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Thermal Performance Modeling and Analysis of a Simple Gas Turbine and Single-Pressure HRSG for Combined Cycle Operation Under Varying Ambient Conditions

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Abstract:

Combined power plants are more efficient in converting thermal energy into electrical energy due to the advancement of technology in the world. A simple gas turbine unit is integrated with a steam unit to form a combined cycle unit, utilizing a Heat Recovery Steam Generator (HRSG) to recover waste heat from the gas turbine exhaust and generate additional power without additional fuel consumption The present study aims to model a simple gas turbine power generation plant along with a single-pressure Heat Recovery Steam Generator (HRSG) and conduct a thermal analysis under varying operating conditions, specifically ambient temperature and atmospheric pressure (1 bar). The study focuses on the gas power generation plant in the Al-Qayyarah district, south of Mosul, which is designed with a generation capacity of up to (125 MW). The objective is to determine the operational parameters of the single-pressure HRSG in relation to its engineering design, should the gas turbine plant be converted into a combined cycle unit. This analysis considers energy balance, the mass flow rate of exhaust gases from the gas turbine, and the steam production rate from the HRSG. The results indicate that an increase in ambient temperature at constant atmospheric pressure negatively impacts the performance of the simple gas turbine unit. Specifically, thermal efficiency, power generation capacity, and exhaust gas mass flow rate decrease, while exhaust gas temperature and fuel consumption rate increase. The results also showed that increasing ambient temperature negatively affects the performance of the combined cycle power generation unit by reducing the gas turbine power output due to lower air density. However, it leads to an increase in exhaust gas temperature, which may slightly improve HRSG effectiveness and steam production but generally results in a decline in overall cycle efficiency and generating capacity. Analysis of the results indicates that a significant amount of thermal energy, approximately 60%, is lost with the exhaust gases from the combustion process in the gas turbine unit. However, this waste heat can

be recovered and utilized to generate additional power and improve overall thermal efficiency by integrating the gas turbine with a steam unit through a Heat Recovery Steam Generator (HRSG), forming a combined cycle power generation system.

Keywords: Gas Turbine Cycle, Ambient Temperature, Exhausts Temperature, HRSG, Combined Cycle

Introduction:

The global demand for energy, particularly in developing countries, has been increasing significantly due to economic growth, industrial expansion, high population growth, and infrastructure development. Thermal power plants, including steam power plants, simple gas turbine units, and diesel generators, play a crucial role in meeting this demand. In recent years, the use of gas turbines for power generation has increased due to their favorable capital cost-to-energy ratio, high operational flexibility, and reliability in simple cycle configurations, as well as their relatively high thermal efficiency. However, in a simple gas turbine unit, a significant portion of the thermal energy generated from hydrocarbon fuel combustion is lost to the environment through exhaust gases, with only (29-38%) of the total energy being converted into electrical power [1].

To address the increasing energy demand and improve efficiency, researchers and engineers have explored advanced technologies for recovering waste heat from gas turbine exhaust. One of the most effective solutions is the Combined Cycle Power Plant (CCPP), which integrates a Brayton cycle (topping cycle) with a Rankine cycle (bottoming cycle) to maximize energy conversion efficiency [2]. The two cycles are connected through a Heat Recovery Steam Generator (HRSG), as shown in (Fig. 1).

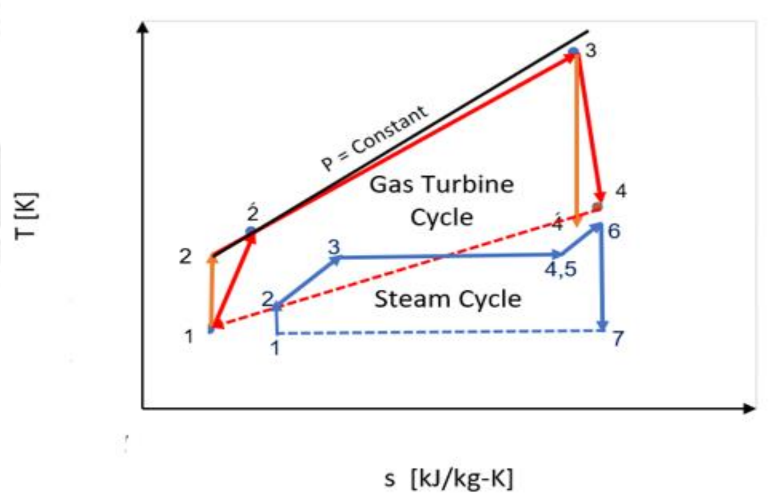

Fig. 1 Shows the T-s diagram of Combined Cycle Power Plant with a Single Pressure HRSG

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Exhaust heat from the gas turbine is recovered in a Heat Recovery Steam Generator (HRSG) to produce steam at suitable pressure and temperature. The produced steam is then used to produce further electricity in steam turbines. HRSGs are classified into single, dual, and triple pressure types depending on the number of drums in the boiler (see Fig. 2).

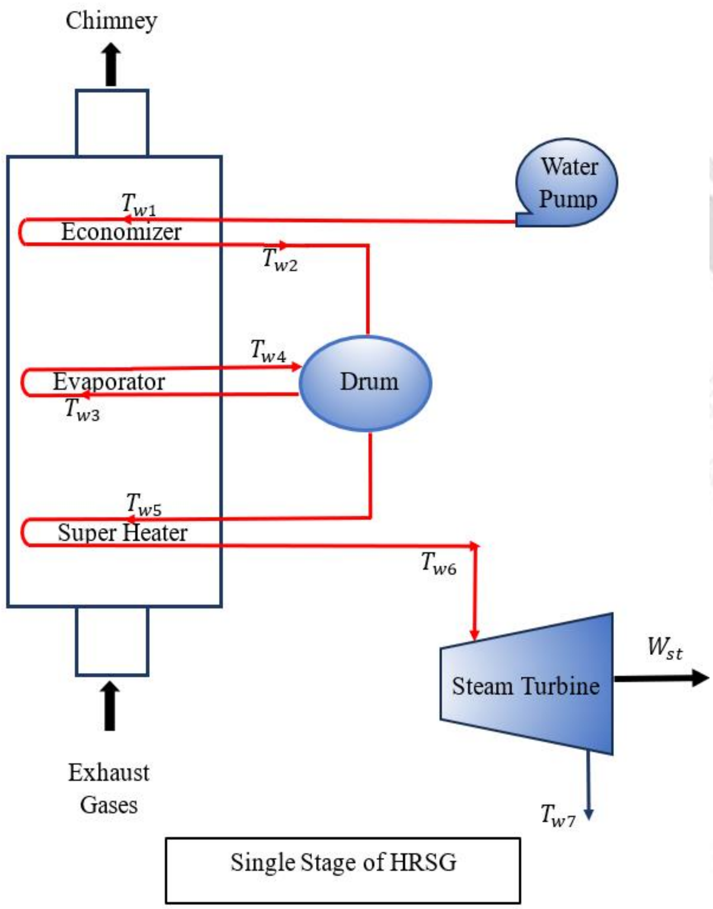

Fig. 2 Shows the Sketch of Combined Cycle Power Plant with a Single Pressure HRSG

The advancement in gas turbine technology has led to the widespread use of combined-cycle generating units based on gas turbines as the primary engine for combined-cycle generating units. The combined-cycle generating unit not only meets the electricity needs and requirements, but also provides heating and cooling for buildings by incorporating absorption chillers. Moreover, the steam generation systems (HRSG). The process of merging the gas unit and forming a combined generation unit reduces fuel consumption and emissions in the simple gas unit [4-5]. Recirculating steam generation (HRSG) systems have emerged as very effective solutions, and have been widely used in cogeneration applications, steam injection for combustion, and most commonly In combined cycle power plants, the waste heat is transferred with the exhaust gases of the gas turbines to the working fluid (water) inside the steam generation system (HRSG), where the fluid is converted into steam for use in the steam turbines of the steam generating unit. This process contributes to high efficiency. The lower energy loss and longer life of combined cycle power plants make them the preferred choice for generating electricity [2-3].

