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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 thermodynamico performance of these plants depends onothe ambient temperature varies considerably from one season to another. The object of the current study is to investigate the effects of these variations on the performance of combined cycle power plants. In this study AL-Mussaib thermal power plant, Iraq has been choseno 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 4 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. The main conclusion drawn from this study were the net power output, thermal and exergy efficiencies of the RCCPP increases as the ambient temperature decreases. The massoflow rate of steamodecreases with the increase of ambient temperature and increases with the increase of compressor pressure ratio and 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: Ambient temperature, Combined cycle, Exergy analysis, HRSG, Repowering, power output

INTRODUCTION

The 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 its 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 to increase their operational life one of the best and a lower expensive ways to secure power sources (Mehrabani *et al*., 2014).

There are two main methods of repowering: full repoweringy 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 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 an HRSG and a gas turbine (or turbines). The idea of using this technique suggested in 1949 for the first time and has been utilized in 1960 (Mehrabani *et al*., 2014).

Some researchers 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 Mehrabani *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). The exergy analysis method is 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 Mehraban *et al*. (2014) have been optimized parallel feed water 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 and Milazzo (2007) and Rohani and Ahmadi (2014) can be mentioned.

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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. The 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 locationoof the power plant plays an important role in its performance. (Arrieta and Lora, 2005) have been shown that the net power output varied from (640-540 MW) when the ambient temperature varied in range (0-35°C) and the temperature of gas about (675°C). Ibrahim and Rahman (2012) has been reported that if the ambient temperature increase from (273-333 K) the total power output will increase about 7%. 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 3 is about 11.6%, reduction in the mass of air as the temperature reaches 45°C. The power output was found to be decreased as the ambient temperature increases. As the ambient temperature reaches 45°C the power output and The thermal efficiency of the CCPP reduced by 25.3 and 8%, respectively. Fellah (2010) has been shown that at an ambient temperature of 15°C the thermal and exergy efficiencies rise it 42.80 and 40.20%, respectively the increases in the total power output about 53.46%. At an ambient temperature of 40°C the thermal and exergy efficiencies drop to 41.11 and 38.60%, respectively total power output rise to 60.98%. In another study (Ibrahim and Rahman, 2013) found that the strong influence of ambient temperature produces a reduction in the power output in the gas turbine unit from 571-487 MW when the ambient temperature increases from 273-323 K.

In this research, fullirepowering 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.

MATERIALS AND METHODS

Repowered combined cycle power plant: The main idea of the CCPP has grown out of the need to improve Al-Mussaib thermal power plant efficiency by utilizing the waste heat in the turbine exhaust gases which 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 have both gas and steam turbines supplying power to the network.

The topping cycle is AL-Khairat gas turbine power plant which represent the largest and the most important stations that producing the electric power in Iraq and located in Karbala 19 km away from the Hindi center, established by the Turkish company Calik Enerjisince 2011 but it was beginning in operating,

