

INFLUENCE OF AMBIENT TEMPERATURE ON THE PERFORMANCE OF REPOWERED COMBINED CYCLE POWER PLANT

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ABSTRACT :

A performance study was conducted on the combined cycle power plant (CCPP), which is one of the most important options for replacement and repowering of the available steam power plant. The thermodynamic performance of these plants depends on the ambient temperature which varies considerably from one season to another. The objective of the current work is to investigate the effects of these variations on the performance of combined cycle power plants. In this study AL-Mussaib thermal power plant has been chosen for repowering, for this purpose, AL-Khairat gas turbines are used and the effect of duct burner is investigated. The repowered combined cycle power plant (RCCPP) consists of four gas turbines and four heat recovery steam generator (HRSG) with one steam turbine was associated with the unit in this model. The theoretical analysis was made according to both first and second laws of thermodynamic analysis. A generalized computer program was prepared by Fortran 90 for this purpose. It is found that the net power output, thermal and exergy efficiencies of the RCCPP increases as the ambient temperature decreases. The mass flow rate of steam decreases with the increase of ambient temperature at constant compressor pressure ratio and turbine inlet temperature (TIT). The exergy destruction in the combustion chamber and HRSG decreases while, the exergy destruction of the condenser increases greatly as the ambient temperature increases

KEY WORDS :- Combined cycle, repowering, exergy analysis, HRSG, ambient temperature .

تأثير درجة حرارة الهواء المحيطة على اداء محطة الطاقة المشتركة

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الخلاصة :-

اشتمل البحث على دراسة اداء محطة توليد الكهرباء والتي تعمل بنظام الدورة المركبة او المؤتلفة (CCPP) والتي تعد واحدة من اهم الخيارات لاستبدال واعادة تأهيل محطة توليد الكهرباء البخارية المتاحة . يعتمد الاداء الترموديناميكي لهذه المحطات على درجة حرارة الهواء المحيط ، والذي يتغير بشكل كبير من فصل الى اخر . وتهدف هذه الورقة الى دراسة تأثير التغير في درجة حرارة الهواء المحيط على اداء هذه المحطات . في هذا البحث تم اختيار محطة المسيب الحرارية لإعادة التأهيل ولهذا الغرض تم استخدام التوربين الغازي لمحطة الخيرات ودراسة تأثير اضافة المحارق الانبوبية ايضا حيث تتكون محطة الطاقة المشتركة او المؤتلفة (CCPP) من اربعة توربينات غازية واربعة مولدات بخار (HRSG) مع توربين بخاري واحد . تم اجراء التحليل النظري وفقا لقانوني الاول والثاني من قوانين الترموديناميك . فقد تم اعداد برنامج حاسوبي شامل باستخدام لغة فورتران 90 لهذا الغرض ، وجد ان معدل القدرة الناتجة وكلا من الكفاءة الحرارية والمتاحية لمحطة الطاقة المشتركة او المؤتلفة (CCPP) تزداد بزيادة درجة حرارة الهواء المحيطة ، ان كتلة البخار تتناقص بزيادة درجة حرارة الهواء المحيط بثبوت كلا من نسبة الانضغاط ودرجة حرارة التوربين الداخلة . ان فواقد كل من غرفة الاحتراق والمولد البخاري (HRSG) تتناقص بزيادة درجة حرارة المحيط بينما في المكثف تزداد .

NOMENCLATURE

LATIN SYMBOLS

C_p	Specific Heat (kJ/kg .K)
ex	Specific Exergy (kJ/kg)
$\dot{E}x$	Total Exergy Rate (kW)
$\dot{E}x_D$	Exergy Destruction Rate (kW)
h	Specific Enthalpy (kJ/kg)
LHV	Lower Heating Value (kJ/kg)
$\dot{m}$	Mass Flow Rate (kg/s)
P	Pressure (bar)
Pr	Pressure Ratio
$\dot{Q}$	Heat Transfer Rate (kW)
s	Specific Entropy (kJ/kg .K)
T	Temperature (°C, K)
$\dot{W}$	Work Rate or Power (kW)

GREEK SYMBOLS

Δ	Change, Difference
γ	Specific Heat Ratio
η	Efficiency
I	First Low of Thermodynamic
II	Second Low of Thermodynamic
ξ	Exergy Ratio

SUBSCRIPTS SYMBOLS

0	Dead or Ambient State
add	Added
ch	Chemical
C	Compressor
CC	Combined Cycle

$Comb$	Combustion
db	Duct Burner
D	Destruction
exh	Exhaust
f	Fuel
g	Exhaust Gases
GT	Gas Turbine
GTC	Gas Turbine Cycle
i	In
j	Number of Component, State
o	Out
p	Pump
ph	Physical
pp	Pinch Point
st	Steam
ST	Steam turbine
sat	Saturated
STC	Steam Turbine Cycle
TD	Temperature Difference

ABBREVIATIONS

AFR	Air-Fuel Ratio
CCPP	Combined Cycle Power Plant
ECO	Economizer
EVA	Evaporator
HRSG	Heat Recovery Steam Generator
RCCPP	Repowered Combined Cycle Power Plant
RH	Reheater
SUP	Superheater
TIT	Turbine Inlet Temperature
TTD	Terminal Temperature Difference

INTRODUCTION :-

The combined cycle power plants (CCPPs) are currently one of the most important options for the construction of new generating capacity as well as for the replacement and repowering of existing steam turbine power plant units and to improve their performances.

Repowering is the conversion of an existing steam power plant into a combined cycle power plant by adding one or more gas turbines and heat recovery steam generator. Repowering of old thermal power plants to enhance their efficiency, increase their output, and to increase their operational life, is one of the best and lower expensive ways to secure power sources (Mehraban et al., 2014).

There are two main methods of repowering: full repowering and partial repowering. Full repowering is the most common way to reconstruct old steam power plants and improve their efficiency. Partial repowering is applied to the modern power plants. In this method the exhaust gases exit from the gas turbine are used to heat the feed water before entering (Naserabad et al., 2015). Full repowering is the most common way of repowering and it is beneficial to the old power plants with a minimum age of 25 years. In this method, an old

boiler is replaced by a HRSG and a gas turbine (or turbines). The idea of using this technique was suggested in 1949 for the first time and has been utilized in 1960 (Mehraban et al., 2014).