Many researchers have focused their studies on improving the performance of Combined Cycle Power Plants (CCPPs) by optimizing the operational parameters of Heat Recovery Steam Generators (HRSGs).

Omer Talal Ibrahim [1] addressed a theoretical study on evaluating the optimal performance of the combined generation unit and applying it to the simple gas generation unit by comparing the use of different types of steam generation systems (HRSGs). He concluded that the use of the combined generation unit significantly improves the energy production and thermal efficiency of the simple gas unit.

Ali Bahbahani [2, 3] et al. conducted a theoretical research and study to improve the heat exchanger tubes in steam generation systems for combined generation units to know the effect of operational factors and engineering variables on the performance of the single-pressure steam generation system. The study of the above researchers focused on knowing the effect of the pinch point temperature difference and the exhaust gas temperature, which was represented by operational factors, and the results obtained were an increase in thermal efficiency and energy efficiency.

Shahd Salem Ibrahim [4] conducted a theoretical study on evaluating the performance of combined gas units using the technology of injecting steam generated from a single-pressure steam generation system. The researcher concluded that an increase in ambient temperature leads to a decrease in air mass flow rate, power generation capacity, and thermal efficiency, while increasing the specific fuel consumption of the unit. Additionally, she observed a significant impact of the approach temperature and pinch point temperature on the performance of the steam generation system and the power output of the steam unit.

Ahmed Shams El-Din Ahmed [6] et al. modeled the components of the evaporator and drum of the steam generation system (HRSG). The researchers focused in the model on the dynamic behavior of water and steam within the steam generation system. The simulation results demonstrated the model's ability to predict the dynamic response of the steam generator drum and evaporator under different conditions, providing insights into improving the operation of the combined cycle generation unit and reducing wear on the equipment.

Hussien Sultan [7] et al. focused their study on evaluating the performance of a gas-fired power plant, in terms of the thermal and economic effects of the ambient temperature on the efficiency and production of the power plant using thermal and economic analyses. The researchers concluded that the ambient temperature significantly affects the efficiency of the gas-fired power plant and its energy output. They also concluded that the total cost of electricity generation decreases when using the combined cycle.

Ighodaro [8] et al. presented a theoretical study that included a thermal and economic analysis of upgrading a gas generating unit by merging it with a steam generating unit, to form a combined generation unit. The researcher used real data for the generating unit in the analysis and concluded that the process of merging the gas and steam generating units increases energy production and raises the overall efficiency of the combined generation unit.

AL Majali [9] et al. studied the performance and efficiency of combined-cycle generating units, which combine gas and steam generation units to maximize thermal efficiency. By systematically examining the effect of key factors on the performance of the combined-cycle unit. The researchers concluded that increasing the gas turbine inlet temperatures and improving the pressures of the steam generator (HRSG) could significantly enhance the thermal efficiency of the combined-cycle unit.

Mostafa A. Elhosseini [10] et al. studied the phenomenon of shrinkage and swelling of working fluid molecules in the parts of steam generators (HRSGs). The researcher's goal was to improve control and management methods in steam generation systems. The researcher reached an improvement in system stability and a reduction in the risks associated with the effects of shrinkage and swelling.

Pathum Thani [11] et al. conducted a study focusing on the optimization of single pressure HRSG systems in a combined cycle unit. The researcher conducted a comprehensive analysis of heat transfer in the parts of the steam generation system (primary superheater, secondary superheater, evaporator, and economizer), and the effect of temperature differences between the pinch point and the approach point. The researcher concluded that as the surface area of the steam generation system increased, the energy output increased but at a decreasing rate. They also highlighted the need for economic optimization. They also concluded that various factors such as electricity prices, heat exchanger costs, capacity factor of the combined cycle unit, and required heat exchange period had an impact on the optimum total area of the steam generation system, and that careful design and selection of factors could significantly improve the overall efficiency and financial feasibility of the combined cycle unit.

The aim of the current research is to conduct a theoretical study by working on conducting a thermal analysis of the operating conditions of the gas unit in the Qayyarah district, south of Mosul city, which operates with a generating capacity of up to (125) megawatts, to know the extent of the impact of these conditions on the performance of the gas generation station and the possibility of converting it into a combined generation station by integrating it with a steam generation station by means of a steam generation system (HRSG) that operates under single pressure.

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Methods of Analysis:

1- Thermodynamic modeling of simple gas turbine cycle based on the T-s diagram (Fig. 3):

Thermodynamic modeling of the simple gas turbine cycle is based on the T-s diagram (Fig. 3). To perform the simulation, several thermodynamic parameters must be known, such as air mass flow rate, temperature, and pressure. Additionally, design specifications, technical cycle characteristics, and the chemical composition of the fuel used in the combustion process are essential input parameters for the analysis.

To facilitate the simulation process, the following assumptions were made:

a- The simple gas turbine cycle operates under steady-state conditions

b- The temperature of the combustion gases entering the gas turbine is assumed to be approximately constant under steady-state conditions, considering regulated cooling and air-fuel ratio control.

c- The rotational speed of the gas turbine remains constant.

d- The efficiency of combustion chamber, air compressor and gas turbine are assumed to be (100, 90, 85) % respectively.

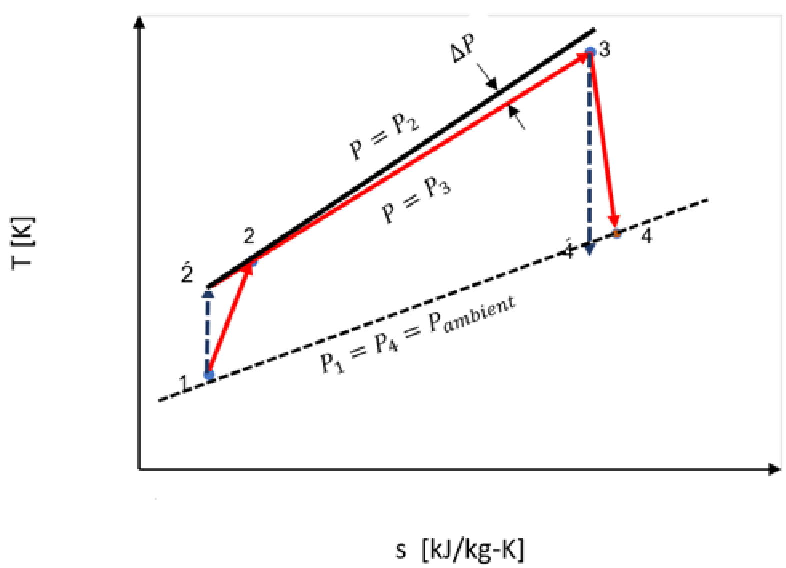

Fig. (3) Shows the T-s Diagram of a Simple Cycle.

