American Journal of Engineering Research (AJER)
e-ISSN : 2320-0847 p-ISSN : 2320-0936
Volume-02, Issue-12, pp-321-333
www.ajer.org

Research Paper Open Access

Exergy Analysis of A Combined Gas/ Steam Turbine Cycle with A Supercharged Boiler

Sayed A. Abdel-Moneim, and Khaled M. Hossin**
* *Prof., Mech. Eng. and Vice Dean, Faculty of Eng., Northern Border Univ.(NBU), Kingdome of Saudi Arabia (KSA).*
*** M Sc., Graduate student*

***Abstract:* -** In this paper,energy and exergy analysis of a combined cyclewith a supercharged boiler was carried out. Acombination of a basic gas turbine and steam cycle with both a supercharged boiler (SB) and a heat recovery boiler (HRB)wasinvestigated. The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the performance of the supercharged boiler combined cycle (SBCC) were studied. Comparisons between the SBCC and the conventional combined cycle were performed. The results indicated that the SBCC gives output power up to 2.1 times of that of the conventional combined cycle when compared at the same values of the operating parameters. However, the SBCCefficiency was found to be lower than the conventional combined cycle. The exergy analysis showed an advantage of SBCC over the conventional combined cycle.

***Keywords*: -** *Thermal power plant; supercharged boiler, combined cycle, energy; exergy; second-law efficiency, exergy destruction*.

NOMENCLATURE

$\overline{C}_P$	specific heat at constant pressure, (kJ/kmol K)
e	flow specific exergy, (kJ/kg)
ExD	exergy destruction rate, (kW)
h	enthalpy, (kJ/kg)
Δh	enthalpy difference, (kJ/kg)
LHV	lower heating value of fuel, (kJ/kmol)
$\dot{m}$	mass flow rate, (kg/s)
m_{HP}	mass of HP steam generated in HRB, (kg/kmol$_{n.g}$)
m_{LP}	mass of LP steam generated in HRB, (kg/kmol$_{n.g}$)
m_{LH}	mass ratio of LP to HP steam
m_{SB}	mass of steam generated in SB, (kg/kmol$_{n.g}$)
M	molecularweight; (kg/kmol)
P	pressure, (bar) or power, (kW)
PR	ST to GT power ratio
q	heat transferred per kg of steam, (kJ/kg)

Greek symbols

μ_1, μ_2, μ_3	steam mass fractions
ε_{HRB}	Efficiency of the HRB
ε_{SB}	Efficiency of the SB
η_{com}	thermal efficiency of combined cycle
η_G	generator efficiency
η_{GC}	thermal efficiency of GT cycle
η_{HRB}	thermal efficiency of HRB steam cycle
η_m	mechanical efficiency
η_P	pump isentropic efficiency,
η_{SB}	thermal efficiency of SB steam cycle
η_{2nd}	second-law efficiency
λ	excess air factor
π_C	compressor presser ratio

Acronyms

T	temperature, (K)
v	specific volume, (m^3/kg)
w	work per kg of steam, (kJ/kg)
W	work, (kJ/$kmol_{n.g}$)
X_a	actual air to fuel ratio, ($kmol_a$/$kmol_{n.g}$)
X_g	amount of product gases, ($kmol_g$/$kmole_{n.g}$)

Subscripts

a	air
com	combined cycle
g	product gases
GC	gas turbine cycle
i	Inlet
$n.g$	natural gas
N	Net
o	Outlet
SC	steam turbine cycle

C	compressor
CON	condenser
CP	condensate pump, circulated pump
EC	economizer
EV	evaporator
FP	feed pump
FWH	surface feed-water heater
GEN	generator
GT	gas turbine
HRB	heat recovery boiler
HP	high pressure
LP	low pressure
NG	natural gas
P	pump
SB	supercharged boiler
SBCC	supercharged boiler combined cycle
SH	superheater
ST	steam turbine
C	compressor
CON	condenser
CP	condensate pump, circulated pump
EC	economizer
EV	evaporator
FP	feed pump
FWH	surface feed-water heater

I. INTRODUCTION

Exergy analysis is a technique based on the first and second laws of thermodynamics which provides an alternative and illuminating means of assessing andcomparing processes and systems rationally and meaningfully. Unlike energy, exergy is not conserved and gets depleted due to irreversibilities in the processes. The performance of energy systems is degraded by the presence ofirreversibilities, and the entropy production is a measure of theirreversibilities that present during a process. In particular, exergy analysis yields efficiencies which provide a true measure of how nearly actual performance approaches the ideal, and identifies more clearly than energy analysis the causes and locations of thermodynamic losses. Consequently, exergy analysis can assist in improving and optimizing designs. Several studies had been carried out by researchers [1-5] to evaluate the performance ofthermal power plants using exergy analysis.

Combined gas/steam turbine cycle power plants are widely used for cogeneration and electricity generation as well. In combined cycles, the gas turbine exhaust heat is utilized through the use of heat recovery boilers (HRBs). The overall efficiency of combined power plants can be improved by: increasing the mean temperature of heat supplied by increasing the inlet gas temperature of the gas turbine and/or decreasing the mean temperature at which heat is rejected [6-8]. Briesch et al. [9] reported that 60 % efficiency can be achievedfor a combined cycle by increasing the gas turbine inlet temperature to 1427ºC.Modeling and optimizing of a dual pressure reheat combined cycle was carriedout by Bassily [10]with introducing a technique to reduce the irreversibility of the steam generator.

One of the applicable methods of saving energy and reducing steam generator size is to supercharge the steam generator by using a gas turbine-driven compressor to furnish combustion air. Developments in metallurgy and pressure vessel technology make it possible to build such a supercharged boiler (SB). The reduction in size and heat transfer surface of a supercharged boiler is due to two reasons. First, as the operating gas-side pressure is increased, the emissivity of the non-luminousradiating gases increases markedly. Second, the higher gas density and available pressure drop permit much higher gas mass flow rates (compared with the conventional steam generator) to be used in the convection section, with higher accompanying convection heat transfer coefficients [11].Mikhael et al. [12] investigated the possibility of utilizing the solar energy for electrical power generation with a hybrid mode of steam generation in a combined power plant incorporating a SB and a HRB.Studies based on the exergy analysis identify the location, the magnitude and the sources of irreversibilities inSBCCs werepresented in [13-15].

In this paper, a supercharged boiler combined cycle (SBCC) is modeled, analyzed and the effect of the different operating parameters are extensively investigated.

