Study the Effect of Using Offset Strip Fin Heat Exchanger on the Efficiency of Combined Gas Turbine with Air Bottom Cycle

Rida Ali Hmouda and Hussain Alrubaie

Abstract— The main objective of this paper is to research on the effectiveness of the development of the industrial gas turbine, which works in from a simple cycle to gas turbines stations that work in a Combined Joule/Joule Cycle (CJJC) (gas / air).This study utilizes the computation of the basic thermal design of the simple cycle gas turbines in order to identify the economical indicators for the thermal process of producing electricity. The computation results shows That the depression of the thermal economical indicators of these stations (η_S=36%) gave rise on both the temperature of the exhaust gases (T_4=528.4°C) and The quantity of the heat contained in the turbine's exhaust discarded in the atmosphere (11.17MW).

To solve these negative problems of the basic thermal design that would increase the quantity of the electrical energy generation and to save the contamination at the lower level. So in this paper the heat exchanger has been designed, which has been considered an important element in the process of utilizing the thermal energy of exhaust gases. The study on the effectiveness of the Combined Joule/Joule Cycle in industrial power plants, Where his study has been designed Offset Strip-Fin heat exchanger has been established by using the mathematical model which was resulted to a thermal efficiency increase up to (η_C=43.5%).

Keywords— Combined Joule/Joule Cycle, Gas turbine, Offset Strip-Fin heat exchanger.

I. Introduction

THE last decade has witnessed remarkable improvements in industrial gas turbine size and performance and the coming years are holding the promise of even more progress in these fields [1]. Due to ever increasing costs of fossil fuels and awareness of the impact on the environment of burning fossil fuels, it is sought to decrease fossil fuel consumption. Various technological developments are employed to lower the fuel consumption and emission of industrial gas turbine units, such as high temperature materials and advanced combustion technologies. However all of modern gas turbine units operate in a simple cycle, i.e., all the heat contained in the turbine's exhaust is discarded to the atmosphere. This lost heat represents anywhere from 65-72 % of the fuel energy, depending on the unit's thermal efficiency. Converting this residual heat into power could provide in excess of clean electrical power of much-needed capacity to electrical power grid. Combined gas and steam turbines cycle looks like logical approach in raising industrial plant efficiency, but this concept is inefficient in areas suffering from water scarcity. The Recently Propose a new design for Combined cycle valid for use in industrial gas turbine. Joule Joule cycles can be combined by an air-gas Offset Strip Fin heat exchanger the exhaust of the primary gas turbine is sent to a heat exchanger, which, in turn, heats the air in the bottom cycle. Air is expanded in the turbine to generate additional power. In comparison to the Joule/Rankin combined cycle, this design does not require bulky steam equipment (boiler, steam turbine, condenser), or a water processing unit [2].

Rida Ali Hmouda is with the Department of Mechanical Engineering University of Misurata, Libya, (e-mail: hmouda11@yahoo.com).

Hussain. Alrubaie was with College of Engineering Technology Hoon, Libya. He is now with the Department of Mechanical Engineering University of Baghdad, Iraq, (e-mail: hussxycc@yahoo.com).

II. The Thermal Analysis of the Simple Cycle Gas Turbine

In present study has been using the following method to procedure thermal analysis of gas turbine at design properties and operating conditions for this unit, Where this method relies on procedure thermal calculation of the basic parts of the gas turbine, in relative terms (per 1 kg of air the inside the compressor). Also been dealing with the turbine blades cooling system, as a separate cycle to extend the cooling air inside the gas turbine [3].

A. Calculate Thermal Economic Indicators for the Gas Turbine Simple Cycle

Air consumption rate of gas turbine (ton/hr), calculated as:

$$G_a = \frac{3600 \times N_e^{GT}}{He \times \eta_M \times \eta_G} \tag{1}$$

Where N_e^{GT} power output (MW), H_e is the specific work for gas turbine (kJ/kg), η_M is the mechanical efficiency and η_G is the efficiency of the generator.