Some researches have been conducted in the fields of repowering and exergy analysis of combined cycles. Full repowering of a steam power plant has been performed by (Mehraban et al., 2014). Be'sat steam power plant in Tehran has been considered as a reference steam power plant. Two different types of HRSGs (single and dual pressure) were used to enhance the efficiency of the CCPP. In a similar manner, practical restrictions of a steam cycle full repowering have been investigated by (Naserabad et al., 2015). Exergy analysis method was used to study the repowered systems. Two types of gas turbines V94.2 and V94.3A were used and effect of duct burner was investigated. Another method of repowering used by (Mehrabani et al., 2014) have optimized parallel feedwater heat recovery for Shahid Rajaei power plant in Tehran considering exergy efficiency. Also extensive researches have been conducted on repowering; among these researchers studies conducted by (Carapellucci et al., 2007, Rohani et al. 2014) can be mentioned. The major operating parameters which influence the CCPP performance are compressor pressure ratio, TIT, pinch point and ambient temperature. Among these variables, the ambient temperature causes the greatest performance variation. Ambient temperature can be simply defined as the temperature of the surrounding or the temperature of the environment. Increases in the ambient temperature can highly affect the gas turbine performance. Therefore, the location of the power plant plays an important role on its performance. Arrieta and Lora., 2005 showed that the net power output varied from (640 to 540 MW) when the ambient temperature varied in range (0-35°C) and the temperature of gas about (675°C). Ibrahim et al., 2012 reported that at the high compressor pressure ratio of about (18), if the ambient temperature increase from (273 to 333 K), the thermal efficiency of the CCPP decreases from (58.5% to 56.4%). Also, the thermal efficiency of the CCPP decreases with the increase of the ambient temperature and air-fuel ratio. A thermal analysis and performance evaluation has been carried out by (Fathi, 2012) to investigate the benefit of applying CCPP on Beijee simple gas turbine. It was found that there is about (11.6%) reduction in the mass of air as the temperature reaches (45°C). The power output was found to decrease as the ambient temperature increases. As the ambient temperature reaches (45°C), the power output and the thermal efficiency of the CCPP were reduced by (25.3 and 8%), respectively. Fellah, 2012 showed that at an ambient temperature of (15°C), the thermal and exergy efficiencies were (42.80 and 40.20%) respectively, and the net power output was about (260.58 MW). At an ambient temperature of (40°C), the thermal and exergy efficiencies dropped to (41.11 and 38.60%), respectively and the net power output decreased to (208.62 MW). In another study, (Ibrahim et al., 2013) found that the strong influence of ambient temperature produces a reduction in the power output in the gas turbine unit from (571 to 487MW) when the ambient temperature increases from (273 to 323 K).

In this work full repowering of AL-Mussaib thermal power plant without omitting feed water heater using a single pressure HRSG with reheat and supplementary firing to recover the energy from the exhaust gases in the gas turbine to produce a superheated steam. The main purpose of the use of the supplementary firing is to avoid the intersection between the gas and steam temperature lines, also to keep the form of the temperature profile, since the temperature and pressure of steam across the HRSG remains constant.

REPOWERED COMBINED CYCLE POWER PLANT (RCCPP) :-

The main idea of the CCPP is to improve Al-Mussaib thermal power plant efficiency by utilizing the waste heat in the turbine exhaust gases which are going to the HRSG to generate a superheated steam. This is known as repowering of the steam power plant. A simplified diagram of the RCCPP plant is shown in **Fig. 1**. This is a combined fluid flow and energy flow diagram, which have both gas and steam turbines supplying power to the network.

AL-Khairat gas turbine power plant represents the largest and the most important station producing the electric power in Iraq, located in Karbala 19 km away from the Hindi center, established by the Turkish company, Çalik Enerji since 2011 but it operates since 2013 on an open basis (Brayton cycle). It 10 units and each unit consists of three main parts: a compressor, a combustion chamber, and a turbine and has a power output of (125 MW). Al-Mussaib thermal power plant is located 64 km south of Baghdad and established by the South Korea company Hyundai in 1983 and it operating since 1989. It consist of 4 units each unit produces (300 MW), which consists of the steam turbine, condenser, pump, closed feedwater heaters and deaerating heater (DA).

HEAT RECOVERY STEAM GENERATOR (HRSG) :-

The task of the HRSG is to transfer heat from the exhaust gases of the gas turbine to the bottoming steam cycle. The HRSG consists of a set of heat exchangers that transfer heat from the flue gases to water. The hot gases enter the HRSG at point (d) and exit at point (g4) as shown in **Fig. 2** . A single pressure HRSG with reheat and supplementary firing is considered for repowering the steam power plant as shown in **Fig. 3**.

THERMODYNAMIC ANALYSIS :-

The thermodynamic analysis of the RCCPP is based on the fundamental of conservation of mass, energy and exergy to compute the energy and exergy contents, thermal and exergy efficiencies and, irreversibility of each component in the system. **Fig. 4** show the temperature-entropy diagram of the RCCPP, the actual and ideal processes are represented in full and dashed line, respectively. The system models are developed on the following assumptions:

1. All the processes are steady state and steady flow.
2. All components are considered adiabatic.
3. The kinetic and potential energy and exergy are assumed to be negligible.
4. Atmospheric condition is taken as pressure 1.013 bar and temperature range (273-318) K.
5. The range of TIT in gas turbine cycle from 1000 K to 1600 K.
6. The combustion efficiency is 98%.
7. The compressor pressure ratio Pr_{air} range is from 5 to 30.
8. Isentropic efficiencies of compressor and gas turbine are 88% to 86%, respectively.
9. The pressure loss at combustion chamber inlet is $\Delta P_{Comb} = 0.25$ bar.
10. The fuel injected into the combustion chamber and duct burner was a natural gas which its composition is given in **Table 1**.
11. Inlet steam temperature and pressure are $T_1 = 538^\circ\text{C}$, $P_1 = 166.713$ bar (AL-Mussiab Operation Manual).
12. Reheat steam temperature and pressure are $T_3 = 538^\circ\text{C}$, $P_3 = 41.38$ bar (AL-Mussiab Operation Manual).

Energy Analysis

Each component of the RCCPP is considered as a control volume in steady state condition. The mass balance and the energy balance for the steady flow process of an open system are given by (Regulagadda, 2010):

$$\sum_{in} \dot{m}_i = \sum_{out} \dot{m}_o \quad (1)$$

$$\dot{Q}_k + \sum_{in} \dot{m}_i h_i = \sum_{out} \dot{m}_o h_o + \dot{W} \quad (2)$$

Compressor (Mehraban, et al., 2014)

$$T_b = T_a \left(1 + \frac{(Pr_{air})^{\left(\frac{\gamma_{air}-1}{\gamma_{air}}\right)} - 1}{\eta_{l,c}} \right) \quad (3)$$

$$\dot{W}_C = \dot{m}_{air} C_{p_{air}} (T_b - T_a) \quad (4)$$

The specific heat of air at constant pressure has been defined as a function of temperature:

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

Combustion Chamber

$$\dot{m}_{air} C_{p_{air}} T_b + \dot{m}_f \eta_{l,Comb} LHV + \dot{m}_f C_{p_f} T_f = \dot{m}_g C_{p_g} T_c \quad (6)$$

$$\dot{m}_g = \dot{m}_{air} + \dot{m}_f \quad (7)$$

$$AFR = \frac{\dot{m}_{air}}{\dot{m}_f} \quad (8)$$

$$P_c = P_b - \Delta P_{Comb} \quad (9)$$

The specific heat of product of combustion at constant pressure has been defined as a function of temperature:

$$C_{p_g} = 0.991615 - \frac{6.99703}{10^4} T + \frac{2.7129}{10^7} T^2 - \frac{1.22442}{10^{10}} T^3 \quad (10)$$

Gas Turbine (Naserabad, et al., 2015)

$$T_d = T_c \left(1 - \eta_{l,GT} \left(1 - \left(1/P_{r_g} \right)^{\left(\frac{\gamma_g-1}{\gamma_g}\right)} \right) \right) \quad (11)$$

$$\dot{W}_{GT} = \dot{m}_g C_{p_g} (T_c - T_d) \quad (12)$$

$$\eta_{l,GTC} = \frac{\dot{W}_{net,GTC}}{\dot{Q}_{GT,add}} \quad (13)$$

$$\dot{W}_{net,GTC} = \dot{W}_{GT} - \dot{W}_C \quad (14)$$

$$\dot{Q}_{GT,add} = \dot{m}_f \eta_{l,Comb} LHV \quad (15)$$

Duct Burner

Applying the energy balance equation to calculate the mass flow rate of the fuel added in the duct burner:

$$\dot{m}_g C_{p,g,d} T_d + \dot{m}_{f,db} \eta_{l,db} LHV = \dot{m}_{exh} C_{p,g,e} T_e \quad (16)$$

$$\dot{m}_{exh} = \dot{m}_g + \dot{m}_{f,db} \quad (17)$$

The duct burner efficiency has been considered as roughly 93%. The fuel injected into the duct burner can have different flow rates depending on the type of the gas turbine used. Therefore, the fuel flow rate must be less than 2 kg/s, since it may lead to burning the superheater pipes (Rohani and Ahmadi, 2014).

