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Research Article

Exergy Analysis of Inlet Air Absorption Cooling Effects on Basic Cogeneration Systems

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Abstract

The use of the electrical energy is increasing in our life and in the world. The electrical energy is lost in the connection lines about 11%, as it is known. By producing the electrical and the heat energy in a cogeneration system to meet the needs, it can be obtained more efficiency in the use of fuel, and that can reduce energy costs. In a cogeneration cycle, by the absorption cooling system, the air entering into the compressor is cooled, and the cooling is obtained from energy of the heat of the exhaust gases. The system in this study is analyzed by using exergy analysis method and 1. and 2. laws of thermodynamics. Some of the heat energy is consumed to cool the air, in this cycle and the remaining heat is used to produce steam. The performance analysis of the whole cycle and also the devices that make up the cycle such as compressor, combustion chamber, turbine and heat exchanger were obtained and discussed. Also exergy losses, exergy efficiency and other performance parameters of the devices were obtained and discussed. The results showed that using absorption cooling (abc) system in a basic cycle made better than the basic one in electrical efficiency. However, because of the absorption cooling (abc) system the exergy efficiency is slightly less than the basic one. The absorption cooling (abc) cogeneration system can be used when the steam demand decreases or electrical demand increases for production more electricity.

Keywords: Cogeneration, Exergy, Absorption.

Giriş Havasının Absorpsiyonlu Soğutma Etkilerinin Temel Kojenerasyon Sistemlerinde Ekserji Analizi

Öz

Elektrik enerjisinin kullanımı hayatımızda ve dünyada giderek artmaktadır. Elektrik enerjisi bilindiği gibi bağlantı hatlarında yaklaşık %11 oranında kaybolmaktadır. Bir kojenerasyon sisteminde elektrik ve ısı enerjisinin ihtiyaca cevap verecek şekilde üretilmesi ile yakıt kullanımında daha fazla verim elde edilebilir ve bu da enerji maliyetlerini azaltabilir. Bir kojenerasyon çevriminde absorpsiyonlu soğutma sistemi ile kompresöre giren hava soğutulur ve soğutma egzoz gazlarının ısısının enerjisinden elde edilir. Bu çalışmada sistem, ekserji analizi yöntemi ve termodinamiğin 1. ve 2. yasaları kullanılarak analiz edilmiştir. Bu çevrimde ısı enerjisinin bir kısmı havayı soğutmak için tüketilir ve kalan ısı buhar üretmek için kullanılır. Tüm çevrimin performans analizi ve çevrimi oluşturan kompresör, yanma odası, türbin ve ısı eşanjörü gibi cihazlar elde edilmiş ve tartışılmıştır. Ayrıca cihazların ekserji kayıpları, ekserji verimliliği ve diğer performans parametreleri elde edilmiş ve tartışılmıştır. Sonuçlar, temel bir çevrimde absorpsiyonlu soğutma (abc) sisteminin kullanılmasının, elektrik verimliliğinde temel olandan daha iyi olduğunu gösterdi. Bununla birlikte, absorpsiyonlu soğutma (abc) sistemi nedeniyle ekserji verimliliği, temel olandan biraz daha düşüktür. Absorpsiyonlu soğutma (abc) kojenerasyon sistemi, daha fazla elektrik üretimi için buhar talebi azaldığında veya elektrik talebi arttığında kullanılabilir.

Anahtar Kelimeler: Kojenerasyon, Ekserji, Absorpsiyon.

1. Introduction

The use of electrical energy is increasing in our life and in the world. The electrical energy is lost in the connection lines about 11%, as it is known. By producing the electrical and the heat energy in a cogeneration system to meet the needs, it can be obtained more efficiency in the use of fuel, and that can reduce energy costs. By using cogeneration system, most of companies can secure to protect themselves for electricity cuts. Also, small scale companies can be used in university campuses, hotels, district heating systems, etc., (ASHRAE, 2000; Karaali, 2015).

In the industry most of the cogeneration plants are between 4 and 25 MW in size. Other cogeneration plants have more power than 25 MW or less power than 4 MW which is said micro cogeneration. That rate can be altered, month by month or on the annual average. Electrical to heat ratio is approximately 1.0, for the cogeneration plant driven by gas engine, approximately 0.6 for the cogeneration plant driven by gas turbine, and less than 0.4 for the cogeneration plant driven by steam power. The total efficiency of a cogeneration plant is the sum of the heat energy and electricity produced by the system to the fuel energy used. In a good designed cogeneration plant, the energy efficiencies are about 80–90% (Horlock, 1997). In cogeneration systems driven by gas turbine, the exhaust gases temperatures are about 850-1550 ^{0}C. Exhaust gases temperatures depends to the kind of fuel used into combustion chamber and on the compression rates. The mechanical energy obtained in the turbine, and at the outlet of the turbine the exhaust gases temperature decreases to 450-850 ^{0}C. The steam is obtained in the HRSG from the heat energy of the exhaust gas after the turbine (Karaali, 2015; Peters et al., 2003).