The Fuel consumption rate of the gas turbine is calculated as follows:

$$BGT = G_t \times G_a \tag{2}$$

Where G_t is The relative rate of fuel consumption for the gas turbine (kg Fuel/kg Air).

The thermal Efficiency is calculated as follows:

$$\eta_S = \frac{3600 \times N_e^{GT}}{BGT \times Q_{CV}} \quad (3)$$

Where (Q_{CV} = 44300 kJ/kg) heating value for the fuel used.

B. Combined Joule/Joule With Air Bottoming Cycle

Like the steam bottoming cycle in a combined cycle, the air bottoming cycle (ABC) may utilize heat rejected from a gas turbine [4]. A general flow sheet diagram of the CJJC is shown in Fig. 1. In the bottom cycle the ambient air (1) is drawn through a filter and is compressed in the air compressor (AC). The compressed air (2_A) is heated in Offset Strip-Fin heat exchanger before it enters (3_A) air turbine. In the turbine the air is expanded while shaft work is generated. After the turbine the air is exhausted to a stack. The work generated in the turbine is sufficient to drive the compressor and a generator (G), (CC) denotes the combustion chamber and (G_{CO}) The relative amount of air drawn from the compressor to cool the turbine blades [3].

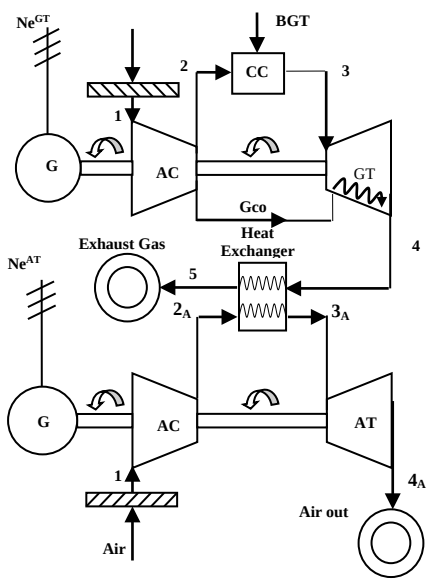

Fig. 1 Combined Joule/Joule with Air Bottoming Cycle.

C. Calculate Thermal Economic Indicators for the Combined Joule/Joule Cycle

The thermal efficiency of the CJJC is calculated as follows:

$$\eta_C = \frac{3600 \times N_e}{BGT \times Q_{CV}} \quad (4)$$

Where N_e (MW) the electrical power produced for CJJC and calculated from the following relationship:

$$N_e = N_e^{GT} + N_e^{AT} \quad (5)$$

Where N_e^{AT} (MW) the electrical power produced for ABC and calculated from the following relationship:

$$N_e^{AT} = \frac{He_A \times \eta_M \times \eta_G \times G_a}{3600} \quad (6)$$

Where H_{eA} is the specific work for ABC (kJ/kg).

III. The Offset Strip Fin Heat Exchanger

In this paper the heat exchanger has been designed, which is considered an important element in the process of exploitation of thermal energy of the exhaust gases, where has been studied and designed cross flow heat exchanger from type of plate fin.

The plate fin heat exchanger (PFHE) are a form of compact heat exchanger consisting of a stack of alternate flat plates called "parting sheets" and fin, brazed together as a block. Plate fin heat exchanger are widely used in various industrial applications because of their compactness. For many years, PFHEs have been widely used for gas separation in cryogenic applications and for aircraft cooling duties. In aerospace applications weight saving is of paramount importance [5].

Salient features of PFHE are discussed by Shah and include the following [6],[7]:

1. Plate fin surface are commonly used in gas-to-gas exchanger applications.
2. The passage height on each side could be easily varied.
3. The maximum operating temperatures are limited by the type of fin-to-plate bonding and the materials employed. PFHE have been designed from cryogenic operating temperatures to about 800°C.

The heat of the gas turbine exhaust is recovered and transferred to the compressed air between the compressor and turbine of the ABC. For this application there are two different types of heat exchanger that can be used: regenerators and recuperators. In a recuperator, heat is transferred through walls that separate the flows. In a regenerator, the heat transfer surface is alternately exposed to the two flows [4].

