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# ÇEVRE ŞARTLARININ GAZ TURBİNLİ KOJENERASYON ÇEVRİMLERİNİN PERFORMANSI ÜZERİNE ETKİLERİ

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**Özet:** Bu çalışmada ekserji analizi metodu kullanılarak çevre şartlarının kojenerasyon çevrimlerinin performansı üzerine etkisi analiz edilmiştir. Bu kojenerasyon çevrimleri basit, hava ön ısıtmalı, hava-yakıt ön ısıtmalı ve giriş havası soğutmalı çevrimlerdir. Ayrıca bu çevrimler değişik hava yakıt kütle oranları için termodinamik açıdan incelenmiş ve performansları birbirleri ile karşılaştırılmıştır. Kompresor giriş havasının 25 $^0$C'den 0 $^0$C'ye soğutulması elektrik güç üretimini 10-15 % civarında artırmaktadır. Ancak ısı gücü giriş havasının soğutulmasından dolayı 10 % civarında düşmektedir. Kompresör giriş havasına su püskürtülerek yaz sezonunda elektrik gücü kolayca 2-7 % civarında artırılabilir. Kompresör giriş havasının basıncı 101.3 kPa'dan 70.18 kPa'ya düşürüldüğünde çevrimlerin ekserji verimleri 11-13 % civarında düşmektedir. Ancak bu basınç düşüşü elektrik gücü üretiminde 25 % civarında düşüşe yol açmaktadır. En yüksek ekserji verimi hava-yakıt ön ısıtmalı çevrimde elde edilmiştir. Dört çevrim arasında en iyi ısı oranı ve ısı ekserjisi basit çevrimde elde edilmektedir. Bu çalışmada tüm çevre şartlarının bu çevrimler üzerindeki etkisi aynı anda göz önüne alınarak analiz edilmiş ve birbiri ile karşılaştırılmıştır.
**Anahtar Kelimler**: Kojenerasyon, Çevre, Verim.

# EFFECTS OF AMBIENT CONDITIONS ON PERFORMANCE OF GAS TURBINE COGENERATION CYCLES

**Abstract:** In this paper, effects of the ambient conditions on performance of cogeneration cycles are analyzed by using exergy analysis method. These cogeneration cycles are the simple cycle, the air preheating cycle, the air-fuel preheating cycle and the inlet air cooling cycle. Also, these cycles are evaluated thermodynamically for different air and fuel mass ratio and the performance of these cycles are compared. The electrical power generation increases about 10-15 % by decreasing the inlet air temperature of the compressor from 25 $^0$C to 0 $^0$C. However, the thermal power decreases about 10 % with decreasing inlet air temperature. During the summer, the electric power can be easily increased 2-7 % by injecting water into the inlet air of the compressor. The exergetic efficiency of cycles is decreased 11-13 %, by decreasing the compressor inlet air pressure from 101.3 kPa to 70.18 kPa. But this change decreases the electric power generation about 25 %. The highest exergy efficiency is obtained for the air-fuel preheated cycle. The simple cycle is the best among the four cycles to obtain high heat rate and heat exergy. The effects of all ambient conditions on these cycles are considered simultaneously, analyzed and compared with each other in this study.
**Keywords:** Cogeneration, Environment, Efficiency.

**Nomenclature**

| | |
|---|---|
| COP | coefficient of performance |
| e | specific exergy (kJ/kg) |
| $\dot{E}$ | exergy flow rate (kW) |
| h | specific enthalpy (kJ/kg), (kJ/kMol) |
| J | generator |
| LHV | lower heating value (kJ/kg) |
| $\dot{m}$ | mass flow rate (kg/s) |
| M | molecular weight (kg/kmol) |
| P | pressure (kPa) |
| $\dot{Q}$ | heat flow rate (kW) |
| R | specific gas constant (kJ/kg K) |
| s | specific entropy (kJ/kg K) |
| T | temperature (K) |
| $\dot{W}$ | power (kW) |
| x | mole fraction (kmol/kmol) |

**Greek letters**

| | |
|---|---|
| η | efficiency |
| $\bar{\lambda}$ | constant |

**Subscripts**

| | |
|---|---|
| C | compressor |
| CC | combustion chamber |
| ch | chemical |
| D | destruction |
| *en* | energy |
| ex | exergy |
| exh | exhaust |
| f | fuel |
| HRSG | heat recovery steam generator |
| i | i. mixture component |
| i, ch | *i.* mixture component, chemical |

| | |
|---|---|
| L | loss |
| ph | physical |
| R | recuperator |
| is | isentropic |
| st | steam |
| T | turbine |
| 0 | environment conditions |