II. DESCRIPTION OF THE CYCLE

Figure 1 shows the present supercharged boiler combined plant which combining the supercharged boiler cycle with the heat recovery cycle. In this SBCC, the compressor supplies pressurized air to the SB (state 2). All combustion takes place in the boiler and steam can be generated at any suitable pressure and temperature(state 20S). The steam generated in the SB is circulated through a separated steam cycle. The steam expands in a steam turbine (ST) with extracted steam fractions during expansion process to heat the water before entering the SB. High-temperature pressurized gas from the boiler is expanded as it flows through the gas turbine (GT). The power so developed supplies the compressor and drives the generator. The hot exhaust gases from the GT pass through a dual pressure HRB to generate steam and next go to the stack (state 9). After the water leaves the condenser (state 1S), it is pumped to the dual pressure HRB, where it is converted to a steam with low and high pressures (states 9S and 7S, respectively). The low pressure steam is mixed with the exhaust steam from the high pressure turbine (HPST) before entering the low pressure turbine (LPST) to expand to the condenser pressure (state 11S).

III. CYCLE ANALYSIS

To evaluate the thermal performance the cycle an analysis of each component based on the following assumptions is carried out:

- Temperature differences and pressure drop through gas and steam pipes are negligible.
- The heat losses and pressure drop for feed-water heaters and condensers are negligible.
- The steam side pressure drop in HRB and SB are negligible.
- Air leakage through gas cycle components is negligible.

The input data and other assumptions used in the present study are listed in Table 1. In the preset study, three values for the gas turbine inlet temperature (T_3) of 1200ºC, 1300ºC, and 1400ºC are investigated. The excess air factor (λ) for the SB is changed from 1.2 to 2.2 within a range of compressor pressure ratios (π_c) from 6 to 30.

III. i. Analysis of the GT cycle

The GT cycle is assumed to operate according to the actual Brayton cycle and the three main processes are as follows:

III. i. i. The compression process in the compressor

The work absorbed by the compressor per kmol of air is determined by,

$$w_C = \overline{C}_{P,a}(T_2 - T_1) \qquad \text{kJ/kmol}_{air} \qquad (1)$$

Where, $\overline{C}_{P,a}$ is calculated at the mean temperature between inlet and outlet of the compressor.

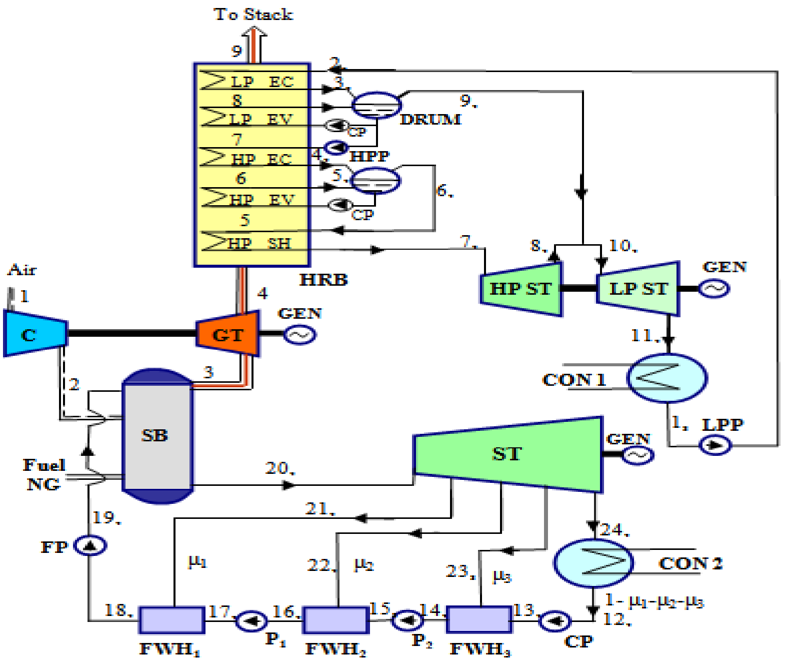

Fig. 1: Schematic diagram for the present SBCC.

III. i. ii. The combustion process in the combustion chamber

In the present study clean natural gas fuel of ultimate analysis as (78.8 % CH_4, 14 % C_2H_6, 6.8% N_2 and 0.4% CO_2 by volume)is used, [11]. The combustion equation based on one kmolof natural gas is:

$$0.788CH_4 + 0.14C_2H_6 + 0.004CO_2 + 0.068N_2 + \lambda.n_{O_2}(O_2 + 3.76N_2) \rightarrow$$
$$1.072CO_2 + 1.996H_2O + (\lambda - 1).n_{O_2}O_2 + (3.76\lambda.n_{O_2} + 0.068)N_2 \qquad (2)$$

where λ is the excess air factor and n_{O_2} is the theoretical O_2 required to burn 1 kmol of natural gas ($n_{O_2} = 1.072 + 1.996/2 - 0.004 = 2.066$ kmol/kmol$_{n.g.}$)

- The energy balance equation for the combustion process based on 1kmol of fuel is:

$$X_a\overline{C}_{P,a}T_2 + LHV + \overline{C}_{P,n.g.}T_{n.g.} = X_g\overline{C}_{P,g}T_3 + m_{SB}(h_o - h_i)/\varepsilon_{SB} \qquad (3)$$

Where, *LHV* is the lower calorific value of the natural gas which given by:

$$LHV = n_{CH_4}.LHV_{CH_4} + n_{C_2H_6}LHV_{C_2H_6} \text{ kJ/kmol}_{n.g.} \qquad (4)$$

and, X_a is the actual amount of air (number of kmoles) per kmol of fuel, and X_g is the amount of product gases per kmol fuel.

The mass flow rates of the fuel and combustion gases are then calculated from the mass flow rate of the air as follows:

$$\dot{m}_{n.g} = \dot{m}_a/(X_aM_a/M_{n.g}) \quad \text{kg/s} \qquad (5\text{-a})$$

$$\dot{m}_g = X_gM_g/M_{n.g}\dot{m}_{n.g} \text{ kg/s} \qquad (5\text{-b})$$

III.i. iii.The expansion process in the gas turbine

In this process, the work done by the GT per kmol natural gas is determined by,

$$W_{GT} = X_g\overline{C}_{P,g}(T_3 - T_4) \text{ kJ/kmol}_{n.g.} \qquad (6)$$

where $\overline{C}_{P,g}$ is also determined at the mean ttemperature between inlet and outlet of the GT.

