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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

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Nomenclature

ABC	air bottoming cycle
C	heat capacity rate
CC	combustion chamber
CHP	combined heat and power (cogeneration)
c_p	specific heat at constant pressure[kJ kg^{-1} K^{-1}]
h	specific enthalpy [kJ kg^{-1}]
$\overline{h}$	enthalpy per mole of species [kJ kmol^{-1}]
i	specific exergy destruction [kJ kg^{-1}]
I	exergy destruction [kJ]
k	specific heat ratio
$\dot{m}$	mass flow rate [kg s^{-1}]
Q_{LHV}	lower heating value [kJ kg^{-1}]
P	pump; Pressure [Pa]
R	specific-gas constant [kJ kg^{-1} K^{-1}]
$\overline{R}$	universal gas constant [kJ kmol^{-1} K^{-1}]
R_c	pressure ratio in topping cycle
r_c	pressure ratio in bottoming cycle
r_m	mass flow ratio of bottoming cycle
s	specific entropy [kJ kg^{-1} K^{-1}]
$\overline{s}$	entropy per mole of species [kJ kmol^{-1} K^{-1}]
SFC	specific fuel consumption [g (kWh)$^{-1}$]
T_0	ambient temperature [K]
TIT	turbine inlet temperature [K]
W	work interaction [kJ]
y	mole fraction in the vapor phase

Greek letters

ε_{rec}	effectiveness coefficient of the air-gas heat exchanger [%]
η	efficiency [%]
η_{II}	second-law efficiency [%]
ψ_{tot}	sum of thermomechanical and chemical stream exergy

Subscripts

2s	isentropic process value
amb	ambient
bot	bottoming cycle component
CV	control volume
e	exit state value
ex	exhaust
i	inlet state value
0	thermomechanical (restricted) dead state
00	environmental (unrestricted) dead state
ch	chemical
comb	combustion chamber
comp	compressor
f	fuel
reacs	reactants
rec	regenerator
top	topping cycle component
tot	total value
turb	turbine

10.5% points to the efficiency of an LM2500PE gas turbine. He also claimed that according to the feasibility study, the ABC is an economical alternative for power generation on both new oil/gas platforms and on the existing platforms with demand for more power. In 1996, Najjar et al. [11], proposed a parametric analysis of a gas turbine with air bottoming cycle using a computer program. He found, besides reducing the cost of hardware installations, it could achieve a higher thermal efficiency than simple gas turbine which does not deteriorate at part load as happens in a simple one. In 1998, Korobitsyn [5] discussed about the advanced gas turbines and compared the conventional steam bottoming cycle with air bottoming cycle. He demonstrated that the ABC shows performance values close to and exceeding those of the steam bottoming cycle. Also Korobitsyn [6] in 2002 concluded that the combination of a gas turbine and an ABC represents a high efficiency Combined Heat and Power (CHP) plant that provides clean and hot air for process needs [2]. The technical-economic analysis showed that an implementation of this scheme at industries that require hot air will result in significant fuel savings and will have a payback time of 3 years. Korobitsyn notes that the ABC allows low-maintenance costs and a short start-up time. Furthermore the plant can be implemented in regions where water resources are limited. In 2004, Poullikkas [2] explained the ABC cycle in his review paper. He referred to the recent studies reported an increase of power by 18–30% depending on the number of intercoolers, and an efficiency growth of up to 10% points.

Although there are advantages in using the gas turbine with air bottoming cycle like low capital cost and simple operation [5], and the ABC can be used for especial applications such as the high-efficiency CHP plant [4], the limited power output due to the need for a large and heavy regenerator (the 45 tons weighing regenerator in a 6.6 MW gas turbine [10]) could be a disadvantage of the ABC cycle.

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

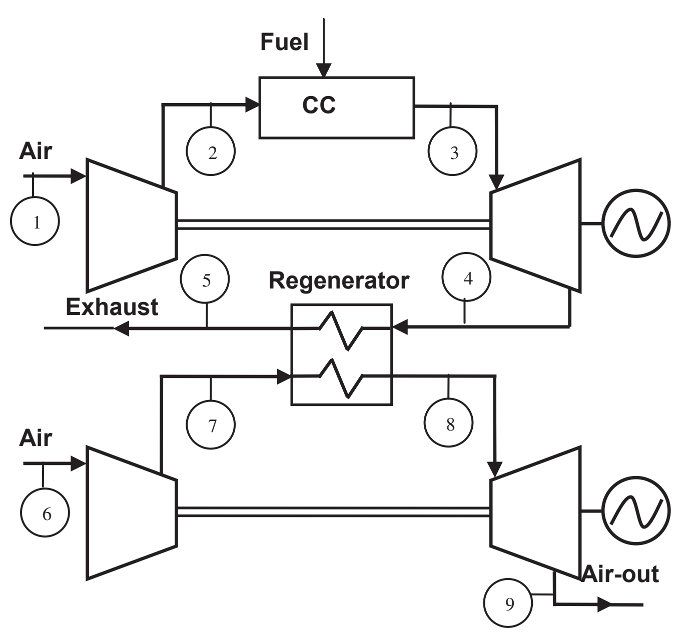


Fig. 1. A gas turbine with air bottoming cycle.

