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The Study of Ambient Temperature Effects on Exergy Destruction of a Heat Recovery Steam Generator

[1]*Abdulkareem Salih Bilal, [2]Mohammed Salih Mohammed

[1,2]Department of Mechanical Engineering, College of Engineering, University of Mosul, 42002, Iraq

*Corresponding Author's E-mail: abdalkareem.enp158@student.uomosul.edu.iq

***Abstract* - Combined cycle power plants (CCPP) are very important to the global power generation industry. The thermodynamic behavior of these cycles is usually studied to determine the optimal configuration and optimal design conditions for any cycle arrangement. In a combined rotary power plant, many factors affect the cycle efficiency and output power. The temperature of the turbine exhaust gas greatly affects the optimal design and configuration of the heat recovery generator. In fact, when the exhaust gas temperature drops, if the number of pressure levels increases, the energy output and heat recovery will increase sharply another important parameter is the ambient temperature. When the ambient temperature increases, the density of the inlet air decreases, so the mass flow through the turbine decreases, and the output power also decreases. In addition, the energy demand for compressing the air in the compressor increases, so the net electrical energy produced by the compressor ranks first in the reduction cycles.**

Keywords: Exergy destruction, Ambient Temperature, HRSG.

I. Introduction

There are many researchers such as Cottas [3] and Moran. [4] Who performed the stress analysis of the combined cycle. Find out the energy loss for each part. Basile [5] studied modeling, numerical optimization, and irreversibility reduction of the combined cycle triple pressure reheating. It considered the seven-point HRSG configuration and analyzed the effect of TIT (turbine inlet temperature) on the pressure point temperature difference. He also studied the effect of TIT on the inlet pressure of high-pressure turbines. Its goal is to reduce the temperature difference at the pressure point. Fiaschi and Manfrida [6] have calculated the energy loss of combined cycle power plants and found that this type of heat recovery steam generator for combined cycle power plant (CCPP) and gas turbine combustion is very important for the global power generation industry. The thermodynamic behavior of these cycles is usually studied to determine the optimal configuration and optimal design conditions for any cycle arrangement. In combined cycle power plants, many parameters affect cycle efficiency and output power. The temperature of the turbine exhaust gas has a significant impact on the optimal design and configuration of the heat recovery generator. In fact, when the exhaust gas temperature decreases, if the number of pressure levels increases, the energy production and heat recovery will increase sharply [1]. Another important parameter is the ambient temperature. When the ambient temperature increases, the density of the inlet air decreases, so the mass flow through the turbine decreases, and the output power also decreases. In addition, the energy demand for compressed air in the compressor increases, so the net electricity generated in the peak cycle decreases. Franco and Casarosa [7] have studied the possibility of increasing the efficiency of combined cycle power plants to more than 60%. money and more. Apply energy and multiplier analysis to study combined cycle power plants, and use operating data to determine the potential for improving system efficiency [8]. Studies have shown that combustion chambers, gas turbines, and heat recovery sensors are the main irreversible sources, accounting for more than 85% of total energy loss. In recent years, many engineers and scientists have suggested that in addition to or instead of traditional energy analysis, exergy analysis can also be performed to better evaluate the thermodynamic performance of a process. The reason is that exergy analysis appears to provide more insights and is more useful in improving efficiency than energy analysis alone. Combined cycle power plants (CCCP) have been widely used in Iran's grid. Therefore, it is very important to evaluate the design and operation of this power plant. The aim of this research is to evaluate the effect of ambient temperature change on the energy loss of a heat recovery steam generator. 125 MW of combined cycle was taken into account in this study and power analyzes are performed using performance data at different ambient temperatures.

II. Heat Recovery Steam Generation Description (HRSG)

The double pressure heat recovery steam generator is a series of heat exchangers, consisting of three heat exchangers (economy, evaporator, and super heater), each with a pressure level (see Figure 1). The economizer is used to heat water close to

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saturation, the evaporator is used to generate saturated steam, and the super heater is used to generate superheated steam. Each heat exchanger is a bundle tube arranged in a straight or zigzag arrangement according to manufacture.[10]

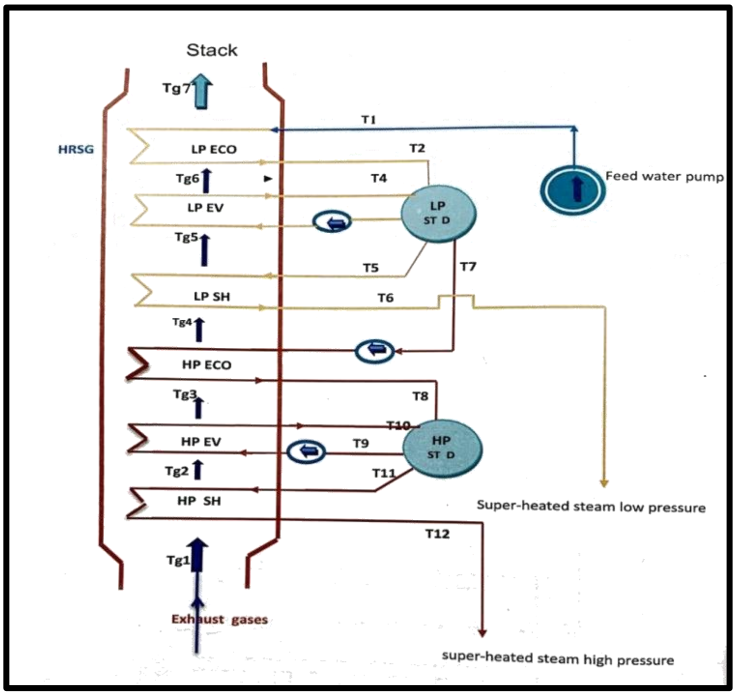

Figure 1: double pressure heat recovery steam generator

Gas turbine

The pressure coming out of the compressor is calculated using the following equation:

$$P_2 = r_{PC} * P_1$$

The ideal temperature (isotropia) of the air leaving the compressor is calculated by the following equation:

$$\acute{T}_2 = T_1 * \left({}^{P_2}/_{P_1}\right)^{\frac{\gamma a-1}{\gamma a}}$$

As for the real temperature of the air leaving the compressor, it is found by the following equation: [1]

$$T_2 = T_1 * \left\{1 + \frac{1}{\eta_{isc}}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma a-1}{\gamma a}} - 1\right]\right\}$$

The work of the compressor is calculated using the following equation:

$$\dot{W}_c = \dot{m}_a * (h_{g2} - h_{g1})$$

The specific heat of the air was created by the characteristics of the program used in terms of temperature or by using the following equation [6]:

$$\text{C\_pa (T)}=1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.49031T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right)$$

While the density of air is calculated through the following equation:

$$\rho_a = \frac{P}{R_a T}$$

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To calculate the air mass flow rate, this equation was used:

$$\dot{m}_a = \rho_a * VA$$

The real combustion equation can be represented as:

$$\lambda\, C_{x1}H_{y1} + (X_{o2}O_2 + X_{N2}N_2 + X_{H2O}H_2O + X_{CO2}CO_2) \longrightarrow Y_{co2}CO_2 + Y_{H2O}H_2O + Y_{O2}O_2 + Y_{N2}N_2$$

The mass of fuel entering the combustion chamber was calculated from the following equation:

$$\dot{m}_f = \frac{\lambda \dot{m}_a M_f}{M_a}$$

While the number of moles of each compound in the combustion products (exhaust gases) was calculated from the following equations:

$$Y_{co2} = (\lambda X_1 + X_{co2})Y_{N2} = X_{N2}$$

$$Y_{H2O} = X_{H2O} + \frac{\lambda Y_1}{2} Y_{O2} = X_{O2} - \lambda X_1 - \frac{\lambda Y_1}{4}$$

For the molar fraction of each combustion product compound only from the following equation:

$$\overline{n_{Pi}} = \frac{Y_i}{\sum Y_i}$$

While the specific heat at constant pressure for the gases leaving the combustion chamber was calculated by the following equation

$$C_{pg}(T) = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.22447T^3}{10^{10}}\right)$$

The mass of the gas mixture entering the gas turbine is the sum of the mass of the air coming from the compressor and the mass of the fuel injected into the combustion chamber, as shown by the equation**:**

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

This chemical reaction results in adding heat to the system with constant pressure. The amount of heat added in the combustion chamber is calculated by the following equation:

$$\dot{Q}_{cc} = \eta_{cc}.\dot{m}_g.(h_{g3} - h_{g2})$$

The optimum temperature of the exhaust gases leaving the gas turbine can be calculated by the following equation:

$$P_3 = P_2 \text{-} 0.02P_2$$

$$P_4 = P_1$$

$$\acute{T}_4 = T_3 * \left({}^{P_4}/{}_{P_3}\right)^{\frac{\gamma_{g-1}}{\gamma_g}}$$

To calculate the temperature of the exhaust gases leaving a gas turbine, we use the following equation:

$$T_4 = T_3 - \eta_{is\,GT}(T_3 - \acute{T}_4)$$

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The work produced by the gas turbine is calculated by the following equation:

$$\dot{W}_{GT} = \dot{m}_g(\mathrm{h_{g3}} - \mathrm{h_{g4}})$$

The network out of the gas turbine is calculated as follows:

$$\dot{W}_{net} = \dot{w}_{GT} - \dot{w}_c$$

Increasing the external ambient temperature from 5 C to 50 C reduces the air mass entering the compressor, the generated power, the thermal efficiency of the simple gas unit, the amount of heat supplied by the fuel to the combustion chamber, and the exhaust gas flow rate decreases by (2.17%),(19.84%) and (12.94%), respectively, while the specific fuel consumption of the installed unit increased by (4.8%) and the temperature of the exhaust gases increased from 505 C to 550 C with the increase in the external ambient temperature from 5 C to 50 C, see figure (2 ,3 , 4 and 5).