The net work for the GT cycle is:

$$W_{N,GC} = (W_{GT}\eta_m - X_aw_C/\eta_m)\eta_G \text{ kJ/kmol}_{n.g.} \qquad (7)$$

The thermal efficiency of the GT cycle is:

$$\eta_{GC} = \frac{W_{N,GC}}{LHV + \overline{C}_{P,n.g}T_{n.g}} \qquad (8)$$

III. ii. Analysis of the Steam Turbine cycles

In the present work a combined cycle shown in Fig. 1, enclosesHRB cycle and SB cycle, is analyzed. Each of these two cyclesis assumed to operateon a Rankinecycle.An energy balance is applied for each component (Control volume) as follows:

III. ii. i. Analysis of the HRB steam cycle

Enthalpy rise in each pump in the cycle is written as:

$$\Delta h_P = v_i(P_o - P_i)/\eta_p \quad \text{kJ/kg} \qquad (9)$$

where P is the pressure in (*KPa*)

The heat added to the steam in each stage of the HRB is:

<u>- Low-Pressure Economizer</u> $\dot{Q}_{LPEC} = (\dot{m}_{HP} + \dot{m}_{LP})(h_{3_s} - h_{2_s})$ kJ/s

or $q_{LPEC} = (1 + m_{LH})(h_{3_s} - h_{2_s})$ kJ/kg$_{HP}$ (10)

wherem_{LH} is the mass ratio of LP to HP steam in the HRB; $m_{LH} = \dfrac{\dot{m}_{LP}}{\dot{m}_{HP}}$

<u>- Low-Pressure Evaporator</u> $q_{LPEV} = m_{LH}(h_{9_s} - h_{3_s})$ kJ/kg$_{HP}$ (11)

<u>- High-Pressure Economizer</u> $q_{HPEC} = h_{5_s} - h_{4_s}$ kJ/kg$_{HP}$ (12)

<u>- High-Pressure Evaporator</u> $q_{HPEV} = h_{6_s} - h_{5_s}$ kJ/kg$_{HP}$ (13)

<u>- High-Pressure Superheater</u> $q_{HPSH} = h_{7_s} - h_{6_s}$ kJ/kg$_{HP}$ (14)

Total heat added in the HRB per kg of HP steam is then:

$$q_{HRB} = q_{LPEC} + q_{LPEV} + q_{HPEC} + q_{HPEV} + q_{HPSH} \text{ kJ/kg}_{\text{HP}} \quad (15)$$

The work of HP and LP pumps per kg of HP steam are:

$$w_{HPP} = \Delta h_{HPP} \text{ kJ/kg}_{\text{HP}} \quad (16)$$

$$w_{LPP} = (1 + m_{LH})\Delta h_{LPP} \text{ kJ/kg}_{\text{HP}} \quad (17)$$

The work of HP and LP stages of the ST per kg of HP steam are:

$$w_{HPST} = h_{7_s} - h_{8_s} \text{ kJ/kg}_{\text{HP}} \quad (18)$$

$$w_{LPST} = (1 + m_{LH})(h_{10_s} - h_{11_s}) \text{kJ/kg}_{\text{HP}} \quad (19)$$

The net work of HRB steam cycle per kg of HP steam is given by:

$$w_{N,HRB} = [(w_{HPST} + w_{LPST})\eta_m - (w_{HPP} + w_{LPP})/\eta_m]\eta_G \text{ kJ/kg}_{\text{HP}} \quad (20)$$

The thermal efficiency of HRB steam cycle is calculated as follows:

$$\eta_{HRB} = \frac{w_{N,SC}}{q_{HRB}} \quad (21)$$

It is obvious that the above equations are based on kg of HP steam. To calculate the mass of HP steam, energy balance between points 4 and 8 in the HRB should be carried out:

$$m_{HP} = \frac{\varepsilon_{HRB} X_g \overline{C}_{P,g}(T_4 - T_8)}{q_{HPSH} + q_{HPEV} + q_{HPEC} + q_{LPEV}} \quad \text{kg/kmol}_{\text{n.g}} \quad (22)$$

where T_8 is determined using the temperature difference at LP pinch point ($\Delta T_{PP,LP}$) as:

$$T_8 = T_{sat}(P_{LP}) + \Delta T_{PP,LP} \quad (23)$$

Then, the net work of HRB steam cycle per kmol natural gas is equal to:

$$W_{N,HRB} = m_{HP} w_{N,HRB} \quad \text{kJ/kmol}_{\text{n.g}} \quad (24)$$

III. ii. ii. Analysis of the SB steam cycle

In order to calculate the fraction of steam required for each surface heater, energy balances for the surface heaters are done.

- Energy balance for surface feed-water heater 1:

$$\mu_1 = \frac{h_{18_s} - h_{17_s}}{h_{21_s} - h_{17_s}} \quad (25)$$

- Energy balance for surface feed-water heater 2:

$$\mu_2 = \frac{(1-\mu_1)(h_{16_s} - h_{15_s})}{h_{22_s} - h_{15_s}} \quad (26)$$

- Energy balance for surface feed-water heater 3:

$$\mu_3 = \frac{(1-\mu_1-\mu_2)(h_{14_s} - h_{13_s})}{h_{23_s} - h_{13_s}} \quad (27)$$

The work of the cycle pumps per kg of steam is:

- Feed pump $w_{FP} = \Delta h_{FP}$ kJ/kg (28)
- Heater pump 1 $w_{P1} = \Delta h_{P1}(1-\mu_1)$ kJ/kg (29)
- Heater pump 2 $w_{P2} = \Delta h_{P2}(1-\mu_1-\mu_2)$ kJ/kg (30)
- Condensate pump $w_{CP} = \Delta h_{CP}(1-\mu_1-\mu_2-\mu_3)$ kJ/kg (31)

Specific work of ST for SB steam cycle (per kg of steam) is:

$$w_{ST,SB} = h_{20_s} - h_{21_s} + (1-\mu_1)(h_{21_s} - h_{22_s}) + (1-\mu_1-\mu_2) \\ (h_{22_s} - h_{23_s}) + (1-\mu_1-\mu_2-\mu_3)(h_{23_s} - h_{24_s}) \quad \text{kJ/kg} \quad (32)$$