<u>Air compressor simulation</u>:

The air pressure resulting from the compression process is calculated using the following mathematical equation:

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$$P_2 = r_{pc} * P_1 \quad (1)$$

In which:

P_1: Ambient Pressure (1 bar)

r_{pc}: Pressure Ratio

P_2: Compressed Air Pressure

From Fig. (3), researchers notice that the compression process of the air coming out of the compressor can be an ideal process at point $2^{'}$. Therefore, the ideal temperature is calculated, and then the actual temperature of the compressed air at point 2 is found, respectively, from the following equations:

$$\acute{T}_2 = T_1 * \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \quad \ldots (2)$$

In which:

$\acute{T}_2$: Isontropic Compressed Air Temperature

T_1 : Ambient Air Temprature (288 K)

$\gamma_a = \frac{Cp_a}{Cv_A}$: Gamma of Compressed Air

$$T_2 = T_1 * \left\{1 + \left(\frac{1}{\eta_{isc}}\right) * \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right]\right\} \quad \ldots (3)$$

In which:

T_2: Actual Compressed Air Temperature

η_{isc} : Mechanical Efficiency of Compressor

The following equation is used to calculate the work done to rotate the air compressor:

$$\dot{W}_c = \dot{m}_a * Cp_a * (T_2 - T_1) \quad \ldots (4)$$

In which:

$\dot{W}_c$: Compressors Work

$\dot{m}_a$: mass flow rate of Compressed Air

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Cp_a : Specific Heeat of Air at Constant Pressure

The following equation is used to calculate the specific heat of air using the properties available in the simulation program (EES) in terms of temperature: [5]

$$C_{pa}(T) = 1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.49031T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right) \quad \dots (5)$$

The following equation is used to calculate the compressed air density rate:

$$\rho_a = \frac{P_2}{R_a * T_2} \quad \dots (6)$$

In which:

ρ_a: Air Density

R_a: Constant Gases

The following equation is used to calculate the compressed air mass flow rate:

$$\dot{m}_a = \rho_a * \dot{v}_a \quad \dots (7)$$

In which:

$\dot{v}_a$: volume flow rate of Air

A_{ic}: Cross Sectional Area of Inlet Air to Compressor

Combustion Chamber Simulation:

This research relies on methane gas as a fuel consumed in the gas power generation unit and its chemical formula is (CH4). Through the combustion equation, the number of moles entering the combustion process with compressed air via the air compressor is found, as well as the number of moles of combustion products is obtained due to its importance in determining the chemical elements in the combustion chamber, which in turn determines the properties of the gases exiting the combustion chamber and entering the gas turbine. The combustion equation for one mole of hydrocarbon fuel combustion can be represented in the stoichiometric form with the amount of compressed air as follows:

$$C_{x1}H_{y1} + \left(x_1 + \frac{y_1}{4}\right)(O_2 + 3.76N_2) \rightarrow x_1CO_2 + y_1H_2O + 3.76N_2 \quad \dots (8)$$

When representing the combustion of one mole of hydrocarbon fuel with an additional amount of compressed air in actual form, it is written as follows:

$$C_{x1}H_{y1} + \lambda\left(x_1 + \frac{y_1}{4}\right)(o_2 + 3.76N_2) \rightarrow x_1CO_2 + \frac{y_1}{2}H_2O + (\lambda - 1)\left(x_1 + \frac{y_1}{4}\right)O_2 + 3.76\lambda\left(x_1 + \frac{y_1}{4}\right)N_2 \quad (9)$$

Both chemical equations (8) and (9) are used to perform all calculations and simulate the combustion chamber of the generating unit in this current research. Through them, the value of the coefficient λ was calculated using the following equation:

$$\Phi = \frac{\left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{stoichiometric}}{\left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{Actual}} \quad (10)$$

Φ : Equivalence Ratio for The Combustion Process

$$\lambda = \frac{1}{\Phi} = \frac{\left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{Actual}}{\left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{stoichiometric}} \quad (11)$$

In which:

λ : Lamda

$\dot{m}_f$: mass flow rate of fuel

The following equation is used to calculate the mass flow of fuel entering the combustion chamber:

$$\dot{m}_f = \frac{\lambda \dot{m}_a M_f}{M_a} \quad \ldots (12)$$

In which:

M_f : Molecular Weight of Fuel

M_a : Molecular Weight of Air

The following equations is used to calculate the number of moles of each element of the combustion products that represent the exhaust gases:

$$Y_{CO2} = x_1 \qquad Y_{H2O} = \frac{y_1}{2} \quad \ldots (13a)$$

$$Y_{O2} = (\lambda - 1)\left(x_1 + \frac{y_1}{4}\right), \qquad Y_{N2} = 3.76 * \lambda\left(x_1 + \frac{y_1}{4}\right) \quad \ldots (13b)$$

Assuming that the temperature at point 2 in Figure (1-3) (T_2) is equal to the temperature of the combustion chamber during mixing of the fuel and compressed air mixture before the combustion process, and that the temperature of the gases exiting the combustion chamber at point 3 is the same as the adiabat flame temperature, which can be calculated using the fluid properties in the program (EES) in the following equation:

$$\Delta H_P = \Delta H_R \qquad \dots(14)$$

The specific heat at constant pressure of the gases exiting the combustion chamber is calculated from the equation: [5]

$$Cp_g(T) = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.2244T^3}{10^{10}}\right) \qquad \dots(15)$$

In which:

Cp_g: specific Heat of Gases at Constant Pressore

To calculate the mass flow rate of the mixture of fuel and compressed air (gases) exiting the combustion chamber and entering the gas turbine, we use the following equation:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \qquad \dots(16)$$

In which:

$\dot{m}_g$: mass flow rate of gases

The chemical reaction resulting from the combustion process is accompanied by high temperature and high pressure, which leads to the addition of additional heat to the system with stable flow and constant pressure, which can be calculated using the following mathematical equation:

$$\dot{Q}_{CC} = \eta_{CC} \cdot \dot{m}_g (h_{g3} - h_{g2}) \qquad \dots(17)$$

In which:

$\dot{Q}_{CC}$: Heat Add at Combustion Chamber

η_{CC} : Efficiency of Combustion Chamber

$(h_{g3} - h_{g2})$: The difference between the exit and entry enthalpy of the combustion chamber.

<u>Gas Turbine Simulation</u>:

For this stage, the ideal temperature of the exhaust gases at point $4'$, which represents an adiabatic process, is calculated using the following laws:

$$P_3 = P_2 - 0.02P_2 \quad ; \quad P_4 = P_1 \qquad \dots(18)$$

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$$\acute{T}_4 = T_3 * \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_4 - 1}{\gamma_4}} \quad \ldots (19)$$

In which:

P_3 : Gas Turbine Inlet Pressure

P_4 : Gas Turbine Exhuast Pressure

γ_4 : Gamma of Exhuast Gases

T_3 : Gases Temperatur Inlet to Gas Turbine

$\acute{T}_4$: Isontropic Exhuast Gases

The following equation is used to calculate the actual temperature of the exhaust gases at point 4:

$$T_4 = T_3 - \eta_{isGT}(T_3 - \acute{T}_4) \quad \ldots (20)$$

In which:

T_4: Exhuast Gases Temperature

η_{isGT}: Mechanical Efficiency of Gas Turbine

To calculate the work produced by the gas turbine, one of the following laws can be used:

$$\dot{W}_{GT} = \dot{m}_g(h_{g3} - h_{g4}) \quad \ldots (21)$$

In which:

$\dot{W}_{GT}$: Work of Gas Turbine

Or using the following law:

$$\dot{W}_{GT} = \dot{m}_g C_{pg4}(T_3 - T_4) \quad \ldots (22)$$

To calculate the network, which is considered as the profit of the gas power generation unit, use the following law:

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_C \quad \ldots (23)$$

In which:

$\dot{W}_{net}$: Net work of Gas Turbine Cycle

To calculate the generating capacity, thermal efficiency and specific fuel consumption rate of a simple gas generating unit, respectively, the following law can be used:

$$PE_{GT} = \dot{W}_{net} \quad ...(24)$$

$$\eta_{thGT} = \frac{PE_{GT}}{\dot{m}_f . LHV_{fuel}} \quad ...(25)$$

$$SFC_{GT} = \frac{\dot{m}_f * 3600}{PE_{GT}} \quad ...(26)$$

In which:

PE_{GT}: Power of Gas Turbine Cycle

η_{thGT} : Thermal Efficiency of Gas Turbine

LHV_{fuel} : Lower Heat Value

SFC_{GT} : Specific Fuel Consumption

The cycle simulation process is done by programming the above equations using the (EES) program and the method of successive substituting after imposing and entering the values of the inputs known to the program such as the thermal factors of the weather conditions such as the air temperature and the atmospheric pressure of the air entering the air compressor and the minimum calorific value of the fuel used in the combustion process and the compression ratio between the air pressure entering the air compressor and exiting it and the pressure gap ratio that occurs in the combustion chamber as a result of friction and imposing and entering some technical specifications into the simulation program represented by the mechanical efficiency of the air compressor and the gas turbine.

