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Performance Augmentation of a Combined Cycle Power Plant with Waste Heat Recovery and Solar Energy

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***Abstract*—**In the present time, energy crises is considered a severe problem across the world. For the protection of global environment and maintain ecological balance, energy saving is considered one of the most vital issues from the view point of fuel consumption. As the industrial sectors everywhere continue efforts to improve their energy efficiency, recovering waste heat losses provides an attractive opportunity for an emission free and less costly energy resource. In the other hand the using of solar energy has become more insistent particularly after the high gross of prices and running off the conventional energy sources. Therefore, it is essential that we should endeavor for waste heat recovery as well as solar energy by making significant and concrete efforts. For these reasons this investigation is carried out to study and analyze the performance of a power plant working by a combined cycle in which heat recovery system generator (HRSG) gets its energy from the waste heat of a gas turbine unit. Evaluation of the performance of the plant is based on different thermal efficiencies of the main components in addition to the second law analysis considering the exergy destructions for the whole components. The contribution factors including the solar as well as the wasted energy are considered in the calculations. The final results have shown that there is significant exergy destruction in solar concentrator and the combustion chamber of the gas turbine unit. Other components such as compressor, gas turbine, steam turbine and heat exchangers having insignificant exergy destruction. Also, solar energy can contribute by about 27% of the input energy to the plant while the energy lost with exhaust gases can contribute by about 64% at maximum cases.

***Keywords*—**Solar energy, environment, efficiency, waste heat, steam generator, performance, exergy destruction.

I. Introduction

INDUSTRIAL waste heat refers to energy that is generated in industrial processes without being put to practical use. Sources of waste heat include hot combustion gases discharged to the atmosphere, heated products exiting industrial processes, and heat transfer from hot equipment surfaces. The exact quantity of industrial waste heat is poorly quantified, but various studies have estimated that as much as 20 to 50% of industrial energy consumption is ultimately discharged as waste heat. While some waste heat losses from industrial processes are inevitable, facilities can reduce these losses by improving equipment efficiency or installing waste heat recovery technologies. Waste heat recovery entails capturing and reusing the waste heat in industrial processes for heating or for generating mechanical or electrical work. Example uses for waste heat include generating electricity, preheating combustion air, preheating furnace loads, absorption cooling, and space heating. The recovered heat can replace fossil energy that would have otherwise been used in the combustion process. Such methods for recovering waste heat can help facilities significantly reduce their fossil fuel consumption, as well as reduce associated operating costs and pollutant emissions [1].

In the other hand solar energy has experienced phenomenal growth in recent years due to both technological improvements resulting in cost reductions and government policies supportive of renewable energy development and utilization. While the cost of solar energy has declined rapidly in the recent past, it still remains much higher than the cost of conventional energy technologies. Like other renewable energy technologies, solar energy benefits from fiscal and regulatory incentives and mandates, including tax credits and exemptions, feed-in-tariff, preferential interest rates, renewable portfolio standards and voluntary green power programs in many countries. Despite the huge technical potential, development and large-scale, market-driven deployment of solar energy technologies world-wide still has to overcome a number of technical and financial barriers [2].

Integrating the solar energy with the waste heat to augment the performance of the thermal power plants has a great interest in the matter of producing electrical energy, which is under the category of Integrated Solar Combined Cycle (ISCC) technology, or hybrid solar-gas. This solution represents an optimum sustainable and efficient combination by integrating a solar field (parabolic trough collectors or a tower and heliostats) with a combined cycle plant, which combines gas and steam cycles together, by depending on the waste heat coming out of gas turbine unit.

The hybridization of technologies confers great advantages to the combined solution: the solar portion operates continuously as designed including when the available radiation fluctuates. The efficiency of the conventional combined cycle plant is improved and fuel consumption is reduced, therefore reducing the CO_2 emissions.

In comparison with the existing Rankin cycle plants, an ISCC plant offers three principal advantages. First, solar energy can be converted to electric energy at a higher efficiency. Second, the incremental unit cost for the larger steam turbine in the integrated plant is less than the overall

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unit cost in a solar-only plant. Third, an integrated plant does not suffer the thermal inefficiencies associated with the daily startup and shutdown of the steam turbine [3].