Net work of SB steam cycle per kg of steam is:

$$w_{N,SB} = \left[w_{ST,SB}\eta_m - \left(w_{FB} + w_{P1} + w_{P2} + w_{CP}\right)/\eta_m\right]\eta_G \text{ kJ/kg} \quad (33)$$

Heat added in the SB per kg of steam is given by:

$$q_{SB} = h_{20_s} - h_{19_s} \text{ kJ/kg} \quad (34)$$

Thermal efficiency of the SB steam cycle is:

$$\eta_{SB} = \frac{w_{N,SB}}{q_{SB}} \quad (35)$$

Net work of the SB steam cycle per kmol natural gas is then:

$$W_{N,SB} = m_{SB} w_{N,SB} \text{ kJ/kmol}_{n.g} \quad (36)$$

where the mass of steam generated in the SB (m_{SB}) is obtained from Eq. (3).
The total net output of the combined cycle per kmol natural gas is:

$$W_{com} = W_{N,GC} + W_{N,HRB} + W_{N,SB} \text{ kJ/kmol}_{n.g} \quad (37)$$

The combined cycle thermal efficiency is then calculated as:

$$\eta_{com} = \frac{W_{com}}{LHV + \bar{C}_{P,n.g} T_{n.g}} \quad (38)$$

Another important parameter for the combined cycle is the power ratio, and it is defined as:

$$PR = \frac{W_{N,HRB} + W_{N,SB}}{W_{N,GC}} \quad (39)$$

The output power produced by each combined cycle in (*kW*) can be determined from the following equation:

$$P_{com} = W_{com}\left(\dot{m}_{n.g} / M_{n.g}\right) \text{kW} \quad (40)$$

Exergy Analysis

The exergy destruction in the different control volumes of the cycle is calculated by applying the exergy balance equation derived by [16-17]. This equation reads:

$$EXD = \sum \dot{m}_i e_i - \sum \dot{m}_o e_o + \sum\left(1 - \frac{T_0}{T}\right)\dot{Q}_{CV} - \sum \dot{W}_{CV} \text{ kW} \quad (41)$$

where,

$\dot{Q}_{CV}$:heat transferred to the control volume, kW

$\dot{W}_{CV}$: rate of work out from the control volume, kW

T : temperature at which heat is transferred, K

T_0: reference temperature and equal to 298K.

The exergy of a flow stream for a given pressure (*P*) and temperature (*T*) is given by:

$$e = \left(h - h_o\right) - T_o\left(s - s_o\right) \quad 42)$$

where, the properties *h* and *s* for steam are obtained from the present code, and for gas are calculated from the ideal gas model as:

$$h - h_o = C_P\left(T - T_o\right) \quad (43)$$

and $s - s_o = C_P \ln\left(\frac{T}{T_o}\right) - R\ln\left(\frac{P}{P_o}\right)$ (44)

The second-law efficiency for each control volume in steady state steady flow (SSSF) process is calculated as:

$$\eta_{2nd} = 1 - \frac{EXD}{\sum Ex_i - \sum Ex_o} \quad (45)$$

IV. NUMERICAL CALCULATIONS

In the present work a FORTRAN computer code is designed includes special subroutines utilizing the governing equations (1 to 45). This code was used to, calculate the thermodynamic properties of the water at each state, perform heat balance for each control volume in the combined cycle, evaluate energy and exergy performance characteristics of the cycle, predict the effect of the different operating parameters on the cycle performance.

V. RESULTS AND DISCUSSIONS

The present results were found based on the following operating data for the cycle following Akiba and Thani [11] as listed in Table 1, .

Table 1 Assumptions of the cycle.

Parameter	Value	Unit
Air mass flow rate	67.9268	kg/s
Ambient temperature.	30	^{o}C
Atmospheric pressure.	1.01325	bar
Compressor isentropic efficiency.*	85	%
GT isentropic efficiency.*	90	%
Gas-side pressure loss in SB.*	6	%
Efficiency of SB.	95	%
GT exhaust gas pressure.	1.05	bar
Pump isentropic efficiency.*	70	%
ST isentropic efficiency.*	87	%
Condenser pressure.	0.075	bar
Efficiency of HRB.	95	%
Pinch point of HRB at HP.*	15	^{o}C
Pinch point of HRB at LP.*	25	^{o}C
LP to HP steam mass ratio.	0.2	-
Mechanical efficiency.*	99	%
Generator efficiency*	98	%

In addition, the following steam conditions at various states in the cycle were considered as listed in Table 2.

Table 2 Steam conditions of the cycle.

Parameter	ST (SB)	HP ST	LP ST	FWH1	FWH2	FWH3
Pressure (bar)	170	50	4.5	59	14	1.9
Temperature (^{o}C)	540	540	T_{sat}	-	-	-

The effect of excess air on the energy and exergy performance characteristics of the SBCC cycle are predicted at fixed T_3 of 1300^{o}C.The energy performance characteristics are plotted against the compressor pressure ratio at different excess air factors are shown in Figs. 2-3. The output power and power ratio are shown in Fig.2 and the combined cycle efficiency is shown in Fig.3.

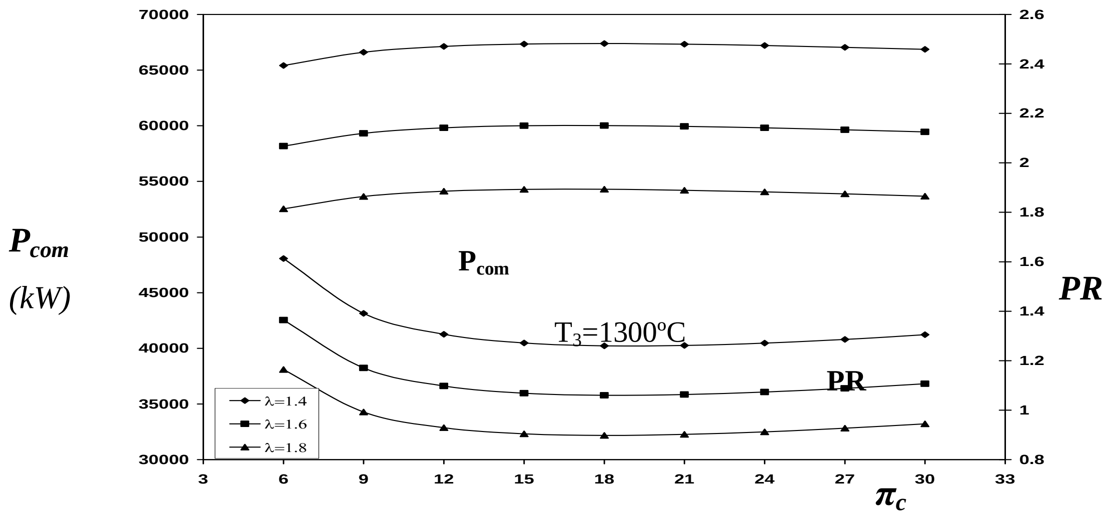

Fig. 2: The SBCC output power and power ratio at fixed T_3 of 1300ºC and at three different excess air factors.