In the present work it was chosen a recuperator because of the following advantages [8].

1. Ease of manufacture.
2. Stationary nature.
3. Uniform temperature distribution and hence less thermal shock.
4. Absence of sealing problem.

There are many forms of fins that can be used with PFHE including offset strip, plain triangular, plain rectangular, wavy, louver, perforated, or pin fin geometries. In this paper has been study offset strip fin type.

The offset strip fin (OSF) is commonly used geometry in PFHE. The OSF is designated as the serrated fin [5]. The fin has a rectangular cross section; it is cut into small strips of length L_f , and every alternate strip is offset by about 50% of the fin pitch in the transverse direction as schematically shown in Fig. 2. Fin spacing *SS*, fin height *H*, fin thickness δ_F, and strip length in the flow direction L_f ,are the major variables of OSF fins. OSF geometry is characterized by high heat transfer area per unit volume, and high heat transfer coefficients. The heat transfer mechanism in OSFs is described by Joshi and Webb [9]. And is as follows. The heat transfer enhancement is obtained by periodic growth of laminar boundary layers on the

fin length, and their dissipation in the fin wakes. This enhancement is accompanied by an increase in pressure drop

Fig. 2. Geometrical description to OSFs core a heat exchanger.

because of increased friction factor. A form drag force, due to the finite thickness of the fins, also contributes to the pressure drop. Fluid interchange between channels is possible. OSFs are used in the approximate Reynolds number range of 500-10,000 [10].

Salient features pertaining to thermal performance include the following [5].

1. Commonly used in air separation plants where high thermal effectiveness at low mass velocities is required.
2. The heat transfer performance of OSFs is increased by a factor of about 1.5 to 4 over plain fins of similar geometry [10].
3. At high Reynolds numbers, the *J* factor decreases, while the friction factor remains constant because of the high form drag. Therefore OSFs are used less frequently for very high Reynolds number applications.
4. They are used at low Reynolds number applications calling for accurate performance predictions, such as some aerospace applications.

A. The Analysis Method of Heat Exchanger

Is known as that both of hot and cold fluids it flows during cross flow heat exchanger and both streams unmixed.

In this study different alternatives have been proposed for the design of the heat exchanger and the selection of a suitable alternative between these alternatives, which are as follows:

The first alternative: a study and design cross flow heat exchanger with single pass for each stream as shown in Fig. 3.

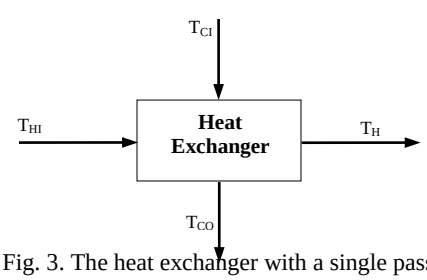

Fig. 3. The heat exchanger with a single pass.

Where it was found by the initial results of study that this alternative needs to Large of heat exchange area in order to get the thermal energy required, so This problem was solved for the method of studying another alternative.

The second alternative: a study and design cross flow heat exchanger with three passes. And so with the aim of increasing the value of log mean temperature difference (LMTD). But the problem here is difficulty in obtaining an equation or a chart to calculate the log mean temperature difference for the heat exchanger. Where the type of flow is a cross flow exchanger whit three passes for each stream and both streams unmixed. To solve this problem we divide the heat exchanger into three exchangers, each of which represents a pass intersects it the hot and cold fluid perpendicular as shown in Fig. 4.

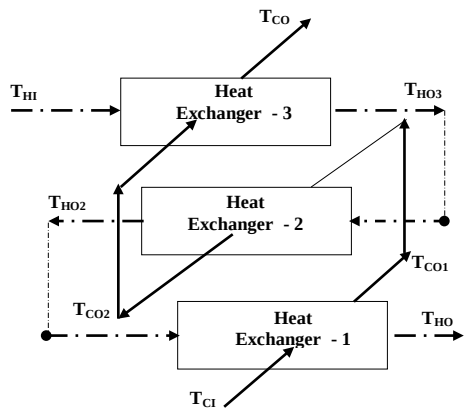

Fig. 4. The heat exchanger with three passes.