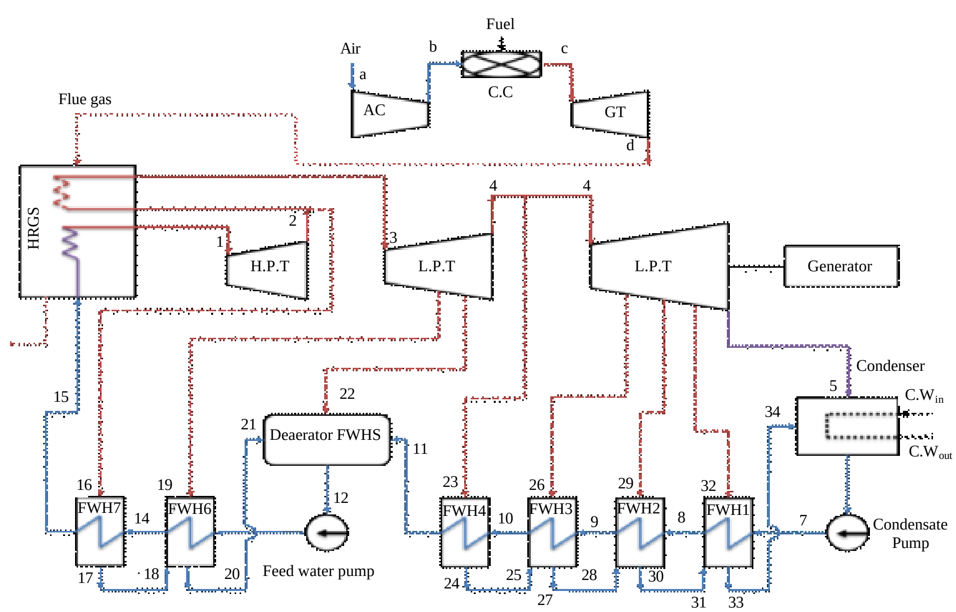

Fig. 1: Schematic diagram of the repowered combined cycle power plant

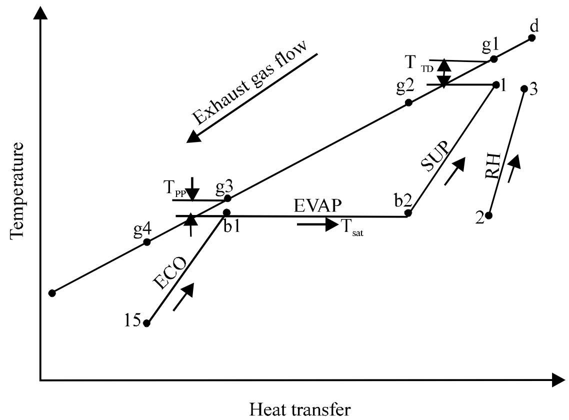

Fig. 2: Temperature profile for a single pressure HRSG with reheat

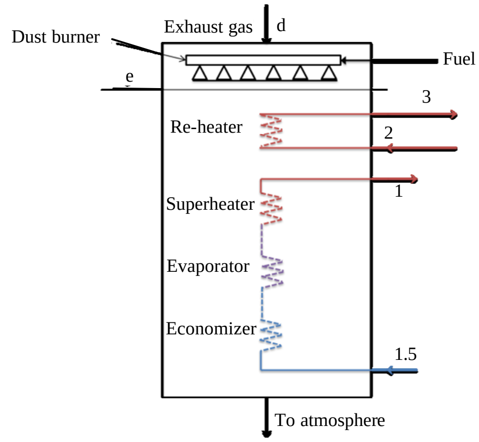

Fig. 3: Schematic diagram of a single pressure HRSG with reheat

since, 2013. Operate on an open basis (Brayton cycle). A number of units are 10 and each unit consists of three main parts: a compressor, a combustion chamber a turbine and has a power output of (125 MW).

The bottoming cycle is Al-Mussaib thermal power plant is located 64 km south of Baghdad and established by the South Korea company Hyundai in 1983 and it has begun in operating, since, 1989. It consist of 4 units, each unit gives 300 MW consists of the steam turbine, condenser, pump, closed feedwater heaters and Deaerating heater (DA).

Heat recovery steam generator: 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 the heat from the gas to water. The hot gases enter the HRSG at the 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. Figure 4 shows 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:

- All the processes are steady state
- The kinetic and potential of energy and exergy are assumed to be negligible

Table 1: Volumetric analysis of natural gas

Substance	Formula	Volumetric analysis (%)	MW (kg/kmol)	LHV (kJ/kg)
Methane	CH_4	75.70	16.04	50020
Ethane	C_2H_6	18.44	30.07	47480
Carbone dioxide	CO_2	2.44	44.01	-
Propane	C_3H_8	2.40	44.09	46360
Nitrogen	N_2	0.72	28.01	-
n-Butane	C_4H_{10}	0.24	58.12	45720

- Atmospheric condition is taken as pressure 1.013 bar and temperature range 273-328 K
- TIT in the gas turbine cycle is range from 1000-2000 K
- The combustion chamber efficiency is 98%
- The compressor pressure ratio Pr_{air} is range from 5-30
- Isentropic efficiencies of compressor and gas turbine are 88-86%, respectively
- The pressure loss at combustion chamber and exhaust are 0.02 and 0.013%, respectively
- The fuel injected into the combustion chamber and duct burner was a natural gas its composition is presented in Table 1
- Inlet steam temperature and pressure T_1 = 538°C, P_1 = 166.713 bar
- Reheat steam temperature and pressure T_3 = 538°C, P_3 = 41.38 bar

