Exergy Analysis of Gas Turbine Air- Bottoming Combined Cycle for Different Environment Air Temperature

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Abstract

In this paper, an exergy analysis of a conventional gas turbine cycle and a conventional gas turbine cycle (topping gas turbine cycle) with air bottoming cycle (ABC) is presented. For carrying out the exergy analysis, a gas turbine cycle with ABC were modeled in detailed and, for each component of cycle properties of working fluid and material flow exergy have been calculated. Results indicates that the second law efficiency of gas turbine with ABC is averagely 6% more than the second law efficiency of simple gas turbine while the intake air temperature of both cycles is the same. Also, by increasing the intake temperature, total irreversibility of both cycles increases.

Keywords: *exergy analysis, irreversibility, second law efficiency, air bottoming cycle.*

1 Introduction

The exhaust stream temperature of open cycle gas turbine is typically around 500 ºC and consequently there is considerable scope for heat recovery applications to improve cycle efficiency. Two basic heat recovery arrangements can be used to improve cycle efficiency [1]:

1. Recuperation cycles, in which the recovered heat is used in the same gas turbine cycle;

2. Bottoming cycles, in which the exhaust is used as a heat source for an essentially independent power cycle.

Recuperative cycles can be divided to gas to gas recuperation, steam injection, evaporation cycles and chemical recuperations. Bottoming cycles consist of a basic exhaust heat recovery arrangement, in which the exhaust is used as a heat source for an essentially independent power cycle. They can be divided to combined cycle and Kalina cycle. In the case of combined cycle (steam bottoming), the need for a high-pressure steam generator, a steam turbine and a condenser might be unfeasible on a small scale [2]. It should be mentioned that special requirements are imposed on water quality, high-pressure equipment and operators of steam plant.

An alternative air bottoming cycle (ABC) consists of a compressor, a heat exchanger and a gas turbine, Fig. 1, [2]. It operates at low and moderate pressure and uses the ambient air as a working fluid. Concept and details of the ABC behind a gas turbine and performance analysis of this combined cycle were described in [2-4]. Recent studies on gas turbine cycle with ABC reported an increase of power by 18-30% depending on the number of intercoolers, and efficiency growth of up to 10% points [5].

In this study, the performance of a gas turbine cycle with ABC is analyzed in terms of exergy destruction ratio, in order to focus better the main locations of losses and to compare the performance of a simple gas turbine cycle with a gas turbine cycle with ABC. Exergy analysis is performed for these two cycles. For carrying out the exergy analysis, a gas turbine cycle with ABC is modeled in detailed and, for each component of cycle, properties of working fluid and material flow exergy are calculated. Results indicate that the second law efficiency of gas turbine with ABC is averagely 6% more than the second law efficiency of simple gas turbine while the intake air temperature of both cycles is the same. Also, by increasing the intake temperature, total irreversibility of both cycles increases.

2 Exergy

Exergy analysis is based on the second law and generally allows process inefficiencies to the better pinpointed than does an energy analysis, and efficiencies to be more rationally evaluated [6]. The second law equations for the system under consideration were derived from a basic exergy balance. For any open system at steady state, an equation for the rate of exergy destruction is [7]:

$$\dot{I}_{tot} = \sum \dot{\Phi}_{Q,j} + \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi + (\dot{W}_{act}) \quad (1)$$

Where $\dot{I}_{tot}$ is the summation of total irreversibility of the system and T_o is the ambient temperature of the system's surrounding. Also $\dot{\Phi}_{Q,j}$ represents the availability transfer Q_j to or from a reservoir at T_j [7]. The mass flow rates of each of the material flows crossing the system boundary is represented by $\dot{m}$ and the specific exergy associated with each of the flows is represented by ψ which is made up of thermo-mechanical and chemical exergy. The chemical exergy derived from a composition imbalance between a substance and its environment. Since fuel is injected into the combustion chamber of a gas turbine cycle the chemical exergy of fuel and combustion products must be evaluated. On a molar basis, the specific total exergy of a material flow can be written as [7]:

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

Where y_i is the mole fraction of species i in the flow and $y_{i,00}$ is mole fraction of species i in the reference environment. The summation in equation (2) is carried out over all species present.