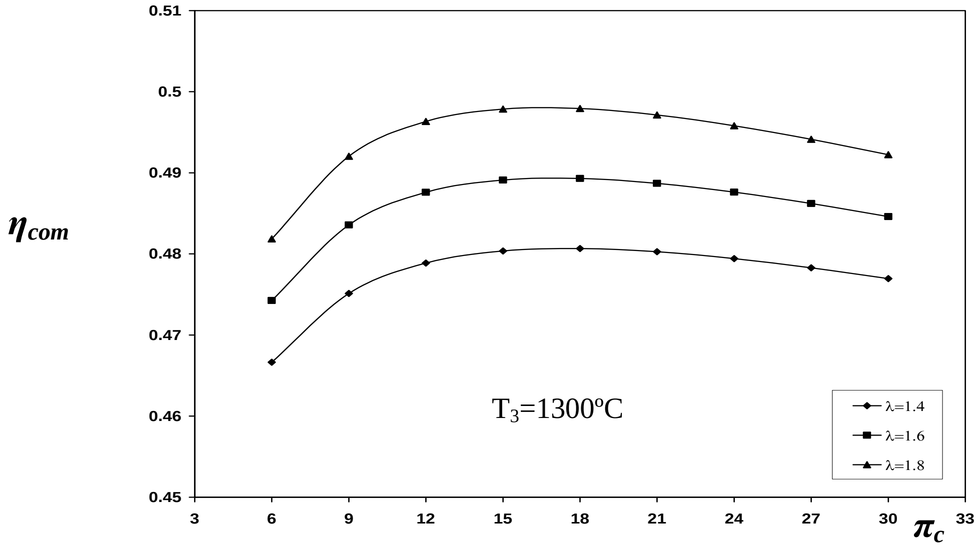

Fig. 3: The SBCC energy efficiency at fixed T_3 of 1300ºC and at three different excess air factors.

The results showed noticeable effects of the excess air factor on the cycle performance. The output power and the power ratio decrease as the excess air factor increases, while the combined cycle efficiency increases. Also, an optimum compressor pressure ratio for the combined cycle efficiency was found depending on the excess air factor. On the other hand, the change of the output power with the compressor pressure ratio is almost small.

The exergy destructions in the cycle components at different excess air factors is shown in Fig.4. It was found that, the exergy destruction in the SB is the major part followed by that in the HRB. Figure 4 shows that the exergy destruction in the SB decreases by increasing the excess air factor. Also, by increasing the excess air factor, the exergy destruction in the HRB is slightly decreased due to the reduction in the temperature difference between the hot gases and cold steam in the HRB. It is clear that the exergy destruction in the compressor is not affected by the excess air factor as the air mass flow rate was fixed constant.

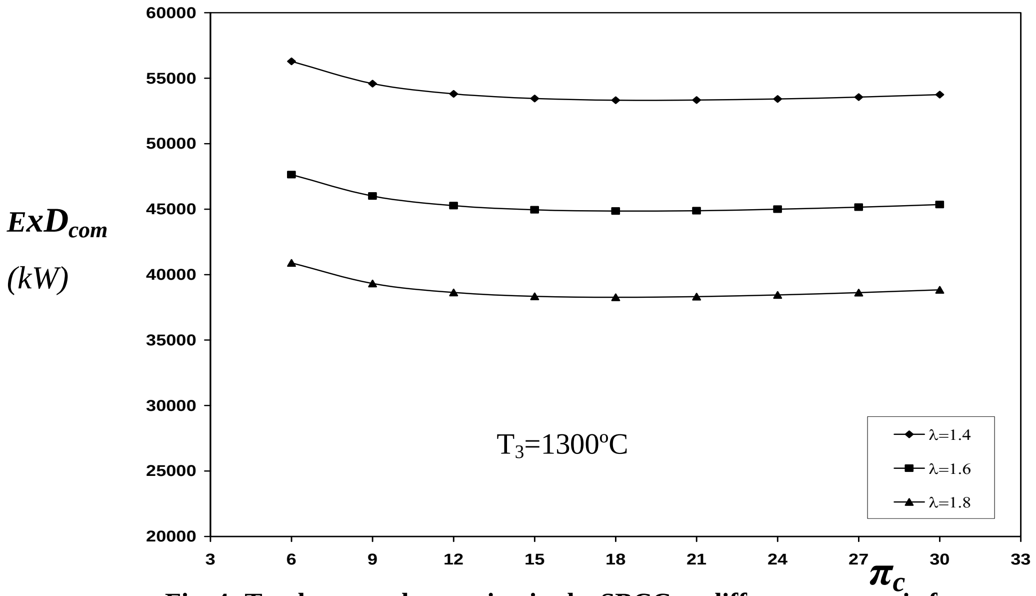

Fig. 4: Total exergy destruction in the SBCC at different excess air factors.

The exergy destructions in the ST, FWHs, and CON2 are decreased by increasing the excess air factor due to thedecrease in the amount of steam generated in the SB, while those for the other components were not affected. Figure 5 shows the relative values of the total exergy destruction in the different components in the combined cycle.

