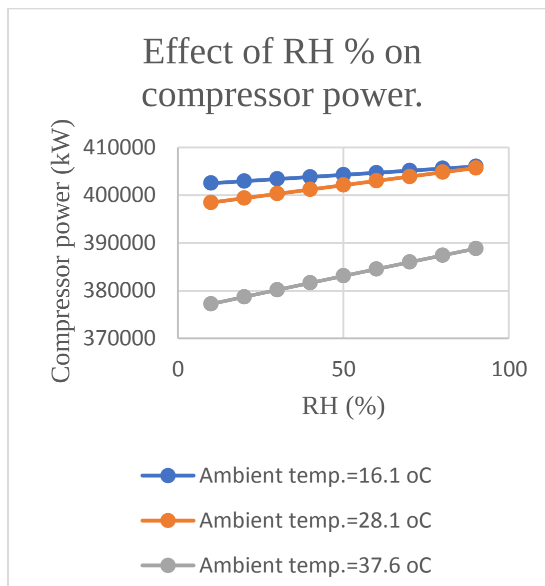

Fig.18 The effect of air relative humidity on compressor power.

As a result of the increase of compressor work and power, compressor isentropic efficiency is decreased with the increase of air relative humidity. At the same air relative humidity ratios, compressor isentropic efficiency at 37.6 °C is lower than that at 28.1 °C and at 16.1 °C as shown in Fig.19.

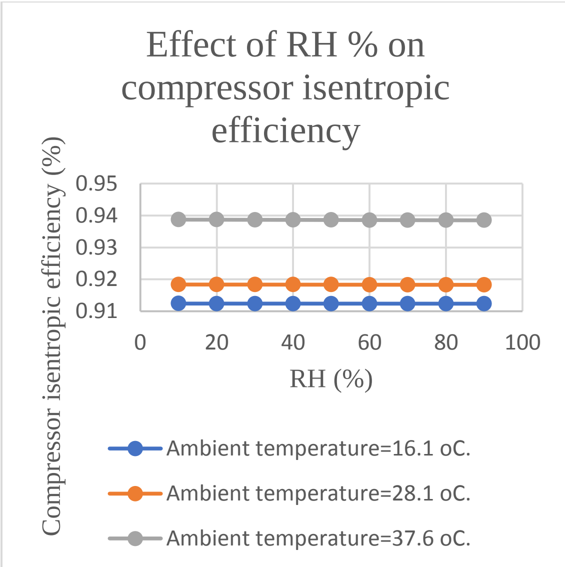

Fig. 19 The effect of air relative humidity ratio on compressor isotropic efficiency.

It is clear in Fig. 20 that with the increase of air relative humidity ratio, compressor specific exergy destruction increases. At the same air relative humidity ratios, compressor specific exergy destruction at 37.6 ^{o}C is higher than that at 28.1 ^{o}C and at 16.1 ^{o}C

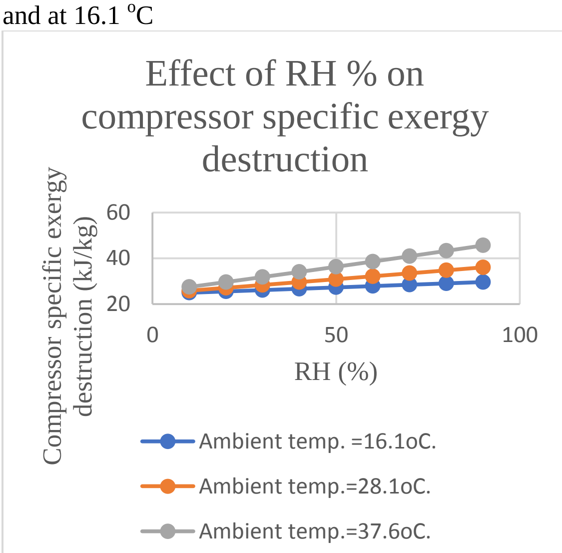

Fig. 20 The effect of air relative humidity on compressor specific exergy destruction

Compressor exergy destruction rate is decreased with the increase of air relative humidity because the dry air mass flow rate is decreased at the same ambient temperature. At the same air relative humidity ratios, exergy destruction rate at 37.6 ^{o}C is lower than that at 28.1 ^{o}C and at 16.1 ^{o}C this is also because the dry air mass flow rate is decreased with the increase of ambient temperature as shown in Fig. 21.

