Exergy analysis of gas turbine with air bottoming cycle

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Abstract

In this paper, the exergy analysis of a conventional gas turbine and a gas turbine with air bottoming cycle (ABC) is presented in order to study the important parameters involved in improving the performance characteristics of the ABC based on the Second Law of thermodynamics. In this study, work output, specific fuel consumption (SFC) and the exergy destruction of the components are investigated using a computer model. The variations of the ABC cycle exergy parameters are comprehensively discussed and compared with those of the simple gas turbine. The results indicate that the amount of the exhaust exergy recovery in different operating conditions varies between 8.6 and 14.1% of the fuel exergy, while the exergy destruction due to the extra components in the ABC makes up only 4.7–7.4% of the fuel exergy. This is the reason why the SFC of the ABC is averagely 13.3% less and the specific work 15.4% more than those of the simple gas turbine. The results also reveal that in the ABC cycle, at a small value of pressure ratio, a higher specific work with lower SFC can be achieved in comparison with those of the simple gas turbine.

1. Introduction

Nowadays, the gas turbine has a large share of the world electricity generation. According to Diesel & Gas Turbine Worldwide's 32nd Annual Power Generation Order Survey [1], the total output of the ordered gas turbines reached about 43 GW from June 2008 to May 2009. In recent years, the performance of industrial gas turbines has been improved due to considerable investment in research and development [2]. Intercooling, improved pressure ratio, and increased turbine inlet temperature (TIT) are widely used to enhance the efficiency of gas turbines [3]. Since the exhaust of a gas turbine has a relatively high temperature and a large mass flow, the exhaust waste heat can be utilized to further improve the performance. Combining the gas turbine cycle (Brayton cycle) with a medium/low-temperature bottoming cycle (like Rankine cycle), known as the conventional combined cycle, is the most effective way to increase efficiency of a gas turbine cycle [2]. The main idea is to use the hot exhaust gases of Brayton cycle, as the heat source of bottoming cycle [4]. Heavy-duty gas turbines in combination with heat recovery steam generators and steam turbines represent the state of the art of this approach [2], but the installation costs of the high-pressure steam generator, the steam turbine, and the condenser might prove to be prohibitive in small-scale power generation [5]. It should be mentioned that special requirements are needed on water quality, high-pressure equipment and the operators of steam plant [6]. The heat recovery steam generator and the steam turbine of the conventional combined cycle plant can be replaced by the air bottoming cycle (ABC). As Fig. 1 shows in the ABC, the exhaust flow of an existing, topping gas turbine is sent to a gas–air heat exchanger, which heats the air in the secondary gas turbine cycle [5]. The air bottoming cycle was patented by Farrell of General Electric Company in 1988 [7]. William Farrell claims that the ABC provides greater thermodynamic efficiency compared to that of the gas turbine alone, while retaining the operational flexibility of the gas turbine. He notes that a steam and gas turbine combined cycle has a number of drawbacks such as difficulties in handling water steam and the need for large capital investment. The especial application of the air bottoming cycle was also invented in Nov. 1988 by ED Alderson [8]. In the presence of a fired gas turbine that is supplied with coal gas fuel, the power output of bottoming cycle may be used for producing compressed air to the oxygen plant which supplies oxygen to the coal gasifier. In 1995, Kambanis [9] showed that an LM2500 gas turbine coupled to the ABC has a considerably better off-design performance, especially at lower power rating levels, where gas turbines run very inefficiently. Also in 1996, O. Bolland [10] found that the ABC adds
A major increase in natural gas as a fuel for power generation is foreseen for industrial countries [12]. Thus, improving the performance of gas turbines will be essential for both new and existing plants in order to reduce fuel consumption and the production of environmental emissions. Considering what has been mentioned above about the efficiency and applications of the air bottoming cycle, it can be regarded as an alternative to achieve this goal, especially in low-power gas turbines. Exergy destruction analysis can be applied to identify the reasons why the

![Diagram](image-url)

**Fig. 1.** A gas turbine with air bottoming cycle.
performance of a gas turbine with ABC is higher than that of a simple gas turbine.