2- Thermodynamic modeling of a heat recovery steam generator based on the schematic diagram in Fig. (2):

In modeling a heat recovery steam generator, many thermodynamic parameters of the exhaust gases from the basic gas turbine mode must be known, such as (mass flow rate, temperature, pressure, and chemical composition), which are inlet parameters of the heat recovery steam generator.

To facilitate the computational analysis of the governing equations in the design of the three types of steam generation system (single-pressure, double-pressure, triple-pressure) and to develop models in general for the compression stage, the following hypotheses were developed:

a- The system operates as an **Unfired HRSG**, meaning the heat available for steam generation is solely derived from the exhaust gases of the gas turbine, without any additional combustion.
b- The analysis assumes steady-state conditions where the thermal properties of the exhaust gases (temperature, pressure, and mass flow rate) remain constant, thereby eliminating thermal fluctuations.
c- The cooling of exhaust gases is considered only due to heat exchange within the HRSG heat exchangers, without any external cooling sources.
d- Radiative heat transfer is neglected due to the small spacing between the heat exchanger tubes, which minimizes its contribution compared to convection and conduction [5].
e- The stack temperature is maintained above 100°C to prevent the condensation of sulfur dioxide (SO_2) into sulfur trioxide (SO_3), which can lead to acidic corrosion of heat exchanger materials. For the single-pressure HRSG system, the pinch point temperature ($T_{P.P}$) is set to 10°C, the approach temperature ($T_{Ap.P}$) is 6°C, and the triple temperature ($T_{tr.P}$) is 35°C.
f- The HRSG design is simulated under steady-state conditions, assuming that the simple gas power generation unit operates at standard ISO conditions, where the mass flow rate of the working fluid remains balanced across the economizer, evaporator, and superheater, i.e., ($\dot{m}_{ss} = \dot{m}_{sat} = \dot{m}_{wa} = \dot{m}_{\upsilon}$).

Energy Balance:

Determining the appropriate operational parameters for the steam generation system—such as the design steam pressure, temperature distribution, mass flow rate of the working fluid, and the amount of heat transferred—is essential for the engineering design of heat exchangers. Achieving an energy balance between the working fluid and the exhaust gases is a fundamental requirement in this process. Under steady-state heat exchange conditions, and assuming negligible heat losses to the surroundings, the heat lost by the exhaust gases is approximately equal to the heat gained by the working fluid. This relationship can be mathematically expressed as follows refer to Fig. 2:

$$\dot{Q}_g = \dot{Q}_{Sh} + \dot{Q}_{Ev} + \dot{Q}_{Eco} \qquad \text{(For Single stage)} \quad \dots (27)$$

In which:

$\dot{Q}_g$: Heat released with exhaust gases

$\dot{Q}_{Sh}$: Heat Obtained at The Super Heater

$\dot{Q}_{Ev}$: Heat Optained at Evaporator

$\dot{Q}_{Eco}$: Heat Optained at Economizer

Based on this, the heat gained in the Single Pressure HRSG steam generation system for the superheater, evaporator and economizer respectively is calculated according to the following equations:

$$\dot{Q}_{Sh} = \dot{m}_{ss}(h_{w6} - h_{w5}) = \dot{m}_g . Cp_g(T_{g1} - T_{g2}) \quad \dots (28)$$

$$\dot{Q}_{Ev} = \dot{m}_{sat}(h_{w4} - h_{w3}) = \dot{m}_g . Cp_g(T_{g2} - T_{g3}) \quad \dots (29)$$

$$\dot{Q}_{Eco} = \dot{m}_{wa}(h_{w2} - h_{w1}) = \dot{m}_g . Cp_g(T_{g3} - T_{g4}) \quad \dots (30)$$

Where:

$\dot{m}_{ss} = \dot{m}_{sat} = \dot{m}_{wa}$: Mass flow rate of working fluid

T_g : Temperature of gases entering HRSG

The energy balance equations are programmed using the Engineering Equation Solver (EES) program to find the properties of the working fluid and thus obtain the other variables of the exhaust gases and the working fluid as shown in the temperature distribution diagram in Fig. 4.

The temperature of the steam leaving the evaporator is found in terms of the steam pressure at the steam tank and the dry fraction value, the inlet temperature of the working fluid (water) is equal to its temperature when leaving the evaporator as follows:

$$T_{w4} = T\,(P_{desHP}, x_2) \quad (31)$$

$$T_{w4} = T_{w3} = T_{w5} \quad (32)$$

In which:

T_w : operating fluid temperature

The following equation is used to calculate the temperature of the exhaust gases passing through the evaporator:

$$T_{g3} = T_{w3} + T_{pp} \quad (33)$$

In which:

T_{pp} : Pinch Points Temperature

The following equation is used to calculate the temperature of the working fluid leaving the economizer:

$$T_{w2} = T_{w3} - T_{App} \quad (34)$$

In which:

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T_{App} : Approach Point Tempersture

The heat content of the working fluid entering and exiting the economizer through the properties in the simulation program (EES) is calculated as follows:

$$h_{w2} = h\,(T_{w2}, x_1) \tag{35}$$

$$h_{w1} = h\,(T_{w1}, P_{des}) \tag{36}$$

In which:

h_w : Enthalpy of Working Fluid

x_1 : Evaporator inlet dry break value

P_{des} : Design Pressure of Working Fluid in HRSG

As for the heat content of the working fluid entering and exiting the evaporator, it is calculated using the following relationships:

$$h_{w3} = h\left(T_{w3\,,}x_1\right) \tag{37}$$

$$h_{w4} = h(T_{w4}, x_2) = h_{w5} \tag{38}$$

In which:

x_2 : Evaporator outlet dry break value

To calculate the temperature and enthalpy of the steam coming out of the superheater, which represents the steam produced, the following relationship can be used:

$$T_{w6} = T_{g1} - T_{trP} \tag{39}$$

in which:

T_{trP} : Tripple Point Temperature

$$h_{w6} = h(T_{w6}, P_{des}) \tag{40}$$

By substituting the values of the thermal properties of the working fluid and the exhaust gases above in the energy balance equations (27), (28), (29) and (30), we have several unknowns of the operating variables, three in number, in two of the energy balance equations, represented by $(\dot{m_v}, Cp_g, T_{g2})$. The value of the variable representing the specific heat of the exhaust gases and the working fluid is taken based on the program characteristic (EES) at absolute temperature using the following relationship:

$$Cp_g = f\left(T_{gbulk}\right) \quad , \quad Cp_w = f(T_{wbulk}) \quad , \quad T_{bulk} = \frac{T_1 + T_2}{2} \tag{41}$$

To calculate the thermal efficiency, generating capacity, benefits of generating capacity of a combined cycle unit and effectiveness of a single-pressure steam generator, we use the following laws, respectively:

$$\eta_{combined\ cycle} = \frac{Power_{combined}}{\dot{m}_f * LHV\_fuel} \tag{42}$$

$$Power_{combined} = P_{simple\ cycle} + P_{steam\ cycle} \tag{43}$$

$$P_{steam\ cycle} = \dot{m}_s * (hw[6] - hw[7]) \tag{44}$$

$$benefit_{power} = \frac{Power_{combined} + Power_{gas\ cycle}}{Power_{gas\ cycle}} \tag{45}$$

$$\varepsilon_{HRSG} = \frac{\dot{m}_{steam} * (h_w[6] - h_w[1])}{\dot{m}_{gases} * Cp_g * \left(T_g[1] - T_g[2]\right)} \tag{46}$$

Where:

$\eta_{combined\ cycle}$: Efficiency of Combined Cycle

$Power_{combined}$: Power of Combined Cycle

$P_{steam\ cycle}$: Power of Steam Cycle

$benefit_{power}$: Benefit from generated power

ε_{HRSG} : Effectiveness of HRSG in Combined Cycle

The marine generation system is simulated by programming the energy balance equations and solving them to find the remaining operational variables using the Newton-Raphson Method for solving, which gives a better approximation to the solution than the successive substitution method. This method is also one of the simulation methods used to solve a set of nonlinear algebraic equations that are included in the Engineering Equation Solver (EES) program.