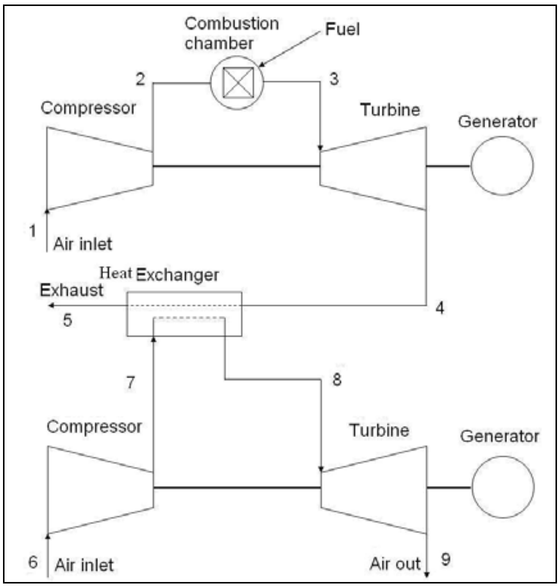

Fig. 1. *Gas turbine with ABC.*

3 Approaches and methodology

3.1 Modeling procedure

The gas turbine cycle with ABC was modeled using each component governing thermodynamic and chemical relations. The model is applied to one set of operating conditions and the effect of varying the inlet temperature (environment) is studied. To simplify the analysis of the system a number of assumptions are made. The assumptions are:

1- Fuel is considered the pure methane and its temperature is 298.15 K.
2- All components of the system are adiabatic and operate at steady state.
3- Working fluid is assumed the ideal gas with variable specific heat and analysis is made for unit mass of the working fluid of the topping gas turbine cycle.
4- Reference environment consist of a gaseous mixture at 1 atm and 298.15 K, composed of 76.62% nitrogen, 20.55% oxygen, 0.03% carbon dioxide and 1.88% water [7].
5- Inlet mass flow rate of the topping gas turbine cycle is constant, because the speed of compressor is constant and the effect of the varying temperature on the mass flow rate is negligible [8].
6- The isentropic efficiency of compressor and turbine are assumed to be constant and equal to 0.85 and 0.87 respectively [3].
7- Combustion efficiency and mechanical efficiency are assumed to be constant and equal to 0.98 and 0.99 respectively [9].
8- Effectiveness coefficient of the air-gas heat exchanger is equal to 0.85 [10].

9- The gas turbine inlet temperatures (TIT) of topping cycle and compression ratio of the topping cycle are 1400 K and 10 respectively.
10- Pressure drop through the air filter before the intake of the compressor, through the combustion chamber and through the both sides of the air-gas heat exchanger are a function of the inlet pressure of the component and according to Fig. 1. are given by [9]:

$$\Delta P_{filter} = 0.02\,P_0 \quad (3)$$

$$\Delta P_{comb} = 0.03\,P_2 \quad (4)$$

$$\Delta P_{H.E.Air} = 0.03\,P_7 \quad (5)$$

$$\Delta P_{H.E.Gas} = 0.02\,P_4 \quad (6)$$

4 Analysis procedure

The Brayton gas topping was analyzed first and the model of this cycle was validated by the experimental data [8]. The air- bottoming cycle is then connected to the topping cycle to construct gas turbine cycle with ABC. By applying the assumptions of section 3.1, the optimum compression ratio and mass flow ratio (mass flow rate of topping cycle to the mass flow rate of the bottoming cycle) of the air bottoming cycle by focusing on the most output work is calculated. The optimum compression ratio, r_c, and mass flow ratio, m_{opt}, of the air bottoming cycle are equal to 4.3 and 1.02 respectively. At this study the exergy analysis is carried out by considering the optimum compression ratio and mass flow ratio of the bottoming cycle. It should be mentioned that the amount of fuel consumption for the both cycles (simple and combined gas turbine cycle) are the same

Working fluid of air bottoming cycle is air. Woking fluid of topping cycle is air at first, but after the combustion chamber the combustion products represent the working fluid. For calculating the chemical exergy, 10 species of combustion products of the methane are calculated by the following equation [11]:

$$CH_4 + f(0.21O_2 + 0.79N_2) \rightarrow \nu_1CO_2 + \nu_2H_2O + \nu_3N_2 + \nu_4O_2 + \nu_5CO + \nu_6H_2 + \nu_7H + \nu_8O + \nu_9OH + \nu_{10}NO \quad (7)$$

Inlet temperature and pressure of air and fuel to the combustion chamber is specified. For calculating the mole fraction of 10 species of equation (7), a set of equations which consist of 4 equations of mass conservation, 6 equations of equilibrium reactions and one energy equation are solved. Solving this set of equations- which consist of linear and non linear equations and are solved using Newtonian method- gives the chemical species of combustion products on molar basis. By determining the species, mechanical and chemical exergy destruction for each component of the system was evaluated. By applying the energy balance and effectiveness coefficient of the heat exchanger, properties of the input to and output from the heat exchanger are evaluated in respect of r_c.