The first industrial gas turbines were used in industry in the 1950s and their designs were influenced by steam turbines and aerodynamic designs. In the late 1950s and early 1960s, lightweight gas turbines were modeled after aircraft engines and were used in power generation, electricity generation, and compression operations in pipelines. Industrial gas turbines are similar to steam turbines with a compression ratio of 12:1, combustion chamber outlet temperature 650-816 ^{0}C and shaft efficiency between 23-27%. In the 1970s, second generation industrial gas turbines were produced with a new approach and shaft efficiency increased to 32-37%. In these, the combustion chamber outlet temperatures have increased and the turbine blades have been made more resistant to high temperatures (Karaali, 2015; Peters et al., 2003).

Karaali and Öztürk (2015), applied some improvement methods for different situations considered for cogeneration systems. These methods of preheating the combustion air and the fuel, injecting the steam into the combustion chamber that produced from the system, and cooling inlet air. In between those methods, preheating the air and the fuel has very good electrical power performance. In addition, the combination of steam injection and preheating of fuel and air was also analyzed. With comparing and combining different improvement methods, they said that the preheating the combustion air and fuel method must be chosen for a cogeneration cycle. In case of a thermal power resource, steam injection method can be considered. For the possibility of cooling the compressor inlet air that does not bring much cost with the possibilities outside the system, it should be included into the cycle (Karaali, 2015). However, if it is possible to inject water into inlet air of the system, absorbent cooling by using some of the heat, or otherwise, in summer working conditions, additional improvement can be possible (Karaali, 2015; Peters et al., 2003).

According to ASHRAE (2000), the altitude of the cogeneration plant to be established, the humidity of the air and the temperature of the average ambient air should be known. In order to prevent NOx formation, demineralized water up to 2% of its mass can be sprayed into the inlet air of the cycle or steam can be injected into the combustion chamber. In addition, it is possible to increase the power by injecting water vapor around 3% of its mass into the compressor outlet air (ASHRAE, 2000).

Cooling the intake air increase the mechanical power and reduces the electrical heat ratio by cooling the inlet air, the density of the air is increased, which for most turbines where the volumetric flow is constant, means increasing the incoming air mass. As a cooling method, it can be applied by injecting water particles into the incoming air (evaporative cooling), absorption cooling or cooling by mechanical means. In evaporative cooling, the density of the incoming air is increased with the injected water particles, and the cooling of the incoming air is ensured by the evaporation of these water particles. In ideal evaporative cooling, the relative humidity of the air should be close to 100% (ASHRAE, 2000; Karaali, 2015). Thus, the temperature of the dry air is reduced by 5-15%. Mechanical cooling is done by chilled water system (more expensive and larger devices), absorption cooling system or direct mechanical cooling. Cooling the inlet air of the compressor increase the mechanical power produced, and reduces the heat power. Also, cooling the inlet air extends the life of the turbine by decreasing the temperature of the exhaust gases of the outlet of the combustion chamber (Karaali, 2015; Peters et al., 2003).

Some factors affect the performance of the gas turbine cogeneration cycles. These factors are altitude, ambient temperature, ambient and turbine inlet outlet pressures. For every 10 0C increase of the inlet temperature decrease the mechanical power produced are about 9%. The increase of 300 meters in altitude decreases the mechanical power produced about 3.5%. The pressure drops of the air pressure, for each kPa in the filters, in the ducts or in the devices reduces the electrical power approximately 2% and pressure loss in the sound breaker, flow or combustion chamber, decreases the mechanical power about 1.2% (Karaali, 2015; Peters et al., 2003).

Elhanan (2006), examined the triple generation (trigeneration) by using waste exhaust gases for absorption cooling in his thermoeconomic analysis in his doctoral thesis. In the combined heat power generation system, seven different situations are considered for different compressor compression ratios (8, 10, 12) and different natural gas fuel prices (0.15, 0.20, 0.25 $/m3) of the ammonia water-cycle absorption refrigeration system. Considering this, he calculated that the system could pay itself back in 7-9 years. In this study, no optimization was made, but thermoeconomic analysis was performed (Elhanan, 2006).