## INTRODUCTION

In a foreseeable future, there is a growing need for power generation with gas turbine and cogeneration systems. Also there is a pressing need to reduce emission because of global warming. Cogeneration systems have many advantages over conventional separate systems for heating and/or cooling, and electric generation such as high efficiency, safe and reliable operation, lower weight per unit power, less environmental emissions, compact size and fast starting time, more economic and dual fuel capability. Improving the performance of gas turbine cogeneration cycles is going to be a major objective in the coming years. In gas turbine systems natural gas or mixed fuels such as alcohols, biomass, refinery residues, naphtha, etc., can be used as fuel. This fuel flexibility is an important advantage for gas turbine systems (ASHRAE, 2000; Boyce, 2002). Main markets for gas turbine systems are industry, district heating systems and others. The criteria for selection of gas turbine cogeneration systems are heat to power ratio, the efficiency and the grade of heat (Horlock, 1997). The main factors that affect high efficiency are increased gas turbine inlet temperature, intercooling, reduced auxiliary power consumption, advanced gas turbine cooling, hydrogen cooled generators, multiple pressure cycle with reheat, better HRSG design, fuel preheating, low compressor inlet air temperature, high compressor inlet air pressure and high compressor inlet air humidity (Jaluria, 2008; Moran and Tsatsaronis, 2000; Najjar, 2001).

Gas turbines are air mass flow machines and their power outputs are very sensitive to the environmental factors. This means location has significant effect on the efficiency and the power output of the cogeneration systems (Pilavachi, 2000). Karaali in his study, conclude that the environmental factors such as ambient temperature, ambient pressure and air humidity are effective parameters on the efficiencies of the cogeneration cycles (Karaali, 2010). The variations in the compressor inlet air temperatures cause significant changes in the electricity and thermal energy power (Kim et al., 2000). Boyce has also indicated in his study that decreasing the compressor inlet air temperature increases the electrical power; for example when the temperature drops from 25 $^0$C to 0 $^0$C the electrical power increases about 14 % but the thermal power decreases about 10 % (Boyce, 2002). Al-Fahed et al., in their articles have shown that the changes in the compressor inlet air temperature cause changes on the performance of the simple cycle cogeneration system. Large temperature differences between day and night, like the Middle East region, the cost changes need to be taken into account in this situation (Al-Fahed et al., 2009). Al-Fahed et al., have found that 1 $^0$C increase in the compressor inlet air temperature leads to a decrease in electricity production around 0.7 %. In the literature, 15 $^0$C cooling of the compressor inlet air by absorption method increases the electric efficiency around 6 % (Al-Fahed et al., 2009). Santo and Gallo have presented that since decreasing the temperature of the compressor inlet air increases the density of the air for the same mass, less energy is spent (Santo and Gallo, 2000). Kehlhofet, et al., have shown that the effects of the compressor inlet air temperature on relative work and cooling inlet air increases electrical power (Kehlhofet, et al., 1999). Ashraf in his study on this issue have found similar results of the effects of evaporative cooling on power (Ashraf, 2001). Al-Fahed et al., obtained similar results with the literature that the simple cogeneration system with evaporative cooling increases the efficiency about 2.5 % (Al-Fahed et al., 2009).

Wang and Chiou have shown in their article that cooling the compressor inlet air from 315 K, to 270 K temperature with a vapor-compression refrigeration chiller, the gas turbine electrical power efficiency increases from 30.45 % to 31.35 % (Wang and Chiou, 2002). If the ambient temperature drops in natural way to 270 K, the electrical efficiency increases from 30.45 % to around 41 %. For steam injection cycle these rates increase from 48 % to about 50 %.

Amel and Cadavid have found in their study that in low ambient temperature conditions the power output is better, but the heat rate is worse than the high ambient temperature conditions. They also found that high relative humidity improves the power of the system (Amel and Cadavid, 2002). Mohapatra and Sanjay have found by using the first law of thermodynamics that decreasing the temperature of the inlet air by 30 $^0$C by using evaporative cooling increases the gas turbine power plant efficiency about 2 % (Mohapatra and Sanjay, 2014). They also found that vapor compression inlet air cooling and evaporative air cooling improve the energy efficiency of the power plant by 4.18 and 4.6 %, respectively. Caresana et al., have studied the effect of the ambient temperature on the global performance and the behavior of the components of the micro turbo gas cogeneration systems (Caresana et al., 2014). They found in their experimental study by using the first law of thermodynamics that decreasing the inlet air temperature increases the electrical efficiency but decreases the thermal efficiency. They also found that 1 $^0$C increasing the inlet air temperature decreases the electrical power by 1.22 %, and the electrical efficiency by 0.51 % while the thermal efficiency increases by 0.7 %. Increasing the specific humidity decreases the molecular weight of air and that affects the temperature. Also, increasing specific humidity increases specific heat and gas constant value at constant pressure. It is possible to increase the electric power with this method at a low cost and it can be easily applied to all types of gas turbines (Ashraf, 2001).