Based on the pre-mentioned realities and justifications about the solar combined cycle power plant, this study is carried out to simulate and analyze the performance of a pre-design power plant site in Misurata city, located in the sun built of Libya. A computer program in visual Basic 6.0 has been developed to formulate the problem where it satisfies a flexibility to change input data and also some design conditions. The calculations are based on the contribution of the solar energy and the wasted energy from the gas turbine on the whole thermal efficiency of the plant.

II. ISCC Plant Layout

The proposed plant consists of a solar field, gas turbine unit and steam turbine unit. Solar field includes a number (M) of rows of the concentrating collectors. These collectors are a cylindrical type with an aperture area of 3m^2 for each one. Each row contains a number (N) of the collectors in series connection as seen in Fig. 1. Also, the drop in collector efficiency due to series connection is considered in the calculations.

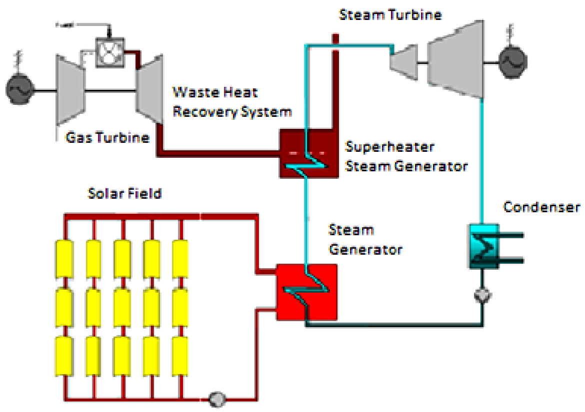

Fig. 1 Description of the proposed ISCC plant

The gas turbine unit consists of a compressor, combustion chamber where the fuel is burned with the air resulting maximum combustion temperature in the cycle. The products of the combustion are expanded adiabatically in the turbine. Actual combustion processes and variable specific heats for the compression processes are considered in the calculations. Steam turbine unit is working with a simple Rankin cycle with high pressure of 90 bars while the condenser pressure is taken as 10 kPa. However, in order to assure a constant power generation from the steam turbine unit, an auxiliary heating system is added at the exit of the solar field in order to produce a saturated steam to the heat recovery system generator at times of less or absence solar radiation. An isentropic efficiency of 80% for the steam turbine is considered in the calculations.

III. Analysis of the ISCC Plant

Each component of the integrated solar combined cycle power plant is treated separately and its mathematical model is developed in the following sections.

A. Solar Concentrator Field

The simulation of the solar concentrator is carried out on the basis of the data input to the model and the given values for the collector parameters. The calculated parameters for the concentrator included: heat losses factor (U_L), collector efficiency factor (F'), and the heat removal factor (F_R). Thus, the absorbed heat from the concentrator can be expressed by [4]:

$$Pr \equiv exp\left(\frac{\bar{s}_T^o}{R}\right) \tag{1}$$

To find the value of the U_L there are many methods, some are analytical and others are semi-empirical. In some analytical solutions, the value of the cover temperature is assumed equal to that of ambient, since it is closed to it, then we calculate the value of the loss coefficient , finally we check the solution by making heat balance around the cover by iterative method, as in [4]-[6].

One of the semi-empirical solutions is given in [7], which depends on Hottel-Woertz, gives reasonable value for this coefficient for a range of temperatures between 60 and 240^{o}C, and heat transfer coefficient between 15 and 60 W/m^2 ^{o}C. Outer of this range the percentage of error is high, about 18%. So by using the following analytical method the accuracy is higher with less percentage of error [8]. This method has been used in the current study as follows:

$$U_L = \left[U_{L1} + U_{L2} + U_{L3}\right]^{-1} \tag{2}$$

where

$$U_{L1} = \left[C(T_1 - T_2)^{0.25} + \frac{\sigma\left(T_1^2 + T_2^2\right)\left(T_1 + T_2\right)}{\frac{1}{\varepsilon_1} + \left(D_1 / D_2\right)\left(\frac{1}{\varepsilon_2} - 1\right)} \right]^{-1} \tag{3}$$