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 higher 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]:

$$\dot{I}_{tot} = \sum Q^o\left(1-\frac{T_o}{T_j}\right) + \sum_{in}\dot{m}\psi - \sum_{out}\dot{m}\psi + \dot{W}_{act} \quad (1)$$

where $\dot{I}_{tot}$ is the total exergy destruction of the system, $\sum_{in}\dot{m}\psi$ is the total stream exergy entering the system, $\sum_{out}\dot{m}\psi$ is the total stream exergy exiting the system, and $\dot{W}_{act}$ is the shaft work transfer rate [14].

Total stream exergy (ψ_{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, $\overline{h}_{i,T}$ and $\overline{h}_{i,T_0}$ are the enthalpy per mole of a species at temperature of T and T_0, $\overline{s}_{i,T,P}$ and $\overline{s}_{i,T_0,P_0}$ are the entropy per mole of a species at inlet flow and thermo-mechanical state respectively, $\overline{R}$ is the universal gas constant [14].

$$\psi_{tot} = \sum_{i=1}^{n} y_i\left(\overline{h}_{i,T} - \overline{h}_{i,T_0} - T_0\left(\overline{s}_{i,T,P} - \overline{s}_{i,T_0,P_0}\right)\right) + \overline{R}T_0\sum_{i=1}^{n} y_i\left(\ln\frac{y_i}{y_{i,00}}\right) \quad (2)$$

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

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

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:

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

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

$$\begin{aligned}&\text{Percentage of fuel exergy} = \frac{\text{exergy destruction of each component}}{\text{fuel exergy}} \times 100\\&\text{Percentage of work output} = \frac{\text{work output of each component}}{\text{fuel exergy}} \times 100\end{aligned} \quad (5)$$

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]:

$$\Delta P_{filter} = 0.02P_0 \quad (6)$$

$$\Delta P_{comb} = 0.03P_2 \quad (7)$$

$$\Delta P_{H.E.Air} = 0.03P_7 \quad (8)$$

$$\Delta P_{H.E.Gas} = 0.02P_4 \quad (9)$$

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).

$$CH_4 + a(O_2 + 3.76N_2) \rightarrow n_{CO_2}CO_2 + n_{H_2O}H_2O + n_{N_2}N_2 + n_{O_2}O_2 + n_{CO}CO + n_{H_2}H_2 + n_HH + n_OO + n_{OH}OH + n_{NO}NO \quad (10)$$