In this paper, the performance of a gas turbine with ABC was studied by means of availability analysis. To carry out exergy analysis, a computer model was introduced to calculate the cycle parameters. For the conventional gas turbine and the gas turbine with ABC, the fuel exergy, specific work output, SFC (specific fuel consumption) and the exergy destruction of the cycle components were calculated. The results indicate that in a gas turbine with ABC the specific work and lower SFC at the same TIT and pressure ratio may be achieved compared to simple gas turbine.

2. Exergy analysis

Exergy analysis is based on the Second Law and generally allows process inefficiencies to be better pinpointed than does an energy analysis [13]. Another important concept in exergy analysis is exergy destruction. Exergy destruction is defined as the quantity of the work capacity of a system lost during a process; this loss occurs due to dissipative effects such as friction, electric resistance, and inelasticity or nonquasistatic processes [14]. The major purpose of exergy analysis is to detect and evaluate quantitatively the irreversibilities of a process [14].

For any open system at steady state conditions, an equation for the rate of total exergy destruction is [14]:

$$I_{tot} = \sum Q^\theta \left( 1 - \frac{T_f}{T_j} \right) + \sum_{in} m_{\psi} - \sum_{out} m_{\psi} + W_{act}$$  \hspace{1cm} (1)

where $I_{tot}$ is the total exergy destruction of the system, $\sum m_{\psi}$ is the total stream exergy entering the system, $\sum m_{\psi}$ is the total stream exergy exiting the system, and $W_{act}$ is the shaft work transfer rate [14].

Total stream exergy ($\psi_{tot}$) for the mixture of the exhaust products can be calculated by Eq. (2), where $y_i$ and $y_{i,00}$ are the mole fraction of a species in a mixture flow and the mole fraction of a species in the atmospheric air respectively, $n_{H,j}$ and $T_j$ are the enthalpy per mole of a species at temperature of $T$ and $T_0$, $s_{i,T,P}$ and $s_{i,T_P,P_0}$ are the entropy per mole of a species at inlet flow and thermo-mechanical state respectively, $R$ is the universal gas constant [14].

$$\psi_{tot} = \sum_{i=1}^{n} y_i \left( n_{H,i} - n_{H,0} - T_0 \left( s_{i,T,P} - s_{i,T_P,P_0} \right) \right) + R T_0 \sum_{i=1}^{n} y_i \left( \ln \frac{y_i}{y_{i,00}} \right)$$  \hspace{1cm} (2)

One typical definition of the second-law efficiency that measures the exergy losses during a process, is given by Eq. (3):

$$\eta_{II} = \frac{\text{useful exergy out}}{\text{exergy input}} = 1 - \frac{\text{exergy destruction}}{\text{exergy input}}$$  \hspace{1cm} (3)

In this study, useful exergy is the work output, and exergy input is the fuel exergy; thus the second-law efficiency can be calculated as follows:

$$\eta_{II} = \frac{\text{work output}}{\text{fuel exergy}}$$  \hspace{1cm} (4)

The exergy destruction of each component and the work output are analyzed in terms of fuel exergy percentage, as follows:

$$\text{Percentage of fuel exergy} = \frac{\text{exergy destruction of each component}}{\text{fuel exergy}} \times 100$$

$$\text{Percentage of work output} = \frac{\text{work output of each component}}{\text{fuel exergy}} \times 100$$

3. Approaches and methodology

3.1. Modeling assumptions

The following assumptions are made in the model to simplify the analysis of the system:

1. Fuel is supposed to be pure methane and its temperature is equal to the ambient temperature.
2. All components of the system are assumed to operate under adiabatic conditions.
3. Working fluid is assumed to be treated as an ideal gas with variable specific heat.
4. The isentropic efficiency of the compressor and the turbine are assumed to be constant and equal to 0.85 and 0.87 respectively [7].
5. The combustion efficiency and the mechanical efficiency are assumed to be constant and equal to 0.98 and 0.99 respectively [15].
6. The effectiveness coefficient of the air-gas heat exchanger is equal to 0.85 [16].
7. The pressure drop through the air filter before the intake of the compressor, through the combustion chamber, and through both sides of the air—gas heat exchanger is a function of the inlet pressure of the component, and is given by the following equalities respectively [15]:

$$\Delta P_{\text{filter}} = 0.02 P_0$$  \hspace{1cm} (6)

$$\Delta P_{\text{comb}} = 0.03 P_2$$  \hspace{1cm} (7)

$$\Delta P_{\text{H.E.Air}} = 0.03 P_7$$  \hspace{1cm} (8)

$$\Delta P_{\text{H.E.Gas}} = 0.02 P_4$$  \hspace{1cm} (9)