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

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

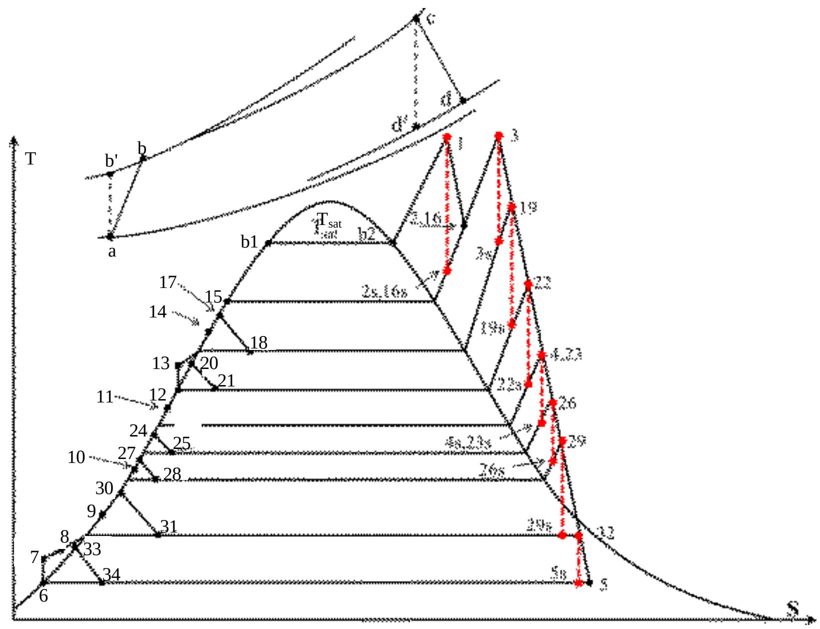

Fig. 4: T-S diagram of the repowered combined cycle power plant

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

Compressor; Mehraban *et al*. (2014):

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

$$\dot{W}_C = \dot{m}_{air}Cp_{air}\left(T_b-T_a\right) \quad (4)$$

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

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

Combustion chamber; Naserabad *et al.* (2015):

$$\dot{m}_{air}Cp_{air}T_b+\dot{m}_f LHV+\dot{m}_f Cp_f T_f = \dot{m}_g Cp_g T_c+\left(1-\eta_{I,C.Ch}\right)\dot{m}_f LHV \quad (6)$$

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

$$P_c = P_b\left(1-\Delta P_{C.Ch}\right) \quad (8)$$

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

$$Cp_g = 0.991615-\frac{6.99703}{10^4}T+\frac{2.7129}{10^7}T^2-\frac{1.22442}{10^{10}}T^3 \quad (9)$$

Gas turbine; Naserabad *et al.* (2015):

$$T_d = T_c\left(1-\eta_{I,GT}\left(1-\left(\frac{1}{Pr_g}\right)^{\left(\frac{\gamma_g-1}{\gamma_g}\right)}\right)\right) \quad (10)$$

$$P_d = P_c\left(1-\Delta P_{exh}\right) \quad (11)$$

$$\dot{W}_{GT} = m_g Cp_g\left(T_c-T_d\right) \quad (12)$$

$$\dot{W}_{net,GTC} = \dot{W}_{GT}-\dot{W}_C \quad (13)$$

Duct burner: Applying the energy balance equation to calculate the mass flow rate of the fuel added in the duct burner (Rohani and Ahmadi, 2014):

$$\dot{m}_g Cp_{g,d} T_d + \dot{m}_{f,db} LHV = \dot{m}_{exh} Cp_{g,r} T_e + \left(1-\eta_{I,db}\right)\dot{m}_{f,db} LHV \quad (14)$$

$$\dot{m}_{exh} = \dot{m}_g + \dot{m}_{f,db} \quad (15)$$

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 >2 kg/sec, since, it may lead to burning the superheater pipes.