$$U_{L2} = \left(D_1 / D_2\right)\left[h_w + \sigma\varepsilon_2\left(T_2^2 + T_a^2\right)\left(T_2 + T_a\right)\right]^{-1} \tag{4}$$

$$U_{L3} = \left(D_1 / D_g\right)\left(t_g / k_g\right) \tag{5}$$

In the previous equation t_g and k_g are the glass thickness and its conductivity factor where the constant (C) in (2) is given by:

$$C = \frac{17.74}{\left(T_{ab} + T_c\right)^{0.4} D_{ab}\left(D_{ab}^{-0.75} + D_c^{-0.75}\right)} \tag{6}$$

A small iteration for the cover temperature (T_c) with a maximum error of 2% has been developed and given by:

$$T_c = T_a + 0.163\left(D_{ab}/D_c\right)^{0.4} h_w^{-0.67} ET \tag{7}$$

where

$$ET = \left[2 - 3\varepsilon_{ab} + \left(1 + 3\varepsilon_{ab}\right)T_{ab}/100\right]\left(T_c - T_a\right). \tag{8}$$

From the above equations we notice that there is no need for a numerical method as in the other analytical methods, just we start with an estimate value for the cover temperature T_1, and then we find the overall loss coefficient. Finally to get sure of the value of T_1 we have to determine the useful energy gain from (1), after finding the value of the removal factor F_R and efficiency factor $F^{'}$ as follows:

$$F' = \frac{U_o}{U_L} \tag{9}$$

where U_o is the overall heat transfer coefficient and is given by:

$$U_o = \left[\frac{1}{U_L} + \frac{D_o}{h_{fi} D_i} + \frac{D_o \ln\left(D_o / D_i\right)}{2K}\right]^{-1} \tag{10}$$

In this equation h_{fi} represents the heat transfer coefficient of the heating fluid inlet to the tube. Then T_1 can be obtained from the following equation:

$$T_1 = T_{fi} + \frac{Q_u / A_a}{\frac{A_r}{A_a} U_l F_R}\left(1 - F_R\right) \tag{11}$$

After that we check the solution by repeating the calculations with this new value of T_1 until the value of the useful energy get fixed. Finally the value of the heating fluid output temperature can be determined from the following equation:

$$T_{fo} = T_{fi} + \frac{Q_u}{\dot{m}_f C_{pf}} \tag{12}$$

B. The Gas Turbine Unit

A gas turbine is an engine which allows the conversion of the energy of fuel in some form of useful power, such as mechanical power. In the ISCC cycle the main purpose of analyzing the gas turbine unit is to evaluate the waste energy within the exhaust gases. Estimating the exhaust gases mass flow and its temperature is the main goal of the GT mathematical solution. The procedure to achieve this goal is to evaluate the compressor, the combustion chamber and turbine performances.

The thermal analysis of the gas turbine unit is carried out by considering air with variable specific heat. Thus, to find the isentropic processes through the compressor or the turbine the relative pressure *(Pr)* is taken in consideration, which has a relationship with the entropy as follows [9]:

$$Pr \equiv exp\left(\frac{\bar{S}_T^o}{R}\right) \tag{13}$$

where

$$\bar{S}_T^o = R\left(a_1 lnT + a_2 T + \frac{a_3}{2}T^2 + \frac{a_4}{3}T^3 + \frac{a_5}{4}T^4 + a_7\right) \tag{14}$$

In the entropy equation the constants a_1 to a_7 are temperature coefficients according to the type of the gas. As shown in these equations the temperature is the main variable, this means that if temperature is known we can find Pr and vice versa. So in the compressor the relative pressure (Pr_2) at the end of the compression process is function of the compressor pressure ratio (CPR) as follows:

$$Pr_2 = Pr_1 \times CPR \tag{15}$$

Then, the exit temperature of the compressor as a function of Pr_2 is calculated where it is used to calculate the enthalpy of air with variable specific heats from the following relation [9]:

$$\bar{h}_T = \bar{R}T\left(a_1 + \frac{a_2 T}{2} + \frac{a_3 T^2}{3} + \frac{a_4 T^3}{4} + \frac{a_5 T^4}{5} + \frac{a_6}{T}\right) \tag{16}$$

Applying combustion equation for one mole of octane (as a fuel used in the present case) with dry air yields:

$$C_8H_{18} + X\ D.A \rightarrow 8CO_2 + 9H_2O + X\ D.A - 12.5O_2 \tag{17}$$

where X is the total number of moles of the dry air supplied per one mole of fuel. Accordingly, for adiabatically combustion, the products temperature is calculated by trial and error. The products are expanded in the gas turbine giving the output power from the turbine.