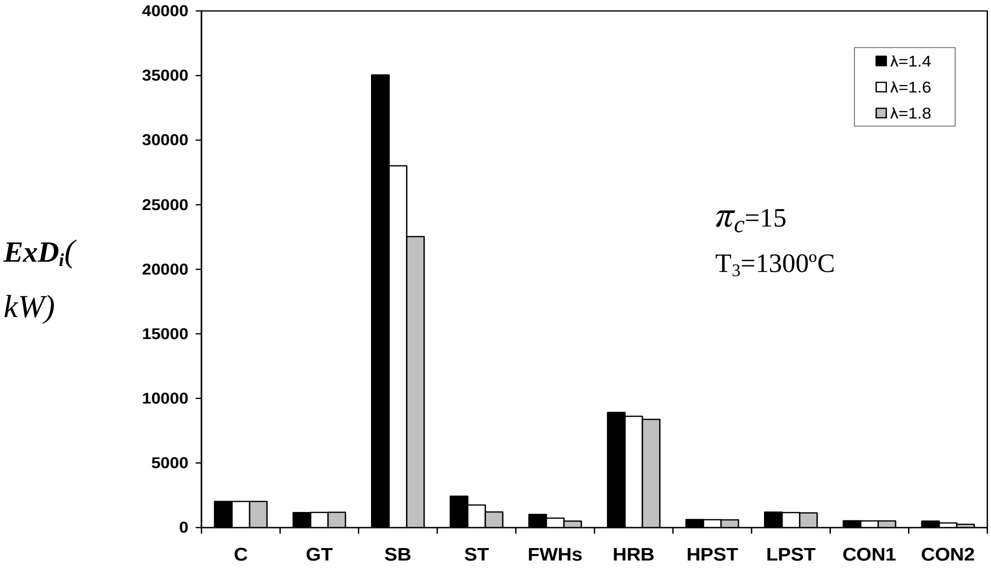

Fig. 5: Exergy destructions in the cycle components at different excess air factors.

Figure 6 shows a plot of the second-law efficiency with the compressor pressure ratio at different excess air factors. The second-law efficiency was increased by increasing the excess air factor. Also, an optimum compressor pressure ratio was found depending on the value of the excess air factor.

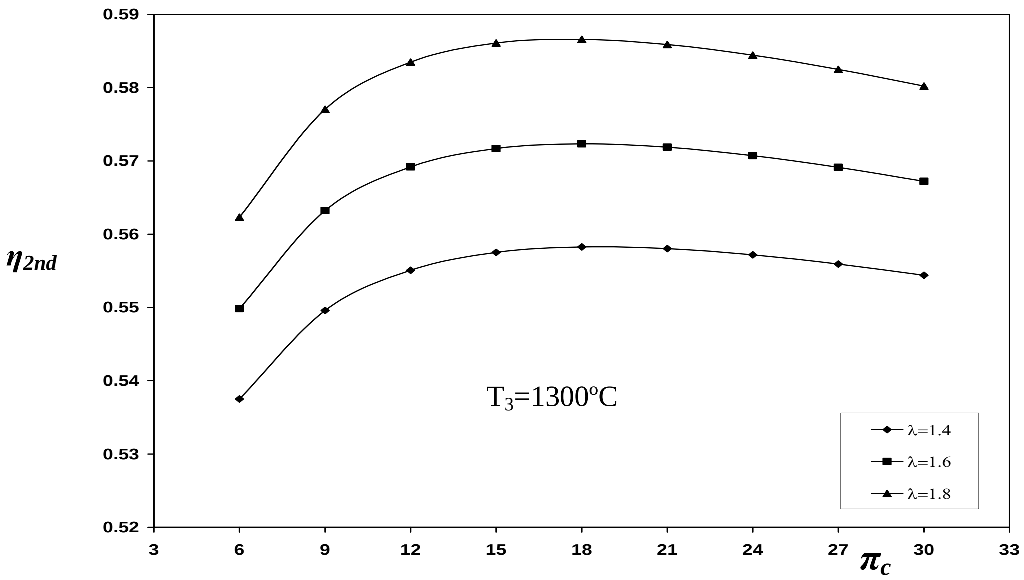

Fig. 6: Second-law efficiency of the SBCC at different excess air factors.

The effect of the turbine inlet temperature on the energy performance and the exergy destruction of the cycle was investigated in the present work. Three different values for T3 of (1200ºC, 1300ºC, and 1400ºC) were studied at a fixed excess air factor of 1.6. Figure 7 shows that the turbine inlet temperatureis strongly affect the combined cycle thermal efficiency.

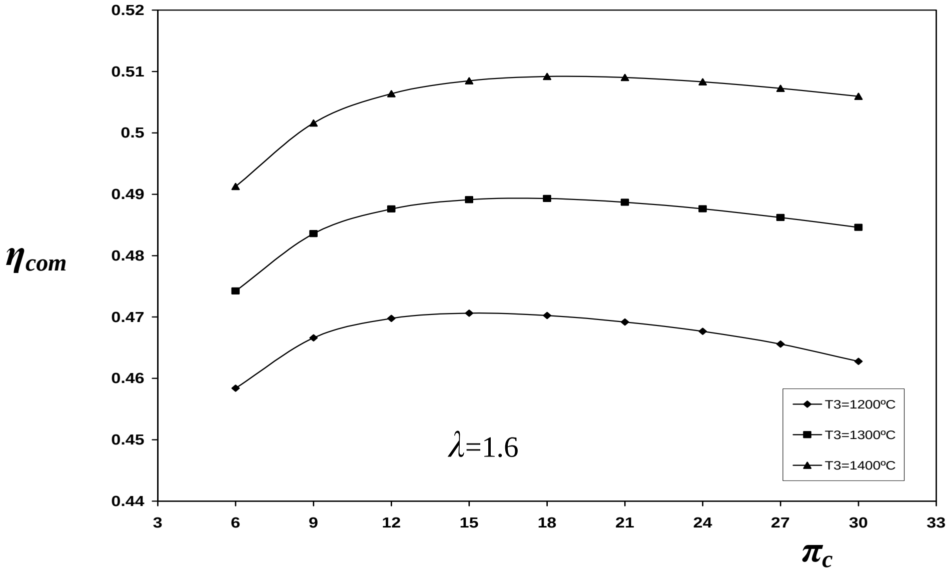

Fig. 7: Thermal performance of the SBCC at different turbine inlet temperatures.

Figure 8 shows the effect of the turbine inlet temperature on the second-law efficiency of the combined cycle. The second-law efficiency is highly affected by the turbine inlet temperature it was strongly increased by the increase in the turbine inlet temperature.