Heat recovery steam generator: Using the energy equation for steam/water and gas in HRSG various sections following equations can be written as follows:

Reheater:

$$\dot{m}_{exh} Cp_g \left(T_e - T_{g1}\right) = \dot{m}_3 \left(h_3 - h_2\right) \quad (16)$$

Super heater:

$$\dot{m}_{exh} Cp_g \left(T_{g1} - T_{g2}\right) = \dot{m}_{st} \left(h_1 - h_{b2}\right) \quad (17)$$

Evaporator:

$$\dot{m}_{exh} Cp_g \left(T_{g2} - T_{g3}\right) = \dot{m}_{st} \left(h_{b2} - h_{b1}\right) \quad (18)$$

Economizer:

$$\dot{m}_{exh} Cp_g \left(T_{g3} - T_{g4}\right) = \dot{m}_{15} \left(h_{b1} - h_{15}\right) \quad (19)$$

To obtain the temperatures in each element, first the above equations should be solved. Then, subsequent to solving these equations, the temperature of water, steam and combustion gases in each element can be determined.

The temperature of the exhaust gases entering the economizer and superheater can be written as follows, respectively (Rohani and Ahmadi, 2014):

$$T_{g3} = T_{sat} + T_{pp} \quad (20)$$

$$T_{g1} = T_1 + T_{TD} \quad (21)$$

The total heat available in the exhaust gases:

$$\dot{Q}_{exh,Gt} = \dot{Q}_{GT,add}\left(1-\eta_{I,GTC} + \dot{Q}_{f,db}\right) \quad (22)$$

Steam turbine: The net power output of the steam cycle is:

$$\dot{W}_{net,STC} = \dot{W}_{ST,act} - \sum \dot{W}_{p,act} \quad (23)$$

The thermal efficiency of the steam cycle is:

$$\eta_{I,STC} = \frac{\dot{W}_{net,STC}}{\dot{Q}_{exh,GT}} \quad (24)$$

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

$$\eta_{I,CC} = \frac{\dot{W}_{net,CC}}{\dot{Q}_{GT,add} + \dot{Q}_{f,db}} \quad (25)$$

$$\dot{W}_{net,CC} = \dot{W}_{net,GTC} + \dot{W}_{net,STC} \quad (26)$$

Exergy analysis: The exergy may be defined as the maximum work that can be achieved by bringing a system into equilibrium with its environment. The exergy can be divided integer 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):

$$\dot{E}_{xQ} + \sum_{in} \dot{m}_i ex_i = \sum_{out} \dot{m}_o ex_o + \dot{E}x_W + \dot{E}x_D \quad (27)$$

Where:

ex	= The specific exergy
$\dot{E}x_Q, \dot{E}x_W$ and $\dot{E}x_D$	= The exergy of heat transfer

Work and the exergy destruction. T is the absolute temperature (K) (0) refers to the dead state. The specific physical exergy for air and combustion gases is given by Naserabad *et al*. (2015).

$$ex_{ph,j} = Cp\left(T_j - T_o - T_o \ln\left(\frac{T_j}{T_o}\right)\right) + RT_o \ln\left(\frac{P_j}{P_o}\right) \quad (28)$$

The specific physical exergy for steam and water is:

$$ex_{ph,j} = h_i - h_o - T_o\left(s_i - s_o\right) \quad (29)$$

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

$$ex_{ch,f} = \xi_f \, LHV \quad (30)$$

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:

$$\xi_f = 1.033 + 0.0169\frac{m}{n} - \frac{0.0698}{n} \quad (31)$$

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):

$$\dot{E}x_{D,\,HRSG} = \left(\dot{E}x_e + \dot{E}x_{15} + \dot{E}x_2\right)_{in} - \left(\dot{E}x_1 - \dot{E}x_3 - \dot{E}x_{g4}\right)_{out} \quad (32)$$

Equation 32 show the term $\dot{E}x_{g4}$ is zero because the exhaust gas from the HRSG cannot be used anymore. The term $\dot{E}x_e = \dot{E}x_4$ 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.

$$\eta_{II,\,HRSG} = \frac{\dot{E}x_1 + \dot{E}x_3 - \dot{E}x_{15} - Ex_2}{\dot{E}x_e} \quad (33)$$

$$\eta_{II,\,CC} = \frac{\dot{W}_{net,\,CC}}{\dot{E}x_f + \dot{E}x_{db}} \quad (34)$$

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 cycles proposed above have been coded using Fortran 90 program and results obtained.