The power generated by the simple gas generating unit is represented by the following**:**

$$Pe_{GT} = \dot{w}_{net} \cdot \eta_{m\ GT}$$

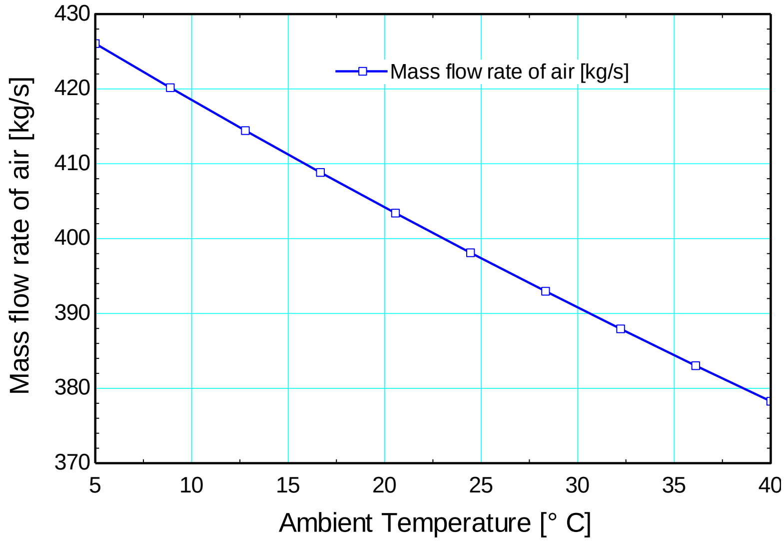

Figure 2: Shows the relationship between the temperature of the external environment and the mass of air

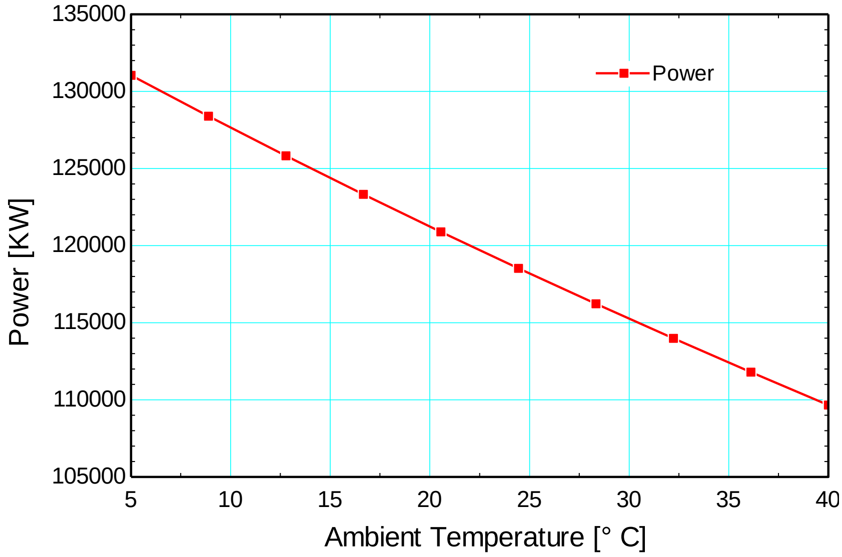

Figure 3: Shows the relationship between ambient temperature and power

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The thermal efficiency of the gas unit is calculated from the following equation:

$$\eta_{th\ GT} = \frac{Pe_{GT}}{\dot{m}_f . LHV}$$

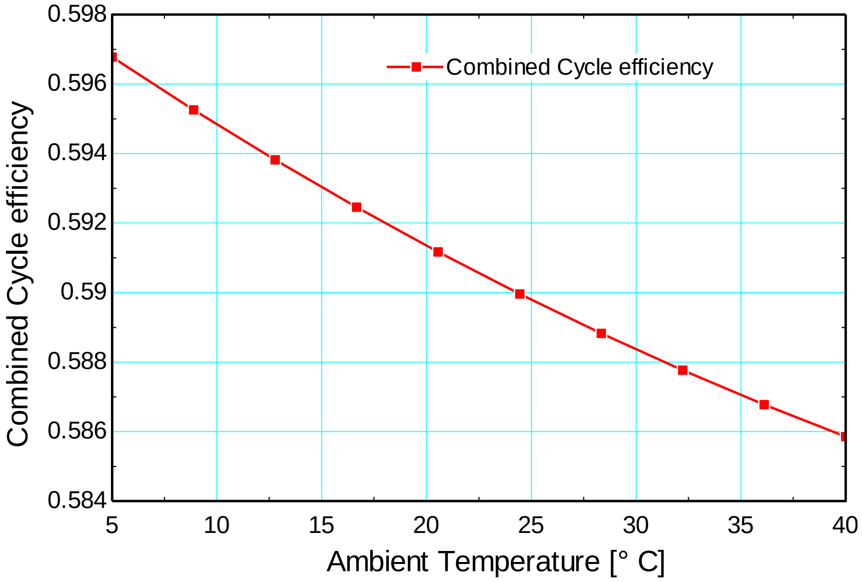

Figure 4: Shows the relationship between the temperature of the external environment and the efficiency of the combined cycle