2. Material and Method

A gas turbine cogeneration system's main device is the gas turbine. It can be seen in Figure 1, air is pressured in a compressor, after that it is burned with the fuel in a combustion chamber. High temperature exhaust gases are produced at the outlet of the combustion chamber give some of its energy in the turbine to produce electricity by the generator. After that, most of its heat energy passes to the water in the HRSG. By that way, electrical energy from the generator and steam or hot water from the HRSG is obtained at the same time.

The steam or hot water are used in drying, heating, or for the process heat needs. Also, the steam or the hot water are used sometimes for district heating, electricity production by using steam turbines, absorption cooling, or other needs. By using and adding some components like steam or water injection, heat exchangers, recuperator, steam turbine added to the main machine, or absorption cooling different systems can be obtained. Also, by using different fluids like CO_2 as the working fluid, different cycles can be obtained. In Figure 2 general diagram of the case of using absorption cooling system in the basic cogeneration system is given.

In Figure 2, the system that the air entering into the compressor is cooled by the absorption device by using and driven the heat of the exhaust is shown. In that system, some of the heats are used to cool the air, the other heats are used to produce steam.

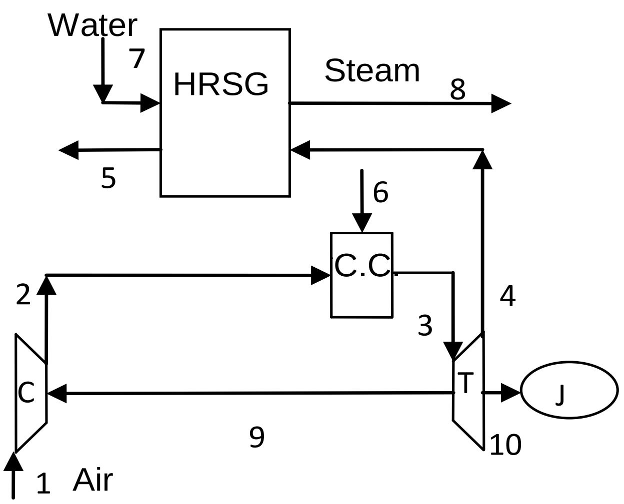

Fig. 1 The diagram of the basic cogeneration system.

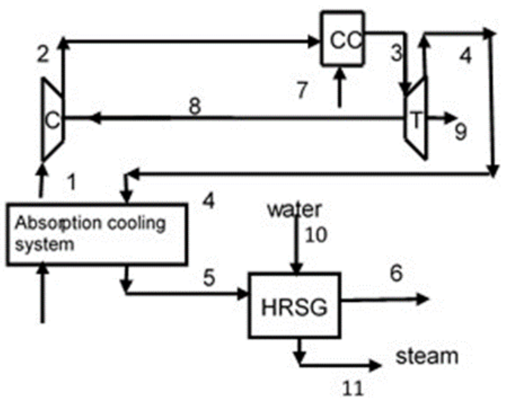

Fig. 2 General diagram of the case of using absorption cooling system in the basic cogeneration system.

Cogeneration plants consist some different components and temperature, chemical composition and pressure changes happen in those components. Also, in the combustion chamber a chemical reaction are obtained. Assumptions made in the calculations and in the analysis of the system in this study are as follows (Moran et al., 2000; Jaluria, 2008). The system works in the continuous regime, the laws of ideal gas mixture are applied to exhaust and air. The fuel is Methane, and taken as ideal gas. Also, combustion is complete, there are no NOx formation. There is no heat loss except from the combustion chamber and are 2% of the fuel upper calorific value. An open system that the properties of the matter is uniformly distributed in each area of the control surface where there is a mass exchange, and the heat and the work exchanges is not change over time which are defined as continuous flow continuous open. For open system and steady state, the first law of thermodynamics,

$$\dot{Q}_{KH} - \dot{W}_{KH} + \sum_g \dot{m}_g \left(h_g + \frac{V_g^2}{2} + gz_g\right) - \sum_c \dot{m}_c \left(h_c + \frac{V_c^2}{2} + gz_c\right) = 0 \tag{1}$$

In steady state, the law of conservation of mass is,

$$\sum \dot{m}_g = \sum \dot{m}_ç \tag{2}$$

In combustion the chemical energies are converted into thermal energy. For this study it is assumed that the combustion reaction takes place ideally and completely. It is also assumed that the natural gas is methane gas to simplify the calculations. the following chemical reaction is taken as a basis,

$$\bar{\lambda} CH_4 + (0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O \rightarrow (1+\bar{\lambda})(X_{N2}N_2 + X_{O2}O_2 + X_{CO2}CO_2 + X_{H2O}H_2O) \tag{3}$$