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

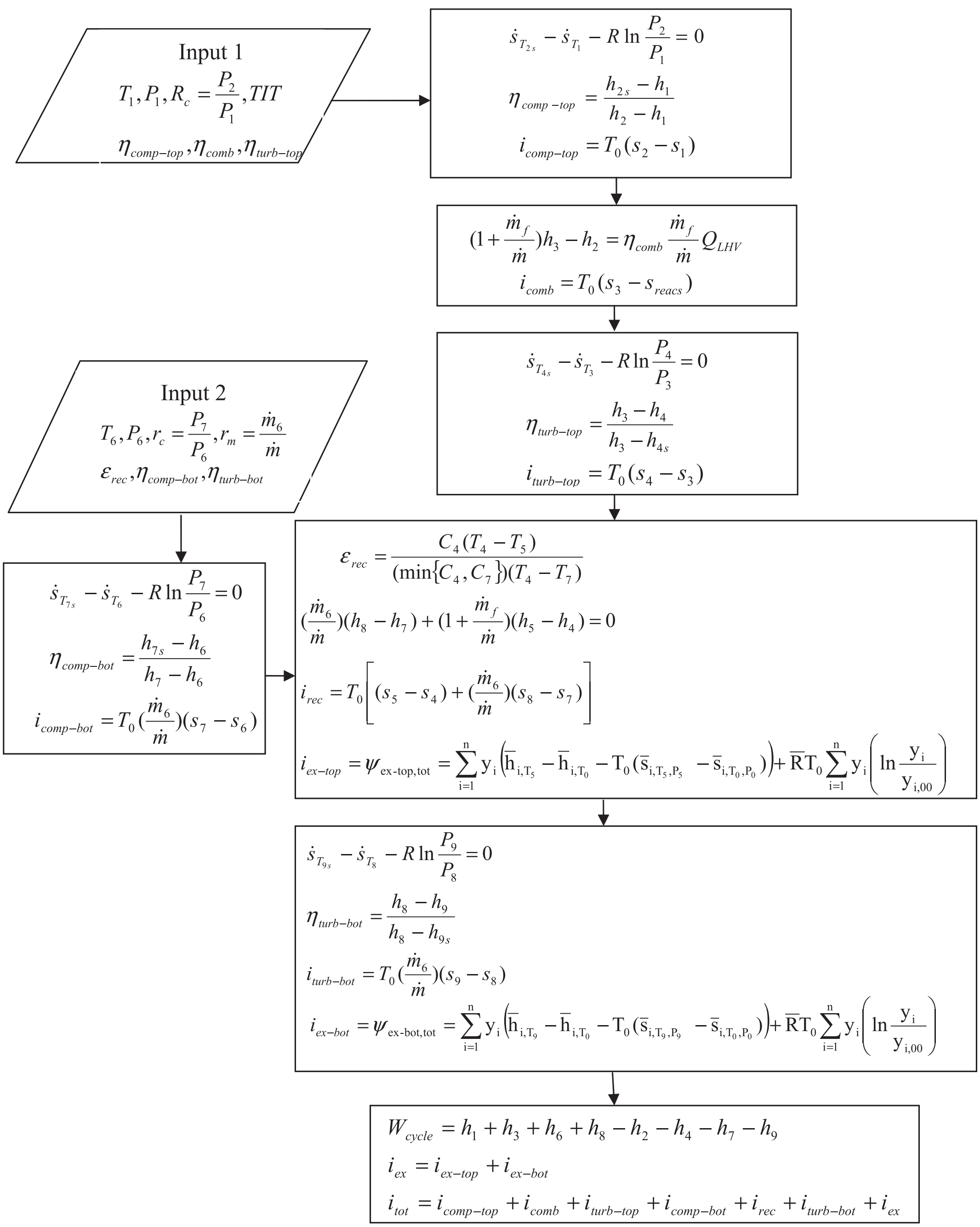
For a simple gas turbine cycle, the calculation procedure is as follows. The pressure drop due to air filtering before the compressor is considered. The inlet air enters the compressor at state 1. Considering an isentropic efficiency of η_{comp} for the compressor and a pressure ratio of R_c, the state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

$$R_c = \frac{P_2}{P_1} \tag{11}$$

$$\left(\dot{S}_{T_{2s}} - \dot{S}_{T_1}\right) - \dot{m}R\mathrm{Ln}\frac{P_2}{P_1} = 0 \tag{12}$$

$$\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{13}$$

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



Flowchart 1. Gas turbine with ABC calculations.

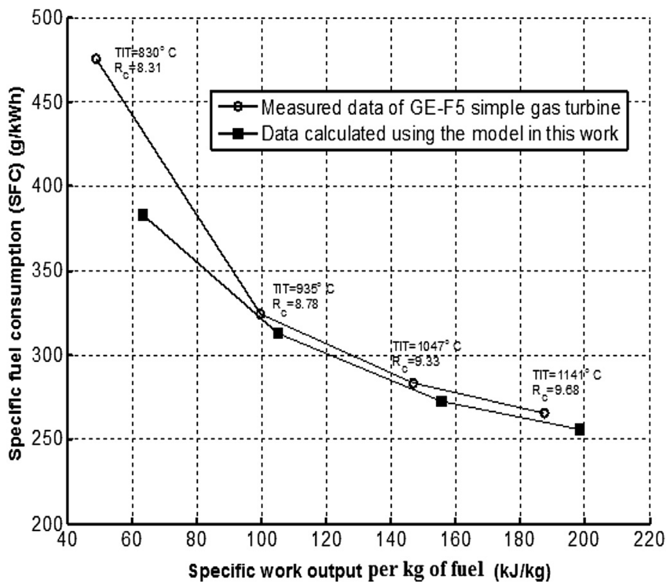


Fig. 2. Comparison between the results of the model and those of the experiments [17] for the GE-F5 simple gas turbine.