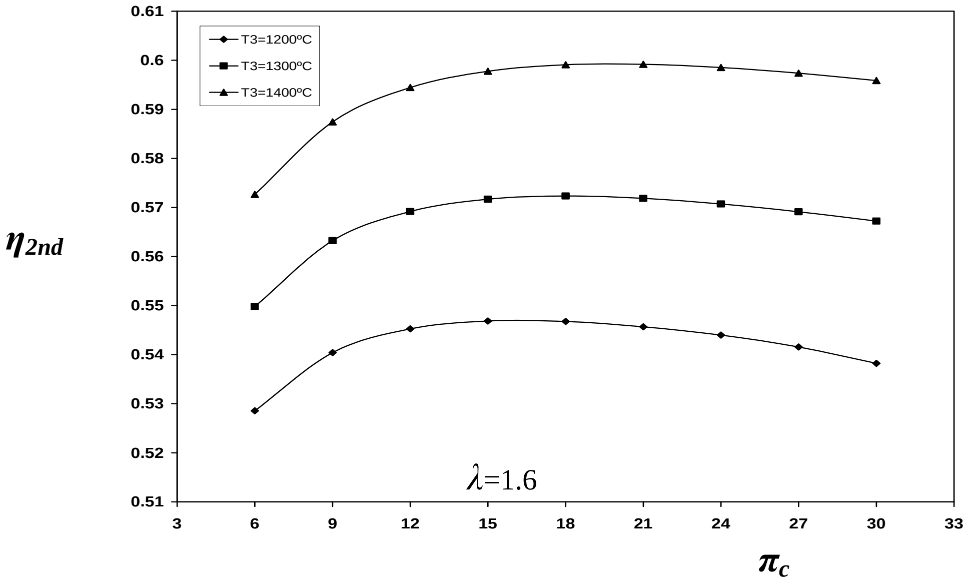

Fig. 8:The second-law efficiency of the SBCC at different turbine inlet temperatures.

A comparison between the SBCC and the conventional combined cycles was carried out to evaluate the performance of these cycles.This comparison was carried out at a fixed air mass flow rate of 67.9268 kg/s, turbine inlet temperature of 1300ºC, and the other parameters are considered as listed in Table 1.

Figure 9 shows a comparison between the thermal efficiency of SBCC and conventional combined cycles. The results showed that the combined cycle efficiency of the SBCC is lower than that of the conventional combined cycle. Also, for the conventional combined cycle, the efficiency is continuously increased by increasing the compressor pressure ratio.

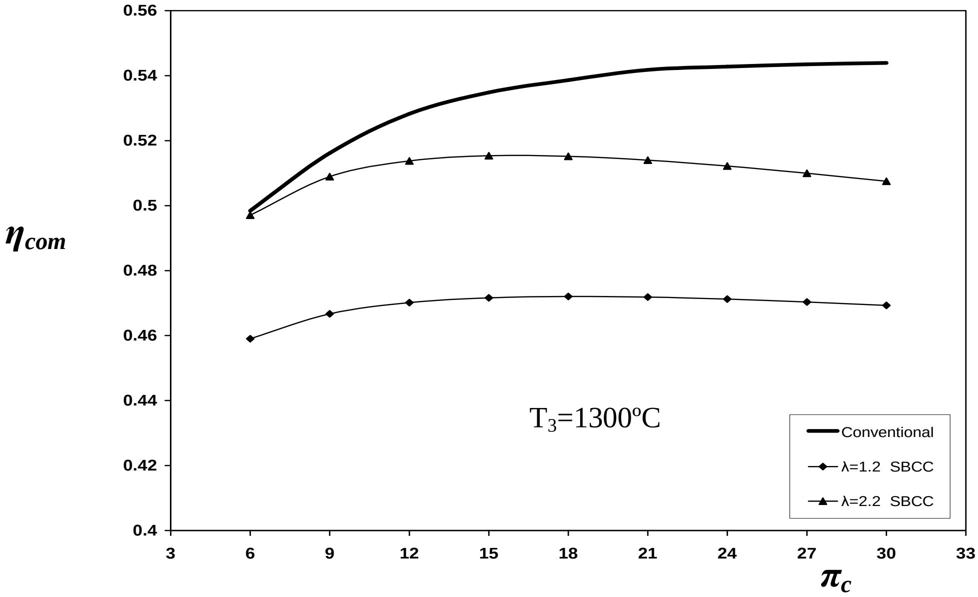

Fig. 9: Comparison between theefficiency of SBCCand conventional combined cycles.

Figure 10 shows a comparison between the second-law efficiency ofSBCC and conventional combined cycles. The second-law efficiency of the SBCC is almost higher than that of the conventional one at excess air factor over 1.2. It was found that 9.5% to 18.5% increase in the second-law efficiency was obtained for the SBCC higher than that for the conventional combined cycle.

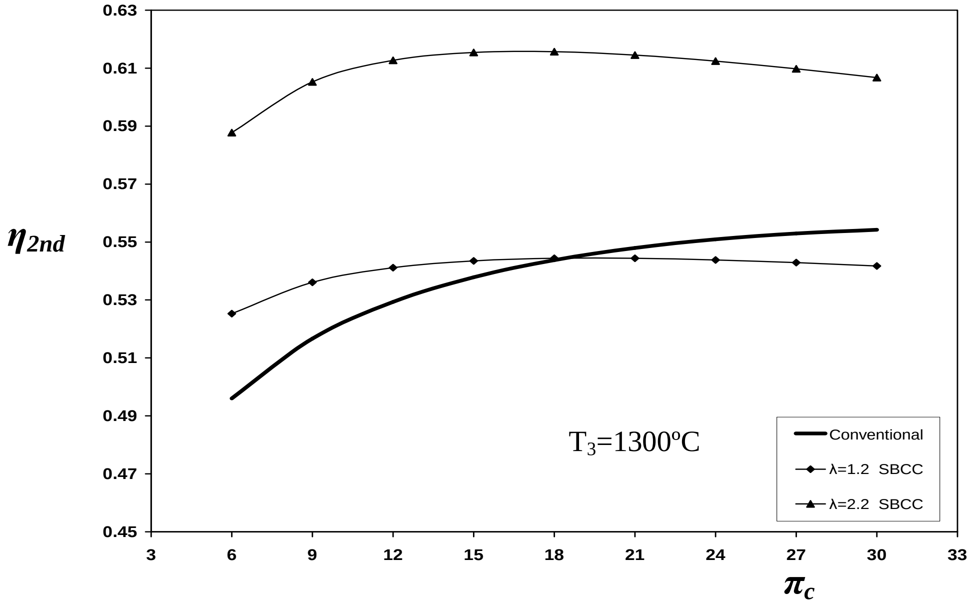

Fig. 10: Comparison between the second-law efficiency for SBCC and conventional combined cycles.