B. The Thermal Analysis of the heat exchanger

The purpose of thermal analysis of the heat exchanger is to determine the relationships for calculating overall heat transfer coefficient a heat exchanger and the expression of the thermal energy that has Transferred from the hot to the cold fluids, depending on the surface area of the heat exchanger, and the log mean temperature difference. So this purpose can be achieved by thermal energy balance equation f or both hot (exhaust gases)and cold (air) fluids.

The Log mean temperature difference (*K*) is calculated as follows [11]:

$$\Delta T_m = \frac{P \times A_z^{0.282} \times (T_{HI} - T_{CI})}{\left[Ln\left(\frac{R \times S^{0.22}}{R \times S^{0.22} + Ln(1 - P \times R)} \right) \right]^{1.282}} \tag{7}$$

$$P = \frac{\Delta T_{\min}}{T_{HI} - T_{CI}} \quad , \quad A_z = \frac{W_{\min}}{W_{\max}} \quad , \quad R = \frac{1}{A_z}$$

Where T_H (*K*) the exhaust gases temperature, T_C (*K*) the air temperature, the index I, O denotes inlet and outlet respectively, $W_{min\ ,\ max}$ (*kW/K*) minimum and maximum heat capacity rate respectively, *S*, dimensionless factor and represents the number of heat transfer units, and is calculated from the following equation:

$$ER = 1 - \exp[(\exp(-S^{0.78} A_z) - 1) R \times S^{0.22}] \tag{8}$$

Where ER Effectiveness of the heat exchanger.

From the geometrical details shown in Fig. 2. Can we get the free flow areas for exhaust gases and air as [12]:

Calculate the free flow area for exhaust gases (m^2) as:

$$Aff_H = (H_H - \delta_F)(1 - N_H \times \delta_F) \times L_H \times NP_H \tag{9}$$

$$N_H = \frac{NF_H}{L_H} \quad , \quad NF_H = \frac{L_H}{S_H}$$

Where H_H height of the fin (m), N_H fin frequency, fins per meter, L_H heat exchanger length (m), S_H pitch distribution fins in the layer that passes through it exhaust gases, NF_H number of fins in the layer that passes through it exhaust gases.

Calculate the free flow area for air (m^2) as:

$$Aff_C = (H_C - \delta_F)(1 - N_C \times \delta_F) \times L_C \times NP_C \tag{10}$$

$$N_C = \frac{NF_C}{L_C} \quad , \quad NF_C = \frac{L_C}{S_C}$$

Where H_C height of the fin (m), N_C fin frequency, fins per meter, L_C heat exchanger length (m), S_C pitch distribution fins in the layer that passes through it air, NF_C number of fins in the layer that passes through it air.

The heat transfer areas (m^2) for the exhaust gases can be obtained as given below:

$$A_H = L_H L_C NP_H [1 + 2N_H (H_H - \delta_F)] \tag{11}$$

Where NP_H number of layers that passes through it exhaust gases.

Similarly heat transfer areas (m^2) for the air can be obtained as given below:

$$A_C = L_H L_C NP_C [1 + 2N_C (H_C - \delta_F)] \tag{12}$$

Where NP_C number of layers that passes through it air.

The heat transfer area through the wall between the layers of the heat exchanger is calculated from the following relation:

$$A_W = L_H L_C NP_H \tag{13}$$

Equivalent hydraulic diameter of the heat exchanger is calculated as follows [9]:

Equivalent hydraulic diameter (m) of the finned passages for exhaust gases and air is given by:

$$Dh_H = \frac{2(SS_H - \delta_F)(H_H - \delta_F)}{[SS_H + (H_H - \delta_F)] + \frac{(H_H - \delta_F)\delta_F}{L_f}} \tag{14}$$

$$Dh_C = \frac{2(SS_C - \delta_F)(H_C - \delta_F)}{[SS_C + (H_C - \delta_F)] + \frac{(H_C - \delta_F)\delta_F}{L_f}} \tag{15}$$