C. The Steam Turbine Unit

The steam turbine is an engine in which a steam flow, at high pressure and temperature, is expanded transforming its energy into kinetic energy, which is as well converted into work by moving the rotational parts of the turbine. At the inlet conditions to the steam turbine, the state of the steam of the point (*i*) is superheated. Accordingly, the state of the steam is given by [10]:

$$h_{i(\text{sup.})} = h_{i(g)} + C_{p,\text{sup.}}\left(T_{\text{sup.}}\right)$$

$$C_{p,\text{sup.}} = R\left(\frac{C_p}{R}\right) \tag{19}$$

$$\left(\frac{C_p}{R}\right) = 4.132 - 0.001559T + 0.5315\times10^{-5}T^2 - 0.4209\times10^{-8}T^3 + 0.1284\times10^{-11}T^4 \tag{20}$$

where R is the gas constant of steam and taken as 0.462 KJ/Kg. K.

$$S_{i(\text{sup.})} = C_{pw} \ln\left(\frac{T_{\text{sup}}}{273.15}\right) + \frac{h_{fg}}{T_{\text{sup.}}} + C_{p,\text{sup.}} \ln\left(\frac{T_{\text{sup.}}}{T_{sat.}}\right) \quad (21)$$

where h_{fg} is the latent heat of the steam and given as a function of temperature by **[** 11].

$$h_{fg} = 2501.897 - 2.07t + 1.192 \times 10^{-3} t^2 - 1.586 \times 10^{-5} t^3 \quad (22)$$

Hence, for an isentropic condition in the steam turbine, the exit point *(i+1)* of steam is usually in wet region where the dryness fraction $\boldsymbol{x}$ can be calculated and is used to calculate the enthalpy of the steam.

D. Heat Recovery Steam Generator

The heat recovery steam generator (HRSG) is an important subsystem of a combined power plant which uses the energy from the exhaust gases of the turbine for transferring heat to water and generating steam at high temperature and pressure. The exhaust gases leave the gas turbine at approximately ambient pressure and at very high temperature (500°C to 600°C). This energy is used for the HRSG to produce steam.

Although there are many configurations of HRSG, most of them are divided in the same number of sections as the steam turbine. There is one section for high pressure (HP), low pressure (LP) and sometimes for an intermediate pressure (IP).

A HRSG is composed basically of individual heat exchangers which exchange the energy from the exhaust gases of the turbine with the water/steam of the Rankin cycle.

The water enters first in the economizer for being pre-heated and then goes through the evaporator where the steam is generated at constant pressure and temperature. Finally the steam is superheated in the superheater. After that, the superheated steam enters the turbine where it is expanded and the power is generated.

The majority of the heat is transferred by convection and for increasing the heat surface finned tubes are used. As the heat transfer on the waterside is much higher than on the exhaust gas side, due to the bigger temperature difference between gas and water than between gas and steam, the fins are used on the gas side to rise the heat transfer.

IV. Exergy Analysis of the ISCC Components

Exergy is a measure of the quality or grade of energy and it can be destroyed in the thermal system. The second law states that part of the exergy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is destroyed within the system due to irreversibilities. The second law of thermodynamics uses an exergy balance for the analysis and the design of thermal systems [12].

Exergy not only can be destroyed by irreversibilities but also can be transferred to a system or from a system, as in losses accompanying heat transfers to the surroundings. Improved resource utilization can be realized by reducing exergy destruction within a system and/or losses.

An objective in exergy analysis is to identify sites where exergy destructions and losses occur and rank order them for significance. This allows attention to be centered on the aspects of system operation that offer the greatest opportunities for improvement [13].