The minimum mass of air required to complete theoretical combustion is called the stoichiometric amount of air. However, for complete combustion more air than the theoretical amount of air is always used. The excess air coefficient is the ratio of the actual amount of air to the theoretical amount of air (Moran et al., 2000; Bejan et al., 1996). Exergy or availability are theoretical maximum value of the useful work. Which can be obtained if equilibrium with the environment is achieved at the end of a reversible process. It has two components, chemical and physical. The perfect gas mixtures physical exergy can be written in molar terms for mixed substances,

$$e_{phy} = (\bar{h} - \bar{h}_0)_{mix} - T_0.(s - s_0)_{mix} = \sum_i x_i \left[\int_{T_0}^{T} \bar{c}_{p0i}(T)dT - T_0.\left(\int_{T_0}^{T} \frac{\bar{c}_{p0i}(T)}{T} dT - \bar{R}\, ln \frac{P_i}{P_0}\right)\right] \tag{4}$$

The chemical exergies are maximum useful works which can be obtained when a substance in the reference state (T_0, P_0) becomes thermodynamic equilibrium in terms of chemical composition with its surroundings (Moran et al., 2000; Bejan et al., 1996). The chemical exergy of the gas mixtures is,

$$\bar{e}_{chem,mix} = \sum_i x_i.\bar{e}_{chem,i} + \bar{R}.T_0.\sum_i x_i.\,ln\, x_i \tag{5}$$

Thus, the total exergy of a flow or control mass can be written as

$$\overline{E} = \overline{E}_{phy} + \overline{E}_{chem} \qquad (6)$$

Table 1. The energy, the entropy and the mass equations of the devices of the absorption cooling (abc) system (Moran et al., 2000; Bejan et al., 1996; Horlock, 1997).

Component	Mass Equation	Energy Equation	Entropy Equation
Compressor	$\dot{m}_1 = \dot{m}_2$	$\dot{m}_1 h_1 + \dot{W}_C = \dot{m}_2 h_2$	$\dot{m}_1 s_1 - \dot{m}_1 s_2 + \dot{S}_{gen,C} = 0$
Turbine	$\dot{m}_3 = \dot{m}_4$	$\dot{m}_3 h_3 = \dot{W}_T + \dot{W}_C + \dot{m}_4 h_4$	$\dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{gen,T} = 0$
HRSG	$\dot{m}_5 = \dot{m}_6$ $\dot{m}_{10} = \dot{m}_{11}$	$\dot{m}_5 h_5 + \dot{m}_{10} h_{10} = \dot{m}_6 h_6 + \dot{m}_{11} h_{11}$	$\dot{m}_5 s_5 + \dot{m}_{10} s_{10} - \dot{m}_6 s_6 - \dot{m}_{11} s_{11} + \dot{S}_{gen,HRSG} = 0$
Combustion Chamber	$\dot{m}_2 + \dot{m}_7 = \dot{m}_3$	$\dot{m}_2 h_2 + \dot{m}_7 h_7 = \dot{m}_3 h_3 + 0.02\dot{m}_7 LHV$	$\dot{m}_2 s_2 + \dot{m}_7 s_7 - \dot{m}_3 s_3 + \dot{S}_{gen,CC} = 0$
Overall Cycle	$\bar{h}_i = f(T_i)$ $\bar{s}_i = f(T_i, P_i)$ $\dot{m}_{air} h_{air} + \dot{m}_{fuel} LHV_{CH4} - \dot{Q}_{Loss,CC} - \dot{m}_{eg.,out} h_{eg.,out} - \dot{W}_T - \dot{m}_{steam}\left(h_{water,in} - h_{steam,out}\right) = 0$ $\dot{Q}_{Loss,CC} = 0.02\dot{m}_{fuel} LHV_{CH4}$		

Table 2. Exergy equations, and exergy efficiency equations of the devices of the absorption cooling (abc) cycle (Moran et al., 2000; Bejan et al., 1996; Horlock, 1997).

Component	Exergy Equation	Exergy Efficiency
Compressor	$\dot{E}_{D,C} = \dot{E}_1 + \dot{W}_C - \dot{E}_2$	$\eta_{ex,C} = \frac{\dot{E}_{out,C} - \dot{E}_{in,C}}{\dot{W}_C}$
Turbine	$\dot{E}_{D,T} = \dot{E}_3 - \dot{E}_4 - \dot{W}_C - \dot{W}_T$	$\eta_{ex,T} = \frac{\dot{W}_{net,T} + \dot{W}_C}{\dot{E}_{in,T} - \dot{E}_{out,T}}$
HRSG	$\dot{E}_{D,HRSG} = \dot{E}_5 - \dot{E}_6 + \dot{E}_{10} - \dot{E}_{11}$	$\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$
Combustion Chamber	$\dot{E}_{D,CC} = \dot{E}_2 + \dot{E}_7 - \dot{E}_3$	$\eta_{ex,CC} = \frac{\dot{E}_{out,CC}}{\dot{E}_{in,CC} + \dot{E}_{fuel}}$
Overall cycle	Exergy efficiency	$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch}$ $\dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0))$ $\dot{E}_{ch} = \frac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k \ln x_k\right\}$ $\eta_{ex} = \frac{\dot{W}_{net,T} + (\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG})}{\dot{E}_{fuel}}$