While the specific fuel consumption rate is calculated through the following equation:

$$SFC_{GT} = \frac{\dot{m}_f * 3600}{Pe_{GT}}$$

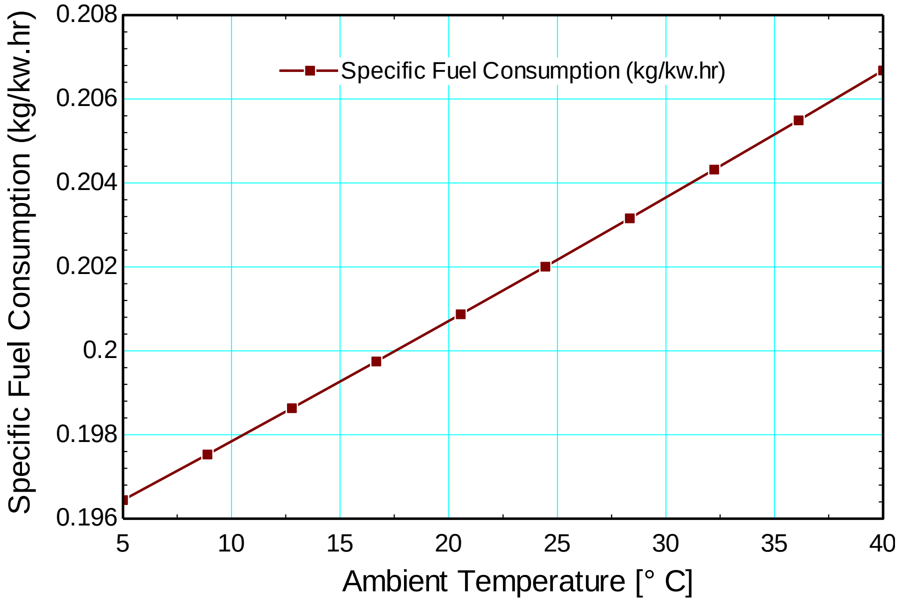

Figure 5: Shows the relationship between ambient temperature and specific fuel consumption

III. Energy Balance

The energy balance program between the working fluid and the exhaust gas is used to obtain the appropriate operating factors for the functioning of the steam generation system, including the design vapor pressure, temperature distribution, mass flow rate, and consumption of the working fluid. The heat obtained is the entry point to find the engineering design of the heat exchanger. The state of thermal equilibrium with the environment is achieved through the process of inversion, and the heat loss from the gas is equal to the heat or vapor produced by the working fluid (water), and is represented by the following formula:

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$$\dot{Q}_g = [\dot{Q}_{SH} + \dot{Q}_{EV} + \dot{Q}_{ECO}]_{HP} + [\dot{Q}_{SH} + \dot{Q}_{EV} + \dot{Q}_{ECO}]_{LP}$$

Accordingly, the heat gained in the high-pressure stage roaster is calculated by the following equation:

$$(\dot{Q}_{SH})_{H.P} = \dot{m}_{ss}(h_{12} - h_{11})_{H.P} = \dot{m}_g . cp_g (T_{g1} - T_{g2})$$

The heat gained in the high-pressure phase evaporator is calculated by the following equation:

$$(\dot{Q}_{EV})_{H.P} = \dot{m}_{sat}(h_{10} - h_9)_{H.P} = \dot{m}_g . cp_g (T_{g2} - T_{g3})$$

In the same way, the heat gained in the economizer of the high-pressure stage is calculated using the following equation:

$$(\dot{Q}_{ECO})_{H.P} = \dot{m}_{wa}(h_8 - h_7)_{H.P} = \dot{m}_g . cp_g (T_{g3} - T_{g4})$$

The heat gained in the low-pressure stage roaster is calculated by the following equation:

$$(\dot{Q}_{SH})_{L.P} = \dot{m}_{ss}(h_6 - h_5)_{L.P} = \dot{m}_g . cp_g (T_{g4} - T_{g5})$$

The heat gained in the low-pressure phase evaporator is calculated by the following equation:

$$(\dot{Q}_{EVA})_{L.P} = \dot{m}_{sat}(h_4 - h_3)_{L.P} = \dot{m}_g . cp_g (T_{g5} - T_{g6})$$

In the same way, the heat gained in the low-pressure economizer is calculated using the following equation:

$$(\dot{Q}_{ECO})_{L.P} = \dot{m}_{wa}(h_2 - h_1)_{L.P} = \dot{m}_g . cp_g (T_{g6} - T_{g7})$$

IV. Exergy Destructions

The thermal regime considered for analysis is provided by some input from the energy source (exergy EF). This input goes to some exergy output (exergy product EP). For a real process, the energy input is always greater than the energy output, and this imbalance is due to the waste of energy; It is useful to distinguish between types of energy waste to study where irreversibility occurs. Two types of external energy waste can be distinguished: internal and external [9]. External energy waste (exergy Eloss) represents the content of exit energy remaining from waste and emissions from production and manifestations, and therefore unused (unused energy is the surplus of used output).External energy destruction (ED) corresponds to wastage of quality due to internal defects in the process and is a direct consequence of system irreversibility. Because of technical deficiencies in the plant, such as friction or lack of insulation, these internal reflections may be technical or structural in nature. The structural damage of electric power E D is determined by the principle and design of the system. Although ED technical energy destruction can be reduced through optimization, structural waste can only be reduced by redesigning the system.

The specific flow exergy of a fluid at any cycle state is given:

$$EXERGY = (h - ho) - To * (s - so)$$

We note that the exergy destruction comp is a direct relationship with the temperature of the external environment, as it increases with increasing temperature, as in Figure 6, and as the opposite is true with the combustion chamber and the turbine, as the relationship is inverse, as the increase leads to a decrease in the Exergy destruction comb.chamber and Exergy destruction turbine as In Figures 7 and 8.