The exergy rate balance for a control volume can be written in the following general form:

$$\frac{dE_{cv}}{dt} = \sum_j \left(1 - \frac{T_o}{T_j}\right)\dot{Q}_j - \left(\dot{W}_{cv} - P_o \frac{dV_{cv}}{dt}\right) + \sum_i \dot{m}_i e_{fi} - \sum_e \dot{m}_e e_{fe} - \dot{E}_d \quad (23)$$

The general equation of exergy destruction for a steady flow system can be expressed by [14]:

$$EXD = T_o\, S_{gen} = T_o \left(\sum m_e\, s_e + \frac{Q_{out}}{T_{b,out}} - \sum m_i\, s_i - \frac{Q_{in}}{T_{b,in}}\right) \quad (24)$$

where S_{gen} is the entropy generation , $T_{b,out}$ and $T_{b,in}$ are the temperatures of the system boundary where heat is transferred out and into the system, respectively.

Based on the developed equation, the entropy generation for each component of the solar combined cycle is given below**.**

A. An Adiabatic Air Compressor & Gas Turbine

$$S_{gen} = m\ (s_e - s_i) = m\ (C_p \ln\frac{T_e}{T_i} - R \ln\frac{P_e}{P_i}) \quad (25)$$

where m is the mass flow rate and represents air mass (m_a) in the compressor, while it is mixture of the fuel and air (m_f + m_a) in the gas turbine, respectively.

B. An Adiabatic Combustion Chamber

$$S_{gen} = S_{prod} - S_{react} = \sum N_P\, \bar{S}_P - \sum N_r\, \bar{S}_r \quad (26)$$

where the air and product gases are at a total pressure (pressure after compressor) and the entropies in (26) are to be calculated at the partial pressure of each components of the mixture.

C. An Adiabatic Steam Turbine

$$S_{gen} = m_s\ (s_e - s_i) \quad (27)$$

where m_s is the steam mass flow rate, whereas s_e and s_i are the ACTUAL entropy at exit and inlet sections of the steam turbine, respectively.

D. Condenser of the Rankin Cycle

$$S_{gen} = m_s\ \ (s_e - s_i + \frac{Q_{cw}}{T_o}) \quad (28)$$

where Q_{cw} is the rate of heat rejected to the cooling water per Kg of steam and the sink temperature is assumed to be T_o.

E. An Adiabatic Heat Exchanger

Including steam generator and solar- Rankin heat exchanger:

$$S_{gen} = m_h\,(s_{e,h} - s_{i,h}) + m_c\,(s_{e,c} - s_{i,c}) \qquad (29)$$

F. Solar Concentrator

The entropy generating (irreversible) process through the solar concentrator assuming that the absorbing plate is at a uniform temperature and steady state conditions is given by [15]:

$$S_{gen,c} = m_f\,(S_{f,out} - S_{f,in}) + \frac{Q_{loss}}{T_O} - \frac{(\tau\,\alpha)_e\,I}{T_{ab}} \qquad (30)$$

Where it is assumed that the concentrator is working between a maximum temperature T_{ab}, and a minimum temperature T_o.

V. Results and Discussion

The simulation results of the main parameters affecting the performance of the integrated solar combined cycle power plant are presented in this section. The input data to the model included:

M_w = 1kg/s, CPR= 14, M =30, N =30, A_c = 3 m^2.

Fig. 2 shows the concentrator efficiency versus the mass ratio (MR) at different values of solar radiation. The mass ratio (MR) means the ratio of the mass of working fluid in the solar loop to the mass of water in the Rankin cycle.

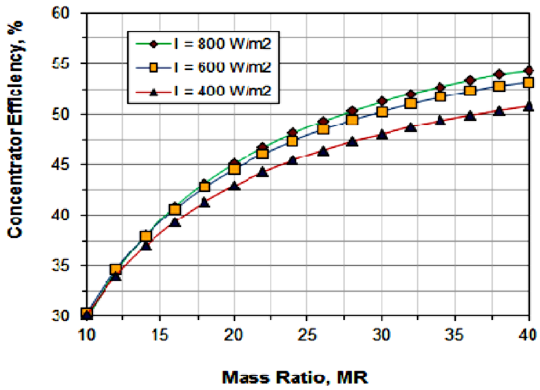

Fig. 2 Concentrator efficiency versus mass ratio (MR) at different values of incident solar radiation