It can be shown from Fig. 5, that the mass flow rate of steam decreases as the ambient temperature increase with a different compressor pressure ratio of decreasing in the mass flow rate and temperature of exhaust gases, i.e., the heat available at exhaust becomes less. Figure 6 show that the mass flow rate of steam decreases as the ambient temperature increase at a constant TIT increases with the increase of TIT at a constant ambient temperature giving more heat available from the exhaust gases. Figure 7 shows the relation between thethermal efficiency of the RCCPP and the ambient temperature for different 3 values of AFR 44, 48 and 52 kg of air/kg of fuel. The thermal efficiency of the RCCPP increases as the ambient

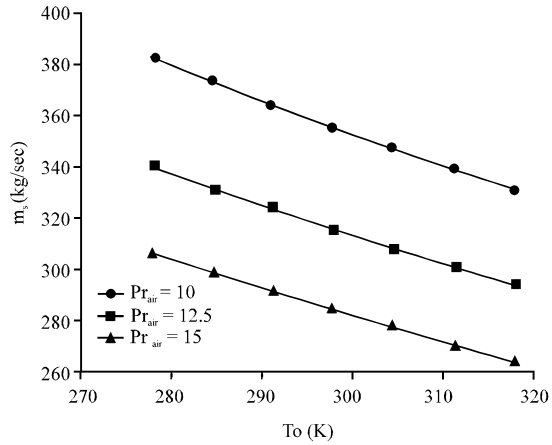

Fig. 5: Effect of ambient temperature o steam mass flow rate with different pressure ratio

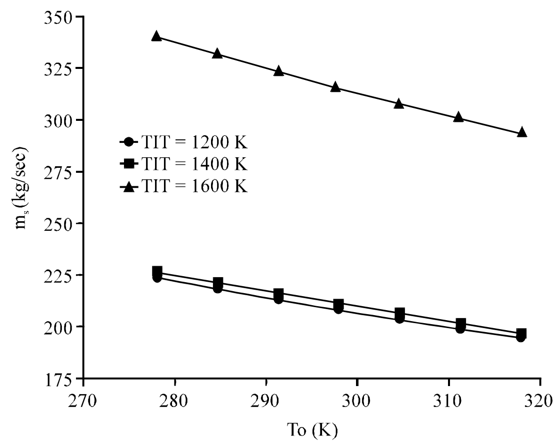

Fig. 6: Effect of ambient temperature on steam mass flow rate with different TIT

temperature increase and AFR decrease, due to a reduction in heatadded to the cycle which is more than the net power output as well as for the exergy of fuel. In other words, the increasesin the steam cycle thermal efficiency morethan the decreases of the gas turbine thermal efficiency the ambient temperature and AFR increases, the reduction in the steam power output less than a reduction in the available heat in the HRSG, therefore, the thermal efficiency of the steam cycle will be increased. The relation between the exergy efficiency at different values of AFR is shown in Fig. 8 which have the same trend of the Fig. 7. It can be seen from Fig. 9 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.

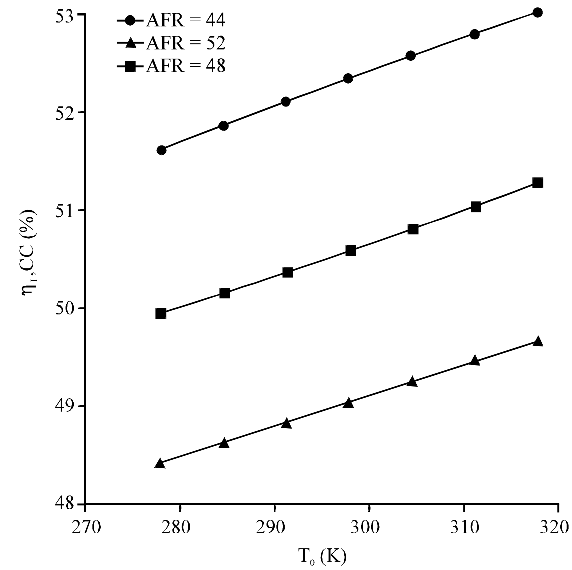

Fig. 7: Effect of the ambient temperature on thermal efficiency of RCCPP for different AFR