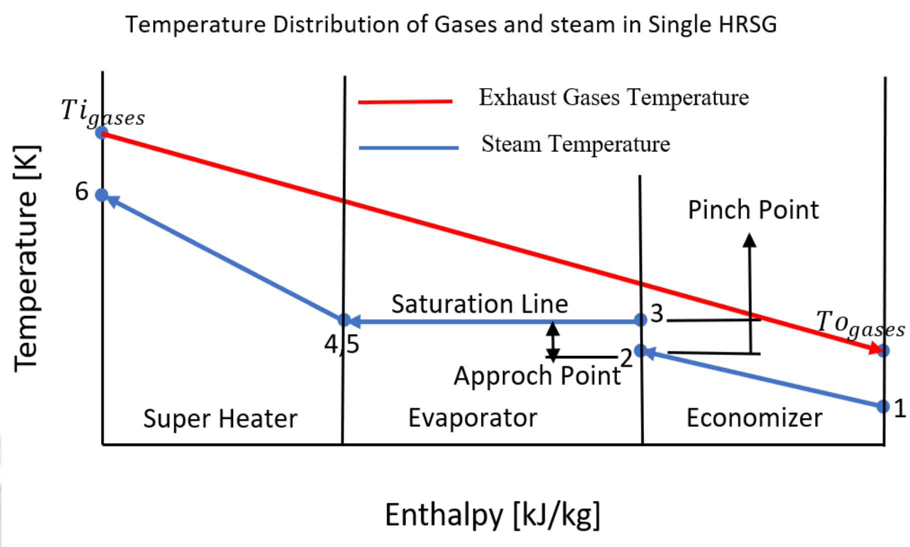

Fig. (4) Temperature Distribution Diagram in a Single-Pressure HRSG System.

Results and Discussion:

The results of the simple gas turbine unit analysis indicate that a significant amount of heat, approximately 60%, is lost with the exhaust gases from the combustion process. The study also confirms that operating conditions, particularly ambient temperature, significantly impact both simple and combined cycle units.

One of the most important results obtained due to the increase in the ambient temperature (288 - 315 K), i.e. approximately (15 - 45 ℃) with a constant pressure (1 bar) bar is the decrease in the flow rate of the compressed air mass entering the combustion chamber (411.3 - 372.5 kg/s) as in Fig. (5) due to the decrease in the relative density of the air, which causes a decrease in the net work done, which represents the difference between the work done by the gas turbine and the air compressor, which leads to a decrease in the generating capacity of the simple gas unit (107074 - 105820 kW) as in Fig. (6). An increase in the equivalence ratio for the combustion process was observed (0.2661 - 0.2938), which represents the ratio between the air mass rate over the fuel mass in the ideal case to the ratio of the air mass rate over the fuel mass rate in the real case as in Fig. (7). From observing Fig. (8), the increase in temperature also led to an increase in the fuel consumption rate for the simple gas unit (0.1933 - 0.1945). The exhaust gas mass flow rate decreased (417.1 - 378.2 kg/s) as in Fig. (5) due to the increase in its temperature (726.4 - 799.8 K) as in Fig. (9). From Fig. (10), we notice a decrease in the thermal efficiency of the simple gas unit (37.25 - 37.02 %) with the increase in ambient temperatures due to the decrease in the net generating capacity.

As for the combined unit, the most important operational factors were obtained, namely the temperature of the exhaust gases and their mass flow rate. Through the analysis of the combined generation unit, one of the most important results obtained was an increase in the net generating capacity (147144 - 157329 kW), as in Fig. (6). The reason is that the increase in the ambient temperature led to an increase in the temperatures of the exhaust gases exiting the gas turbine and entering the steam generation system, which increased the temperature of the produced steam, as in Fig. (9). From Fig. (10), we notice an increase in the thermal efficiency of the combined generation unit (51.19 - 55.04 %), and the reason is due to the increase in the generating capacity of the combined unit affected by the increase in the mass flow rate of the steam produced from the steam generation system (48.01 - 52.56 kg/s), as in Fig. (11). We also notice an increase in the extent of utilization of the generating capacity (0.3742 - 0.4868) as a result of the increase in the generating capacity of the combined unit and a decrease in the generating capacity of the simple gas unit as a result of the increase in the ambient temperature. From observing Fig. (12), we can see the effectiveness of the steam generation system in the combined generation unit, as it increases (0.788 - 0.8445) due to the increase in the temperature of the exhaust gases passing through it as a result of the increase in the ambient temperature.

Conclusions:

One of the most important conclusions reached in the current research:

1- An increase in ambient temperature at constant atmospheric pressure negatively impacts the performance of the simple gas turbine unit. Specifically, thermal efficiency, power generation capacity, and exhaust gas mass flow rate decrease, while exhaust gas temperature and fuel consumption rate increase.
2- That a significant amount of thermal energy, approximately 60%, is lost with the exhaust gases from the combustion process in the gas turbine unit. However, this waste heat can be recovered and utilized to generate additional power and improve overall thermal efficiency by integrating the gas turbine with a steam unit through a Heat Recovery Steam Generator (HRSG), forming a combined cycle power generation system.
3- That increasing ambient temperature negatively affects the performance of the combined cycle power generation unit by reducing the gas turbine power output due to lower air density. However, it leads to an increase in exhaust gas temperature, which may slightly improve HRSG effectiveness and steam production but generally results in a decline in overall cycle efficiency and generating capacity.