Heat Recovery Steam Generator

Using energy equation for steam/water and gas in HRSG various sections, the following energy balance principle can be written as:

Reheater section:

$$\dot{m}_{exh} C_{p,g} (T_e - T_{g1}) = \dot{m}_3 (h_3 - h_2) \quad (18)$$

Superheater section:

$$\dot{m}_{exh} C_{p,g} (T_{g1} - T_{g2}) = \dot{m}_{st} (h_1 - h_{b2}) \quad (19)$$

Evaporator section:

$$\dot{m}_{exh} C_{p,g} (T_{g2} - T_{g3}) = \dot{m}_{st} (h_{b2} - h_{b1}) \quad (20)$$

Economizer section:

$$\dot{m}_{exh} C_{p,g} (T_{g3} - T_{g4}) = \dot{m}_{15} (h_{b1} - h_{15}) \quad (21)$$

To obtain the temperatures in each element, first the above equations should be solved. The temperature of the exhaust gases entering the economizer and superheater can be written as follows, respectively (Rohani and Ahmadi, 2014):

$$T_{g3} = T_{sat} + T_{pp} \quad (22)$$

$$T_{g1} = T_1 + T_{TD} \quad (23)$$

The total heat available in the exhaust gases after duct burner (Naserabad, et al., 2015):

$$\dot{Q}_{exh,GT} = \dot{Q}_{GT,add} (1 - \eta_{l,GTc}) + \dot{Q}_{f,db} \quad (24)$$

Where:

$$\dot{Q}_{f,db} = \dot{m}_{f,db} LHV \eta_{l,db} \quad (25)$$

Steam Turbine

The net power output of the steam cycle is:

$$\dot{W}_{net,STC} = \dot{W}_{ST,act} - \sum \dot{W}_{p,act} \quad (26)$$

The thermal efficiency of the steam cycle is:

$$\eta_{I,STC} = \frac{\dot{W}_{net,STC}}{\dot{Q}_{exh,GT}} \quad (27)$$

The thermal efficiency of the combined cycle is given by (Mansouri, et al., 2012):

$$\eta_{I,CC} = \frac{\dot{W}_{net,CC}}{\dot{Q}_{GT,add} + \dot{Q}_{f,db}} \quad (28)$$

$$\dot{W}_{net,CC} = \dot{W}_{net,GTC} + \dot{W}_{net,STC} \quad (29)$$

Exergy Analysis

The exergy may be defined as the maximum work that can be achieved by bringing a system into equilibrium with its environment (Shapiro, 2006). The exergy can be divided into four distinct components: kinetic, potential, physical and chemical exergy. The kinetic and potential exergy are considered negligible. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state through purely physical processes. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion process (Naserabad et al., 2015).

The exergy balance for the steady state flow of an open system is given by (Kaviri et al., 2012):

$$\dot{E}x_Q + \sum_{in} \dot{m}_i ex_i = \sum_{out} \dot{m}_o ex_o + \dot{E}x_W + \dot{E}x_D \quad (30)$$

where ex is the specific exergy, $\dot{E}x_Q$, $\dot{E}x_W$ and $\dot{E}x_D$ are the exergy of heat transfer, work and the exergy destruction, T is the absolute temperature (K), and (0) refers to the dead state. The specific physical exergy for air and combustion gases is given by (Naserabad et al., 2015).

$$ex_{ph,j} = Cp \left(T_j - T_0 - T_0 \ln \left(\frac{T_j}{T_0} \right) \right) + RT_0 \ln \left(\frac{P_j}{P_0} \right) \quad (31)$$

The specific physical exergy for steam and water is:

$$ex_{ph,j} = h_i - h_o - T_0(s_i - s_o) \quad (32)$$

The chemical exergy of the fuel is given in simplified form by the following relation:

$$ex_{ch,f} = \xi_f LHV \quad (33)$$

Where ξ_f is the fuel exergy factor based on the LHV of the fuel.

For gaseous fuel with C_nH_m , the following experimental equation is used to calculate ξ_f :

$$\xi_f = 1.033 + 0.0169 \frac{m}{n} - \frac{0.0698}{n} \quad (34)$$

The exergy flow balance of the whole HRSG is considered as an open system, therefore, the exergy destruction of the HRSG is given by (Kaviri et al., 2012):

$$\dot{E}x_{D,HRSG} = (\dot{E}x_e + \dot{E}x_{15} + \dot{E}x_2)_{in} - (\dot{E}x_1 + \dot{E}x_3 + \dot{E}x_{g4})_{out} \quad (35)$$

In equation (35) the term $\dot{E}x_{g4}$ is zero because the exhaust gas from the HRSG cannot be used anymore. The term $\dot{E}x_e = \dot{E}x_d$ is the exergy of hot gases inlet the HRSG when the HRSG operate without supplementary firing. $\dot{E}x_{15}$ is the exergy of feedwater inlet the HRSG, $\dot{E}x_1$ is the exergy of steam leaving the HRSG. $\dot{E}x_2$ and $\dot{E}x_3$ are the exergy of inlet and outlet reheated steam, respectively.

$$\eta_{II,HRSG} = \frac{\dot{E}x_1 + \dot{E}x_3 - \dot{E}x_{15} - \dot{E}x_2}{\dot{E}x_e} \quad (36)$$

$$\eta_{II,CC} = \frac{\dot{W}_{net,CC}}{\dot{E}x_f + \dot{E}x_{f,db}} \quad (37)$$