Calculate the Exergy Destruction in the Compressor as in the following equation:

$$ED_{comp} = \dot{m}_a (ex_2 - ex_1) + w_{comp}$$

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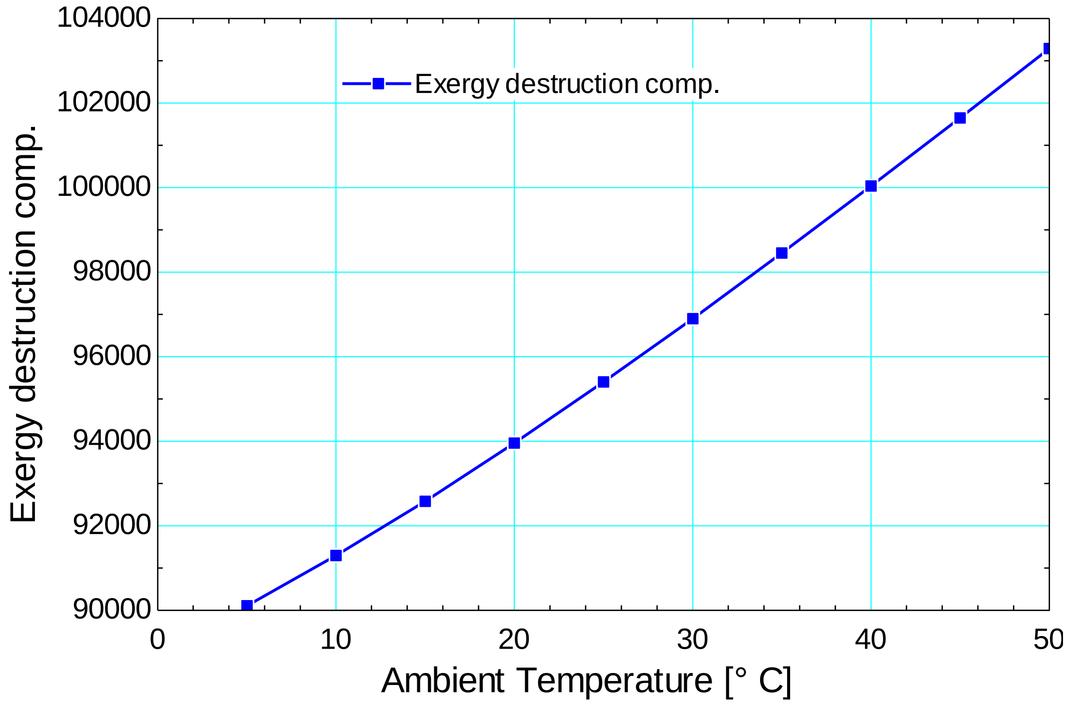

Figure 6: shows the relationship between ambient temperature and Exergy destruction comp

Calculate the combustion chamber energy destruction using the following equation:

$$ED_{cc} = \dot{m}_g\,(ex_3 - ex_2) - E_{fuel}$$

$$E_{fuel} = LCV * \dot{m}_g$$

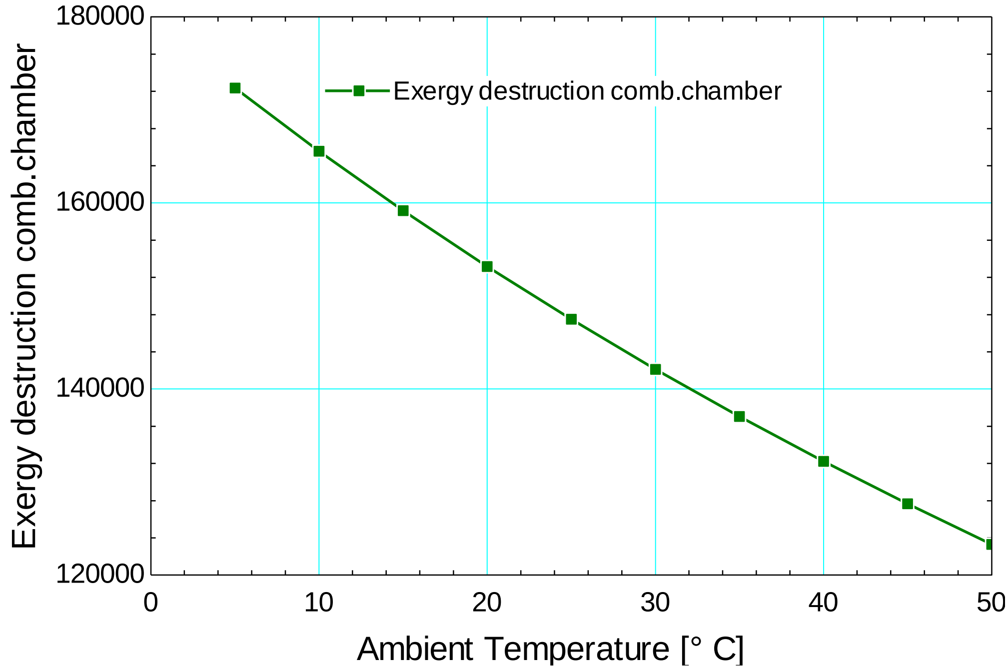

Figure 7: shows the relationship between the ambient temperature and the Exergy destruction comb chamber