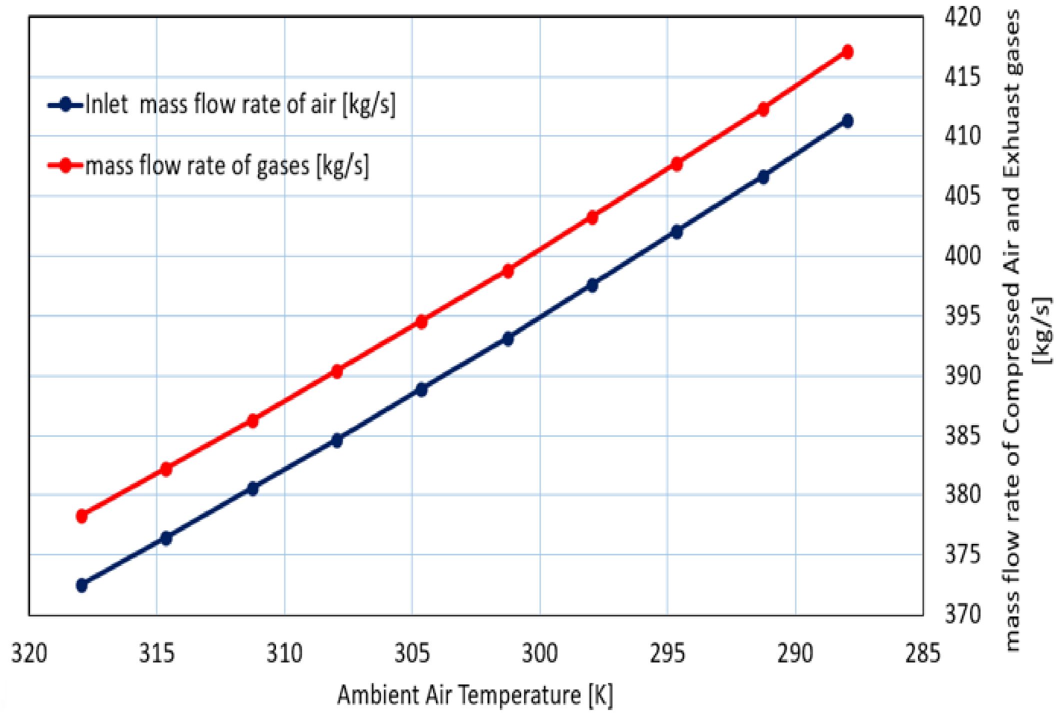

Fig. (5) Effect of Ambient Air Temperature to Mass Flow Rate of Compressed Air and Exhausts Gases.

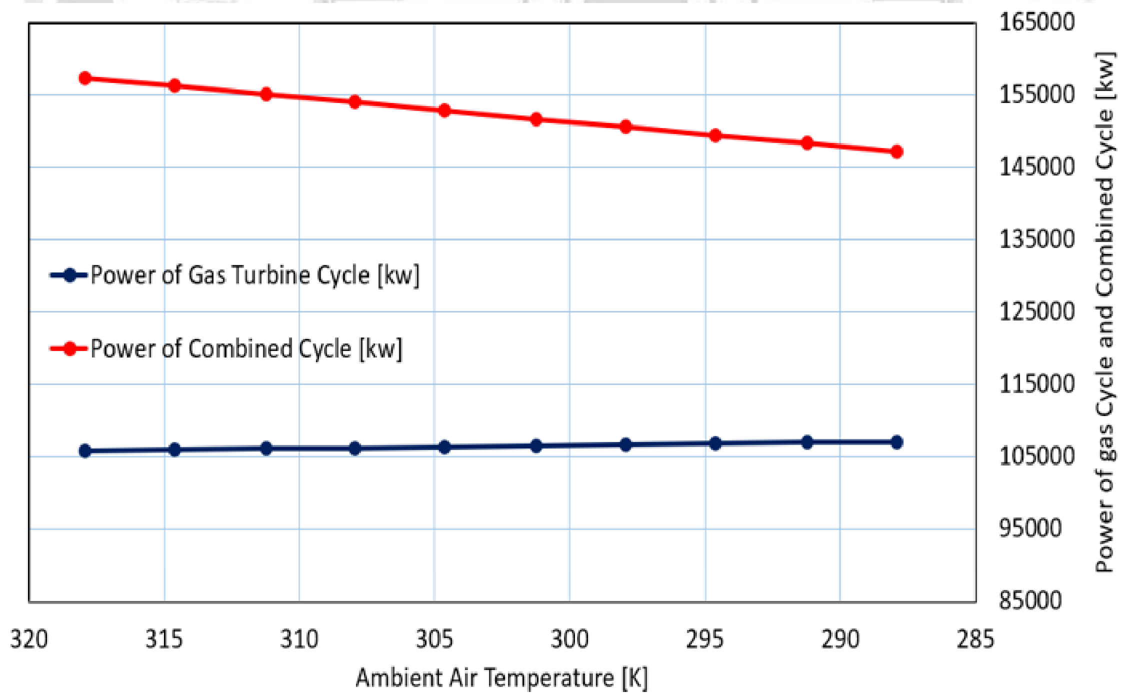

Fig. (6) Effect of Ambient Air Temperature on The Power of The Cycles.

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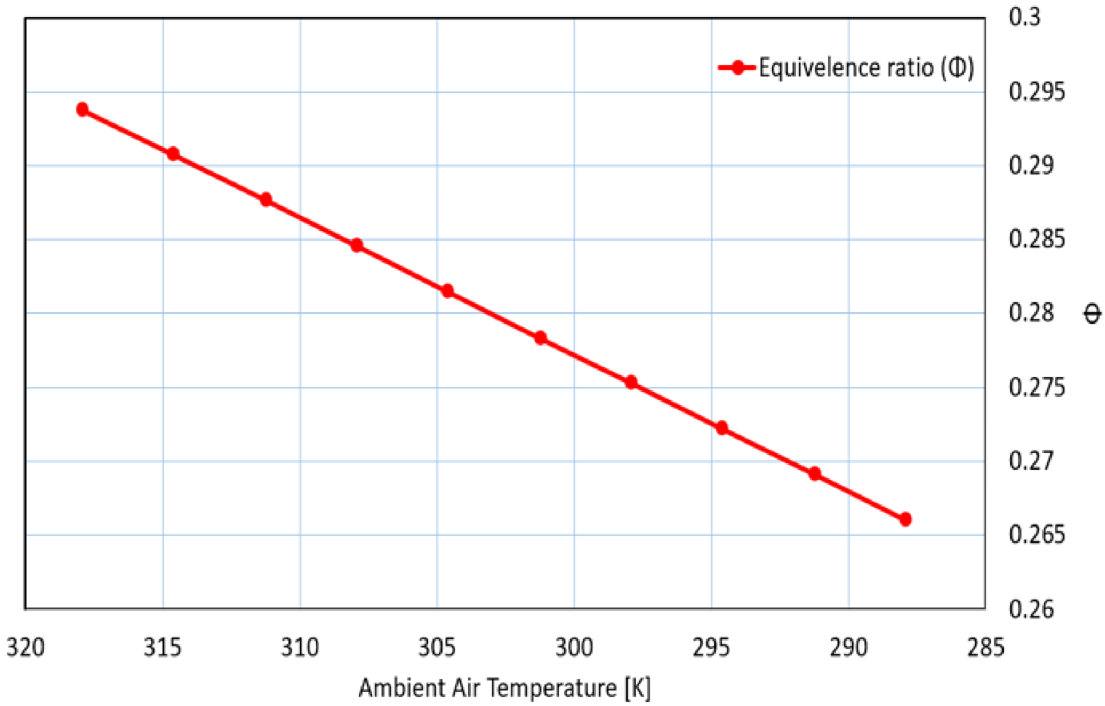

Fig. (7) Effect of Ambient Air Temperature to Equivalence Ratio.