In general, the second law efficiency in this study is defined by:

η = (net exergy rate of the product/ net exergy rate of input) (8)

The net rate of the product is the net power produced, and the net exergy rate of input is calculated by subtracting the exergy rates of all exit streams from the exergy rates of all input streams. In this investigation, the input stream is taken to be fuel and the exit stream is the combustion products from the stack exhaust gases.

In this study the exergy destruction ratio of each component of cycle is defined by:

Exergy destruction ratio = (irreversibility of each component/ net exergy of inlet fuel) (9)

5 Results & discussions

Fig. 2 shows the output work for a conventional gas turbine cycle and a gas turbine cycle with ABC. Since the exhaust heat of topping cycle is recovered by the bottoming cycle, output work increases. This figure illustrates that output work of a conventional gas turbine cycle while the inlet temperature of compressor is 253 K(-20 ºC) is equal to output work of a gas turbine cycle with ABC while the inlet temperature of compressor is 316 K (43 ºC). So, this arrangement of gas turbine cycle with ABC can be used as a method to compensate the output work of a gas turbine cycle while the inlet temperature of compressor increases.

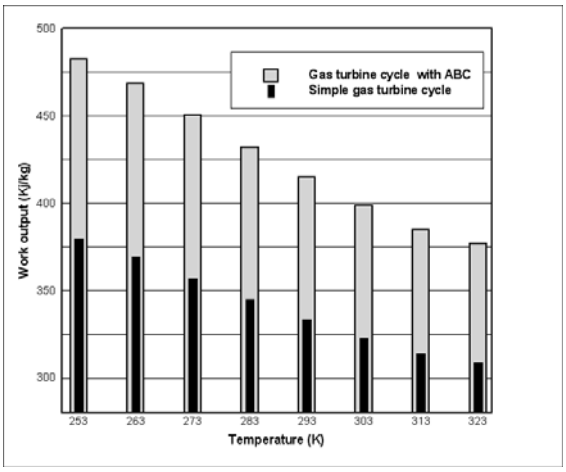

Fig. 2. *Work output for simple and combined gas turbine cycle.*

Fig. 3 shows that the thermal efficiency of a gas turbine cycle with ABC is averagely 9% than the conventional gas turbine cycle.

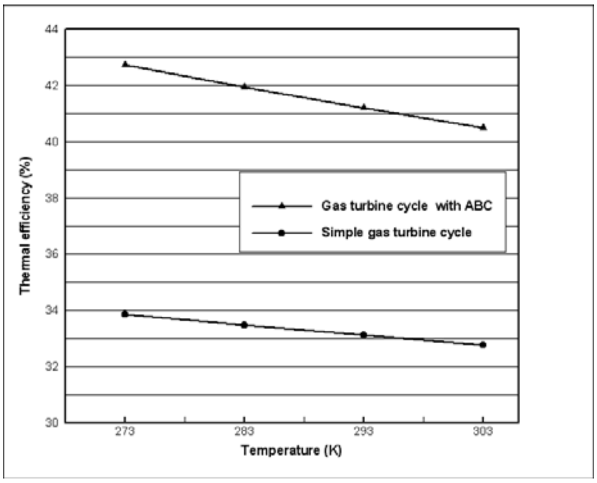

Fig. 3. *Thermal efficiency for simple and combined gas turbine cycle.*

Fig. 4 shows that the exergy destruction ratios of the compressors of the gas turbine cycle with ABC are more than the simple the gas turbine cycle. Since the gas turbine cycle with ABC has two turbines; the turbine irreversibility of gas turbine cycle with ABC is more than the conventional gas turbine, Fig. 5. The source of irreversibility in both the compressor and turbine is mainly frictional and thermal losses in the flow [12]. For a given isentropic efficiency, specific heat at constant pressure, isentropic exponent and pressure ratio the lost exergy is independent of inlet temperature of turbine and compressor [7,13]. At this investigation the calculations refer to variable specific heat at constant pressure; so by increasing the inlet temperature, specific heat at constant pressure and isentropic exponent increase which increase the exergy destruction and exergy destruction ratios of turbine and compressor, Fig. 4 & Fig. 5.