Where: $SS_H = \frac{1}{N_H} - \delta_F \quad , \quad SS_C = \frac{1}{N_C} - \delta_F$

The overall heat transfer coefficient through the heat exchanger is calculated as following [12]:

The heat transfer coefficient ($W/m^2.K$) for exhaust gases can be obtained in terms of Colburn J factor as following :

$$\alpha_H = \frac{J_H \times G_H \times C_{PH}}{Aff_H \times (Pr_{fh})^{\frac{2}{3}}} \tag{16}$$

Where Colburn (J_H) factor for the exhaust gases is calculated as follows [9]:

$$J_H = 0.21(Re_{fh})^{-0.4} \left(\frac{L_f}{Dh_H}\right)^{-0.24} \left(\frac{\delta_F}{Dh_H}\right)^{0.02} \tag{17}$$

Where Pr_{fh} Prandtl number, Re_{fh} Reynolds number.

The heat transfer coefficient ($W/m^2.K$) for air can be obtained in terms of Colburn J factor as following :

$$\alpha_C = \frac{J_C \times G_C \times C_{PC}}{Aff_C \times (Pr_{fc})^{\frac{2}{3}}} \tag{18}$$

Where Colburn (J_C) factor for the air is calculated as [9]:

$$J_C = 0.21(Re_{fc})^{-0.4} \left(\frac{L_f}{Dh_C}\right)^{-0.24} \left(\frac{\delta_F}{Dh_C}\right)^{0.02} \tag{19}$$

Where Pr_{fc} Prandtl number, Re_{fc} Reynolds number.

Thus, the overall heat transfer coefficient is calculated as:

$$\frac{1}{UA} = \frac{1}{\alpha_H A_H} + \frac{\delta_W}{\lambda_W A_W} + \frac{1}{\alpha_C A_C} \tag{20}$$

Where δ_W wall thickness (m), λ_W thermal conductivity of the wall metal ($W/m.K$).

The thermal energy (kW) exchanged within the heat exchanger is calculated from the following relation:

$$Q = UA\Delta T_m \tag{21}$$

The pressure loss coefficient from exhaust gases side is calculated as follows:

$$\zeta_H = \zeta_{Gas} = \frac{\Delta P_{TH}}{P_H} \tag{22}$$

$$\Delta P_{TH} = \Delta P_{FH} + \Delta P_{CH} \tag{23}$$

Pressure drop (Pa) during passages for exhaust gases is calculated as [9]:

$$\Delta P_{FH} = \frac{4 f_H L_C \rho_H U_{FH}^2}{2 Dh_H} \tag{24}$$

$$f_H = 1.12(Re_{fh})^{-0.36} \left(\frac{Lf}{Dh_H}\right)^{-0.65} \left(\frac{\delta_F}{Dh_H}\right)^{0.17} \tag{25}$$

$$\Delta P_{CH} = 0.5(\rho_{HO} U_{HO}^2 - \rho_{HI} U_{HI}^2) \tag{26}$$

The pressure loss coefficient from air side is calculated as:

$$\zeta_C = \zeta_{Air} = \frac{\Delta P_{TC}}{P_C} \tag{27}$$

$$\Delta P_{TC} = \Delta P_{FC} + \Delta P_{CC} \tag{28}$$

Pressure drop during passages for air is calculated as [9]:

$$\Delta P_{FC} = \frac{4 f_C L_H \rho_C U_{FC}^2}{2 Dh_C} \tag{29}$$

$$f_C = 1.12(Re_{fc})^{-0.36}\left(\frac{Lf}{Dh_C}\right)^{-0.65}\left(\frac{\delta_F}{Dh_C}\right)^{0.17} \tag{30}$$

$$\Delta P_{CC} = 0.5(\rho_{CO} U_{CO}^2 - \rho_{CI} U_{CI}^2) \tag{31}$$

Where ρ_H , ρ_C density of the fluid (kg/m^3) at the mean temperature of both gases and air respectively, ρ_{HI} , ρ_{HO} , ρ_{CI} , ρ_{CO} , density the entry and exit of gases and air to the heat exchanger respectively, U_{FH} , U_{FC} velocity in narrowest cross section for both gases and air (m/s), U_{HI} , U_{HO} , U_{CI} , U_{CO} velocity the entry and exit of gases and air to the heat exchanger respectively, P_H or P_4 (Pa) gas pressure, P_C or P_{2A} (Pa) air Pressure, ΔP_{CH} , ΔP_{CC} (Pa) Pressure drop due to Change in velocity for gases and air respectively.