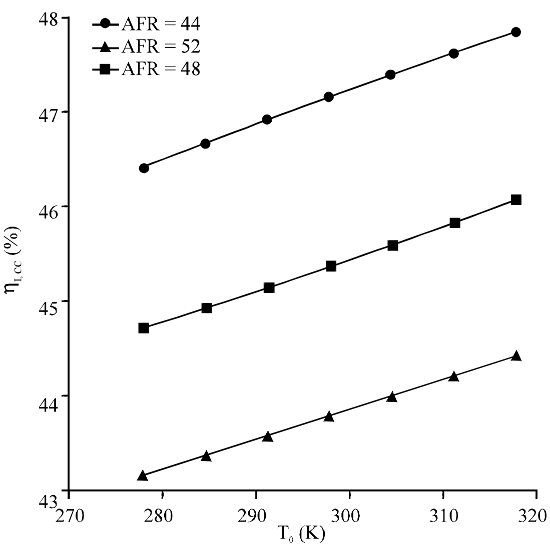

Fig. 8: Effect of the ambient temperature on exergy efficiency of RCCPP for different AFR

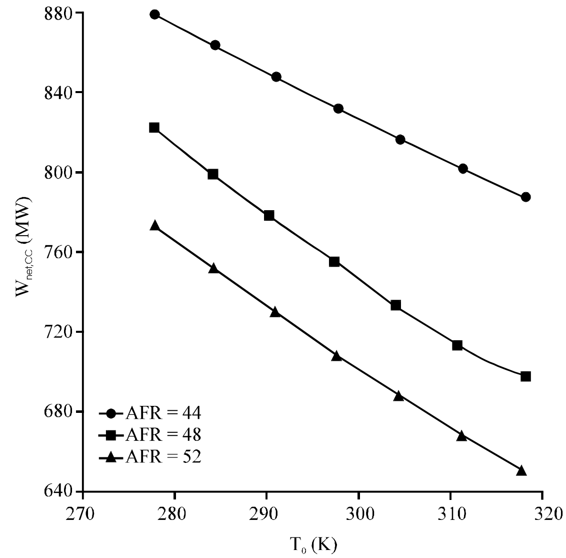

Fig. 9: Effect of ambient temperature on net power output of RCCPP for different AFR

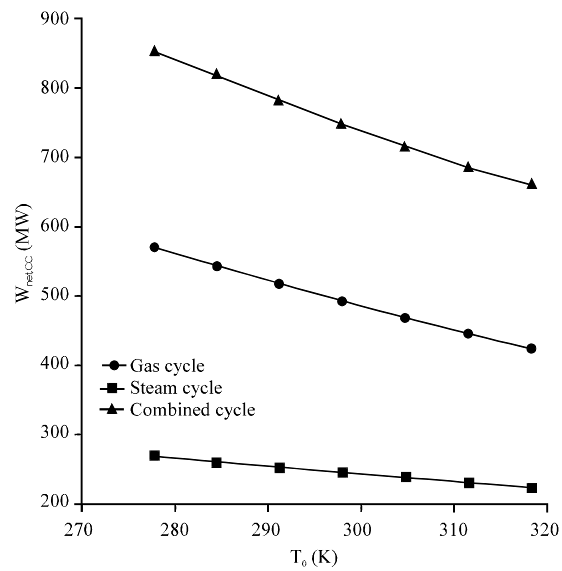

Fig. 10: Effect of ambient temperature on net power output

The effect of ambient temperature on the power output of gas, steam and rrepowered combined cycles shown in Fig. 10. It is found the power output from the gas, steam and repowered combined cycles decreases when the ambient temperature increase the density of the air decreases with the increases of the ambient temperature. So, the fuel mass flow rate will be decreased 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 be decreased, also, the heat available in the HRSG decreases as the mass flow rate of the exhaust gases decrease and consequently the total power output of the RCCPP becomes less. Figure 11 shows that the thermal efficiency of the gas cycle decreases when the ambient temperature increase the power output becomes less the thermal efficiency of the steam cycle increases with the increase of the ambient temperature the reduction in the power output of the steam cycle less than that of the heat available in the exhaust gases which is required to produce the steam.