3.2. Combustion process

To calculate the chemical exergy of the mixture of the combustion products, the mole fractions of the products must be determined. The combustion equation is given by Eq. (10).

$$\text{CH}_4 + a(\text{O}_2 + 3.76\text{N}_2) \rightarrow n_{\text{CO}_2} \text{CO}_2 + n_{\text{H}_2}\text{O}_2 + n_{\text{N}_2} \text{N}_2 + n_{\text{O}_3} \text{O}_2 + n_{\text{CO}_2} \text{CO} + n_{\text{H}_2} \text{H}_2 + n_{\text{H}} \text{H} + n_{\text{O}_2} \text{O}_2 + n_{\text{OH}_2} \text{OH} + n_{\text{NO}_2} \text{NO}$$  \hspace{1cm} (10)
The moles fractions of the species of the products can be calculated by considering atomic mass conservation equations (4 equations), equations of chemical equilibrium reactions (8 equations), and the calculation are applied for a given initial temperature of reactants in a specific temperature of combustion. This combustion temperature means Turbine Inlet Temperature (TIT). In the model there is a special subroutine which manages the species calculation of the combustion products.

3.3. Method of calculations

For a simple gas turbine cycle, the calculation procedure is as follows. The pressure drop due to air filtering before the compressor is considered. The inlet air enters the compressor at state 1. Considering an isentropic efficiency of $\eta_{comp}$ for the compressor and a pressure ratio of $R_c$, the state 2 at the compressor outlet can be calculated as:

$$R_c = \frac{P_2}{P_1}$$

(11)

$$\left( \frac{\dot{m}_r}{m} \right) h_1 - h_2 = \eta_{comb} \frac{\dot{m}_r}{m} Q_{UNF}$$

(12)

$$i_{comp-out} = T_0 (s_2 - s_1)$$

(13)

For a specified TIT, the exhaust temperature and the specific work of the cycle can be obtained as follows. Having considered the combustion chamber pressure drop, combustion chamber

\[\text{Flowchart 1. Gas turbine with ABC calculations.}\]
efficiency, mechanical compressor efficiency, mechanical turbine efficiency and mass flow rate of the fuel, we have:

\[ H_3 - H_2 = \eta_{\text{comb}} n_{\text{comb}} Q_{\text{LHV}} \quad (14) \]

\[ (S_{4u} - S_{f1}) - RLH_P^4 P_4 = 0 \quad (15) \]

\[ \eta_{\text{turb}} = \frac{h_3 - h_4}{h_3 - h_4} \quad (16) \]

\[ W_{\text{cycle}} = h_1 + h_3 - h_2 - h_4 \quad (17) \]

Similarly, the calculation model for a gas turbine with air bottoming cycle (Fig. 1) can be developed; the procedure is given in Flowchart 1.

For both cycles, the exergy destruction of each component is calculated in the model from the exergy balance for a steady-state open system mentioned in section 2.

In this work, MATLAB 7.3 software was employed to perform the calculations. Each device of a power plant cycle is defined as a function in MATLAB; these functions are used to calculate the parameters related to that device. Tables of thermodynamic properties [14] available for air, combustion products and steam are used in the program.

### 3.4. Model validations

To validate the model, the results are compared to the data obtained from the experiments performed by Ghazikhani et al. [17] for the GE-F5 simple gas turbine, and also to the results obtained by Najjar et al. [11] for the ABC cycle. Fig. 2 shows a comparison of model results with those of the experiments [17] for the SFC against specific work, where a good agreement is observed. Fig. 3 shows
the comparison between the results of the model with those of Najjar et al. [11] for the overall efficiency in an ABC cycle. As TIT is increased the efficiency is also increased. In this case, the difference between the two results is averagely less than 5% which indicates a good agreement.