IV. Results And Discussion

From the mathematical model of essential design for simple cycle gas turbine, has been studied the effect of turbine inlet temperature (TIT), and the impact of the pressure ratio in the compressor (PR_C) on each of the thermal efficiency and specific work. Fig. 5. Shows that increasing the TIT leads to an increase in the thermal efficiency of the cycle due to increased specific work and increase the electrical power produced that increase the thermal efficiency of the cycle. We also note that increasing the PR_C increases the thermal efficiency. Where due to Increasing the PR_C Leads to raise the temperature of the air that comes out of the compressor, and thus reduce the fuel consumption and increase the thermal efficiency of the cycle. Where it is noted that the efficiency up to (η_S=36.06%) when (TIT=1200°C , PR_C=15).

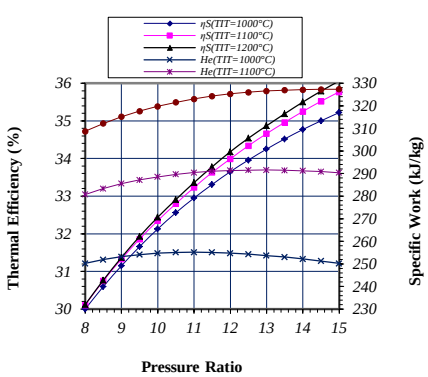

Fig. 5. Efficiency and specific work as function of compressor pressure ratio.

Also has been used mathematical model for ABC to study the effect of the effectiveness of the heat exchanger, as well as study the effect of the pressure ratio in the bottoming cycle (PR_{CA}) on thermal efficiency of the combined cycle.

Fig. 6. Shows that increase the effectiveness of the heat exchanger leads to increases the efficiency of the cycle. And the reason for this is that the increased effectiveness of the heat exchanger leads to raise the temperature of the air that comes out of the heat exchanger, and thereby increase both of specific work and the electrical power produced for ABC, Which in turn contributes to increase the thermal efficiency of the combined cycle. It also notes from the Fig. 6. That increasing the PR_{CA} leads to increasing the thermal efficiency of the cycle due to increased specific work. After that with the increase in the PR_{CA}, we note a decrease in thermal efficiency due to raise the temperature of the air that comes out of the compressor. The reason for this is due to the decrease air density and increased its velocity, leading to increase the pressure loss from the air side of the heat exchanger.

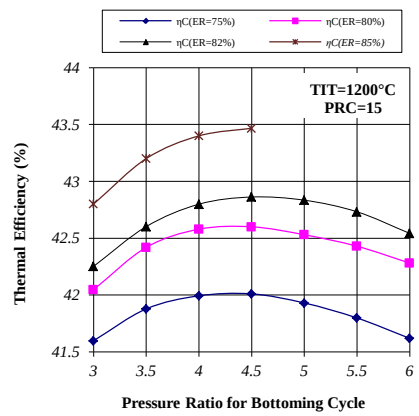

Fig. 6. Optimization of the overall pressure ratio in ABC.

Fig. 6. Also shows the optimum value of the pressure ratio for bottom cycle, which gives the highest efficiency for the combined cycle as is clear from the Fig. 6. that this value is (PR_{CA}=4.5) at all the values of the heat exchanger effectiveness. As can be seen from the Fig. 6. That values efficiency dropped at (PR_{CA}>4.5 , ER=85%) due to the decline in the value of the log mean temperature difference of the heat exchanger. The maximum value of the efficiency have been achieved at (η_C=43.5 % ,TIT=1200°C , PR_C=15, ER=85%).