The destruction of the excessive power of the turbine can also be calculated by the following equation

$$ED_t = \dot{m}_g(ex_3 - ex_4) - w_t$$

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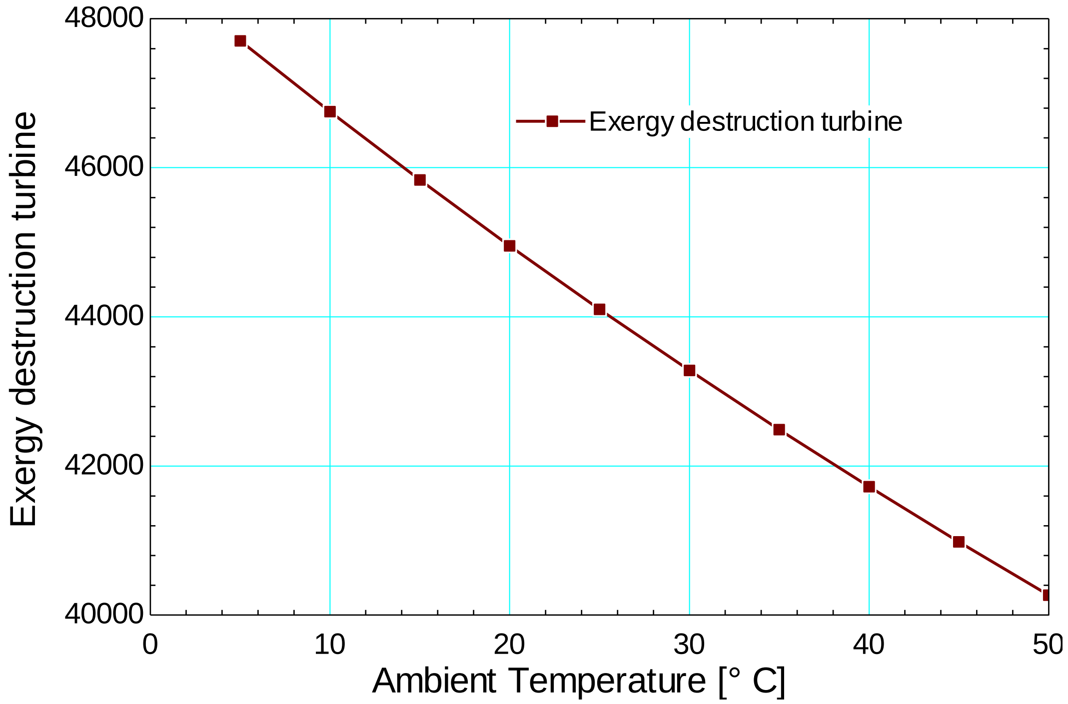

Figure 8: Shows the relationship between the temperatures of the outer ocean with the Exergy destruction turbine

The destruction of the economizer at low pressure can also be calculated using the following equation

$$ED_{lp\_eco} = ED_{g6} - ED_{g7} + ED_{w1} - ED_{w2}$$

The destruction of the evaporator at low pressure can also be calculated using the following equation

$$ED_{lp\_eva} = ED_{g5} - ED_{g6} + ED_{w3} - ED_{w4}$$

The destruction of the super heater at low pressure can also be calculated using the following equation

$$ED_{lp\_sh} = ED_{g4} - ED_{g5} + ED_{w5} - ED_{w6}$$

The destruction of the economizer at high pressure can also be calculated using the following equation

$$ED_{hp\_eco} = ED_{g3} - ED_{g4} + ED_{w7} - ED_{w8}$$

The destruction of the evaporator at high pressure can also be calculated using the following equation

$$ED_{hp\_eva} = ED_{g2} - ED_{g3} - ED_{w10} + ED_{w9}$$

The destruction of the super heater at high pressure can also be calculated using the following equation

$$ED_{hp\_sh} = ED_{g1} - ED_{g2} + ED_{w12} - ED_{w11}$$

And to calculate the energy-induced destruction in the total energy recovery in energy recovery would be

$$ED_{HRSG\_TOT} = ED_{lp\_eco} + ED_{lp\_eva} + ED_{lp\_sh} + ED_{hp\_eco} + ED_{hp\_eva} + ED_{hp\_sh}$$

Figures 9 and 10 shows the relationships between the available exergy efficiency and the destructive exergy in each part of the heat recovery evaporator with the temperature of the outside environment, as some parts increase directly and others inversely.