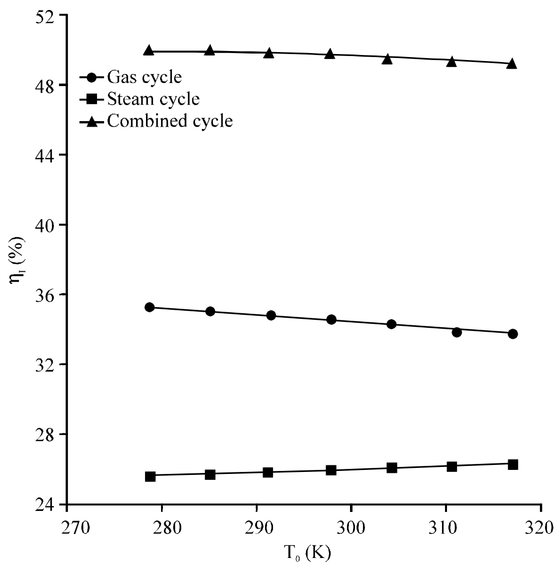

Fig. 11: Effect of ambient temperature on thermal efficiency

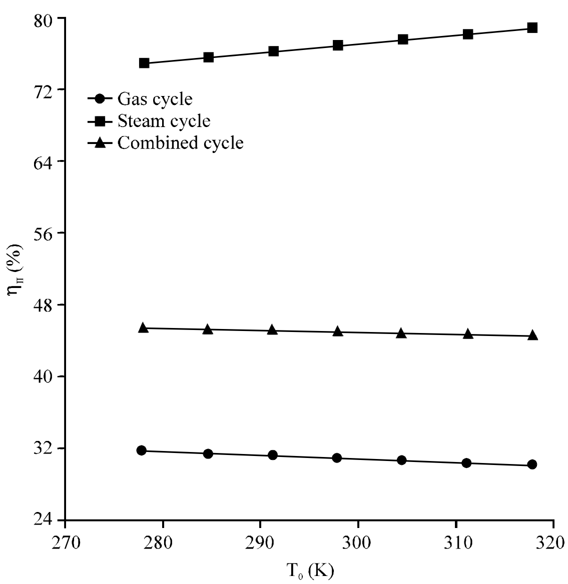

Fig. 12: Effect of ambient temperature on exergy efficiency

Even though the thermal efficiency of the steam cycle increase, the thermal efficiency of the RCCPP will be decreased, due to decrease in the net power output of the RCCPP. Figure 12 shows that the exergy efficiency of the gas cycle decreases when the ambient temperature increase. For the steam cycle the exergy efficiency increases the exergy loss of the exhaust gases becomes less. While the exergy efficiency of the combined cycle decreases with the increase of the ambient temperature

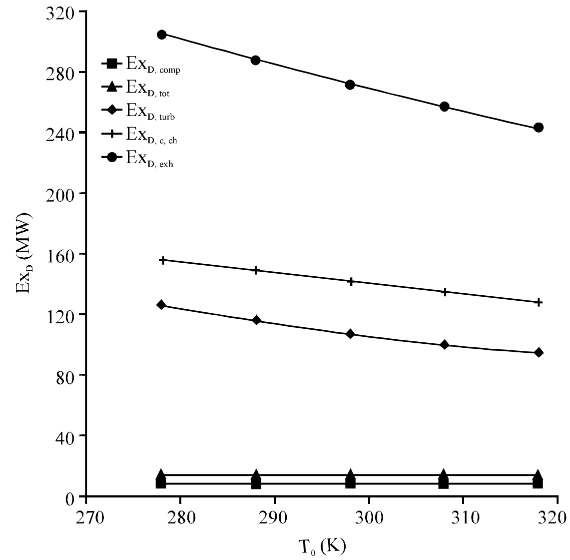

Fig. 13: Effect of ambient temperature on exergy destruction rate of gas cycle component

because the net power output of the RCCPP becomes less. In addition, to that the exergy of the fuel increases with the increase of the ambient temperature.

The ambient temperature has also effected 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. 13. 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, due to the mass flow rate and temperature of the exhaust gases decreases with the increase of the ambient temperature while the rate of exergy destruction for turbine and compressor increasing slightly. Figure 14 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. 15. It can be seen that the exergy destruction of all components, expect condenser increased as ambient temperature increases 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