efficiency, mechanical compressor efficiency, mechanical turbine efficiency and mass flow rate of the fuel, we have:

$$H_3 - H_2 = \eta_{comb} \dot{m}_f Q_{LHV} \tag{14}$$

$$\left(\dot{S}_{T_{4s}} - \dot{S}_{T_3}\right) - RLn\frac{P_4}{P_3} = 0 \tag{15}$$

$$\eta_{turb} = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{16}$$

$$w_{cycle} = h_1 + h_3 - h_2 - h_4 \tag{17}$$

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

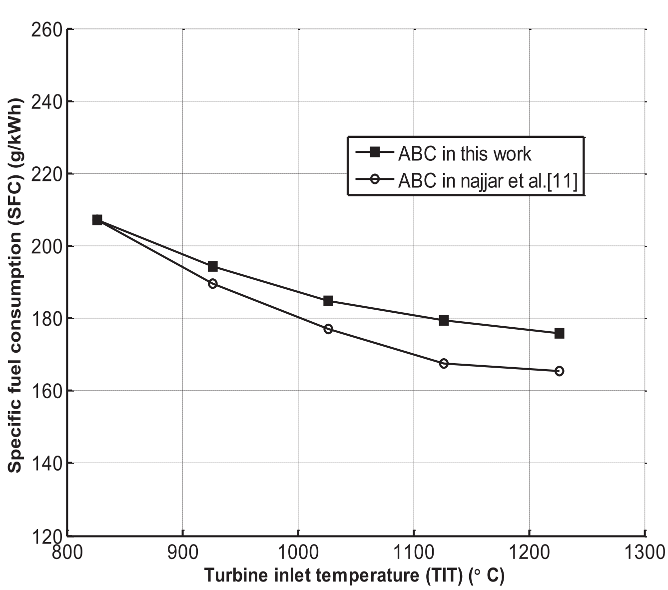


Fig. 3. Comparison between the results of the model and those of Najjar el al [11]. for a gas turbine with ABC.

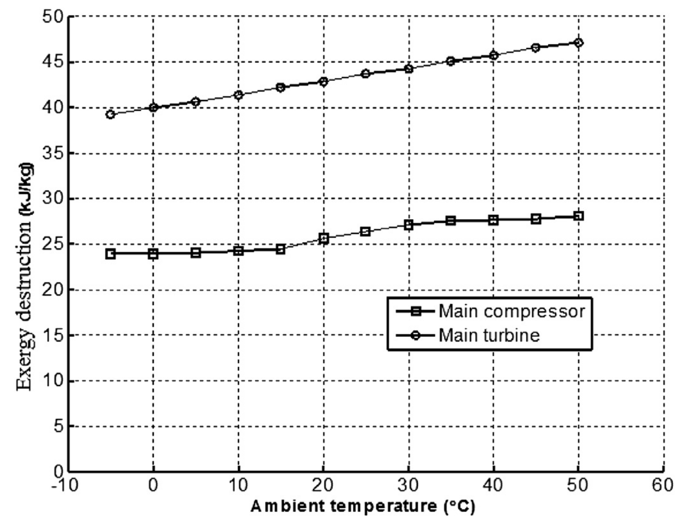


Fig. 4. Exergy destruction of the main turbine and compressor in ABC and simple gas turbine cycle ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

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

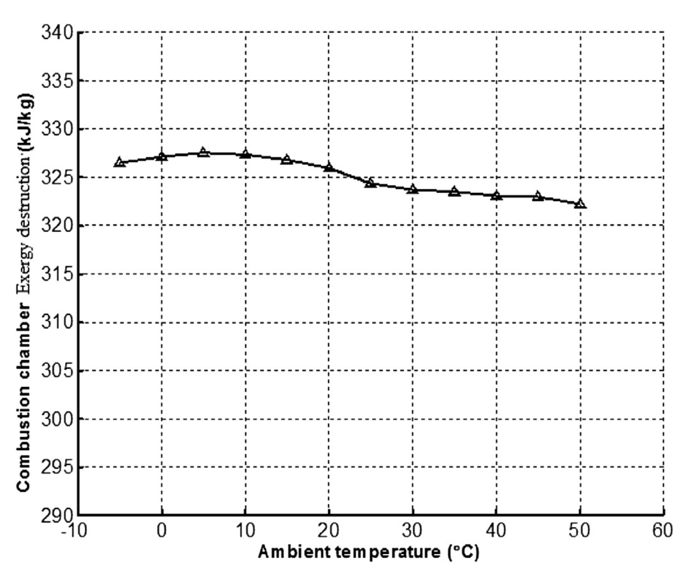


Fig. 5. Exergy destruction of the combustion chamber in ABC and simple gas turbine cycle ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