Exergy equation for open systems (input and output mass quantities are equal to each other) is,

$$\sum_i \dot{m}_i h_i - \sum_i T_0 S_i - \sum_j \dot{m}_j h_j + \sum_j T_0 S_j + \sum \dot{Q}_k - \sum \dot{Q}_k \frac{T_0}{T_k} - \dot{W} = \dot{E}_{loss} \qquad (7)$$

In table 1, energy, entropy and mass equations of the devices of the absorption cooling (abc) system are given.

In table 2, the exergy, and the exergy efficiency equations of the devices of the absorption cooling (abc) system are given.

3. Results and Discussion

In Figure 3, variation of exergy efficiency and electrical efficiency of the absorption cooling (abc) and basic cogeneration system with compressing ratio are given. As can be seen that, performance of the absorption cooling (abc) system is better than the basic one. The exergy efficiency of the absorption cooling (abc) and the basic systems are almost the same. However, the electrical efficiency of the absorption cooling (abc) is higher than the basic cogeneration system about 4-5%. Since the heat energy of the exhaust's gases are used for cooling the inlet air of the cycle the exergy efficiency of the absorption cooling (abc) system is slightly less (about 0.2%) than the basic cycle.

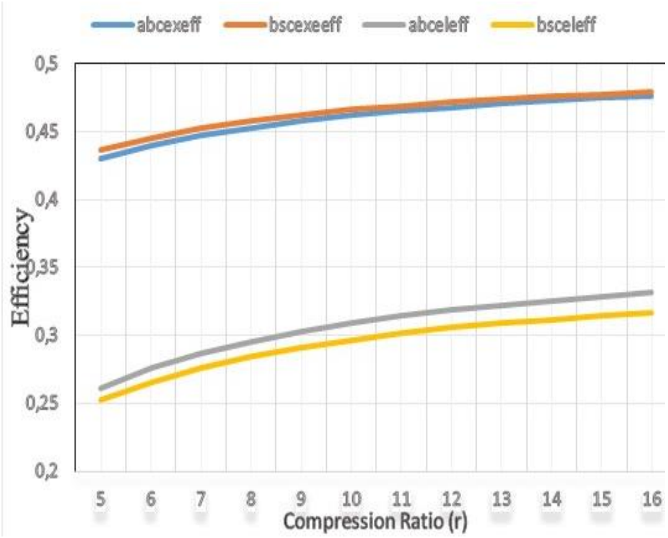

Fig. 3 Variation of exergy efficiency and electrical efficiency of the absorption cooling (abc) and basic cogeneration system with compressing ratio.

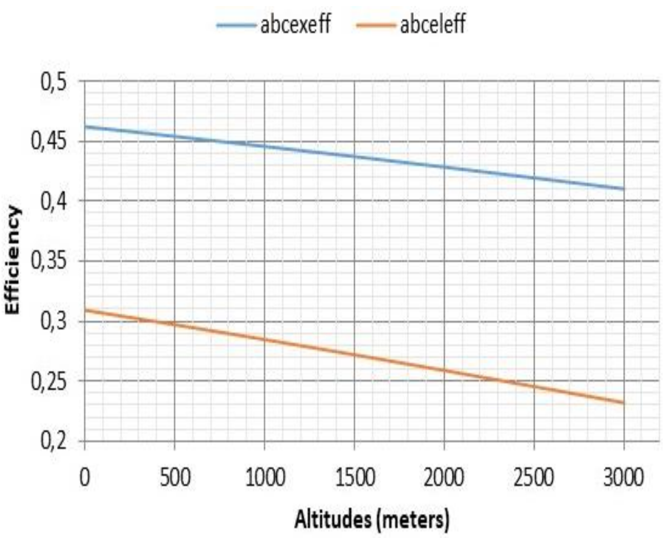

Fig. 4 Variation of exergy efficiency and electrical efficiency of the absorption cooling (abc) system with altitudes.