It is seen from the figure that the efficiency increases as MR increases due to the decrease of the thermal losses which occurred at the higher mass flow rate through the solar field. Maximum efficiency of about 54% is achieved in the figure. Looking at the variation of the incident solar radiation, it is seen that the higher efficiency curve occurred at the higher value of the incident solar radiation. This is interpreted by the higher increase of the incident solar radiation which is meeting by small increase in the inlet fluid temperature where there is no energy storage. So, the figure shows that the variation of incident solar radiation represents a dominant influence in the concentrator efficiency.

Fig. 3 shows the solar contribution factor versus the mass ratio, MR at different values of solar radiation. The solar contribution factor is defined by the amount of the heat taken by the solar field relating to the total amount of heat supplied to the steam cycle. It is shown from the figure that the solar contribution increases as the mass ratio increases due to the enhancement of the concentrator efficiency which occurred at the higher values of MR.

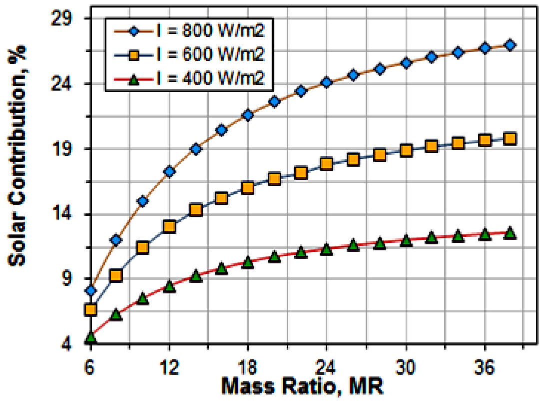

Fig. 3 Solar contribution factor versus mass ratio (MR) at different values of incident solar radiation

Also, the solar contribution increases with increase of the solar radiation which seem to be logically. Maximum contribution of about 27 % is attained at MR = 36 and I = 800 W/m^2 while the corresponding minimum value is 13 % at the same MR, but with I = 400 W/m^2. Other factors such as, gas turbine efficiency (42%), steam turbine efficiency (43%), exhaust gases contribution (63%), auxiliary contribution (10%) are resulted from the simulation results under the given conditions.

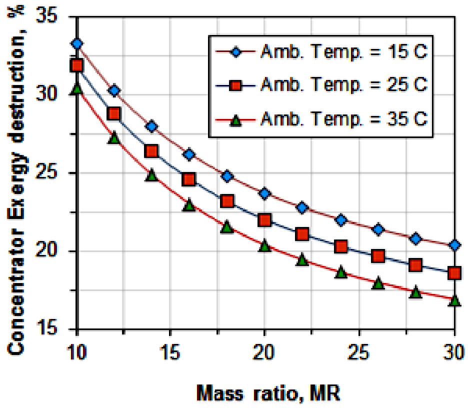

Fig. 4 Exergy destruction of the solar concentrator at different surrounding temperatures

The exergy destruction percent (as a ratio of the incident solar radiation, I = 800 W/m^2) versus MR with different surrounding temperature (T_o) is presented in Fig. 4. As seen, the value of exergy destruction decreases as MR increases referring to the decrease of thermal losses accompanying to the higher value of MR. Also, as the surrounding temperature increases, the exergy destruction decreases . Maximum exergy destruction (33% from the incident radiation) occurred at minimum MR and lower surrounding temperature as seen in the figure.

Fig. 5 presents the gas turbine efficiency versus CPR for burning of one kmole of octane (C_8H_{18}) with excess air ratio of 200% and ambient temperature of 298K. The overall efficiency is presented also in the figure. Here in this figure, the gas turbine efficiency increases as CPR increases due to the higher output of the turbine which occurred at the higher values of combustion temperature at the end of the combustion process. But, this temperature is limited also by the metallurgical conditions that the turbine blades can withstand.