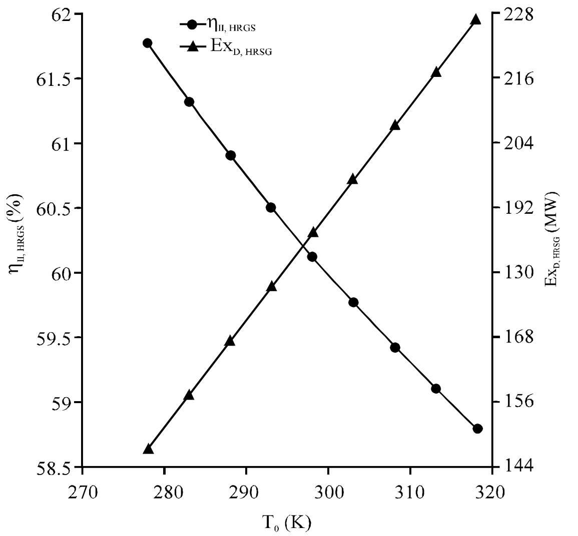

Fig. 14: Effect of ambient temperature on exergy efficiency and exergy destruction rate of HRSG

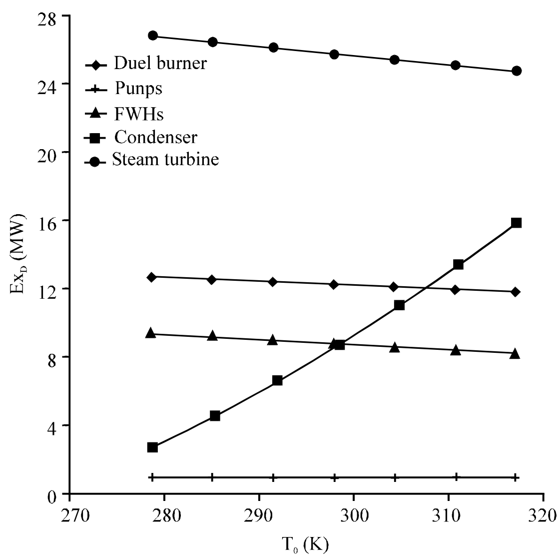

Fig. 15: Effect of ambient temperature on exergy destruction rate of steam cycle component

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. Figure 16 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. 15.

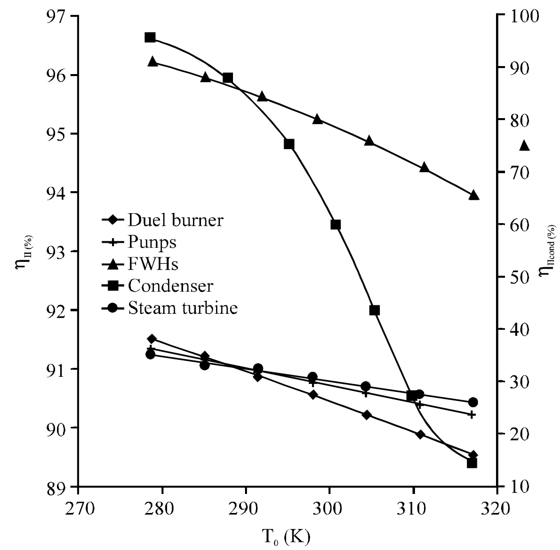

Fig. 16: Effect of ambient temperature on exergy efficiency of steam cycle component

CONCLUSION

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: the decreases of ambient temperature increase the net power output from the RCCPP as well as the thermal efficiency vice-versa. The amount of steam generated in the HRSG decreases with the increase of ambient temperature and increases with the increase of compressor pressure ratio and TIT. 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. The exergy efficiency of the HRSG increases and the exergy destruction in the HRSG decreases as the ambient temperature increases.

NOMENCLATURE

Latin symbols:

Cp = Specific heat at constant pressure (kJ/kg.k)
ex = Specific exergy (kJ/kg)
$\dot{E}x$ = Total exergy rate (kW)
$\dot{E}x_D$ = Exergy dsestruction rate (kW)
h = Specific enthalpy (kJ/kg)
LHV = Lower heating value (kJ/kg)
$\dot{m}$ = Mass flow rate (kg/sec)
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
C.Ch = Combustion Chamber
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
ph = Physical
pp = Pinch point
st = Steam
sat = Saturated
STC = Steam Turbine Cycle
TD = Temperature Difference

Abbreviations:
AFR = Air-Fuel Ratio
CCPP = Combined Cycle Power Plant
HRSG = Heat Recovery Steam Generator
RCCPP = Repowered Combined Cycle Power Plant
TIT = Turbine Inlet Temperature
TTD = Terminal Temperature Difference

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