RESULTS AND DISCUSSION :-

The influence of ambient temperature on the performance of the gas turbine power plant and the RCCPP is presented in this section. Modeling of cycle components and governing equations developed for proposed cycles have been coded using Fortran 90 program. To validate the Fortran 90 code for performance analysis of the RCCPP, the code was tested using data from (Ibrahim et al., 2012) model. In this model the CCPP consists of: axial compressor, combustion chamber, gas turbine, single pressure HRSG (without reheat and supplementary firing), steam turbine, condenser and feed water pump. The results for comparison between the simulated net power output of Fortran 90 code and (Ibrahim et al., 2012) model together with the effect of different values of AFR and ambient temperature are presented in **Fig. 5**. It is clear that the net power output of the CCPP increases with the increase of the ambient temperature and with the decrease of the AFR. This is because the CCPP power output increases with the decrease of the exhaust gases mass flow rate, which leads to increase the steam mass flow rate with decrease of the AFR; therefore, the net power output of the CCPP increases with the increase of the ambient temperature, because the increase of the steam turbine power output are greater than the decreases in the gas turbine power output. The simulation results were satisfactory compared with the (Ibrahim et al., 2012) model of the CCPP. It can be shown from **Fig. 6** that the mass flow rate of steam decreases as the ambient temperature increase with a different compressor pressure ratio for ($TIT=1400K$, $T_{pp}=10^\circ C$ and $T_{TD}=25^\circ C$). Increasing the ambient temperature reduces the density of the air and consequently reduces the air mass flow into the compressor for the same volumetric flow rate, which results in decreased mass flow rate and temperature of exhaust gases i.e., the heat available at exhaust becomes less. **Fig. 7** shows the relation between the mass flow rate of steam and ambient temperature with different TIT for ($Pr_{air}=12.5$, $T_{pp}=10^\circ C$ and $T_{TD}=25^\circ C$). It is clear that the mass flow rate of steam decreases as the ambient temperature increase at a constant TIT, while increases with the increase of TIT at a constant ambient temperature giving more heat available from the exhaust gases. The heat available in the HRSG becomes more due to increases the temperature and mass flow rate of the exhaust gases. **Fig. 8** shows the relation between the thermal efficiency of the RCCPP and the ambient

temperature with different three values of AFR (44, 48 and 52) kg of air/kg of fuel for ($Pr_{air}=12.5$, $T_{pp}=10^{\circ}\text{C}$ and $T_{TD}=25^{\circ}\text{C}$). It is found that the thermal efficiency of the RCCPP increases as the ambient temperature increase and AFR decrease. For a given AFR, when the ambient temperature increase the mass flow rate of air and fuel decrease to achieve the desired AFR as a result, the exhaust gases temperature increases and gas mass flow rate decreases, so causing the power output and the thermal efficiency of the gas turbine to fall. The amount of steam generated in the HRSG decreases with the increase of the ambient temperature, because the heat available in the exhaust gases becomes less due to decrease the mass flow rate of the exhaust gases. Whilst the thermal efficiency of steam cycle increases even with the decrease of the amount of steam generated in the HRSG, because the reduction in the steam power output is less than a reduction in the heat available in the HRSG, therefore, the thermal efficiency of the steam cycle increase. Consequently, when the ambient temperature increase and AFR decrease the increase of the steam cycle thermal efficiency is more than the decrease of the gas turbine thermal efficiency. The relation between the exergy efficiency at different values of AFR is shown in **Fig. 9** for ($Pr_{air}=12.5$, $T_{pp}=10^{\circ}\text{C}$ and $T_{TD}=25^{\circ}\text{C}$) which have the same trend of the **Fig. 8**. For a given AFR, the exergy efficiency of the RCCPP increases with the increase of the ambient temperature, because the exergy of fuel becomes less due to decrease the fuel mass flow. **Fig. 10** shows the relation between the net power output from the RCCPP and the ambient temperature with different AFR for ($Pr_{air}=12.5$, $T_{pp}=10^{\circ}\text{C}$ and $T_{TD}=25^{\circ}\text{C}$). It can be seen from this figure that the net power output of the RCCPP decreases as the ambient temperature and AFR increase, due to the decreases of TIT resulting from the reduction in the mass flow rate of the fuel thus the power output from the gas turbine becomes less and consequently the net power output of the RCCPP becomes less and less. The effect of ambient temperature on the power output of gas, steam and repowered combined cycles, is shown in **Fig. 11** for ($TIT=1400\text{K}$, $Pr_{air}=12.5$, $T_{pp}=10^{\circ}\text{C}$ and $T_{TD}=25^{\circ}\text{C}$). It is found the power output from the gas, steam and repowered combined cycles decreases when the ambient temperature increase. The power output from the gas turbine decreases with the increase of the ambient temperature because the air mass flow rate decreases with the increase of the ambient temperature due to decrease the density of air. So, the fuel mass flow rate will decreases since the TIT kept constant and thus the mass flow rate of the exhaust gases becomes less. The amount of the steam generated in the HRSG will decrease also, because the heat available in the HRSG decreases as the mass flow rate of the exhaust gases decrease leading to decrease the steam turbine power output and consequently the net power output from the RCCPP becomes less. **Fig. 12** shows that the thermal efficiency of the gas cycle decreases when the ambient temperature increase, because the power output becomes less, while the thermal efficiency of the steam cycle increases with the increase of the ambient temperature, because the reduction in the power output from the steam cycle less than that of the heat available in the exhaust gases which is required to produce the steam. The thermal efficiency of the RCCPP will be decreased even though increase the thermal efficiency of the steam cycle, because the decrease of the gas turbine thermal efficiency is more than the increase of the steam cycle thermal efficiency. **Fig. 13** shows that the exergy efficiency of the gas cycle decreases when the ambient temperature increase. For the steam cycle the exergy efficiency increases, because the reduction of the exergy of the exhaust gases (resulting from the decrease of the mass flow rate of the exhaust gases) across the HRSG less than that of the power output from the steam cycle. While the exergy efficiency of the repowered combined cycle decreases with the increase of the ambient temperature, because the net power output of the RCCPP becomes less. In addition to that, the exergy of the fuel decreases with the increase of the ambient temperature, because the fuel mass flow rate decreases with the increase of the ambient temperature. The ambient temperature has also effect on the exergy destruction rate of each component in the gas turbine cycle for a constant compressor pressure ratio and TIT as shown in **Fig. 14** for ($TIT=1400\text{K}$, $Pr_{air}=12.5$, $T_{pp}=10^{\circ}\text{C}$ and $T_{TD}=25^{\circ}\text{C}$). It can be observed that the exergy destruction of the combustion chamber is

higher than the other parts of gas turbine but the exergy destruction of combustion chamber and exhaust gases decreases with the increase of ambient temperature, because the mass flow rate of the exhaust gases decreases with the increase of the ambient temperature while the rate of exergy destruction for turbine and compressor increasing slightly. **Fig. 15** shows that the HRSG exergy destruction decreases with the increase of the ambient temperature because the exergy of the steam outlet from the HRSG becomes more. While, the exergy efficiency of the HRSG increases with the increase of the ambient temperature, because the HRSG exergy destruction becomes less. The effect of the ambient temperature on the exergy destruction rate of the steam cycle components is shown in **Fig. 16**. It can be seen that the exergy destruction of all components, except condenser increased as ambient temperature increases, because the temperature difference between the cooling water and the steam increases as the ambient temperature increase. The exergy destruction of the duct burner decreases with the increase of the ambient temperature, due to the available heat in the reheat section becomes more and the fuel needed for the supplementary firing becomes less. Therefore, the steam produced decreases, the exergy destruction of the steam turbine decreases even with the increasing of the difference between the turbine inlet and outlet temperature. **Fig. 17** shows that the exergy efficiency of all components decreases with the increases of the ambient temperature. The maximum reduction occurs in the condenser and other components didn't have a large exergy difference. The results are consistent with those shown in **Fig. 16**.