In Figure 4, variation of exergy efficiency and electrical efficiency of the absorption cooling (abc) system with altitudes are given. As it is showed those, exergy efficiency and electrical efficiency are decreasing with increasing altitudes. From zero meters to 3000 meters the decreases in exergy efficiency are from

0.462 to 0.4105 which means about 11% decrease. Also, from zero meters to 3000 meters the decreases in electrical efficiency are from 0.3092 to 0.2314 which means about 25% decrease. The reasons of these decreases are the altitude increasing, decreases the density of the air so the compressor work increases.

In Figure 5, variation of the efficiencies of the compressor, the turbine, the combustion chamber, and the heat recovery steam generator of the absorption cooling (abc) system with altitudes are given. From zero meters to 3000 meters the decreases in efficiency of the compressor are from 0.9216 to 0.8218 which means about 11% decrease. The alteration in the efficiency of the turbine can be neglected. But, from zero meters to 3000 meters the decreases in the efficiency of the combustion chamber are from 0.9216 to 0.8218 which means about 1% decrease, and the decreases in the efficiency of the HRSG are from 0.6706 to 0.6463 which means about 3.6% decrease.

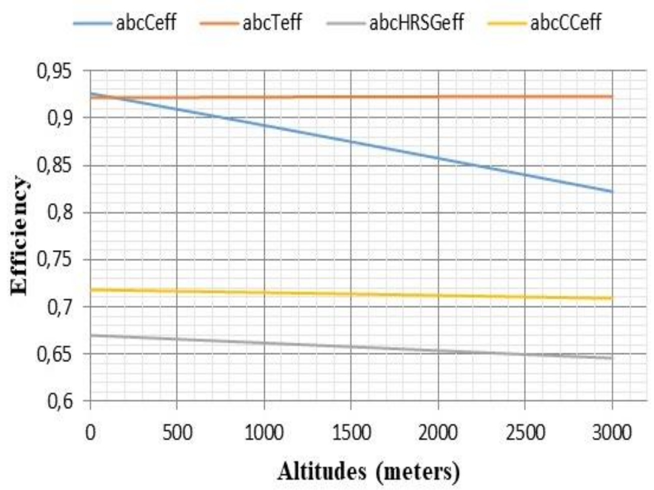

Fig. 5 Variation of efficiencies of the compressor, the turbine, the combustion chamber, and the heat recovery steam generator of the absorption cooling (abc) system with altitudes.

In Figure 6, variation of the exergy efficiency and electrical efficiency of the absorption cooling (abc) system with compression ratio are given. As it is showed this, the exergy efficiency also the electrical efficiency is increasing with the increasing compression ratio. The increasing in the exergy efficiency is from 0.4304 to 0.4766 which means about 11% increase, and the increasing in the electrical efficiency is from 0.2614 to 0.3313 which means about 26% increase.

In Figure 7, variation of the efficiencies of the compressor, the combustion chamber, the turbine and the heat recovery steam generator of the absorption cooling (abc) system with compression ratio are given. As can be seen that, increasing from 5 to 16 the compression ratio increases the efficiency of the compressor from 0.9113 to 0.9348 which means about 2.6% increase. The alteration in the efficiency of the turbine is from 0.9775 to 0.8881 which means about 9% decrease. The reason is more work are spending for compressing which is decreasing the turbine obtained work. However, increasing from 5 to 16 the compression ratio increases the efficiency of the combustion chamber from 0.6535 to 0.7559 which means about 16% increase, and the increases in the efficiency of the HRSG are from 0.6535 to 0.6782 which means about 3.8% increase.

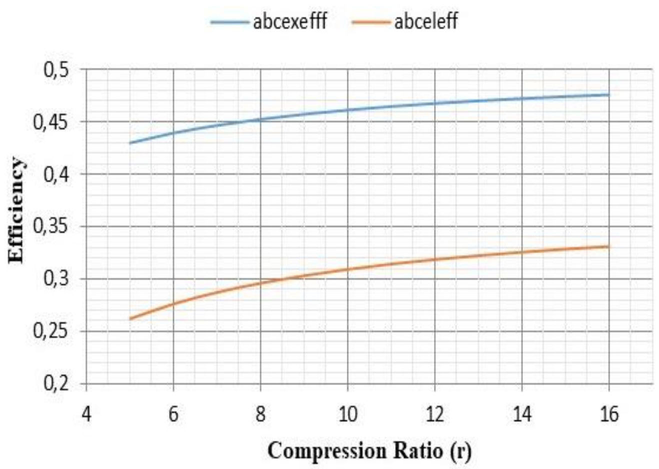

Fig. 6 Variation of exergy efficiency and electrical efficiency of the absorption cooling (abc) system with compression ratio.