In the literature there are many studies to analyze the effect of ambient conditions on cogeneration cycles efficiency and performance. These studies generally are based on the first law efficiency of thermodynamics. But second law analysis must be included for analyzing the effect of the ambient condition on the performance of cogeneration cycles. Also, possible competitive cycles and parameters must be included in this evaluation and they should be compared with each other.

These points are the motivation of this presented study. In this study, the effect of the ambient conditions and the evaporative cooling of the compressor inlet air on the performance of the cogeneration cycles are analyzed by using exergy analysis method. The four cogeneration cycles which are analyzed in this study are called simple, air preheating, air-fuel preheating and inlet air cooling cycles. Also, thermodynamic evaluations of the cycles, for different air and fuel mass ratio are studied and the performance of the cycles is compared.

**DESCRIPTION OF THE CYCLES**

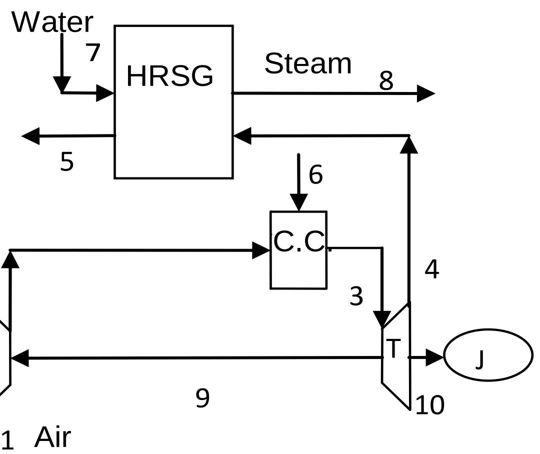


**Figure 1.** Simple cycle.

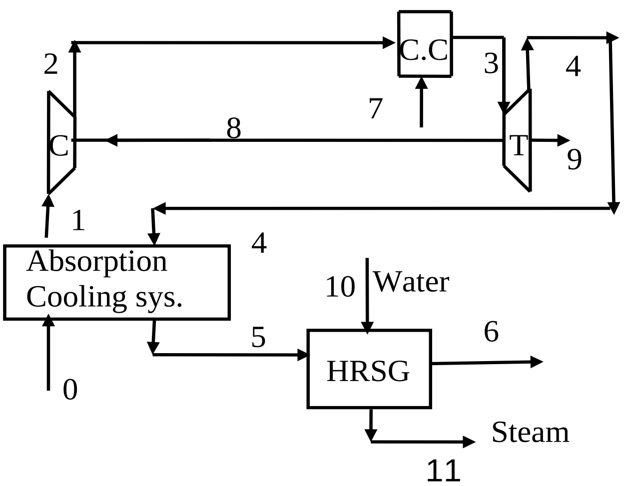


**Figure 2**. Inlet air cooling cycle.

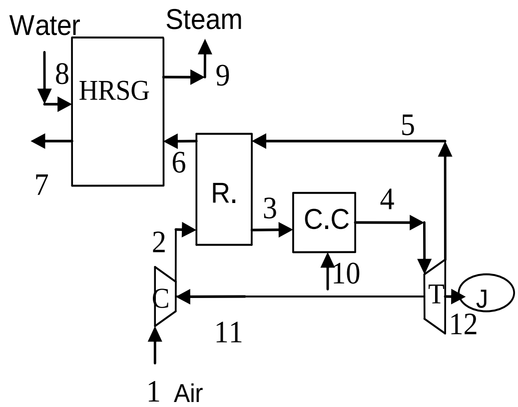


**Figure 3.** Air preheated cycle.

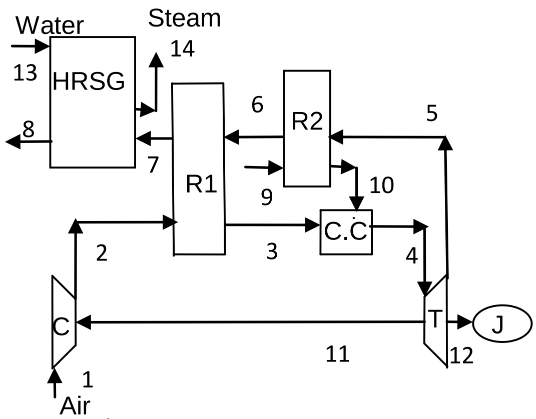


**Figure 4.** Air-fuel preheated cycle.

In the inlet air cooling cycle and in the simple cycle compressed air (Fig. 1-2) enter the combustion chamber. In the recuperated cycles (Fig. 3-4) compressed air heated by hot exhaust gases in the recuperator and then enters the combustion chamber. The hot gases which exit from the combustion chamber are expanded at the gas turbine. And then from the gas turbine hot gases are the source of the heat energy of the recuperators and the heat recovery steam generator (Khaliq and Kaushik, 2004; Sue and Chuang, 2004). These cycles are fueled with natural gas, but to simplify the modeling approach the following assumptions are introduced; the pressure losses in the combustion chamber, the recuperator and the HRSG are known as 0,05 for each component. The environmental conditions are fixed and defined as $T_0$ = 298.15 K and $P_0$ = 101.3 kPa. The main capacity of the air compressors are $m_1$ = 91.4 kg/s, the HRSG $m_s$ = 14 kg/s saturated steam at 2000 kPa. The gas turbine electricity power is 30 MW (electric power is equal to the mechanic power obtained from the gas turbine), and the combustion chamber's fuel is $m_f$ = 1.64 kg/s methane. The working fluid assumed as ideal gas, cogeneration systems operates at steady state, natural gas is taken as methane and modeled as an ideal gas. The combustion is complete and $N_2$ is inert, and the heat loss for the combustion chamber is 2 % of the fuel's LHV, and all other components operate without heat loss (Bejan et al., 1996; Karaali and Ozturk, 2015). Kinetic and potential energy effects are ignored. For avoiding corrosive sulfuric acid formation in the exhaust, the outlet temperature of the HRSG is taken as 400 K. The compressor and the turbine operate adiabatically.