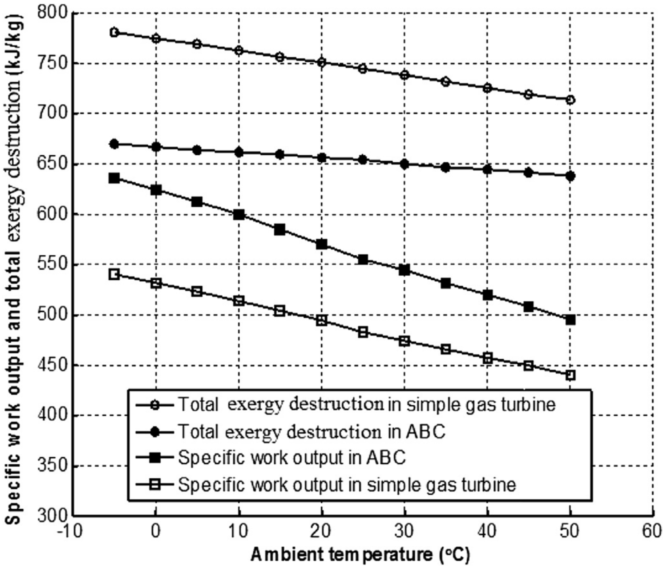


Fig. 6. Comparison between work output and total exergy destruction variations in ABC and simple gas turbine ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

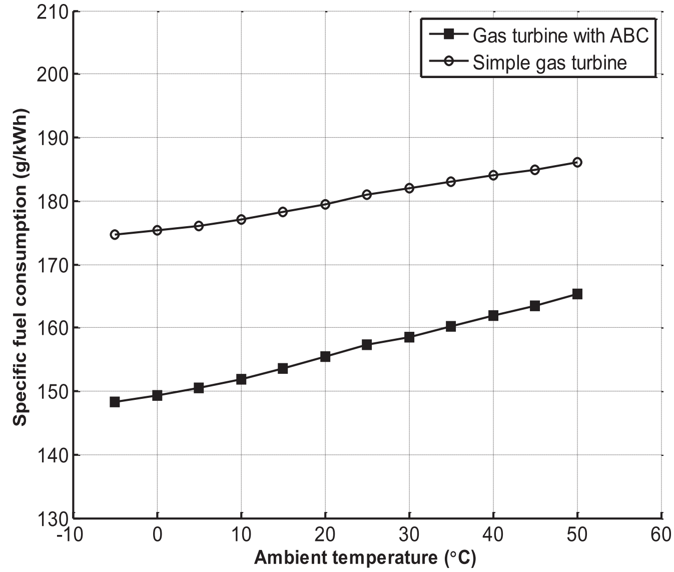


Fig. 8. SFC of ABC cycle and simple gas turbine cycle ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

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]:

$$i_{turb} = T_0 c_p \ln\left[(1-\eta_{turb})\left(\frac{P_1}{P_2}\right)^{(k-1)/k} + \eta_{turb}\right] \tag{18}$$

and

$$i_{comp} = T_0 c_p \ln\left[\frac{1}{\eta_{comp}} - \left(\frac{1}{\eta_{comp}} - 1\right)\left(\frac{P_1}{P_2}\right)^{(k-1)/k}\right] \tag{19}$$

respectively, where i_t and i_c are the exergy destruction of the turbine and compressor, and η_t and η_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

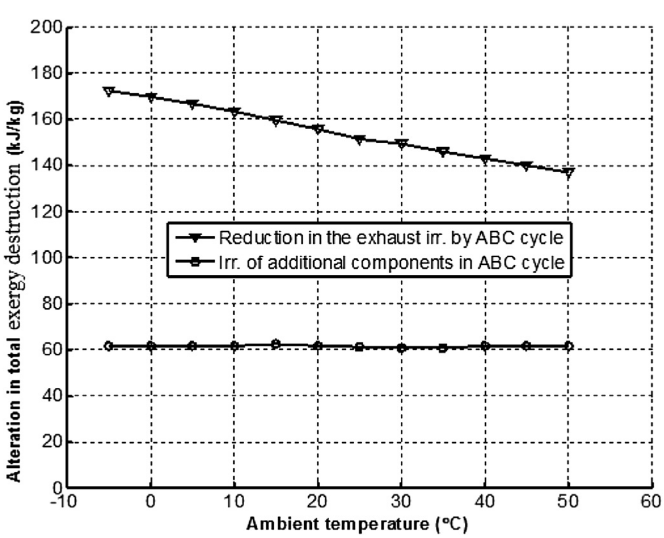


Fig. 7. Comparison between exhaust exergy destruction reduction by ABC cycle and the exergy destruction created by additional components in ABC cycle ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