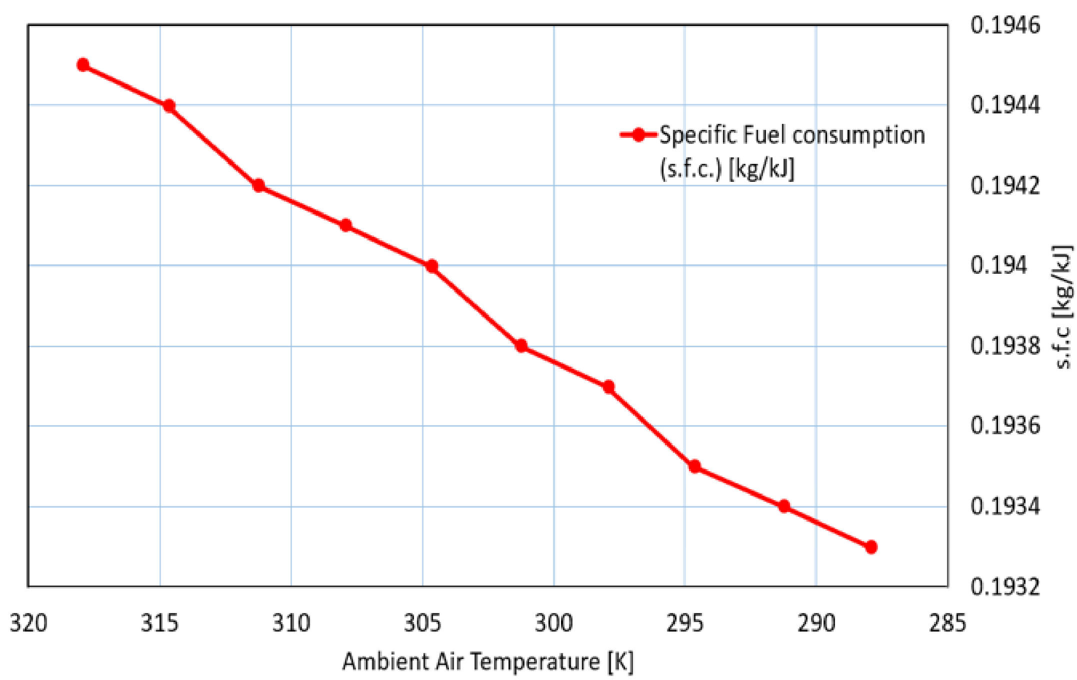

Fig. (8) Effect of Ambient Air Temperature to Specific Fuel Consumption.

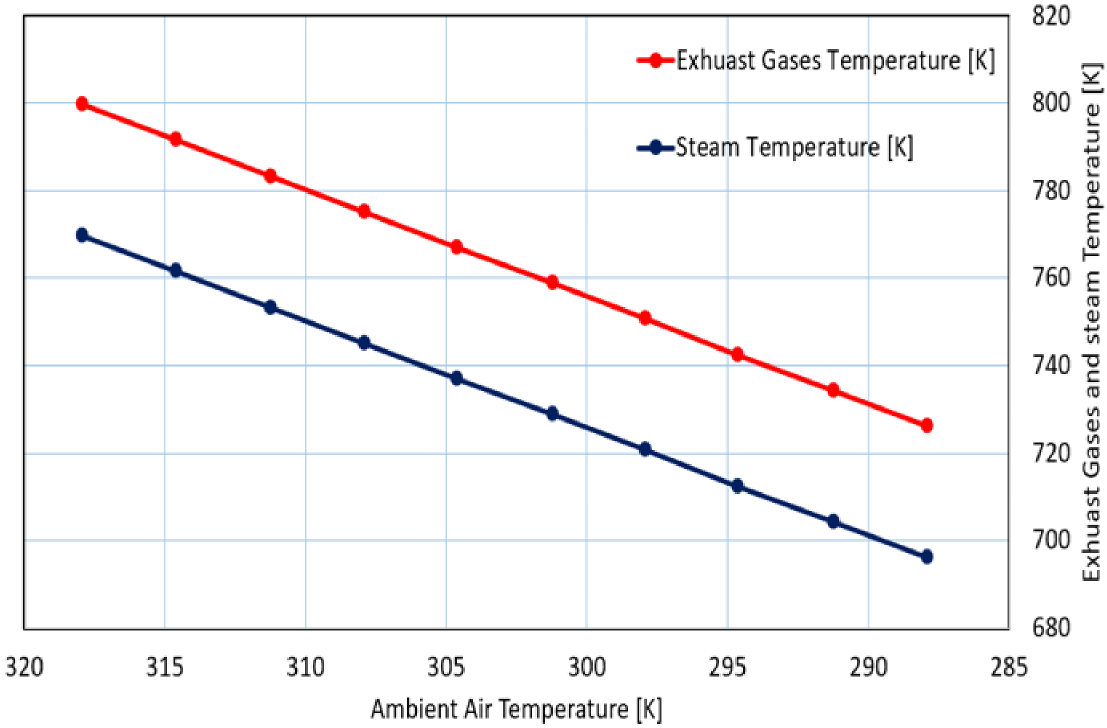

Fig. (9) Effect of Ambient Air Temperature to Exhausts Gases and Steams Temperature.

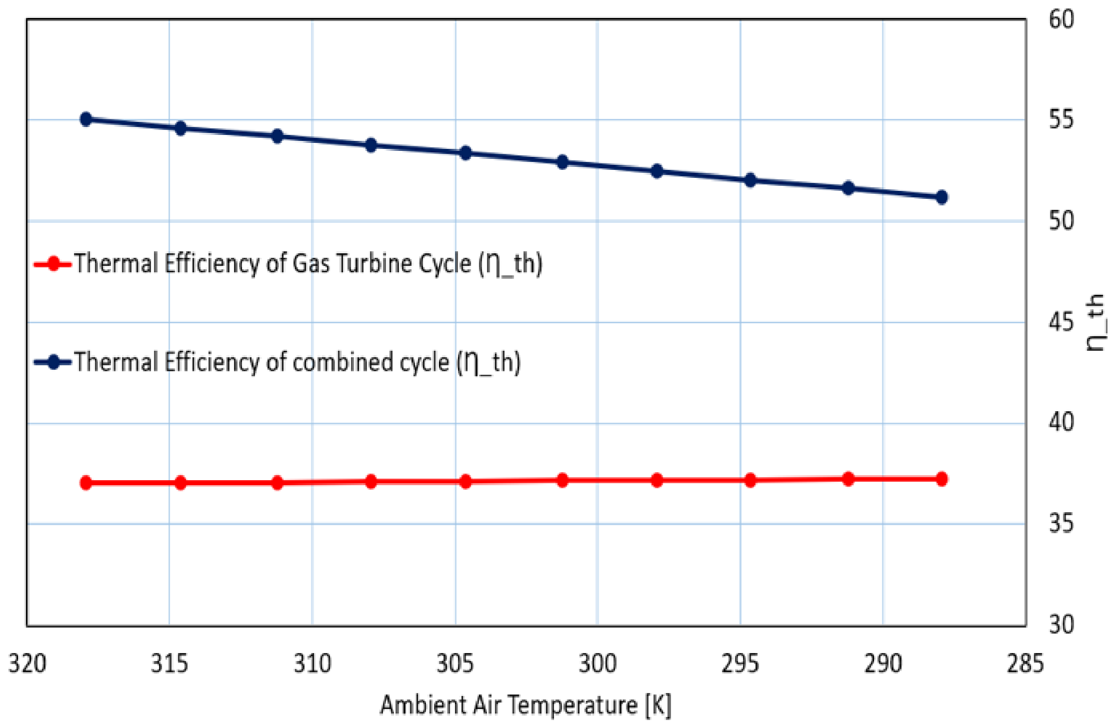

Fig. (10) Effect of Ambient Air Temperature to Thermal Efficiency of Cycle.