## ANALYSES OF THE CYCLES

The thermodynamic analysis of each of the components introduced at the previous section will be done and mathematical modeling will be explained in this section. The compressor outlet, the combustion chamber outlet, the recuperator exhaust side outlet, the heat recovery steam generator inlet exhaust side and the gas turbine isentropic temperatures are calculated by inserting specific entropy expressions for $N_2$, $O_2$, $CO_2$ and $H_2O$ from (Bejan et al., 1996). The entropies of the streams are calculated from the same reference, as well. The thermodynamic model and the step-by-step calculation procedure for the air preheated cycle (CGAM cycle) are explained as follows. Specific enthalpies and specific entropies are calculated for each stream from the equations of the tables in (Bejan et al., 1996; Karaali and Ozturk, 2015).

**Table 1.** The mass, the energy and the entropy equations of the components of the air preheated cycle.

| Component | Mass Equation | Energy Equation | Entropy Equation |
|---|---|---|---|
| Compressor | $\dot{m}_1 = \dot{m}_2$ | $\dot{m}_1 h_1 + \dot{W}_C = \dot{m}_2 h_2$ | $\dot{m}_1 s_1 - \dot{m}_1 s_2 + \dot{S}_{gen,C} = 0$ |
| Combustion Chamber | $\dot{m}_3 + \dot{m}_{10} = \dot{m}_4$ | $\dot{m}_3 h_3 + \dot{m}_{10} h_{10} = \dot{m}_4 h_4 + 0.02\dot{m}_{10} LHV$ | $\dot{m}_3 s_3 + \dot{m}_{10} s_{10} - \dot{m}_4 s_4 + \dot{S}_{gen,CC} = 0$ |
| Recuperator | $\dot{m}_2 = \dot{m}_3$ <br> $\dot{m}_5 = \dot{m}_6$ | $\dot{m}_2 h_2 + \dot{m}_5 h_5 = \dot{m}_3 h_3 + \dot{m}_6 h_6$ | $\dot{m}_2 s_2 + \dot{m}_5 s_5 - \dot{m}_3 s_3 - \dot{m}_6 s_6 + \dot{S}_{gen,R} = 0$ |
| Turbine | $\dot{m}_4 = \dot{m}_5$ | $\dot{m}_4 h_4 = \dot{W}_T + \dot{W}_C + \dot{m}_5 h_5$ | $\dot{m}_4 s_4 - \dot{m}_5 s_5 + \dot{S}_{gen,T} = 0$ |
| HRSG | $\dot{m}_6 = \dot{m}_7$ <br> $\dot{m}_8 = \dot{m}_9$ | $\dot{m}_6 h_6 + \dot{m}_8 h_8 = \dot{m}_7 h_7 + \dot{m}_9 h_9$ | $\dot{m}_6 s_6 + \dot{m}_8 s_8 - \dot{m}_7 s_7 - \dot{m}_9 s_9 + \dot{S}_{gen,HRSG} = 0$ |

**Table 2.** The exergy and the exergy efficiency equations of the components of the air preheated cycle.

| Component | Exergy Equation | Exergy Efficiency |
|---|---|---|
| Compressor | $\dot{E}_{D,C} = \dot{E}_1 + \dot{W}_C - \dot{E}_2$ | $\eta_{ex,C} = \frac{\dot{E}_{out,C} - \dot{E}_{in,C}}{\dot{W}_C}$ |
| Combustion Chamber | $\dot{E}_{D,CC} = \dot{E}_3 + \dot{E}_{10} - \dot{E}_4$ | $\eta_{ex,CC} = \frac{\dot{E}_{out,CC}}{\dot{E}_{in,CC} + \dot{E}_{fuel}}$ |
| Recuperator | $\dot{E}_{D,R} = \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6$ | $\eta_{ex,R} = \frac{\dot{E}_{out,air,R} - \dot{E}_{in,air,R}}{\dot{E}_{out,exhaust,R} - \dot{E}_{in,exhaust,R}}$ |
| Turbine | $\dot{E}_{D,T} = \dot{E}_4 - \dot{E}_5 - \dot{W}_C - \dot{W}_T$ | $\eta_{ex,T} = \frac{\dot{W}_{net,T} + \dot{W}_C}{\dot{E}_{in,T} - \dot{E}_{out,T}}$ |
| HRSG | $\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9$ | $\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$ |