Fig. 7. Shows the effect of the effectiveness of the heat exchanger on both thermal efficiency and specific work for topping gas turbine cycle, as is clear from the Fig. 7. That the highest value for the efficiency of the cycle be at (η_C=43.5% , ER=85%), while the highest value for the specific work be at (He=326.4 kJ/kg , ER=85%).

As can be seen from the Fig. 7. Also increase the effectiveness of the heat exchanger leads to increased specific work, due to increasing the number of layers that passes through it both of exhaust gases and air in the heat exchanger. This in turn leads to a drop in the velocities of each of the exhaust gases and the air passing through these layers, and thus lower of the losses in the pressure from side of the

exhaust gases and from side of the air in the heat exchanger, and thus increase the specific work for the cycle.

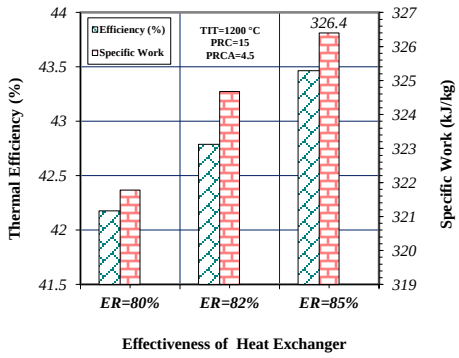

Fig. 7. Efficiency and specific work as function of Effectiveness of OSF heat exchanger.

Also has been studied the effect of effectiveness of the heat exchanger on the size of the heat exchanger used. Fig. 8. Shows that increasing the effectiveness of the heat exchanger leads to increase the size of the heat exchanger.

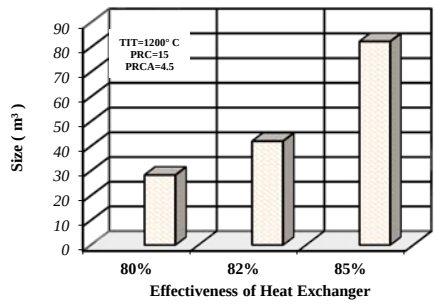

Fig. 8. The size at different values of Effectiveness.

Also has been studied the effect of turbine inlet temperature TIT on the thermal efficiency for the combined cycle and simple cycle at different pressure ratios as shown in Fig. 9. Where notes that by increasing the TIT, leads to increases thermal efficiency for both simple cycle and combined cycle, also notes that increasing the pressure ratio increases the thermal efficiency of simple cycle and combined cycle.

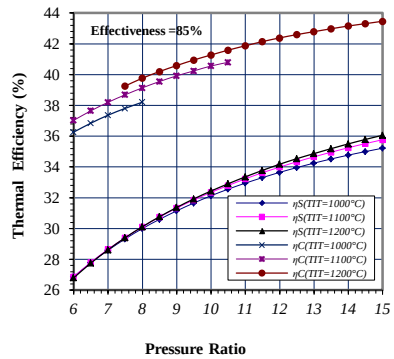

Fig. 9. Comparison between efficiency of simple and combined cycle.

V. Conclusions

The conclusion is that the convert industrial gas turbines into combined Jule/Jule cycle is an economical alternative for power generation. The pressure ratio was found optimal at 15 in topping cycle and 4.5 in bottoming cycle. A recuperator is recommended instead of a regenerator for the heat transfer from the gas turbine exhaust gas to the compressed air of the ABC. The combined Joule/Joule cycle efficiency was calculated to 43.5 percent. The simple cycle gas turbine efficiency was calculated to 36 percent thus Implementation of the air bottoming cycle at a gas turbine adds 7.5 percent points to the simple cycle efficiency, and a rise in the power output of 20.5 to 32 percent depending on the compression ratio.

When compared with the steam bottoming cycle, the ABC showed performance values close to and exceeding those of the steam bottoming cycle, while featuring a simpler and robust design, smaller dimensions, this combined system is very much less costly than that using steam by dispensing with boiler, steam turbine, condenser, pumps, water treatment plant, cooling towers, etc.

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