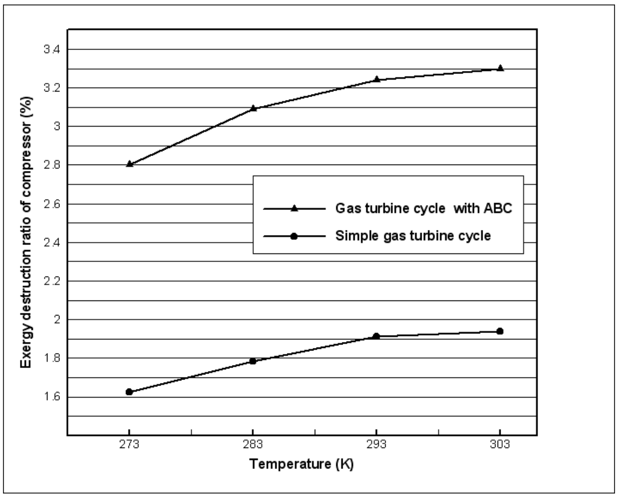

Fig. 4. *Exergy destruction ratio of compressor for simple and combined gas turbine cycle.*

Fig. 6 shows that combustion chamber has a major contribution in irreversibility of both cycles. It should be mentioned that the irreversibility and the amount of exergy destruction in the combustor are mainly attributed to a chemical reaction during the combustion process [12]. The Effect of the environment temperature on the irreversibility of combustion chamber is very little. Since the amount of injected fuel into the combustion chamber of both cycles is the same, so the combustor irreversibility of both cycles is the same, too. While the environment temperature is 298 K, the least exergy destroys. It can be explained that while the temperature of inlet air and fuel is the same (293 K), entropy generation of air- fuel mixing minimizes and the least exergy destruction occurs. By increasing the environment temperature the amount of injected fuel to combustion chamber decreases which results in increasing the exergy destruction ratio of combustor.

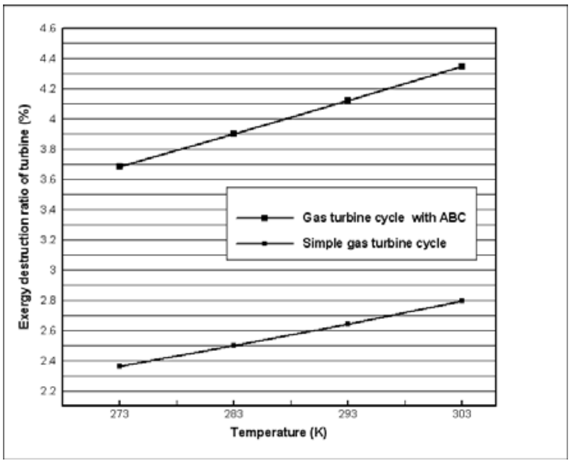

Fig. 5. *Exergy destruction ratio of turbine for simple and combined gas turbine cycle.*

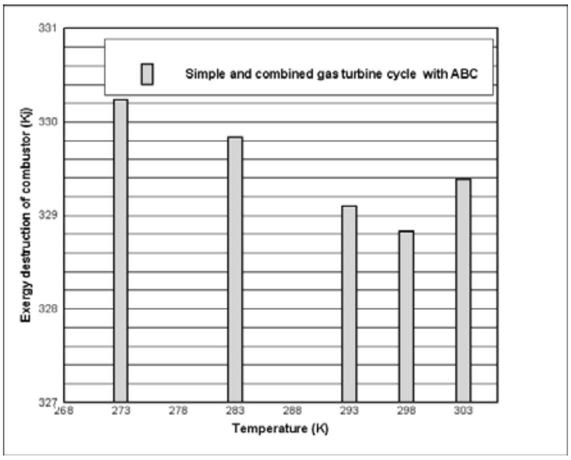

Fig. 6. *Exergy destruction of combustor for simple and combined gas turbine cycle.*

Fig. 7 shows the exergy destruction ratio of air-gas heat exchanger. This figure illustrates that the contribution of heat exchanger irreversibility among the total irreversibility of gas turbine with ABC is little. By increasing the environment temperature, the irreversibility of air-gas heat exchanger decreases. It can be explained that by increasing the environment temperature, the temperature difference between hot (gas) and cold (air) streams in both sides of the heat exchanger decreases which results in reduction of heat exchanger irreversibility and exergy destruction ratio.