In Table 1 and in Table 2 the mass, energy, entropy, exergy, and the exergy efficiency equations of the components of the air preheated cycle are given.

$$\bar{h}_i = f(T_i) \tag{1}$$

$$\bar{s}_i = f(T_i, P_i) \tag{2}$$

$$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch} \tag{3}$$

$$\dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0)) \tag{4}$$

$$\dot{E}_{ch} = \frac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k \, ln x_k\right\} \tag{5}$$

The chemical reaction in the combustion chamber can be written as follows (Bejan et al., 1996).

$$\bar{\lambda}CH_4 + [0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O] \rightarrow (1+\bar{\lambda})[X_{N2}N_2 + X_{O2}O_2 + X_{CO2}CO_2 + X_{H2O}H_2O] \tag{6}$$

$$X_{N2} = \frac{0.7748}{1+\bar{\lambda}} \tag{7}$$

$$X_{CO2} = \frac{(0.0003+\bar{\lambda})}{1+\bar{\lambda}} \tag{8}$$

$$X_{H2O} = \frac{(0.019+2\bar{\lambda})}{1+\bar{\lambda}} \tag{9}$$

$$X_{O2} = \frac{(0.2059-2\bar{\lambda})}{1+\bar{\lambda}} \tag{10}$$

Excess air rate is

$$Ear = \frac{\dot{m}_{airgiven}}{\dot{m}_{airtheoretical}} \tag{11}$$

Heat loss of the combustion chamber can be written as,

$$\dot{Q}_{Loss,CC} = 0.02\dot{m}_{fuel}LHV_{CH4} \tag{12}$$

- **Absorption cycle**

For the absorption cycle COP is taken as 0,70 for LiBr-water

- ***Overall Balance Equations for the Cycles***

The overall energy balance of the systems is,

$$\dot{m}_{air}h_{air} + \dot{m}_{fuel}LHV_{CH4} - \dot{Q}_{Loss,CC} - \dot{m}_{eg.,out}h_{eg.,out} - \dot{W}_T - \dot{m}_{steam}\left(h_{water,in} - h_{steam,out}\right) = 0 \tag{13}$$

Energy and exergy efficiencies of the cycles are (Lazzaretto and Tsatsaronis, 2006; Karaali and Ozturk, 2015).

$$\eta_{en} = \frac{\dot{W}_{net,T}+\dot{Q}_U}{\dot{Q}_{fuel}} \tag{14}$$

$$\eta_{ex} = \frac{\dot{W}_{net,T}+(\dot{E}_{steam,HRSG}-\dot{E}_{water,HRSG})}{\dot{E}_{fuel}} \tag{15}$$

## RESULTS AND DISCUSSION

### Effect of the Compressor Inlet Air Temperatures on the Working Conditions of the Cycles

The effect of the compressor inlet air temperature on the performance of the cycles is summarized in the Figs 5 to 11. In Figure 5, the variations of the electric power with excess air rate for different values of the compressor inlet air temperatures are shown. As it can be seen in this figure, increasing excess air rate increases electric power for the air preheated and the air-fuel preheated cycles.

The maximum electric power for the simple cycle is obtained in the interval of 2.5-3 of the excess air rate. It is also observed in this figure that increasing ambient temperature causes a decrease in the electrical power generation for all cycles. The air-fuel preheated cycle is better than the other cycles as far as the electric power generation in cold climates or winters. Decreasing the compressor inlet air temperature decreases the compressor work and thus it increases the electric power generation of the cycle.

In Figure 6 variations of heat power with excess air rate for different values of the compressor inlet air temperature is given. It can be seen in this figure that, decreasing the compressor inlet air temperature decreases the thermal power. Increasing the excess air rate decreases the thermal power of the systems for all cycles faster than the compressor inlet air temperature. The simple cycle seems to be the best among the other cycles in generating heat power in cold climates or winters.

In Figure 7 it can be seen that decreasing the compressor inlet air temperature, increases the rate of the electric to heat energy. The reason for this result is that the low inlet air temperature of the compressors increases the electrical power and reduces the thermal output. Increasing the excess air rate increases the electric power generation so it increases the electric to heat energy ratio. This increase in the recuperated cycles is greater than the others.