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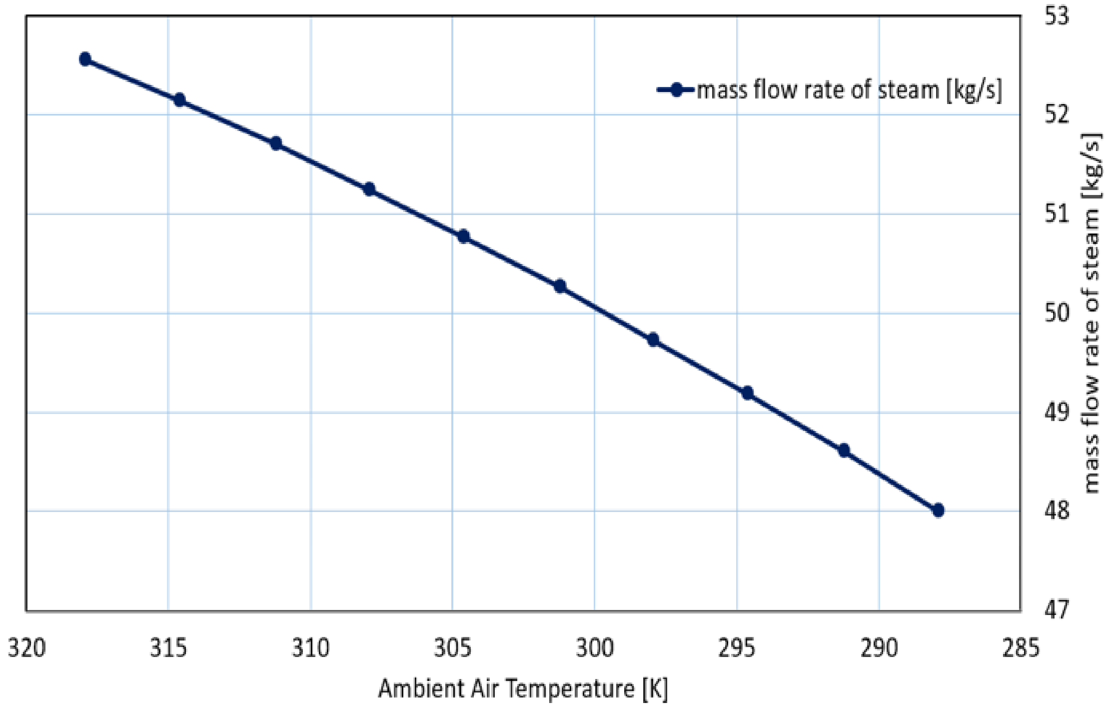

Fig. (11) Effect of Ambient Air Temperature to Mass Flow Rate of Steam.

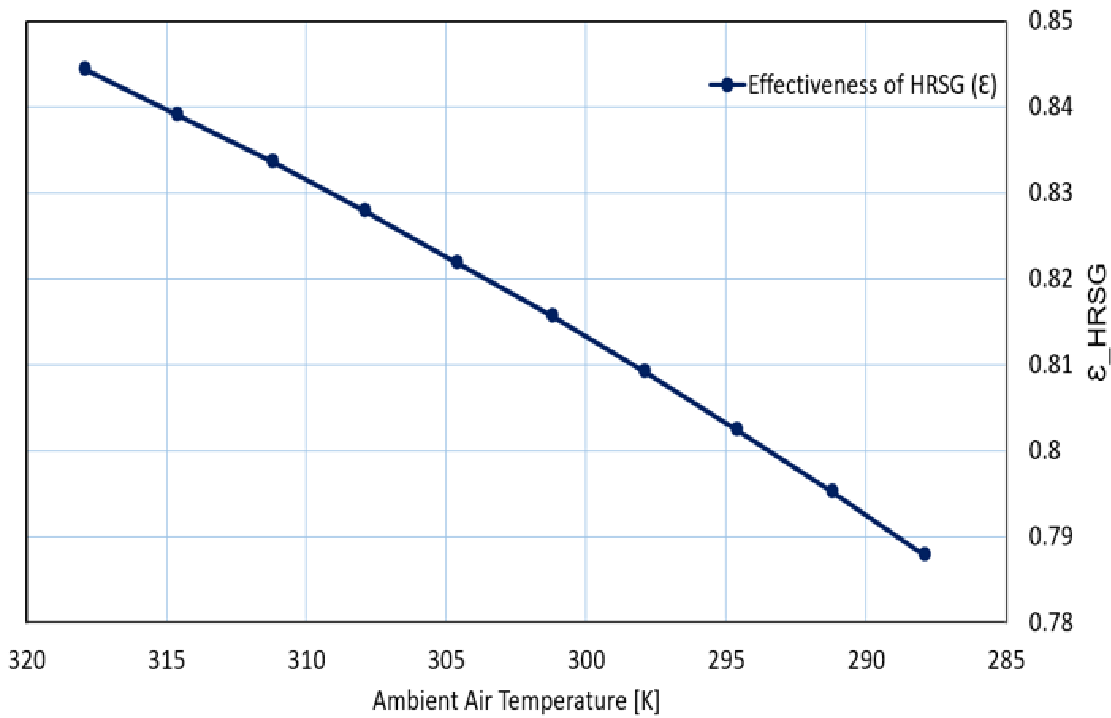

Fig. (12) Effect of Ambient Air Temperature to Effectiveness of HRSG.

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ARTICLE
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نمذجة وتحليل الأداء الحراري لتوربين غازي بسيط ومولد بخاري أحادي الضغط لاستعادة الحرارة للعمل في دورة مشتركة في ظل ظروف محيطة مختلفة

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الخلاصة:

تعتبر محطات الطاقة المشتركة أكثر كفاءة في تحويل الطاقة الحرارية إلى طاقة كهربائية بسبب تقدم التكنولوجيا في العالم. تم دمج وحدة توليد طاقة غازية بسيطة مع وحدة توليد طاقة بخارية لتكوين وحدة توليد طاقة مركبة، وذلك باستخدام مولد البخار (HRSG) لاستعادة الحرارة المهدرة من غازات العادم للتوربين الغازي وتوليد طاقة إضافية دون استهلاك إضافي للوقود. يهدف البحث الحالي إلى نمذجة محطة توليد توربينية غازية بسيطة ومولد بخار لاستعادة الحرارة (HRSG) أحادي الضغط، لإجراء تحليل حراري لهما تحت تأثير الظروف التشغيل المتمثلة بدرجات الحرارة المحيطة وضغط جوي (1) بار. تم العمل على محطة توليد الطاقة الغازية في ناحية القيارة جنوب مدينة الموصل، والمصممة بطاقة توليد تصل إلى (125) ميغاواط، وذلك للحصول على العوامل التشغيلية لمولد البخار لاستعادة الحرارة (HRSG) أحادي الضغط، والتي تتعلق بالتصميم الهندسي إذا ما تم تحويل المحطة الغازية إلى وحدة توليد مشتركة، مع الأخذ بنظر الاعتبار التوازن الطاقي، معدل التدفق الكتلي لغازات العادم من الوحدة الغازية، البخار الناتج من مولد البخار باستعادة الحرارة (HRSG) أحادي الضغط. وقد أظهرت النتائج أن زيادة درجات الحرارة المحيطة عند ضغط ثابت لها تأثير على أداء الوحدة الغازية البسيطة، حيث انخفضت الكفاءة الحرارية والقدرة التوليدية ومعدل التدفق الكتلي لغازات العادم، وارتفعت درجات حرارة غاز العادم ومعدل استهلاك الوقود. كما أظهرت النتائج أيضًا أن زيادة درجة الحرارة المحيطة تؤثر سلبًا على أداء وحدة توليد الطاقة ذات الدورة المركبة، وذلك من خلال تقليل إنتاج الطاقة للتوربين الغازي بسبب انخفاض كثافة الهواء. إلا أنها تؤدي إلى زيادة درجة حرارة غاز العادم، الأمر الذي قد يكون طفيفًا تحسين فعالية (HRSG) وإنتاج البخار ولكنه يؤدي إلى انخفاض في كفاءة الدورة الإجمالية وقدرة التوليد. وبالنظر إلى النتائج نلاحظ أن هناك فقدان كبير للحرارة مع الغازات الناتجة عن عملية الاحتراق في الوحدة الغازية يقدر بحوالي 60% ولكن من الممكن الاستفادة منه في توليد قدرة إضافية وزيادة الكفاءة الحرارية بعد دمجها مع وحدة بخارية باستخدام منظومة توليد البخار (HRSG) وتكوين وحدة توليد طاقة مركبة.

الكلمات الدالة: دورة الغازية، درجة حرارة المحيط، درجة حرارة العادم، منظومة توليد البخار ، الدورة المركبة.