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

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** M Sc., Graduate student

Abstract: - In this paper, energy and exergy analysis of a combined cycle with a supercharged boiler was carried out. A combination of a basic gas turbine and steam cycle with both a supercharged boiler (SB) and a heat recovery boiler (HRB) was investigated. 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 SBCC efficiency 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

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\overline{C}_p$</td>
<td>specific heat at constant pressure, (kJ/kmol K)</td>
</tr>
<tr>
<td>$e$</td>
<td>flow specific exergy, (kJ/kg)</td>
</tr>
<tr>
<td>$ExD$</td>
<td>exergy destruction rate, (kW)</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy, (kJ/kg)</td>
</tr>
<tr>
<td>$\Delta h$</td>
<td>enthalpy difference, (kJ/kg)</td>
</tr>
<tr>
<td>$LHV$</td>
<td>lower heating value of fuel, (kJ/kmol)</td>
</tr>
<tr>
<td>$m$</td>
<td>mass flow rate, (kg/s)</td>
</tr>
<tr>
<td>$m_{HP}$</td>
<td>mass of HP steam generated in HRB, (kg/kmol$_{n.g}$)</td>
</tr>
<tr>
<td>$m_{LP}$</td>
<td>mass of LP steam generated in HRB, (kg/kmol$_{n.g}$)</td>
</tr>
<tr>
<td>$m_{LP}$</td>
<td>mass ratio of LP to HP steam</td>
</tr>
<tr>
<td>$m_{SB}$</td>
<td>mass of steam generated in SB, (kg/kmol$_{n.g}$)</td>
</tr>
<tr>
<td>$M$</td>
<td>molecular weight; (kg/kmol)</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure, (bar) or power, (kW)</td>
</tr>
<tr>
<td>$PR$</td>
<td>ST to GT power ratio</td>
</tr>
<tr>
<td>$q$</td>
<td>heat transferred per kg of steam, (kJ/kg)</td>
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Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$\mu_1, \mu_2, \mu_3$</td>
<td>steam mass fractions</td>
</tr>
<tr>
<td>$\varepsilon_{HRB}$</td>
<td>Efficiency of the HRB</td>
</tr>
<tr>
<td>$\varepsilon_{SB}$</td>
<td>Efficiency of the SB</td>
</tr>
<tr>
<td>$\eta_{com}$</td>
<td>thermal efficiency of combined cycle generator efficiency</td>
</tr>
<tr>
<td>$\eta_G$</td>
<td>thermal efficiency of GT cycle</td>
</tr>
<tr>
<td>$\eta_{HRB}$</td>
<td>thermal efficiency of HRB steam cycle</td>
</tr>
<tr>
<td>$\eta_m$</td>
<td>mechanical efficiency</td>
</tr>
<tr>
<td>$\eta_P$</td>
<td>pump isentropic efficiency,</td>
</tr>
<tr>
<td>$\eta_{SB}$</td>
<td>thermal efficiency of SB steam cycle</td>
</tr>
<tr>
<td>$\eta_{2nd}$</td>
<td>second-law efficiency</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>excess air factor</td>
</tr>
<tr>
<td>$\pi_C$</td>
<td>compressor pressure ratio</td>
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</table>

Acronyms

www.ajer.org
Exergy analysis is a technique based on the first and second laws of thermodynamics which provides an alternative and illuminating means of assessing and comparing 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 of irreversibilities, and the entropy production is a measure of their irreversibilities 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 of thermal 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 achieved for a combined cycle by increasing the gas turbine inlet temperature to 1427°C. Modeling and optimizing of a dual pressure reheat combined cycle was carried out 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-luminous radiating 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 in SBCCs were presented 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

$I$. INTRODUCTION

Exergy analysis is a technique based on the first and second laws of thermodynamics which provides an alternative and illuminating means of assessing and comparing 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 of irreversibilities, and the entropy production is a measure of their irreversibilities 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 of thermal power plants using exergy analysis.