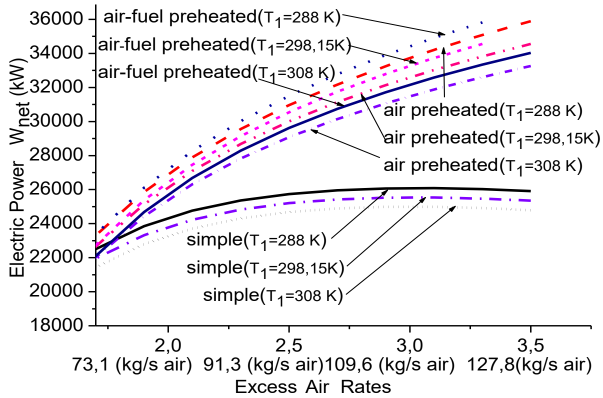


**Figure 5.** Variations of electric power with excess air rate for different values of the compressor inlet air temperatures.

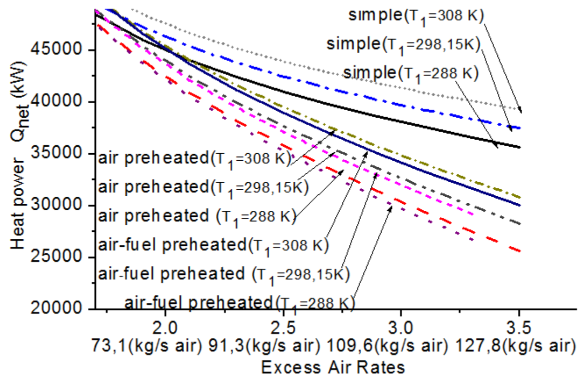


**Figure 6.** Variations of heat power with excess air rate for different values of the compressor inlet air temperature.

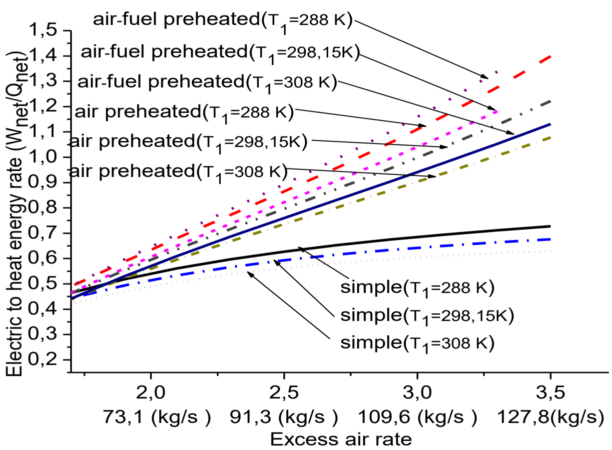


**Figure 7.** Variations of electric to heat energy rate with excess air rate for different ambient temperatures.

In Figure 8 the variations of the electric and the heat exergy power with the ambient temperature for fixed excess air rate at 2.5 is given. Decreasing ambient temperature increases the electric power generation but decreases the heat exergy. The electric power decreases about 23 % and the heat exergy increases about 20 % for the increasing ambient temperature for all cycles. The air-fuel preheated cycle provides better electricity production and the simple cycle is better for the heat exergy power production among the other cycles.

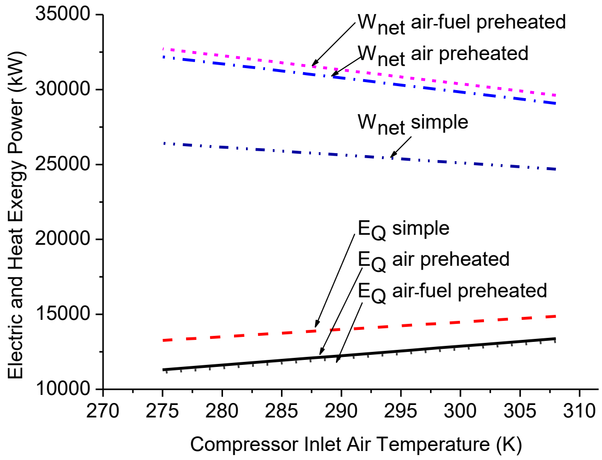


**Figure 8.** Variations of electric and heat exergy power with ambient temperatures (for fixed excess air rate at 2.5).

In Figure 9 the variations of the exergetic efficiency with respect to the ambient temperature for the excess air rates fixed at 2.5 are given for the simple, the air preheated and the air-fuel preheated cogeneration cycles. The exergy efficiency of the systems decreases with the increasing inlet air temperature. The reason for this is that more power consumption is needed for compressing the air. For the simple cycle, the decrease in the exergy efficiency is less than the air preheated and the air-fuel preheated cycles. The thermal power of the simple cycle is better than the air preheated and the air-fuel preheated cycles. The reason for this is that the simple cycle has not recuperation process. However, recuperation has a significant increase in the electrical power of the air preheated and the air-fuel preheated cycles.