In Figure 8, variation of the exergy efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio are given. It is seen that, increasing the compressing ratio from 5 to 16 makes the exergy efficiency an increase for the absorption cooling (abc) cycle about 9%. However, increasing the value of the excess air rates from 1.3 to 3.5 give a maximum exergy efficiency of 0.4804 at excess air rate about 2.1. That point is the optimum excess air rate that the system must run.

In Figure 9, variation of the compressor efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio are given. As can be seen that, the increases in the excess air rates from 1.3 to 3.5 does not make any effect on the compressor efficiency. However, the increases in the compression ratio from 5 to 16 increases the absorption cooling (abc) system's compressor efficiency about 2%.

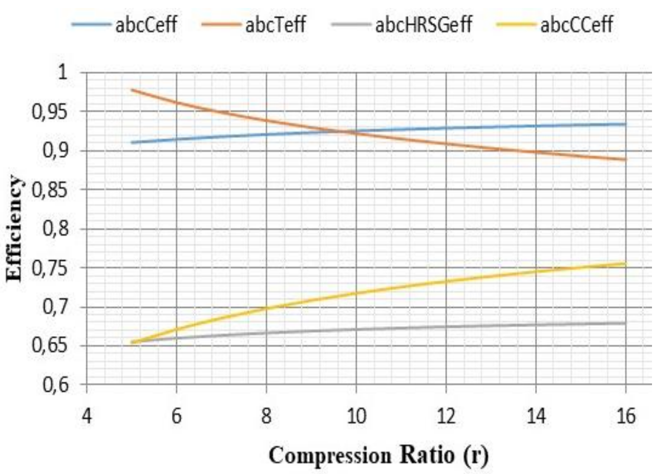

Fig. 7 Variation of efficiencies of the compressor, the combustion chamber, the turbine and the heat recovery steam generator of the absorption cooling (abc) system with compression ratio.

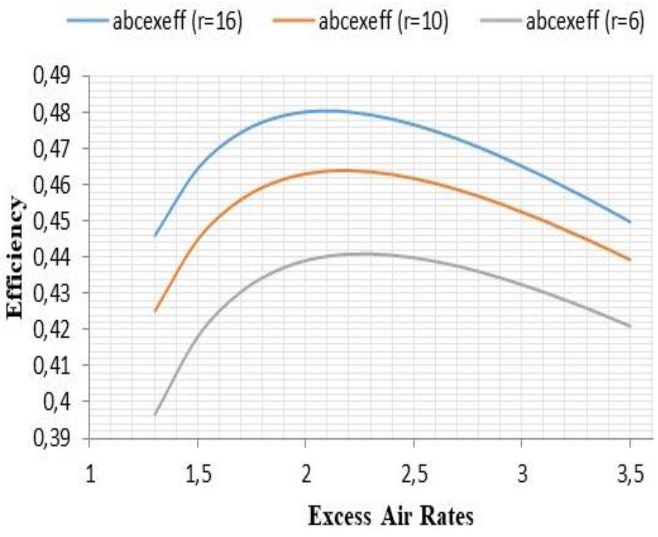

Fig. 8 Variation of the exergy efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio.

In Figure 10, variation of the combustion chamber efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio are given. It is shown that, combustion chamber efficiency is increasing (about 5-8%) with the decreasing excess air rates and is increasing (about 7-13%) with compression ratio.

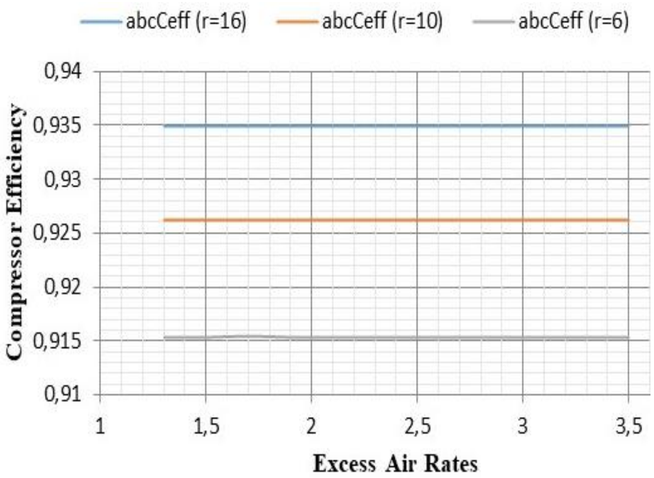

Fig. 9 Variation of the compressor efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio.

In Figure 11, variation of the turbine efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio are given. It is shown that, the turbine efficiency is increasing (about 9-22%) with the increasing excess air rates and is increasing (about 0-12%) with compression ratio.