CONCLUSIONS :-

An analysis based on the first and second law of thermodynamics has been performed to find the effect of ambient temperature variation on the performance of the RCCPP. The following conclusions can be drawn:

1. The decrease of ambient temperature increases the net power output from the RCCPP as well as the thermal efficiency, and vice-versa.
2. The amount of steam generated in the HRSG decreases with the increase of ambient temperature at constant compressor pressure ratio and TIT.
3. The net power output decreases linearly with increases in the ambient temperature and AFR while, the thermal and exergy efficiencies of the RCCPP increases linearly with the increases of ambient temperature and decrease in the AFR.
4. The exergy efficiency of the HRSG increases and the exergy destruction in the HRSG decreases as the ambient temperature increases.

Table 1: Volumetric Analysis of Natural Gas.

Substance	Formula	Volumetric Analysis (%)	MW (kg/kmol)	LHV (kJ/kg)
			(Shapiro, 2006)	
Methane	CH ₄	75.7	16.04	50020
Ethane	C ₂ H ₆	18.44	30.07	47480
Carbone dioxide	CO ₂	2.44	44.01	-
Propane	C ₃ H ₈	2.4	44.09	46360
Nitrogen	N ₂	0.72	28.01	-
n-Butane	C ₄ H ₁₀	0.24	58.12	45720

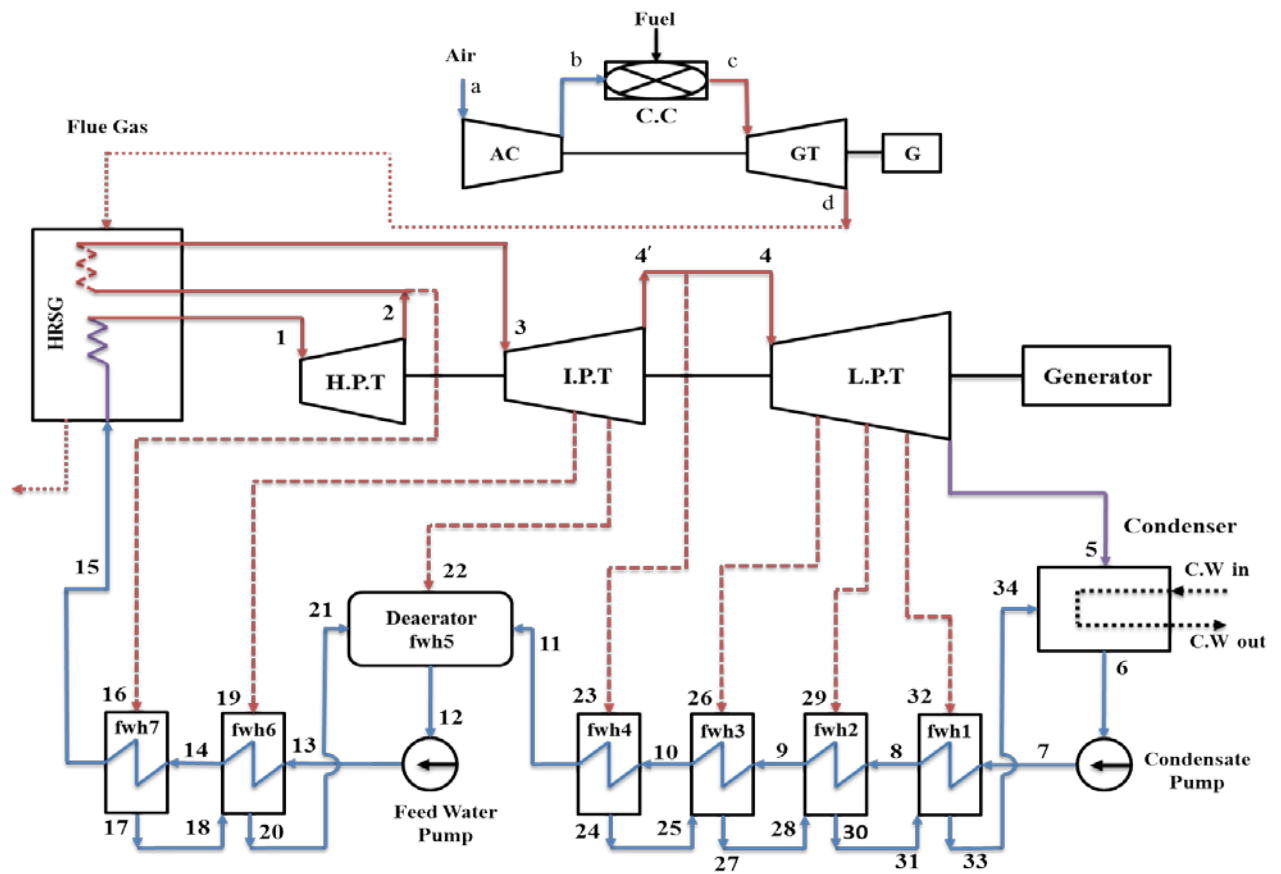


Fig. 1: Schematic diagram of the repowered combined cycle power plant .

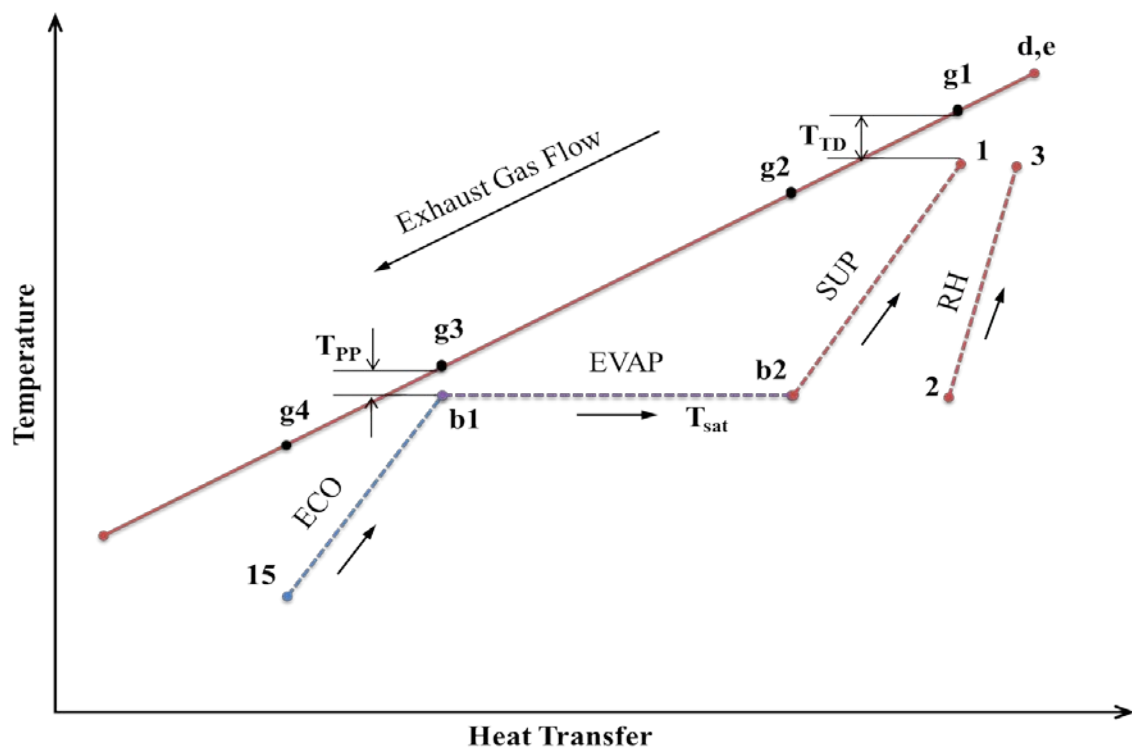


Fig. 2: Temperature profile for a single pressure HRSG with reheat.