4. Results and discussion

Fig. 4 shows that the exergy destruction of the main turbine and compressor of the ABC and the simple gas turbine increases when the ambient temperature is increased. To explain this, it can be reasoned that for an adiabatic turbine and compressor with constant specific heat, we have [14]:

$$i_{\text{turb}} = T_0c_p \ln \left[ \frac{1}{\eta_{\text{turb}}} \left( \frac{P_1}{P_2} \right)^{\frac{(k-1)}{k}} + \eta_{\text{turb}} \right]$$

(18)

and

$$i_{\text{comp}} = T_0c_p \ln \left[ \frac{1}{\eta_{\text{comp}}} \left( \frac{1}{\eta_{\text{comp}}} - 1 \right) \left( \frac{P_1}{P_2} \right)^{\frac{(k-1)}{k}} \right]$$

(19)

respectively, where $i_{\text{t}}$ and $i_{\text{c}}$ are the exergy destruction of the turbine and compressor, and $\eta_i$ and $\eta_c$ are the turbine and compressor isentropic efficiencies.

These equations show that the internal irreversibilities of the turbine and compressor are only functions of the pressure ratio and isentropic efficiencies for given values of $c_p$, $k$ and $T_0$. Therefore, by increasing the ambient temperature, $c_p$ increases, which leads to an increase in the amount of the above mentioned irreversibilities.

Fig. 5 shows the exergy destruction of the combustion chamber in both cycles. The exergy destruction of the combustion chamber is mainly attributed to the chemical reaction during the combustion.
process. Since the amount of air/fuel ratio and injected fuel into the combustion chamber for both cycles is identical, the combustion chamber exergy destruction of the both cycles for a given ambient temperature is the same.

The exergy destruction of combustion comprises of two phenomena; the difference between the temperatures of fuel and inlet air during the mixing process and the difference between the temperatures of reactants and products in the combustion chamber.

As Fig. 5 shows, increasing the low ambient temperature results in an increase in the exergy destruction. This is due to the increased difference between the reactor discharge and the reaction zone property. At higher ambient temperatures, the exergy destruction reduces due to the smaller difference between reactant and product species.

Fig. 6 shows that the total exergy destruction of the simple gas turbine is greater than that of the combined gas turbine. This is because of the exhaust exergy recovery in the ABC cycle. It should be noted that the exhaust exergy in a simple gas turbine is completely wasted by the mixing of the combustion products into the atmosphere. By converting the simple gas turbine to the ABC cycle, the total exergy destruction decreases; it can be said that the total exergy destruction is decreased by the exhaust exergy recovery and increased by adding new components. Because of the lower total exergy destruction in the ABC cycle, a greater amount of fuel exergy will be converted into work compared to the simple gas turbine cycle as shown in Fig. 6.

Fig. 7 shows the comparison between the simple gas turbine exhaust exergy destruction reduction and the exergy destruction created by new components. As Fig. 7 shows, the exergy destruction following the addition of the new components is much lower than the amount of exhaust exergy destruction reduced in the new cycle. This explains why the SFC is lower in ABC compared to a simple gas turbine. This is confirmed in Fig. 8. The SFC of the ABC is averagely 13.3% less than that of the simple gas turbine. As indicated in Fig. 8, this improvement is greater at lower ambient temperatures. In fact a lower ambient temperature leads to a higher temperature difference inside the regenerator, which intern influences the SFC by increasing the exergy recovery as well as the exergy destruction, however, the exergy recovery is dominant.

Fig. 9 shows that in the simple gas turbine, 55% of the fuel exergy has been approximately lost in the exhaust and combustion chamber; the figure also shows that about 6% of the fuel exergy is associated with the exergy destruction of the turbine and the compressor, and the remainder of the fuel exergy accounts for the work output of the cycle.