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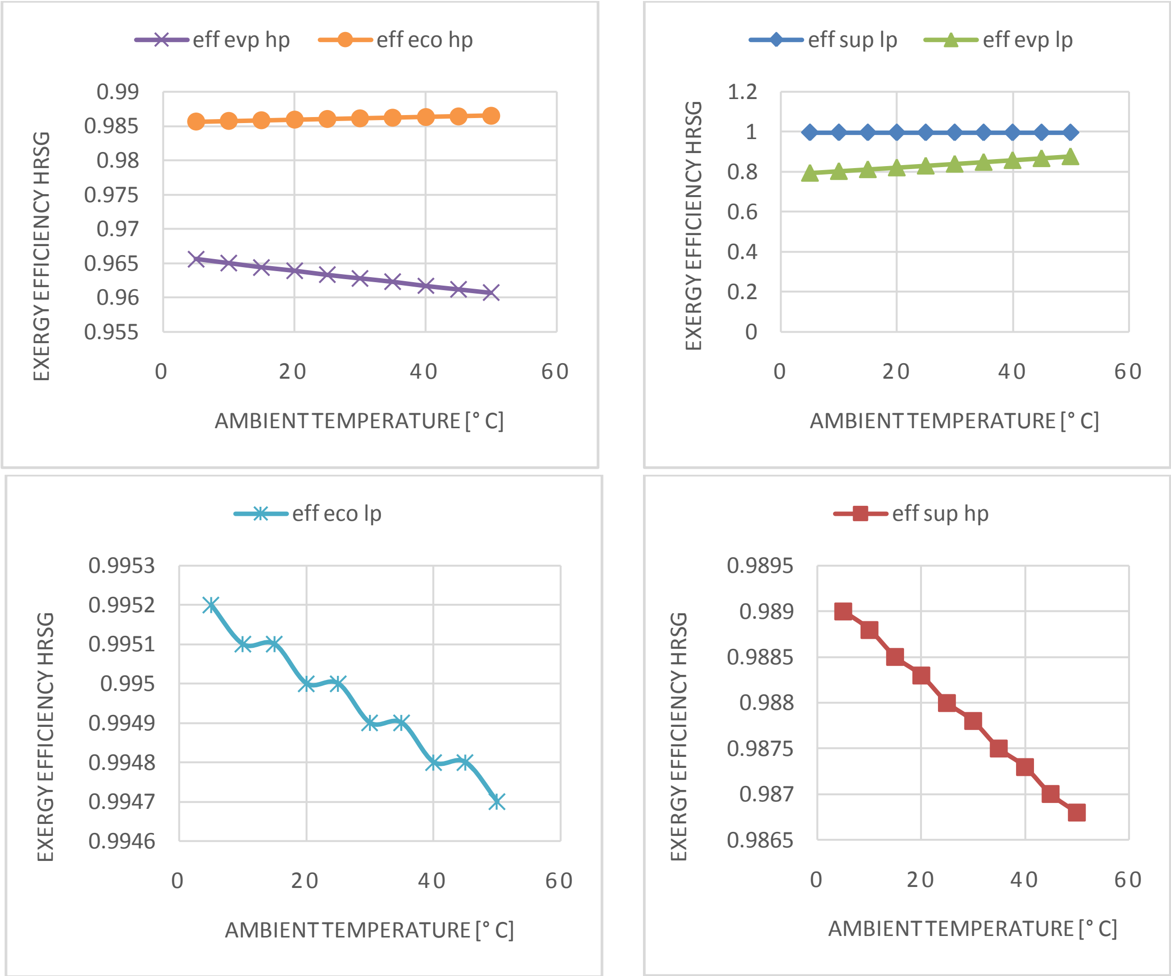
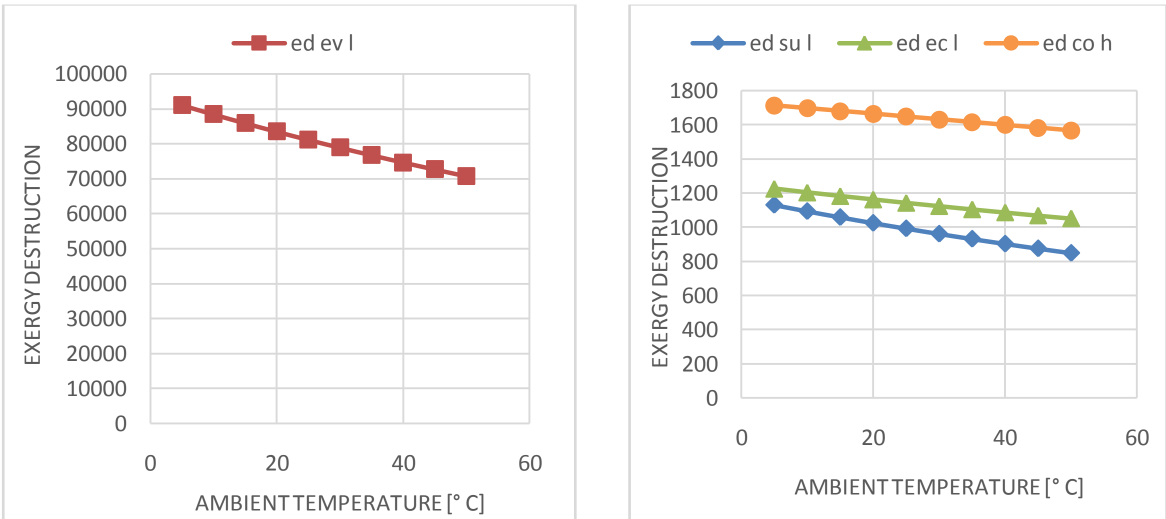


Figure 9: Shows the relationship between the ambient temperature and the available exergy efficiency