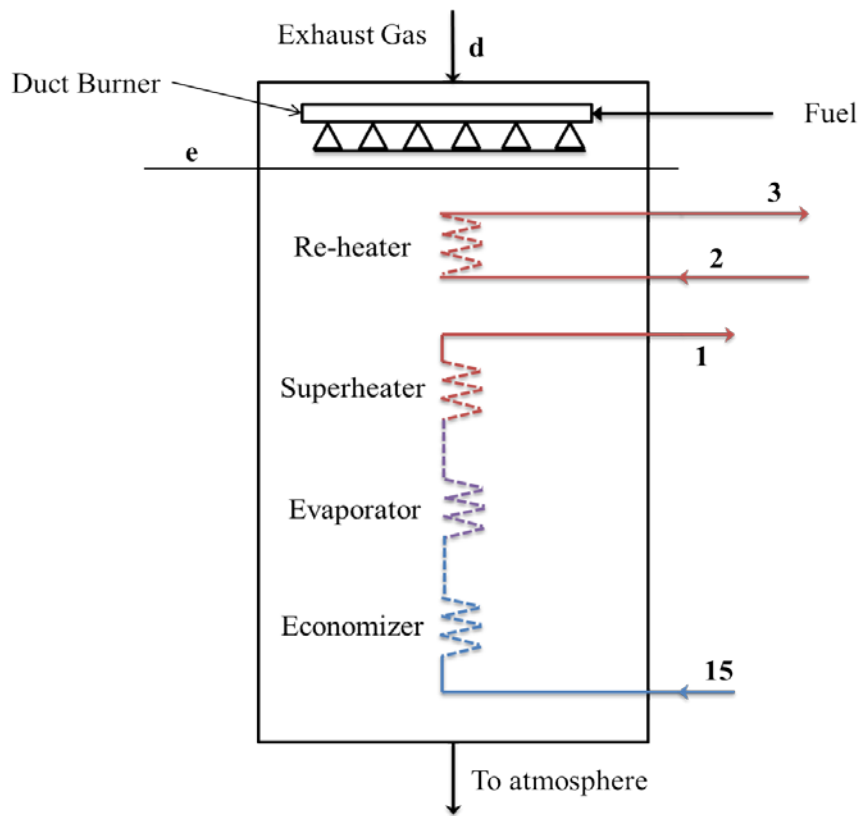


Fig 3: Schematic diagram of a single pressure HRSG with reheat.

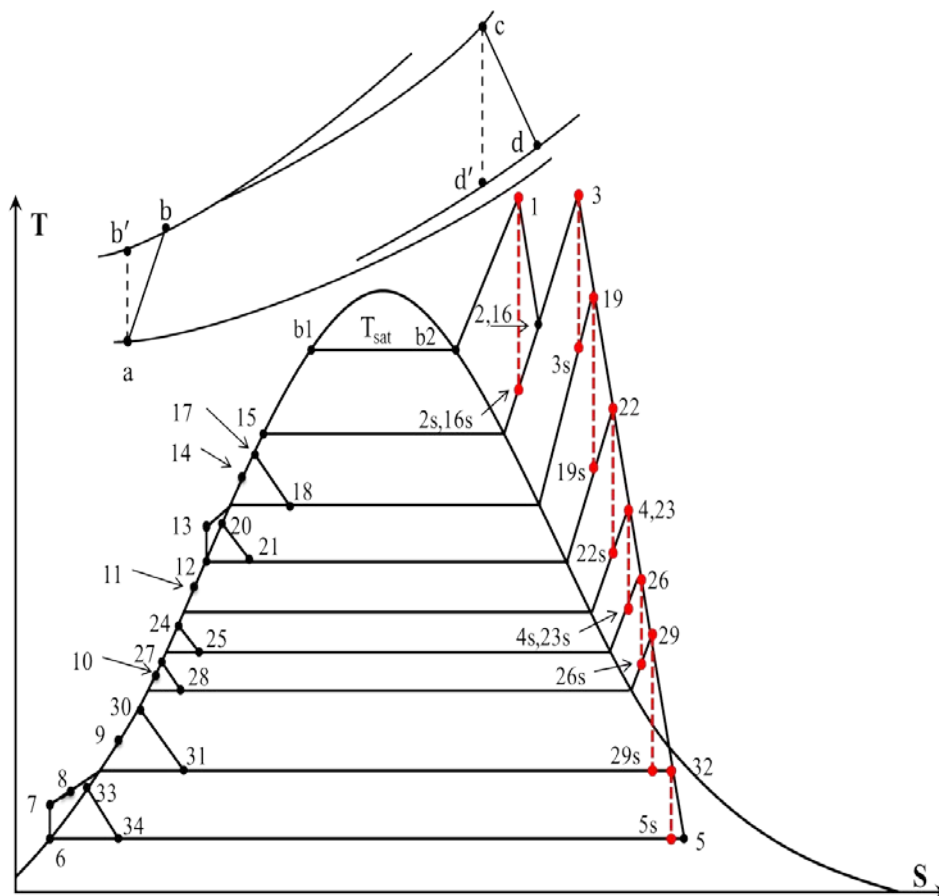


Fig. 4: T-s diagram of the repowered combined cycle power plant.

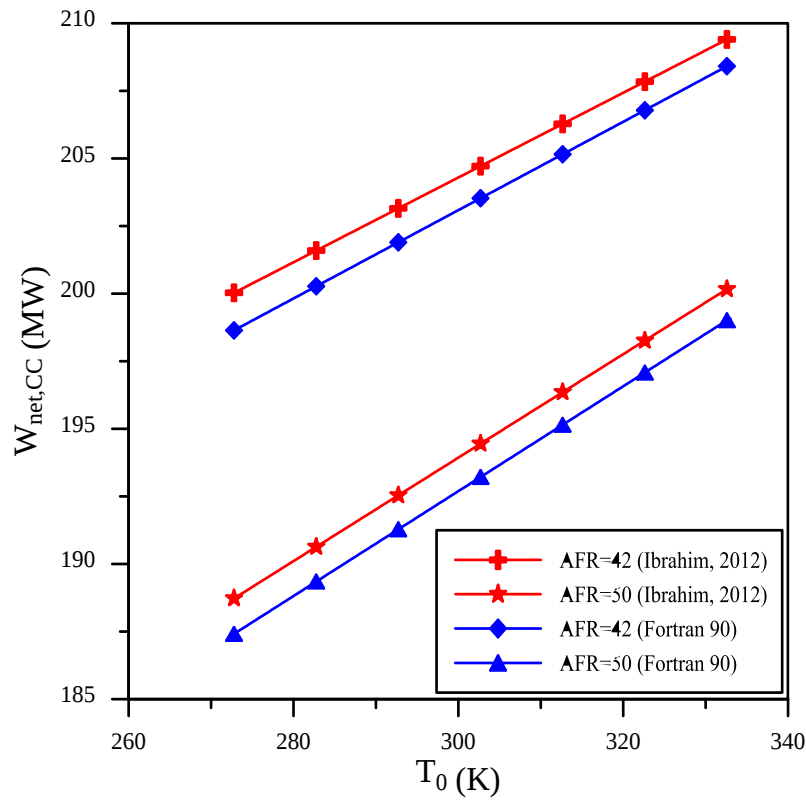


Fig. 5: Comparison between the simulated net power output of Fortran 90 code and (Ibrahim et al., 2012) model with the effect of different values of AFR and ambient temperature.