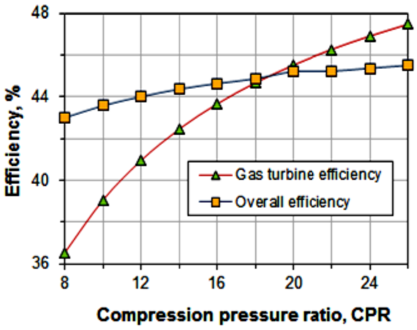

Fig. 5 Gas turbine efficiency and overall efficiency versus CPR

The overall efficiency is presented versus CPR also in Fig. 5, where its value increases as CPR increases due to the increase of power output from gas turbine with higher value of CPR. But, one can find from the figure that there is a convergence between the gas turbine and overall efficiencies up to CPR =19. This trend is because the overall efficiency is a function of gas turbine efficiency and steam turbine efficiency where the later decreased as CPR increases.

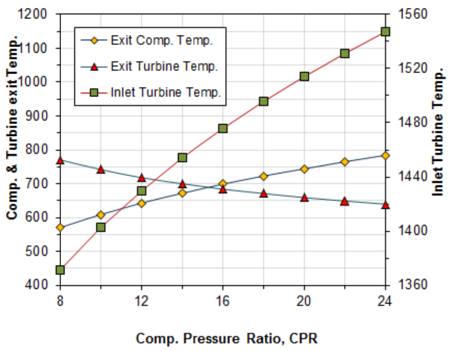

Fig. 6 Various temperatures of gas turbine unit versus compression pressure ratio, CPR

Fig. 6 presents various temperatures concerning the gas cycle versus CPR. It is clear from the figure that the temperature of air entering the combustion chamber increases as CPR increases. This leads to increase the maximum temperature in the cycle. On the other hand, the exit temperature of the products decreases as CPR increases due to the decrease of heat rejected at higher CPR. Maximum temperature of 1550K at CPR = 24 is attained in the figure.

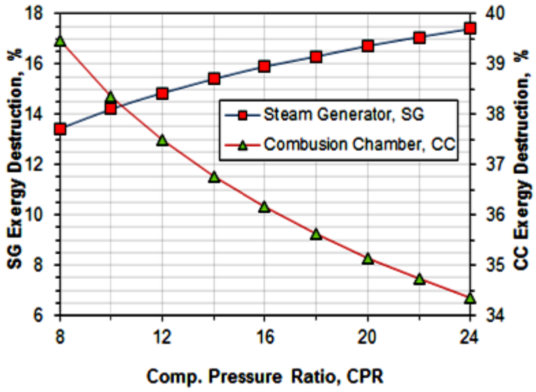

Fig. 7 Exergy destruction of combustion chamber and steam generator versus CPR

Fig. 7 shows the variation of exergy destruction for the main components of gas turbine unit (combustion chamber and steam generator). The exergy destruction of combustion chamber decreases as CPR increases as seen. This is related to the increase of air temperature entering the combustion chamber compared to the increase of the maximum temperature. For example, the air temperature increases by about 56°C when CPR increases from 12 to 16, while the maximum temperature increases by about 46°C for the same increase of CPR. Maximum exergy destruction with value of 39% occurred at CPR = 8 as seen in Fig. 7. However, the exergy destruction for steam generator decreases as CPR increases which is related to the decrease of product temperature entering the steam generator. Maximum exergy destruction (17% from input heat to gas turbine) in steam generator occurred at minimum CPR as seen in the figure. The exergy destruction for the other components of the integrated combined power plant is not presented here where their values are not significant and ranges from 1% to 5%.

VI. Conclusions

In this paper the thermodynamic analysis of an integrated solar combined cycle is carried out to augment the performance of a thermal power plant by waste heat and solar energy. This study is based on the first and second law of thermodynamics, in which an exergy analysis of the ISCC components is carried out. Here, the following can be concluded:

- The integrated solar combined cycle can be operated by high efficiency compared to the gas turbine and steam turbine efficiency under the same operating conditions. The solar energy can contribute by about 27% of the energy supplied to the Rankin cycle.
- The wasted energy of the gas turbine can contribute by about 64%.

- Maximum exergy destruction can be occurred in the combustion chamber i.e. 39% of the input energy to gas turbine while the exergy destruction in the concentrating collector is 33% from the incident solar radiation.

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