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In this paper, a supercharged boiler combined cycle (SBCC) is modeled, analyzed and the effect of the different operating parameters are extensively investigated.
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 present study, three values for the gas turbine inlet temperature ($T_3$) of 1200°C, 1300°C, and 1400°C are investigated. The excess air factor ($\lambda$) for the SB is changed from 1.2 to 2.2 within a range of compressor pressure ratios ($\pi_c$) 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,

$$w_c = C_{p,a}(T_2-T_1) \text{ kJ/kmol air}$$

where, $C_{p,a}$ is calculated at the mean temperature between inlet and outlet of the compressor.
In the present study clean natural gas fuel of ultimate analysis as (78.8 % CH₄, 14 % C₂H₆, 6.8% N₂ and 0.4% CO₂ by volume) is used, [11]. The combustion equation based on one kmol of natural gas is:

\[ 0.788CH_4 + 0.14C_2H_6 + 0.004CO_2 + 0.068N_2 + \lambda n_{O_2} (O_2 + 3.76N_2) \rightarrow \]

\[ 1.072CO_2 + 1.996H_2O + (\lambda - 1)n_{O_2} + 3.76\lambda n_{O_2} + 0.068N_2 \]

where \( \lambda \) 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 of fuel).

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

\[ X_a C_{P,g} T_2 + LHV + C_{P,n.g} T_{n.g} = X_g C_{P,g} T_3 + m_{SB} (h_{v} - h_i) / \varepsilon_{SB} \]

Where, \( LHV \) is the lower calorific value of the natural gas which given by:

\[ LHV = n_{CH_4} LHV_{CH_4} + n_{C_2H_6} LHV_{C_2H_6} \ (kJ/kmol_{n.g}) \]

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:

\[ \dot{m}_{n.g} = \dot{m}_a / (X_a M_a / M_{n.g}) \]

\[ \dot{m}_g = X_g M_g / M_{n.g} \dot{m}_{n.g} \]

(5-a)

(5-b)

III. iii. The expansion process in the gas turbine

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

\[ W_{GT} = X_g C_{P,g} (T_3 - T_4) \ (kJ/kmol_{n.g}) \]

where \( C_{P,n.g} \) is also determined at the mean temperature between inlet and outlet of the GT.

The net work for the GT cycle is:

\[ W_{N,GC} = (W_{GT} / \eta_m - X_a W_C / \eta_m) \eta_G \ (kJ/kmol_{n.g}) \]

(7)

The thermal efficiency of the GT cycle is:

\[ \eta_G = \frac{W_{N,GC}}{LHV + C_{P,n.g} T_{n.g}} \]

(8)

III. ii. Analysis of the Steam Turbine cycles

In the present work a combined cycle shown in Fig. 1, encloses HRB cycle and SB cycle, is analyzed. Each of these two cycles is assumed to operate 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:

\[ \Delta h_p = v_p (P_o - P) / \eta_p \ (kJ/kg) \]

(9)

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

- **Low-Pressure Economizer**: \( \dot{Q}_{LPEC} = (\dot{m}_{LP} + \dot{m}_{LP}) (h_{3_2} - h_{3_1}) \) kJ/s

or \( q_{LPEC} = (1 + m_{LP}) (h_{3_2} - h_{3_1}) \) kJ/kg_{LP}\( \)

(10)

where \( m_{LP} \) is the mass ratio of LP to HP steam in the HRB: \( m_{LP} = \dot{m}_{LP} / \dot{m}_{HP} \)

- **Low-Pressure Evaporator**: \( q_{LP} = m_{LP} (h_{4_2} - h_{4_1}) \) kJ/kg_{LP}\( \)

(11)

- **High-Pressure Evaporator**: \( q_{HPEV} = h_{5_3} - h_{5_2} \) kJ/kg_{HP}\( \)

(12)

- **High-Pressure Superheater**: \( q_{HPSH} = h_{6_3} - h_{6_2} \) kJ/kg_{HP}\( \)

(13)
The net work of SB steam cycle per kg of steam is:

$$\eta_{SB} = \frac{w_{N,SC}}{q_{SB}}$$  \hspace{1cm} (21)