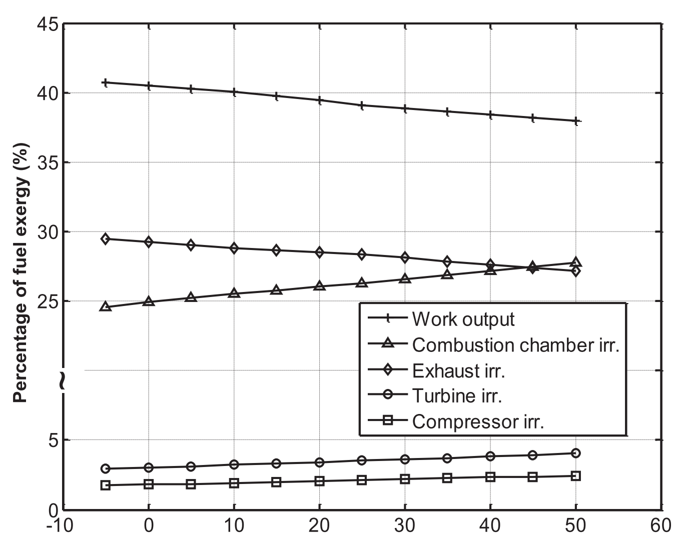


Fig. 9. Simple gas turbine: work output, exergy destruction of components as a percentage of fuel exergy ($R_c = 25$, $r_c = 6$ and TIT = 1400 °C).

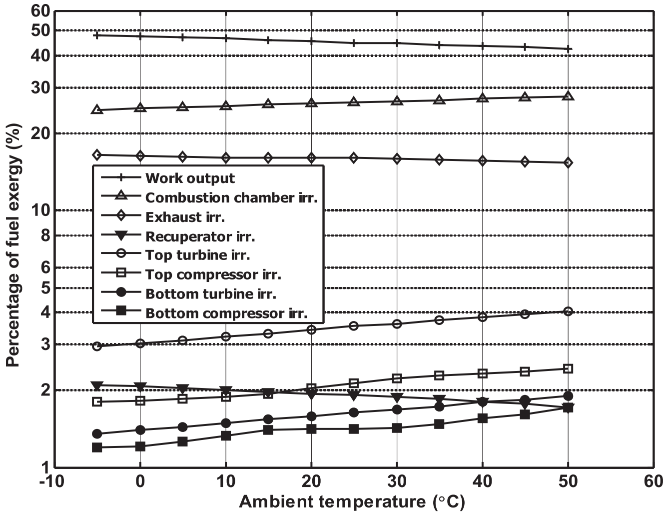


Fig. 10. Gas turbine with ABC: work output of the cycle and exergy destruction (irreversibility) of components as percentages of fuel exergy (R_c = 25, r_c = 6 and TIT = 1400 °C).

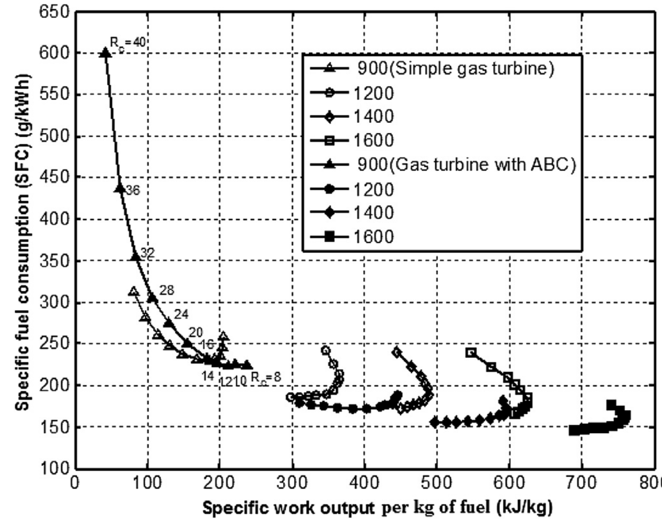


Fig. 12. Variations of SFC against specific work output at different R_cs and TITs in simple gas turbine and gas turbine with ABC (r_c = 6, T_{amb} = 25 °C).