Fig. 10 shows the percentage of work output and exergy destructions in each component from fuel exergy consuming in the gas turbine with air bottoming cycle. The decrease in the exhaust exergy destruction and the increase in the work output of the ABC is the only significant change compared to the simple gas turbine (compare Fig. 10 with Fig. 9). With the exhaust exergy recovery in the ABC, approximately 12.3% of fuel exergy is recovered; 4.9% accounts for the exergy destruction of the additional components in the ABC, while 7.4% results in the increase of the work output (compare Fig. 10 with Fig. 9).

It is also understood from Fig. 10 that the amount of exergy destruction of the regenerator is higher than that of the other components are added to provide ABC from simple gas turbine (i.e.,
the turbine and the compressor in the bottoming cycle). Fig. 10 also shows that with the increase in the ambient temperature, the exergy destruction of the regenerator decreases due to the low temperature difference in the regenerator at high ambient temperatures. Exergy destruction terms of the bottoming cycle compressor and turbine are less than those of the simple gas turbine as in the former, the temperature difference between the outlet and inlet is smaller (compare Fig. 10 with Fig. 9).

As Fig. 11 shows, the simple gas turbine has lower second-law efficiency for all given Rs and TITs except for the small value of TIT and pressure ratios higher than 14. This is due to poor exhaust exergy at the low TIT, making the ABC cycle inefficient. The figure also illustrates that the ABC has the advantage of higher specific works and second-law efficiencies at lower pressure ratios and TITs.

Fig. 12 shows the SFC in the ABC and the simple gas turbine for different TITs and pressure ratios. It is interesting to note that in the figure, the maximum specific work has been shown for all operating conditions.

As shown in the figure, in the simple gas turbine there are two SFCs for a given specific work output at some points. For the ABC, this situation occurs for TITs above 1400 °C. The trend reveals that the effect of the pressure ratio on the SFC of the gas turbine with ABC is less important than that of the simple gas turbine. This means that for a given TIT in a simple gas turbine, as the pressure ratio increases the specific work has been shown for all operating conditions.

As shown in the figure, the simple gas turbine has lower second-law efficiency for all given Rs and TITs except for the small value of TIT and pressure ratios higher than 14. This is due to poor exhaust exergy at the low TIT, making the ABC cycle inefficient. The figure also illustrates that the ABC has the advantage of higher specific works and second-law efficiencies at lower pressure ratios and TITs.

### Table 1

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As shown in Table 2, the isentropic efficiency of the turbine is the most effective parameter to improve the cycle performance of simple and ABC gas turbines. It can be noted that the parameters of the gas turbine with ABC are generally more sensitive than those of

### Table 2

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As shown in Table 2, the isentropic efficiency of the turbine is the most effective parameter to improve the cycle performance of simple and ABC gas turbines. It can be noted that the parameters of the gas turbine with ABC are generally more sensitive than those of
the simple gas turbine. This is because in the ABC, by increasing the
turbine efficiency at constant pressure ratio, the exhaust temper-
Figure 5-1ature is increased in the topping cycle at a constant expansion ratio,
increasing the possibility of exergy recovery in the recuperator. This
is affected to the all parameters in the ABC cycle (i.e., more sensi-
tivity of the ABC cycle). But in the simple gas turbine increasing
turbine efficiency at constant pressure ratio just exhaust temper-

ture increased.

6. Conclusions

In this work the components exergy destruction of the simple
and ABC gas turbine cycles in different operating conditions are
investigated. While the regenerator has the largest contribution in
the exergy destruction of the air bottoming cycle, the recovery of
the exhaust exergy in a gas turbine with ABC still plays the most
important role in increasing the cycle performance. This makes the
SFC of the gas turbine with ABC to be decreased averagely by 13.3%
and the specific work to be increased by 15.4% compared to those of
the simple gas turbine. The exergy analysis shows that the SFC of
the ABC is typically smaller compared to that of the simple gas
turbine for almost all TITs. At higher TITs; however the pressure
ratio of the gas turbine with air bottoming cycle has no signi-
ficant effect on fuel economy. This can provide conditions under
which we can have a lower pressure ratio with an acceptable SFC.

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