Fig. 8 shows that the exhaust exergy of gas turbine cycle with ABC is less than the simple gas turbine cycle. It can be concluded that the by connecting an air bottoming cycle to a simple gas turbine cycle, exhaust exergy is recovered and more work is produced. Also Fig. 8 shows that by increasing the environment temperature the exhaust exergy of both cycles decrease. Also the reduction of exhaust exergy of gas turbine cycle with ABC is less that the simple gas turbine cycle. It can be explained that by increasing the environment temperature the cold stream (air) temperature of heat exchanger increases which results in decreasing the temperature difference between hot and cold streams and consequently results in decreasing the rate of heat transfer irreversibility.

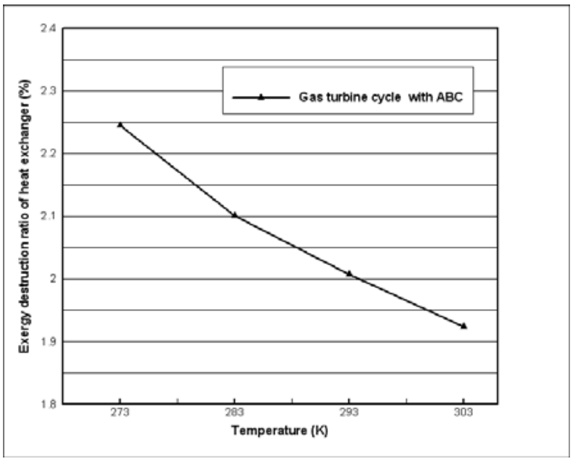

Fig. 7. *Exergy destruction of heat exchanger for combined gas turbine cycle.*

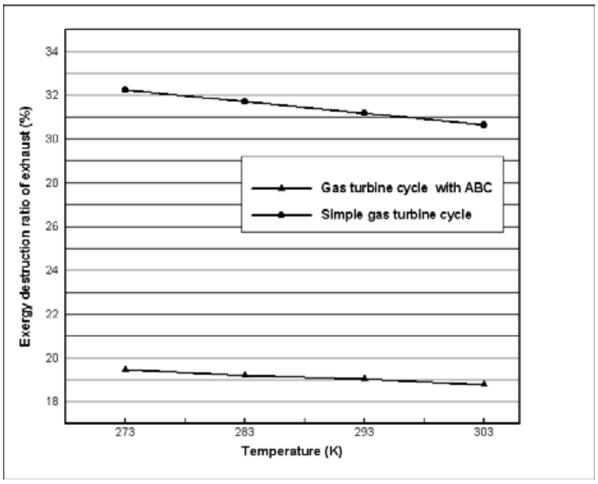

Fig. 8. *Exergy destruction ratio of exhaust for simple and combined gas turbine cycle.*

Fig. 9 shows the effect of varying the environment temperature (T_0) on the second law efficiency of the simple gas turbine cycle and the gas turbine cycle with ABC. This figure shows that the second law efficiency of gas turbine cycle with ABC is averagely 6% more than the second law efficiency of simple gas turbine cycle. It can be explained that bottoming cycle recovers exhaust exergy of the topping cycle; so exergy destruction of exhaust stream decreases and the net power output of the cycle increases and consequently, second law efficiency of gas turbine cycle with ABC exceeds the second law efficiency of the simple gas turbine cycle.

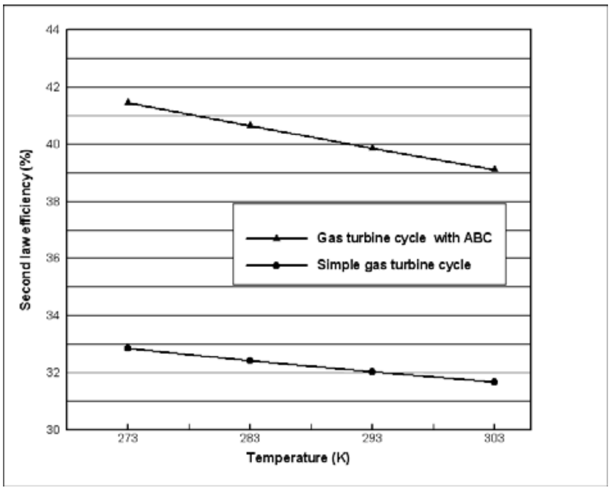

Fig. 9. *Second law efficiency of simple and combined gas turbine cycle.*

6 Conclusion

In this study, exergy- based analysis of a gas turbine air- bottoming combined cycle was carried out to investigate the effect of varying environment air temperature on the exergy destruction of this cycle. By connecting a heat exchanger to a simple gas turbine to recover the exhaust exergy, net work output increases which results in developing the second law efficiency by averagely 6%.

7 References

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