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 compressor 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.,

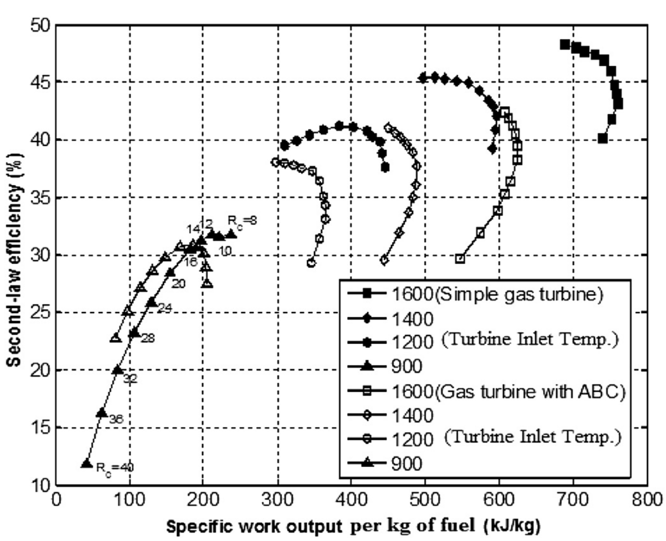


Fig. 11. Variations of second-law efficiency against specific work output at different R_cs and TITs (Turbine Inlet Temperature) in simple gas turbine and gas turbine with ABC (r_c = 6, T_{amb} = 25 °C)

Table 1
The specific fuel exergy and the percentages of specific work and components exergy destruction for simple and ABC gas turbines at different operating conditions ($r_c = 6$).

R_c	TIT (°C)	T_{amb} (°C)	$\psi_{ch.f}$ (kJ/kg)	Simple W (%)	I_{comp} (%)	I_{comb} (%)	I_{turb} (%)	I_{ex} (%)	ABC W (%)	I_{comp} top (%)	I_{comp} bot (%)	I_{comb} (%)	I_{turb} top (%)	I_{turb} top (%)	I_{rec} (%)	$I_{ex\ top}$ (%)	$I_{ex\ bot}$ (%)
10	900	5	749	30.9	2.6	34.2	3.5	28.8	35.3	2.6	2.2	34.2	3.5	2.5	1.8	12.3	5.6
		20	714	29.8	3.1	35.2	3.8	28.1	32.7	3.1	2.5	35.2	3.8	2.8	1.7	13.1	5.1
		35	683	29.2	3.1	36.4	4.2	27.1	30.7	3.2	2.6	36.4	4.3	3.1	1.7	13.5	4.5
	1200	5	1186	32.6	1.7	29.7	2.2	33.8	40.5	1.6	1.4	29.7	2.2	1.7	2.5	11.7	8.7
		20	1151	32	1.9	30.6	2.4	33.1	39.1	1.9	1.5	30.6	2.4	1.8	2.4	12.1	8.2
		35	1119	31.6	1.9	31.7	2.6	32.2	38.1	1.9	1.6	31.7	2.5	2.0	2.3	12.3	7.6
	1400	5	1505	33	1.3	27.4	1.7	36.6	41.9	1.3	1.1	27.4	1.7	1.3	2.8	12.2	10.3
		20	1469	32.5	1.5	28.2	1.9	35.9	40.9	1.5	1.2	28.2	1.9	1.4	2.7	12.4	9.8
		35	1437	32.2	1.5	29.2	2	35.1	40.2	1.5	1.2	29.2	2.0	1.5	2.6	12.5	9.3
25	900	5	552	34.3	4.3	31.3	7.5	22.6	33.2	4.3	3.0	31.2	7.6	3.5	1.2	13.1	2.9
		20	506	31.8	5	32.3	8.6	22.3	28.2	5.1	3.5	32.3	8.7	4.0	1.0	14.7	2.5
		35	461	28.8	6	33.3	10	21.9	23.0	6.0	3.9	33.3	10.0	4.7	1.1	16.0	2.0
	1200	5	984	39.6	2.4	27.3	4.1	26.6	44.4	2.4	1.7	27.3	4.1	1.9	1.7	11.0	5.5
		20	937	38.6	2.7	28.1	4.5	26.1	42.3	2.7	1.9	28.1	4.6	2.1	1.6	11.6	5.1
		35	891	37.4	3.1	29	5	25.5	40.4	3.1	2.0	29.0	5.0	2.4	1.6	11.9	4.6
	1400	5	1299	40.9	1.8	25.2	3.1	29	47.3	1.8	1.3	25.2	3.1	1.4	2.0	10.9	7.0
		20	1251	40.1	2	26	3.4	28.5	45.8	2.0	1.4	26.0	3.4	1.6	1.9	11.3	6.6
		35	1204	39.2	2.3	26.9	3.7	27.9	44.4	2.3	1.5	26.9	3.7	1.7	1.8	11.5	6.2
40	900	5	428	31.2	6.1	30	11	21.3	26.3	6.1	3.8	30.0	11.5	4.6	1.2	14.9	1.6
		20	377	26.8	7.2	31	14	21.4	17.8	7.3	4.7	31.0	13.7	5.5	1.2	17.5	1.3
		35	326	20.9	8.9	32.1	17	21.6	7.5	9.0	5.5	32.1	16.6	6.7	1.4	20.3	0.9
	1200	5	856	41.2	3	26.1	5.8	23.9	44.3	3.0	1.9	26.1	5.8	2.3	1.3	11.1	4.2
		20	804	39.7	3.4	27	6.4	23.5	41.3	3.4	2.2	27.0	6.5	2.6	1.2	11.9	3.9
		35	752	37.9	3.9	27.9	7.2	23.1	38.5	3.9	2.4	27.9	7.3	2.9	1.3	12.5	3.3
	1400	5	1169	43.3	2.2	24.2	4.2	26.1	48.4	2.2	1.4	24.2	4.2	1.6	1.7	10.6	5.7
		20	1115	42.3	2.4	25	4.6	25.7	46.5	2.5	1.6	25.0	4.6	1.8	1.6	11.1	5.3
		35	1063	41.2	2.7	25.8	5.1	25.2	44.9	2.8	1.7	25.8	5.1	1.9	1.6	11.4	4.8