Finally, the present predictions for the SBCC were correlated in terms of the investigated operating parameters. New correlation form was obtained respectively, for the combined cycle efficiency, second-law efficiency, and the total exergy destruction ratio (the total exergy destruction to the total exergy input) with different correlating coefficient as listed in Table 3. This correlation form is,

$$\Phi = a_0 \pi_C{}^{a_1} \lambda^{a_2} \left(T_3 / T_0\right)^{a_3} \qquad (46)$$

Where, the variable Φ is one of η_{com} , η_{2nd} ,or EXD^{*}_{com} and the coefficients a_0, a_1, a_2, and a_3 are listed in Table 3, and $T_3$and T_0 are temperatures in (K).

The obtained correlationis valid within the ranges of the operating parameters of (6 $\leq\pi_c\leq$ 30, 1200ºC$\leq T_3\leq$ 1400ºC, and 1.2$\leq\lambda\leq$2.0).

Table 3 Coefficients of the correlation46.

Variable	a_0	a_1	a_2	a_3	% DEV_{max}
η_{com}	0.15798	1.137E-2	0.13231	0.62025	±2.57
η_{2nd}	0.15204	1.698E-2	0.18693	0.71240	±2.76
EXD^{*}_{com}	2.60315	-2.806E-2	-0.30428	-1.04413	±4.75

VI. CONCLUSIONS

In the present work, a thermodynamic analysis ofa supercharged boiler combined cycle was carried out. The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the performance of the cycle were investigated. A comparison between the SBCC and the conventional cycle performance was also carried out. The preset study leads to the following conclusions:

1. The largest values of the output power for the SBCC are predicted at a minimum excess air factor and a maximum turbine inlet temperature.
2. The SBCChas higher values of the output power ranging from 1.6 to 2.1 times that for the conventional combined cycle.
3. The values of the combined cycle thermal efficiency of the SBCC are lower than that of the conventional cycle.
4. For a turbine inlet temperature of 1300^{o}C, optimum compressor pressure ratios which give maximum efficiencies are predicted for the SBCC. While, for the conventional cycle, the efficiency is continuously increased with the compressor pressure ratio.
5. The maximum exergy losses were found in the supercharged boiler and the heat recovery boiler. Therefore, research effortsare recommended to minimize losses in these components.
6. Lower values of the total exergy destruction in the SBCC were found at the higher excess air factor over 1.2.
7. Exergy destruction ratio,ranges from 31% to 43%, was found for SBCC, while values from 43% to 52% were obtained for the conventional combined cycle.
8. Higher values for the second-law efficiency were found for SBCC compared with that for the conventional combined cycle. An enhancement ranging from 9.5% to 18.5% in the second-law efficiency for SBCC was foundcompared with that for conventional cycle.
9. New correlation was obtained to correlate the combined cycles performance characteristics with the different operating parameters (turbine inlet temperature, the excess air factor, and the compressor pressure ratio).

REFERENCES

[1] M. J., Ebadi andM. Gorji-Bandpy, *Exergetic Analysis of Gas Turbine Plants*, Int. Journal of Exergy Research. 2(1), 2005, 31-39.

[2] P. O. Ayoola1, and N. A. Anozie, *A Study of Sections Interaction Effects on Thermodynamic Efficiencies of a Thermal Power Plant*, British Journal of Applied Science & Technology, ISSN: 2231-0843, *3(4),2013, 1201-12143.*

[3] W. Goran and G. Mei. On Exergy and Sustainable Development-Part 1: Conditions and Concepts. Exergy International Journal. 1(3), 2001, 128-145.

[4] S. Sengupta, A. Dattaand S. Duttagupta. Exergy Analysis of a Coal-Based 210mw ThermalPower Plant. Int. Journal Energy Research,31(1), 2007, 14-28.

[5] A. Mohammad, A. Pouria and H. Armita, Energy, Exergy and Exergoeconomic Analysis ofa Steam Power Plant, Int. Journal Energy Research. 33(5), 2008, 499-512.

[6] M. S. Briesch, R. L. Bannister, I. S. Diakunchak and D.J. Huber, *A Combined Cycle Designed to Achieve Greater Than 60 Percent Efficiency*, ASME J. Engineering for Gas Turbines and Power, 117, 1995, 734-741.

[7] A.M. Bassily, *Modeling, Numerical Optimization, and Irreversibility Reduction of a Dual-Pressure Reheat Combined-Cycle*, Applied Energy, 81, 2005, 127-151.

[8] M. Akiba and E.A. Thani,*Thermodynamic Analysis of New Combination of Supercharged Boiler Cycle and Heat Recovery Cycle for Power Generation*, ASME J. Engineering for Gas Turbines and Power, 118, 1996, 453-460.

[9] N.N.Mikhael, K.K.A. Moradand A.M.I. Mohamed, *Design Criterion of Solar-Assisted Combined-Cycle Power Plants with Parabolic Through Concentrators*, Port-Said Engineering Research J., 4,2000, 80-101.

[10] M. Ghazikhani, H. Takdehghan and A. Moosavi, *Exergy Analysis of Gas Turbine Air-Bottoming Combined Cycle for Different Environment Air Temperature*, Proceedings of 3rd International Energy, Exergy and Environment Symposium, 2007.

[11] C. Koch, F. Cziesla and G. Tsatsaronis, *Optimization of Combined Cycle Power Plants Using Evolutionary Algorithms*, Chemical Engineering and Processing, 2007.

[12] Y. Kwon, H. Kwan, S. Oh,Exergoeconomic Analysis of Gas Turbine Cogeneration System, Int. Journal of Exergy, 1, 2001,31-40.

[13] H. Jericha, and F. Hoeller,*Combined Cycle Enhancement*,ASME J. Engineering for Gas Turbines and Power, 113,1991, 198-202.

[14] O. Bolland, *A Comparative Evaluation of Advanced Combined Cycle Alternatives*,ASME J. Engineering for Gas Turbines and Power, 113, 1991, 190-197.

[15] B. Seyedan, P.L.Dhar, R.R. Guar and G.S. Bindra, *Optimization of Waste Heat Recovery Boiler of a Combined Cycle Power Plant*, ASME J. Engineering for Gas Turbines and Power, 118,1996, 561-564.

[16] G.V. Wylen, R. Sonntag and C. Borgnakke, *Fundamentals of Classical Thermodynamics*,(4th Ed.), (John Wiley & Sons, Inc., 1994).

[17] Y. Sanjay, O. Singh and B. N. Prasad, *Energy and Exergy Analysis of Steam Cooled Reheat Gas–Steam Combined Cycle*, Applied Thermal Engineering, 2007.