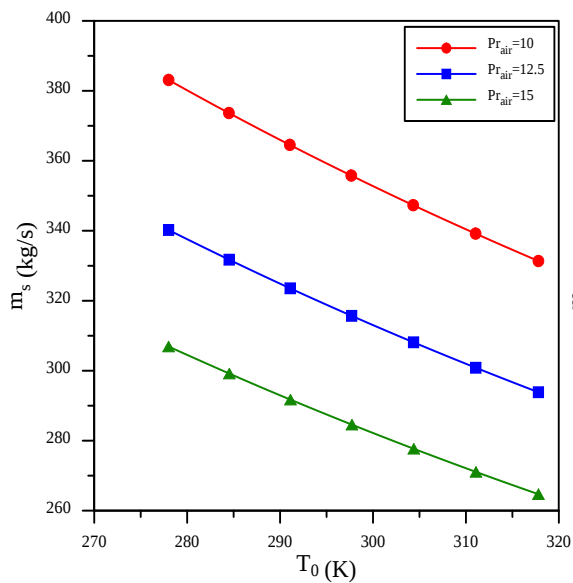


Fig. 6: Effect of ambient temperature on steam mass flow rate with different pressure ratio.

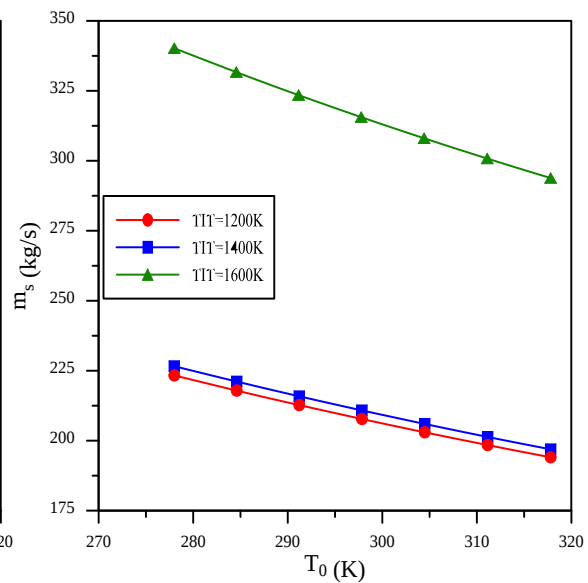


Fig. 7: Effect of ambient temperature on steam mass flow rate with different TIT.

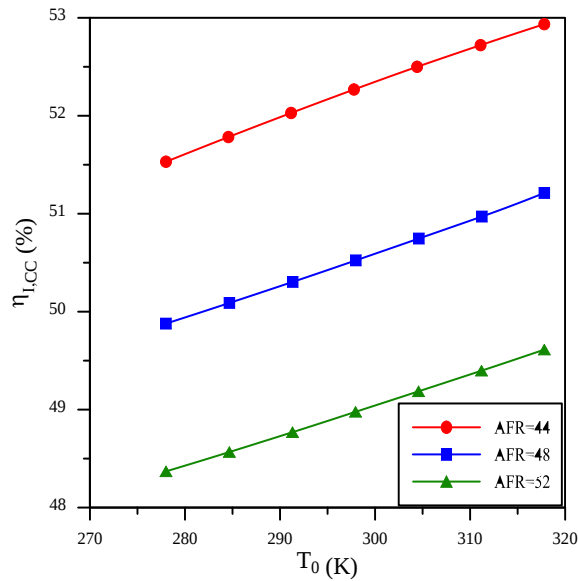


Fig. 8: Effect of ambient temperature on thermal efficiency of RCCPP for different AFR.

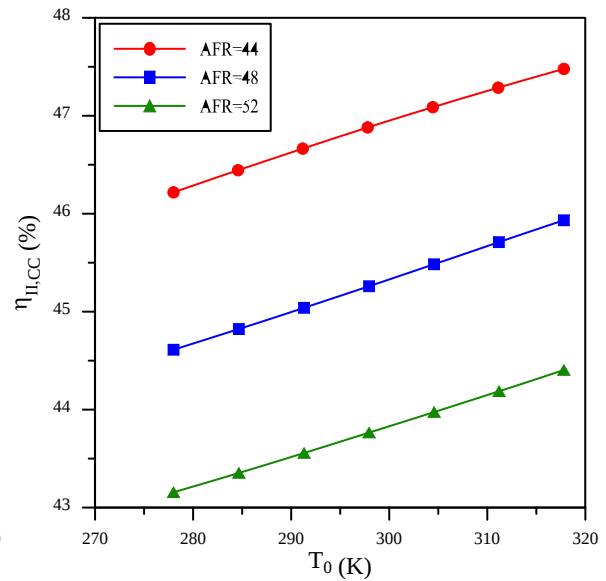


Fig. 9: Effect of ambient temperature on exergy efficiency of RCCPP for different AFR.

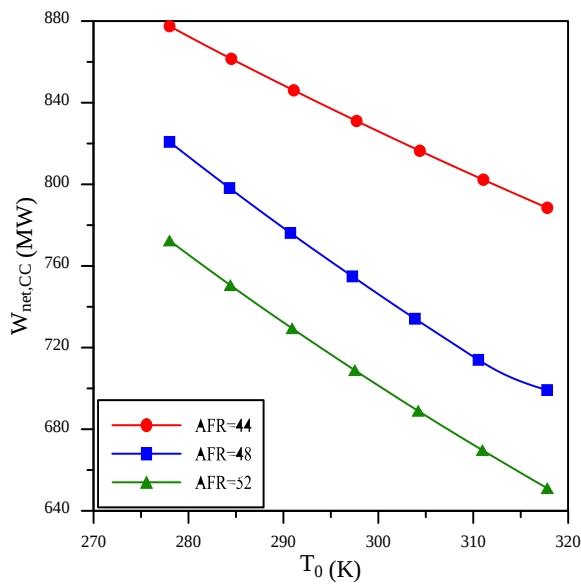


Fig. 10: Effect of the ambient temperature on net power output of RCCPP for different AFR.

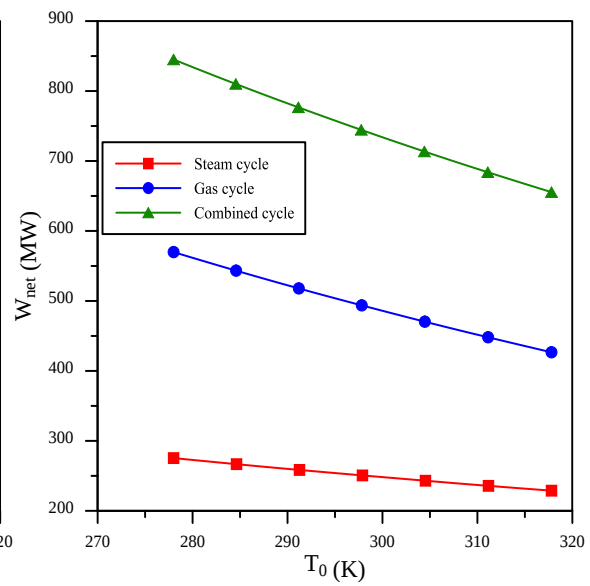


Fig. 11: Effect of ambient temperature on net power output.