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).

Table 2
The sensitivity analysis of the selected parameters η_{comp}, η_{turb}, and ε_{rec} influencing on η_{II}, W, I_{tot}, and SFC ($T_{amb} = 25$ °C).

TIT		1200 °C			1400 °C		
Item		η_{comp}	η_{turb}	ε_{rec}	η_{comp}	η_{turb}	ε_{rec}
Base data		0.88	0.89	0.85	0.88	0.89	0.85
Absolute increase		1%	1%	1%	1%	1%	1%
Simple	η_{II}	0.69	2.14	–	0.52	1.91	–
	W	1.05	2.14	–	0.81	1.91	–
$R_c = 10$	I_{tot}	0.05	−0.99	–	0.05	−0.90	–
	SFC	−0.68	−2.10	–	−0.68	−2.10	–
ABC	η_{II}	1.08	2.07	0.31	0.67	1.88	0.33
	W	1.50	2.07	0.31	1.01	1.88	0.33
$R_c = 10$, $r_c = 6$	I_{tot}	−0.34	−1.33	−0.20	−0.20	−1.31	−0.22
	SFC	−1.07	−2.03	−0.31	−1.07	−2.03	−0.31
Simple	η_{II}	1.05	2.82	–	0.72	2.33	–
	W	1.78	2.82	–	1.27	2.33	–
$R_c = 25$	I_{tot}	0.10	−1.70	–	0.09	−1.51	–
	SFC	−1.04	−2.75	–	−1.04	−2.75	–
ABC	η_{II}	1.49	2.59	0.18	1.08	2.15	0.27
	W	2.23	2.59	0.18	1.64	2.15	0.27
$R_c = 25$, $r_c = 6$	I_{tot}	−0.33	−1.88	−0.11	−0.34	−1.80	−0.22
	SFC	−1.47	−2.53	−0.18	−1.47	−2.53	−0.18
Simple	η_{II}	1.47	3.53	–	0.92	2.72	–
	W	2.53	3.53	–	1.69	2.72	–
$R_c = 40$	I_{tot}	0.14	−2.20	–	0.12	−1.92	–
	SFC	−1.45	−3.41	–	−1.45	−3.41	–
ABC	η_{II}	2.12	3.48	0.28	1.35	2.49	0.22
	W	3.18	3.48	0.28	2.12	2.49	0.22
$R_c = 40$, $r_c = 6$	I_{tot}	−0.34	−2.32	−0.18	−0.35	−2.10	−0.17
	SFC	−2.08	−3.36	−0.28	−2.08	−3.36	−0.28

As Fig. 11 shows, the simple gas turbine has lower second-law efficiency for all given R_cs 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 is improved; i.e., the simple gas turbine can recover more of the exergy of the combustion products by means of more pressure ratio. However, for the ABC, the higher exhaust temperature is used to recover in the bottoming cycle; therefore, the higher pressure ratio is not as important as in the simple gas turbine. A detailed quantitative comparison between the simple and the ABC gas turbine is given in Table 1. The table presents the fuel exergy consumption by the percentage of specific work and the components irreversibilities corresponding to a wide range of variations related to the operating conditions.

5. Sensitivity analysis

As Table 2 shows, 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 temperature 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 sensitivity of the ABC cycle). But in the simple gas turbine increasing turbine efficiency at constant pressure ratio just exhaust temperature 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 significant 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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