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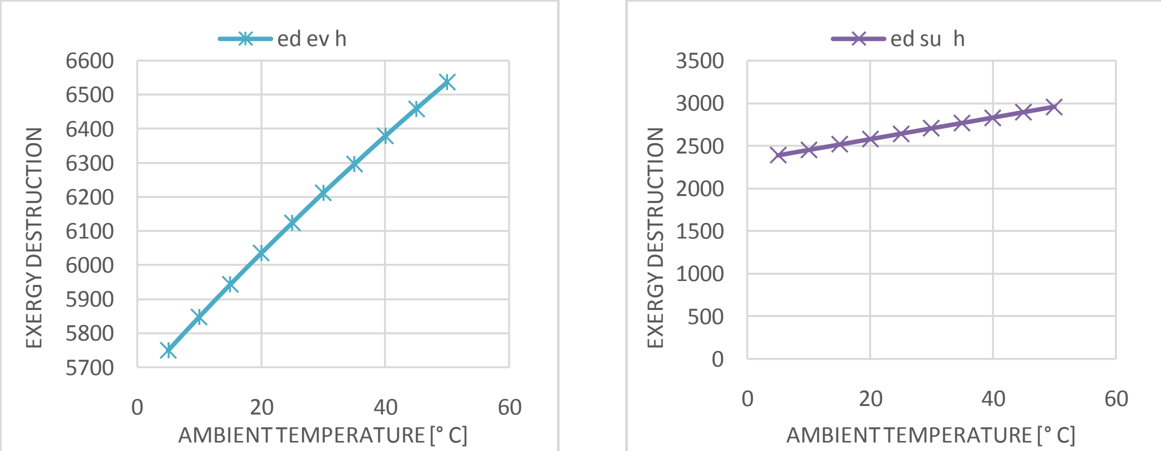

Figure 10: Shows the relationship between the temperature of the external environment and the destructive exergy

Figures 11 and 12 describe the relationship between the external ambient temperature with the total destructive oxygen of the heat recovery evaporator and its total efficiency, as the relationship is inverse with the destructive elixir that decreases with the increase and direct with the efficiency increases with the increase.

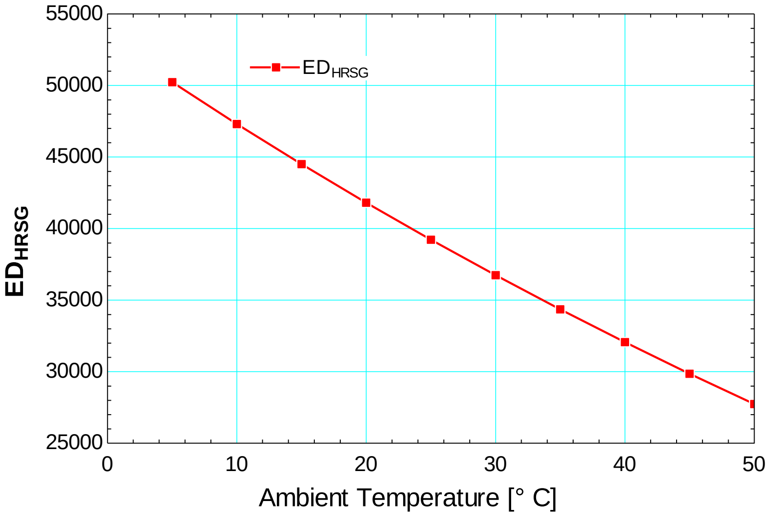

Figure 11: Shows the relationship between the temperature of the external environment and the destructive exergy total

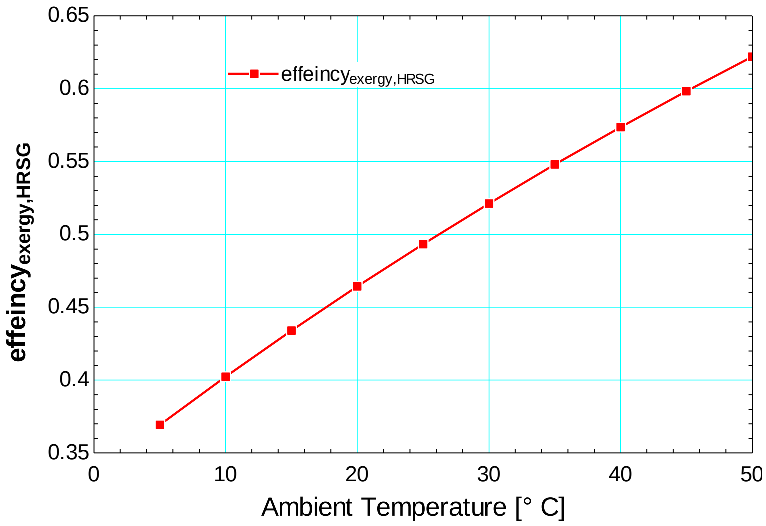

Figure 12: Shows the relationship between the ambient temperature and the available exergy total efficiency

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V. Conclusion

In this paper, the energy loss is calculated for each part of the heat recovery steam generator. The energy loss of the first HP-EVP has been shown to be greater than the energy loss for training the other components for the combustion and non-stress conditions. The effect of changes in ambient temperature on the rate of energy loss of a heat recovery steam generator under three different ambient temperatures are calculated. The rate of heat recovery energy loss (HRSG) has been shown to be the smallest for design conditions (ie5 °C). Exergy analysis also shows that using duct stoves will increase total energy loss. It can also be concluded that by increasing the inlet flue gas temperature, the steam cycle and the efficiency of the steam turbine are also increased.

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