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

$$m_{HP} = \frac{\varepsilon_{HRB}X_g \overline{C}_p, g (T_4 - T_8)}{\frac{q_{HPSH} + q_{HPEV}}{2} + q_{HPEC}} \text{ kg/kmol}_g$$  \hspace{1cm} (22)

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

$$T_8 = T_{sat}(P_{LP}) + \Delta T_{PP,Lp}$$  \hspace{1cm} (23)

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

$$W_{N,HRB} = m_{HP}w_{N,HRB} \text{ kJ/kmol}_g$$  \hspace{1cm} (24)

### III. 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:
  $$\mu_1 = \frac{h_{18_s} - h_{17_s}}{h_{21_s} - h_{17_s}}$$  \hspace{1cm} (25)

- Energy balance for surface feed-water heater 2:
  $$\mu_2 = \frac{(1 - \mu_1)(h_{16_s} - h_{15_s})}{h_{22_s} - h_{15_s}}$$  \hspace{1cm} (26)

- Energy balance for surface feed-water heater 3:
  $$\mu_3 = \frac{(1 - \mu_1 - \mu_2)(h_{14_s} - h_{13_s})}{h_{23_s} - h_{13_s}}$$  \hspace{1cm} (27)

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

- Feed pump $w_{FP} = \Delta h_{FP}$ kJ/kg
- Heater pumps 1 $w_{P1} = \Delta h_{P1}(1 - \mu_1)$ kJ/kg \hspace{1cm} (28)
- Condensate pump $w_{CP} = \Delta h_{CP}(1 - \mu_1 - \mu_2 - \mu_3)$ kJ/kg \hspace{1cm} (29)

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

$$w_{ST,SB} = h_{20_s} - h_{21_s} + (1 - \mu_1)(h_{21_s} - h_{22_s}) + (1 - \mu_1 - \mu_2)(h_{22_s} - h_{23_s}) + (1 - \mu_1 - \mu_2 - \mu_3)(h_{23_s} - h_{24_s}) \text{ kJ/kg}$$  \hspace{1cm} (30)

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

\[ q_{SB} = h_{20s} - h_{19s} \text{ kJ/kg} \]  

Thermal efficiency of the SB steam cycle is:

\[ \eta_{SB} = \frac{W_{N,SB}}{q_{SB}} \]  

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

\[ W_{N,SB} = m_{SB} W_{N,SB} \text{ kJ/kmol}_n \]  

The total net output of the combined cycle per kmol natural gas is:

\[ W_{com} = W_{N,GC} + W_{N,HRB} + W_{N,SB} \text{ kJ/kmol}_n \]  

The combined cycle thermal efficiency is then calculated as:

\[ \eta_{com} = \frac{W_{com}}{LHV + C\gamma_{P,n,g}T_{n,g}} \]  

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

\[ PR = \frac{W_{N,HRB} + W_{N,SB}}{W_{N,GC}} \]  

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

\[ P_{com} = W_{com} \left( \frac{m_{n,g}}{M_{n,g}} \right) \text{ kW} \]  

**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:

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

where,

\[ \dot{Q}_{CV} : \text{heat transferred to the control volume, kW} \]
\[ \dot{W}_{CV} : \text{rate of work out from the control volume, kW} \]
\[ T : \text{temperature at which heat is transferred, K} \]
\[ T_0: \text{reference temperature and equal to 298K}. \]

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

\[ e = (h - h_o) - T_o (s - s_o) \]  

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:

\[ h - h_o = C_p \left( T - T_o \right) \]  
\[ s - s_o = C_p \ln \left( \frac{T}{T_o} \right) - R \ln \left( \frac{P}{P_o} \right) \]