In Figure 12, variation of the heat recovery steam generator efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio are given. It is shown that, the HRSG efficiency is increasing (about 39%) with the increasing excess air rates and is increasing (about 1%) with compression ratio.

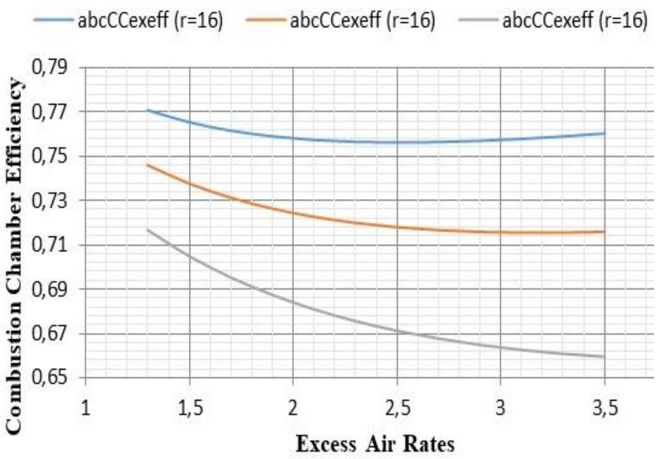

Fig. 10 Variation of the combustion chamber efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio.

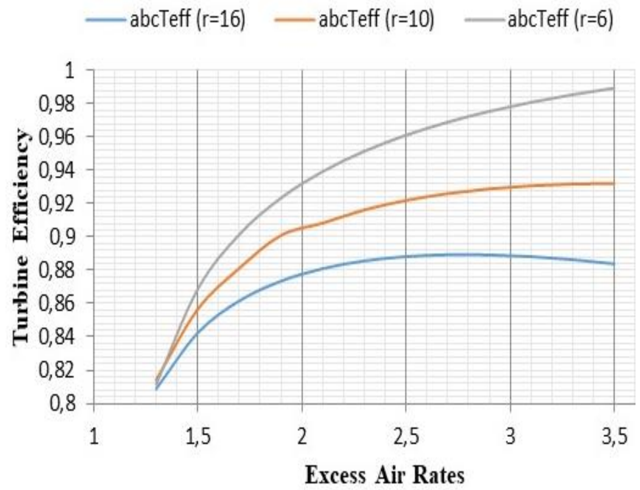

Fig. 11 Variation of the turbine efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio.

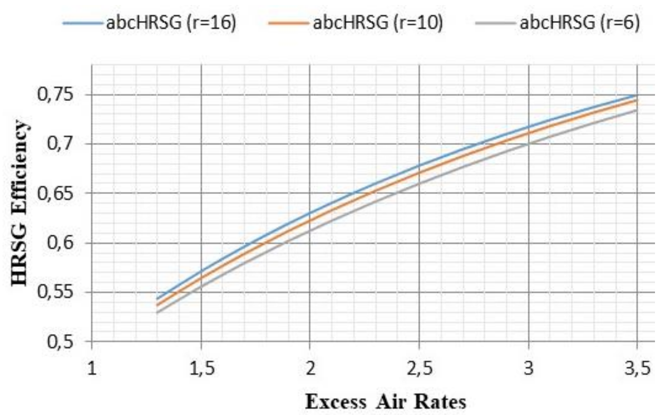

Fig. 12 Variation of the heat recovery steam generator efficiency of the absorption cooling (abc) system with excess air rates for different compression ratio.

4. Conclusions and Recommendations

The results showed that using absorption cooling (abc) system in a basic cycle made better than the basic one in electrical efficiency. However, because of the absorption cooling (abc) system the exergy efficiency is slightly less than the basic one, the absorption cooling (abc) cogeneration system can be used when the steam demand decreases or electrical demand increases for production more electricity. For every device of the absorption cooling (abc) cogeneration system's energy and exergy analyses are done. The results of the calculations are given in results section. The performance of the absorption cooling (abc) cogeneration system is depending on altitudes, compression ratio, excess air rates and atmospheric temperature, mainly. Less atmospheric temperature, less altitudes and higher compression ratio can increase the exergy efficiency and electrical efficiency. However, excess air rates and compression ratio are very effective on the efficiencies and for excess air rates, 2.1 excess air rate value gives the optimum so that these two factors should be optimized. For all optimum working conditions, a thermodynamic and thermoeconomic optimization should be done. Further studies in these optimizations can found in the literature (Karaali and Ozturk, 2015; Jaluria, 2008; Tozlu, 2021; Karaali and Ozturk, 2017).

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