In Figure 10, it can be seen that increasing compressor inlet air temperature increases thermal exergy while increasing excess air rates decreases thermal exergy. This decrease is more significant for the air preheated and the air-fuel preheated cycles than the simple cycle.

In Figure 11, the variations of the exergetic efficiency with the excess air rate for different ambient temperatures are given. It is seen in this figure that decreasing the ambient temperature and increasing the excess air rates increases the exergy efficiency of the air preheated and the air-fuel preheated cycles. For the simple cycle the exergy efficiency reaching the maximum value for the excess air rates around 2.25 and then decreases. For the simple cycle, the exergy efficiency is 15 % less than the air-fuel preheated cycle for ambient temperature at 288 K and for the excess air rate at 3.5.

It should be indicated that increasing the excess air rate decreases the outlet temperature of the combustion chamber and this leads to decrease of the turbine outlet temperature. The results are compared with the studies cited in the Introduction and found in good agreement.

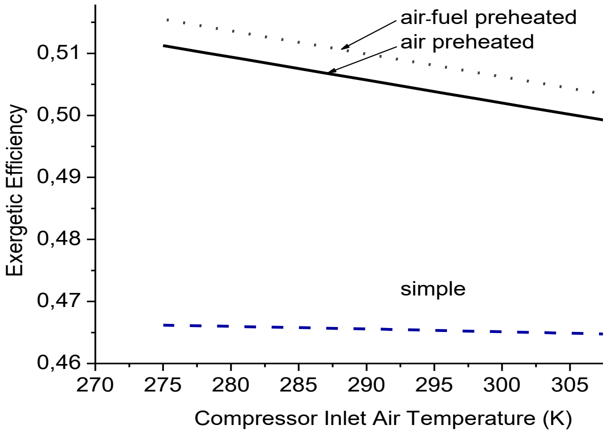


**Figure 9.** Variations of exergetic efficiency with ambient temperature (for fixed excess air rates at 2.5).

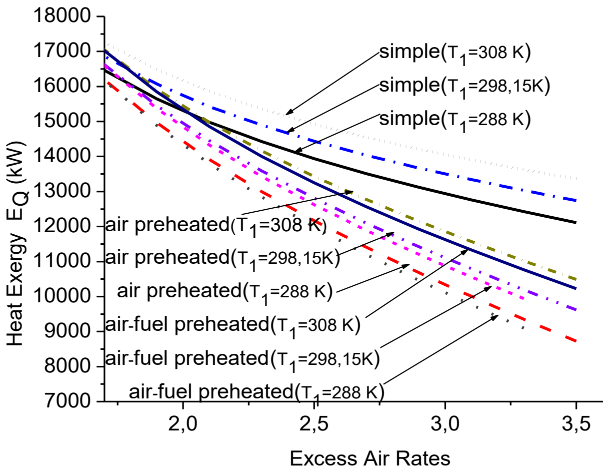


**Figure 10.** Variations of heat exergy with excess air rate for different ambient temperatures.

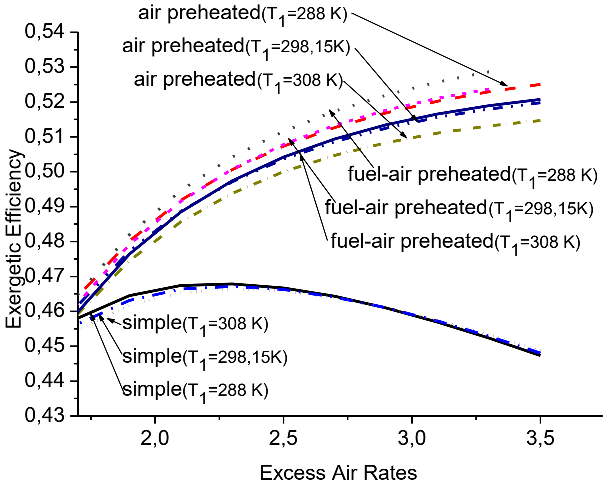


**Figure 11.** Variations of exergetic efficiency with excess air rate for different ambient temperatures.

**The Effects of the Compressor Inlet Air Humidity Ratio on the Working Conditions of the Cycles**

The process of water spraying into compressor inlet air lowers the outlet temperature of the combustion chamber and that process reduces the heat energy production of the cycles and the compressor work consumptions of the cycles. These results of this section are presented in Figs. 12 to 14.

In Fig. 12, it can be seen that increasing the specific humidity, increases the electric power of the systems. During the summer times by injection water into the inlet air of the compressor to increase the specific humidity, the electric power can be easily increased by 2-7 %. However, this method decreases the heat energy power. The simple cycle has better performance for the heat energy power production and the air-fuel preheated cycle has better performance for electric power production.