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

\[ \eta_{2nd} = 1 - \frac{EXD}{\sum \dot{E}_{x_i} - \sum \dot{E}_{x_o}} \]
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>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air mass flow rate</td>
<td>67.9268</td>
<td>kg/s</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>30</td>
<td>°C</td>
</tr>
<tr>
<td>Atmospheric pressure</td>
<td>1.01325</td>
<td>bar</td>
</tr>
<tr>
<td>Compressor isentropic efficiency</td>
<td>85</td>
<td>%</td>
</tr>
<tr>
<td>GT isentropic efficiency</td>
<td>90</td>
<td>%</td>
</tr>
<tr>
<td>Gas-side pressure loss in SB.</td>
<td>6</td>
<td>%</td>
</tr>
<tr>
<td>Efficiency of SB.</td>
<td>95</td>
<td>%</td>
</tr>
<tr>
<td>GT exhaust gas pressure</td>
<td>1.05</td>
<td>bar</td>
</tr>
<tr>
<td>Pump isentropic efficiency</td>
<td>70</td>
<td>%</td>
</tr>
<tr>
<td>ST isentropic efficiency</td>
<td>87</td>
<td>%</td>
</tr>
<tr>
<td>Condenser pressure</td>
<td>0.075</td>
<td>bar</td>
</tr>
<tr>
<td>Efficiency of HRB.</td>
<td>95</td>
<td>%</td>
</tr>
<tr>
<td>Pinch point of HRB at HP.</td>
<td>15</td>
<td>°C</td>
</tr>
<tr>
<td>Pinch point of HRB at LP.</td>
<td>25</td>
<td>°C</td>
</tr>
<tr>
<td>LP to HP steam mass ratio.</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>Mechanical efficiency</td>
<td>99</td>
<td>%</td>
</tr>
<tr>
<td>Generator efficiency</td>
<td>98</td>
<td>%</td>
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</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Parameter</th>
<th>ST (SB)</th>
<th>HP ST</th>
<th>LP ST</th>
<th>FWH1</th>
<th>FWH2</th>
<th>FWH3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure (bar)</td>
<td>170</td>
<td>50</td>
<td>4.5</td>
<td>59</td>
<td>14</td>
<td>1.9</td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>540</td>
<td>540</td>
<td>T_{sat}</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

The effect of excess air on the energy and exergy performance characteristics of the SBCC cycle are predicted at fixed T_{3} of 1300°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 1300°C and at three different excess air factors.](image-url)
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.

The exergy destructions in the ST, FWHs, and CON2 are decreased by increasing the excess air factor due to the decrease 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.

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 temperature is 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.

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 1300°C, 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 the efficiency of SBCC and conventional combined cycles.

Figure 10 shows a comparison between the second-law efficiency of SBCC 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.

![Second-law efficiency comparison](image)

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,

\[
\Phi = a_0 \pi_c^{a_1} \lambda^{a_2} \left( T_3 / T_0 \right)^{a_3}
\]

where, the variable \( \Phi \) is one of \( \eta_{com} \), \( \eta_{2nd} \), 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 correlation is valid within the ranges of the operating parameters of $(6 \leq \pi_c \leq 30, 1200^\circ C \leq T_{\text{in}} \leq 1400^\circ C, \text{and } 1.2 \leq \varepsilon \leq 2.0)$.

<table>
<thead>
<tr>
<th>Variable</th>
<th>$a_0$</th>
<th>$a_1$</th>
<th>$a_2$</th>
<th>$a_3$</th>
<th>% DEV$_{\text{max}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{\text{com}}$</td>
<td>0.15798</td>
<td>1.137E-2</td>
<td>0.13231</td>
<td>0.62025</td>
<td>±2.57</td>
</tr>
<tr>
<td>$\eta_{2\text{nd}}$</td>
<td>0.15204</td>
<td>1.698E-2</td>
<td>0.18693</td>
<td>0.71240</td>
<td>±2.76</td>
</tr>
<tr>
<td>$\text{EXD}_{\text{com}}^*$</td>
<td>2.60315</td>
<td>-2.806E-2</td>
<td>-0.30428</td>
<td>-1.04413</td>
<td>±4.75</td>
</tr>
</tbody>
</table>

VI. CONCLUSIONS

In the present work, a thermodynamic analysis of a 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 present 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 SBCC has 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 $1300^\circ C$, 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 efforts are 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 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 found compared 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