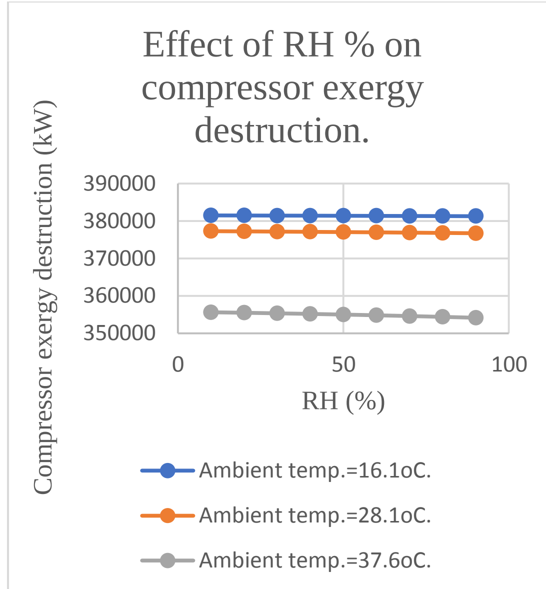

Fig.21 The effect of air relative humidity ratio on compressor exergy destruction rate.

As a result of the increase of compressor specific exergy destruction with the increase of air relative humidity ratio, compressor second law efficiency is decreased. at the same air relative humidity ratios, compressor second law efficiency at 37.6 ^{o}C is lower than that at 28.1 ^{o}C and at 16.1 ^{o}C as it is clear in Fig. 22.

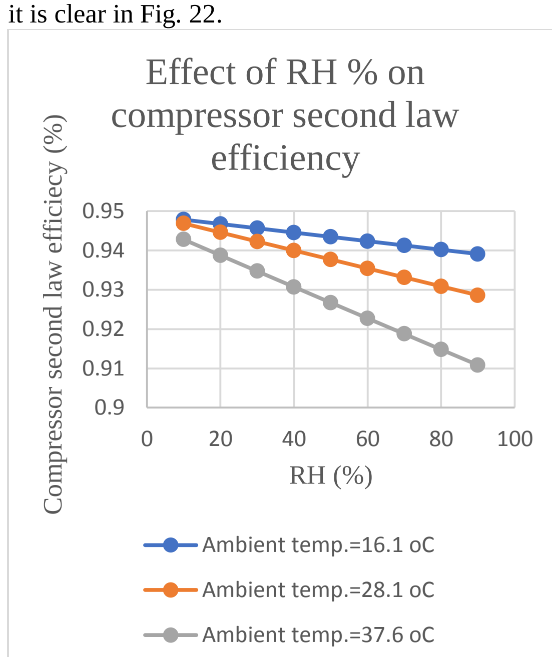

Fig.22 The effect of air relative humidity ratio on compressor second law efficiency.

As a result of the above-mentioned relations between compressor performance and the increase of air relative humidity ratio, gas turbine net power is decreased with the increase of air relative humidity ratios. At the same air relative humidity ratios, Gas turbine (GT) net power at 37.6 °C is lower than that at 28.1 °C and at 16.1 °C as it is clear in Fig. 23.

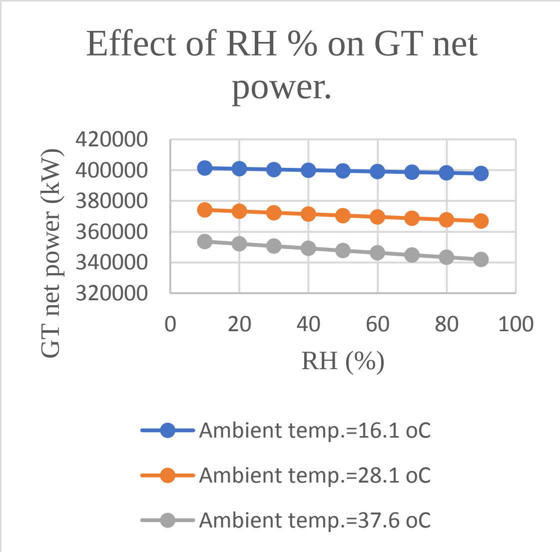

Fig. 23 effect of air relative humidity ratio on GT net power.

Also, gas turbine (GT) efficiency is decreased with the increase of air relative humidity ratio. At the same air relative humidity ratios, Gas turbine efficiency at 37.6 °C is lower than that at 28.1 °C and at 16.1 °C as it is obvious in Fig. 24.