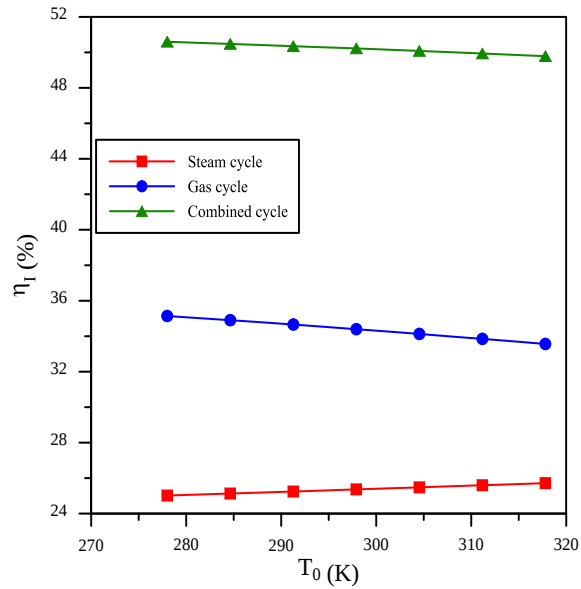


Fig. 12: Effect of ambient temperature on thermal efficiency.

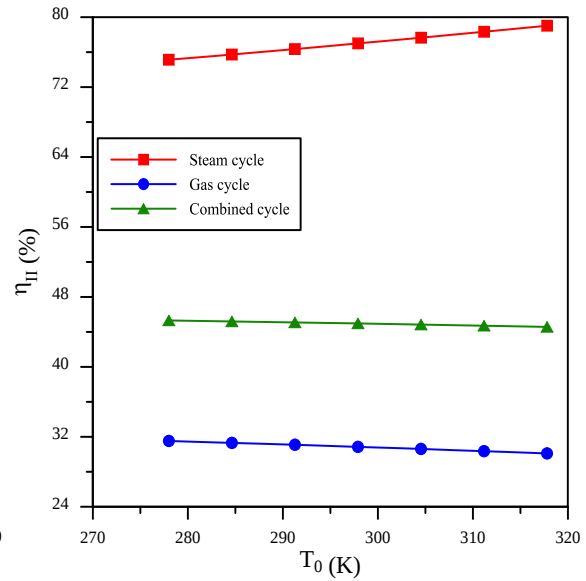


Fig. 13: Effect of ambient temperature on exergy efficiency.

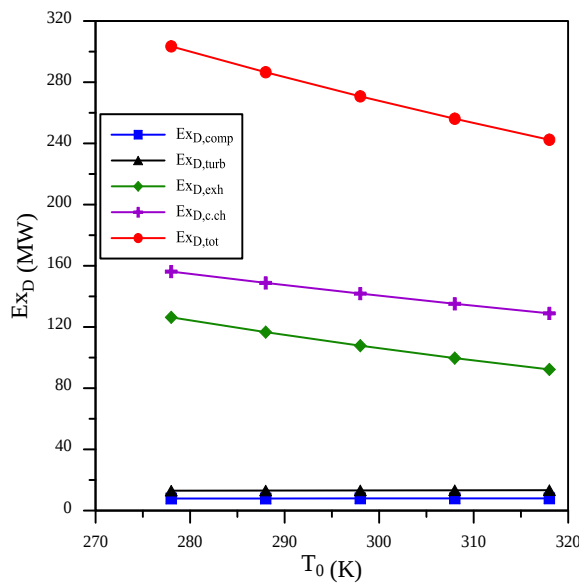


Fig. 14: Effect of ambient temperature on exergy destruction rate of gas cycle components.

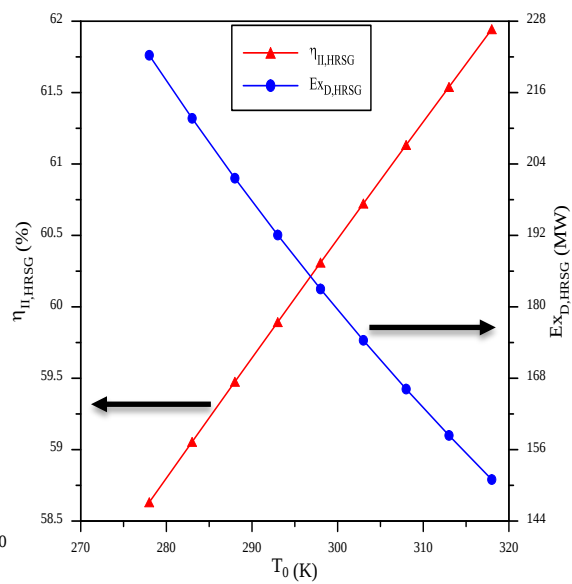


Fig. 15: Effect of ambient temperature on exergy destruction and efficiency of HRSG.

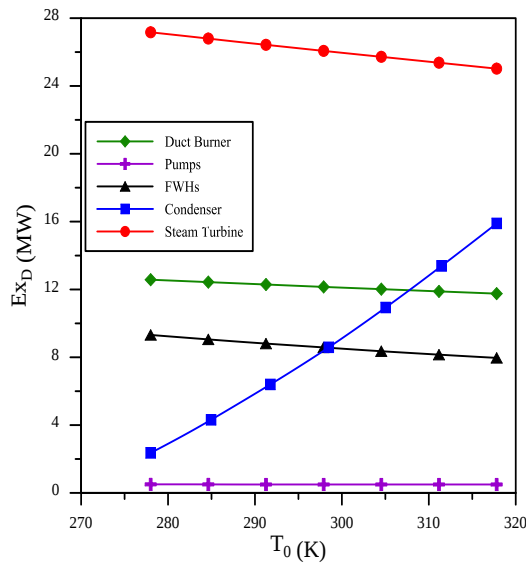


Fig. 16: Effect of ambient temperature on exergy destruction of steam cycle components.

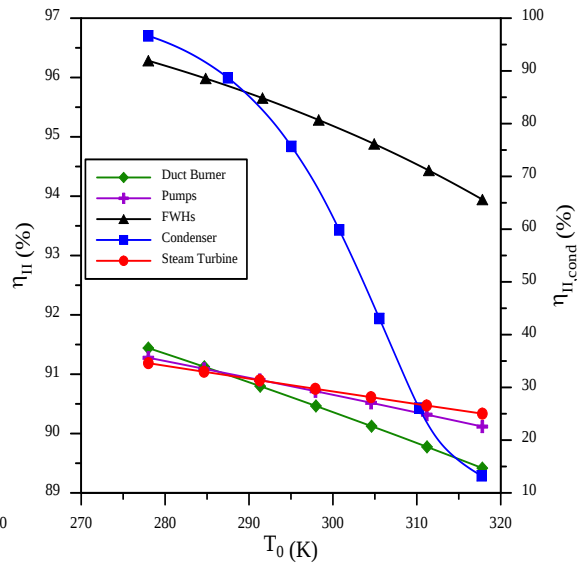


Fig. 17: Effect of ambient temperature on exergy efficiency of steam cycle components.

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