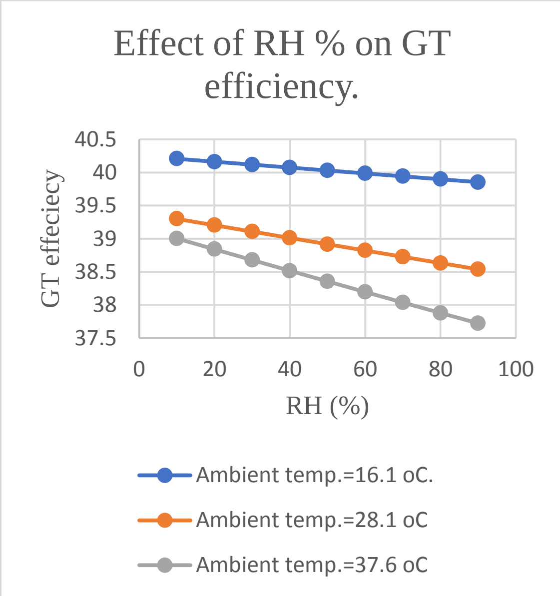

Fig.24 The effect of air relative humidity ratio on GT efficiency.

As gas turbine (GT) efficiency decreases with the increase of air relative humidity ratio (RH), the gas turbine (GT) net heat rate is increased. at the same air relative humidity ratios, the gas turbine (GT) net heat rate at 37.6 °C is higher than that at 28.1 °C and at 16.1 °C as it is illustrated in Fig. 25.

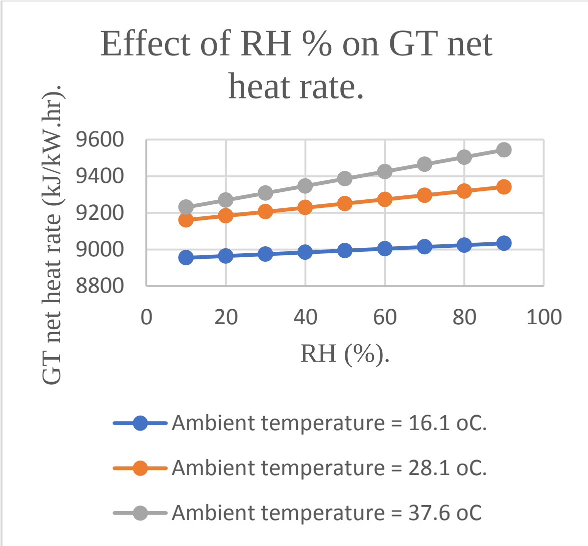

Fig. 25 the effect of air relative humidity ratio (RH) on gas turbine (GT) net heat rate.

7. Conclusions

In this study, the effect of ambient temperature and air relative humidity on the performance of Beni-suef combined cycle power plant (CCPP)has been presented based on energy and exergy analysis. The second law analysis has supplied us by additional and significant information about the performance level of the power plant.

Regarding the effect of ambient temperature on combined cycle power plant performance, the following main conclusions have been obtained:

1. An increase of ambient temperature by 1 °C leads to a decrease in combined cycle power plant (CCPP) gross power output, efficiency and second law efficiency by 7.0446 MW, 0.111 % and 0.0835% respectively and also leads to an increase in combined cycle power plant (CCPP) net heat rate by 11.989 kJ/kW hr.
2. The largest proportion of exergy destruction was at the gas turbines combustion chambers (>86%) in all the studied ambient temperature conditions and its percentage is increasing with the increase of ambient temperature while the exergy destruction itself is decreasing with the increase of ambient temperature as air and fuel mass flow rates decreases.
3. At 16.1 °C ambient temperature condenser exergy destruction proportion came after the combustion chambers (CCs) exergy destruction proportion by a value of 3.78005188 % then the heat recovery steam generators (HRSGs), the gas turbines (GTs) turbine sections and the gas turbines (GTs)

compressors by values of 3.7733183%, 2.7615059% and 1.780288% respectively.

4. While at 28.1 °C ambient temperature, heat recovery steam generators (HRSGs), the gas turbines (GTs) turbine sections, the gas turbines (GTs) compressors and condenser proportions came after the combustion chambers (CCs) exergy destruction proportion by values of 4.5782482%, 3.241486%, 1.925412% and 1.261667379 % respectively.
5. And at 37.6 °C ambient temperature, heat recovery steam generators (HRSGs), the gas turbines (GTs) turbine sections and gas turbines (GTs) compressors proportions came after the combustion chambers (CCs) exergy destruction proportion by values of 5.4185299 %, 3.5311368 % and 1.966382 % respectively. While condenser proportion is small (about 0.4908 %).

Regarding the effect of air relative humidity ratio on gas turbine (GT) performance, the following main conclusions have been obtained:

1. At 16.1 °C ambient temperature an increase in air relative humidity (RH) by 10 % leads to a decrease in the gas turbine (GT) output net power and efficiency by 439.883kW and 0.04407% respectively also leads to an increase compressor specific work, compressor exergy destruction and gas turbine (GT) net heat rate by 1.078827 kJ/kg, 466.219561 kW and 9.901 kJ/kW.hr respectively.
2. At 28.1 °C ambient temperature an increase in air relative humidity (RH) by 10 % leads to a decrease in the gas turbine (GT) output net power and efficiency by 904.904 kW and 0.09505% respectively and also leads to an increase in compressor specific work, compressor exergy destruction and gas turbine (GT) net heat rate by 2.332692 kJ/kg, 974.4538447 kW and 22.593 kJ/kW.hr, respectively.
3. At 37.6 °C ambient temperature an increase in air relative humidity (RH) by 10 % leads to a decrease in the gas turbine (GT) output net power and efficiency by 1450.5kW and 0.16002 % respectively and also leads to an increase in compressor specific work, compressor exergy destruction and gas turbine (GT) net heat rate by 4.036437 kJ/kg and 1634.241454 kW and 39.153 kJ/kW.hr respectively.

8. Nomenclature

h specific enthalpy, kJ/kg
H total enthalpy, kJ
I exergy destruction rate, MW
$\dot{m}$ mass flow rate, kg/s
P pressure, bar or MPa
$\dot{Q}$ heat transfer rate, MW
s specific entropy, kJ/kg K
$\dot{S}$ rate of total entropy, kJ/s K
T temperature, ^{oC} or K
T_H temperature of high temperature reservoir, K
T_L temperature of low temperature reservoir, K
T_o surrounding temperature or ambient temperature, oC or K.
W total work, MJ
$\dot{W}$ power, MW
$\dot{X}$ rate of total exergy, MW
$\dot{X}_{des}$ rate of total exergy destruction, MW

9. Abbreviation

CCPP combined cycle power plant.
CCs combustion chambers.
CEPs condensate extraction pumps.
DB duct burning.
DP dual pressure.
FWPs feed water pumps.
GT gas turbine.
GTG gas turbine generator
HRSG heat recovery steam generator.
HP high pressure section.
HP-EVP high pressure evaporator
HPT high pressure turbine section.
IP Intermediate pressure section.
IPT Intermediate pressure turbine section.
LP low pressure section.
LPT low pressure turbine section.
LHV lower heating value of fuel.
rc compression ratio.
RH air relative humidity.
STC steam turbine cycle.
ST steam turbine.
TIT turbine inlet temperature.

10. Greek letters

η_{th} thermal efficiency
η_{11} exergy efficiency
ψ specific exergy
ξ ratio between fuel specific exergy to fuel LHV.

11. Subscripts

c condenser
Comp compressor
des destruction
gen generation
H high temperature or source temperature (as in T_H and Q_H)
in inlet
L low temperature or sink temperature (as in T_L and Q_L)
out outlet
rev reversible
T turbine
u useful
o dead state or state of surrounding
1 initial or inlet state
2 final or exit state

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تحليل أداء محطة بني سويف المركبة(٤*١٢٠٠)م و.

ملخص:

تعتبر محطة بني سويف المركبة واحدة من اكبر محطات الطاقة واكثرها تطورا في مصر و حتي في العالم.وهي تتكون من اربع وحدات كل وحدة تتكون من تربينتين غازيتين وغلايتين لاستعادة الطاقة من النوع ثلاثي مستويات الضغط وتربينة بخارية وثلاث مولدات كهربائية.تنتج المحطة حوالي ٤٨٠٠ م.و. (١٢٠٠م.و.\وحدة) بكفاءة تزيد عن ٦٠% في عند الظروف التصميمية للمحطة .

يتأثر أداء اي محطة طاقة مركبة بالظروف المحيطة بالمحطة مثل درجة حرارة الوسط المحيط والضغط الجوي والرطوبة النسبية للمكان الذي تم تركيب المحطة فيه ودرجة حرارة دخول مياة التبريد.الهدف من الدرسة هو معرفة مدي تأثير درجة حرارة الوسط المحيط و رطوبة الهواء النسبية علي اداء محطة بني سويف المركبة عن طريق جراء تحليل للطاقة والاكسيرجي لمكونات محطة الطاقة.

تم عمل تحليل للطاقة وللاكسيرجي عند درجات الحررة ١٦.١ و ٢٨.١ و٣٧.٦ درجة مئوية علي وحدة من وحدات محطة بني سويف المركبة باستخدام بيانات التشغيل .ايضا تم دراسة تأثيرالرطوبة النسبية للهواء علي أداء التربينة الغازية عند درجات الحررة ١٦.١ و ٢٨.١ و٣٧.٦ درجة مئوية ورطوبة نسبية متغيرة من ١٠ % وحتي ٩٠%.