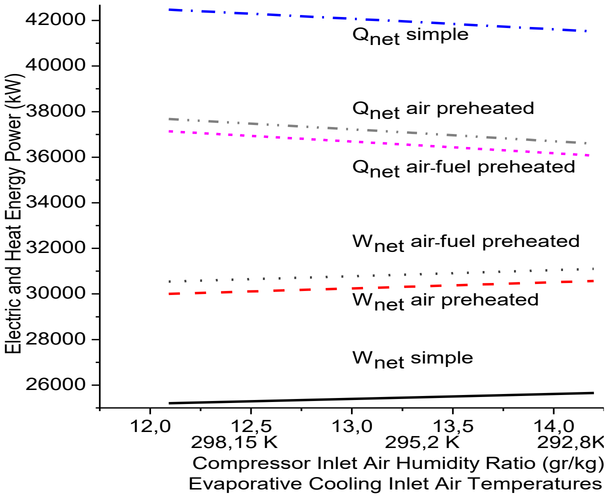


**Figure 12.** Variations of electric and heat energy power with compressor inlet air humidity ratio for various cycles.

In Figure 13 it is shown that increasing the inlet air specific humidity of the simple, the air preheated and the air-fuel preheated cogeneration power cycles decreases the thermal exergy. This happens because the temperature of the air exiting the compressor is low and thus less exergy in the recuperator is obtained. The simple cycle is the best one for heat exergy power and it is about 12 % better than other cycles.

In Figure 14 it is seen that increasing the specific humidity of the compressor inlet air increases the exergy efficiency of the cycles. The exergy efficiency of the simple cycle is not as good as the others and that cannot be understood using the first law of thermodynamics only.

**The Effects of the Compressor Inlet Air Pressure (Altitude) on the Working Conditions of the Cycles**

Compressor inlet pressure (altitude) has an important effect on the operating conditions and on the performance of the cycles. Increase in the altitude affects the air pressure and therefore the working conditions of the cycles are changed. The effects of the compressors inlet air pressure on the cycle performance are presented in Figures 15 to 18.

In Figure 15 it can be seen that decreasing the compressor inlet air pressure (decreasing the ambient pressure or increasing the altitude of the installed systems) decreases the electrical power obtained. The reason for this is that the compressor work increases with decreasing the compressor inlet air pressure and that decreases the power of the turbine but increases the thermal power. The heat power increases about 10 % and the electric power decreases about 25 % by increasing the altitude from sea level to 3000 m.

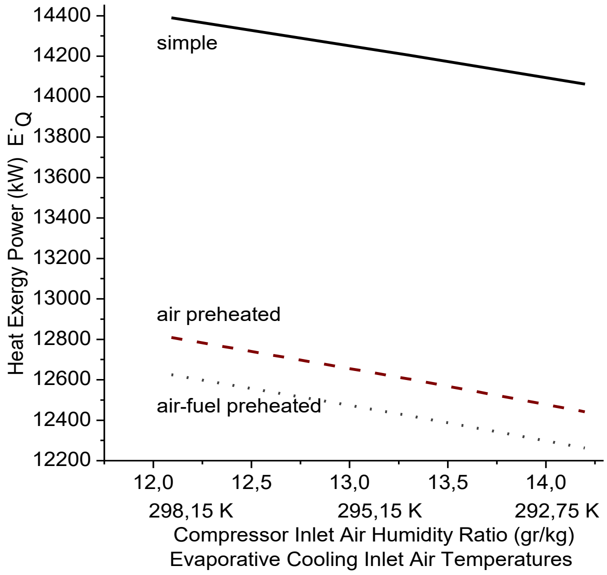


**Figure 13.** Variations of heat exergy power with compressor inlet air humidity ratio.

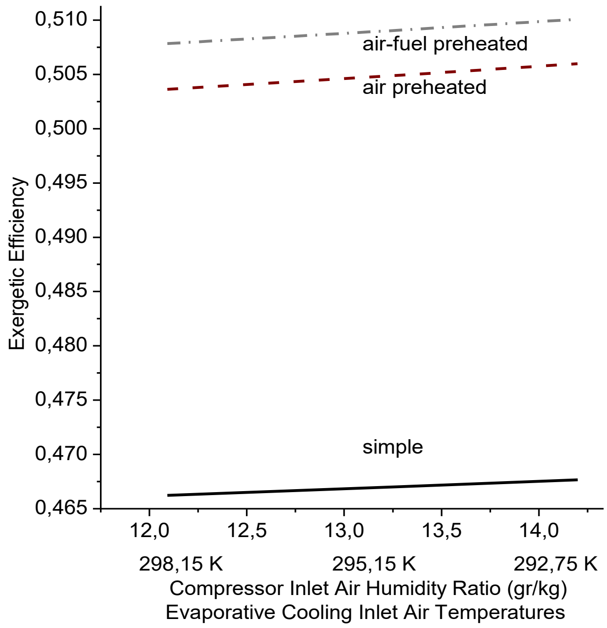


**Figure 14.** Variations of exergetic efficiency with compressor inlet air humidity ratio for different cycles.

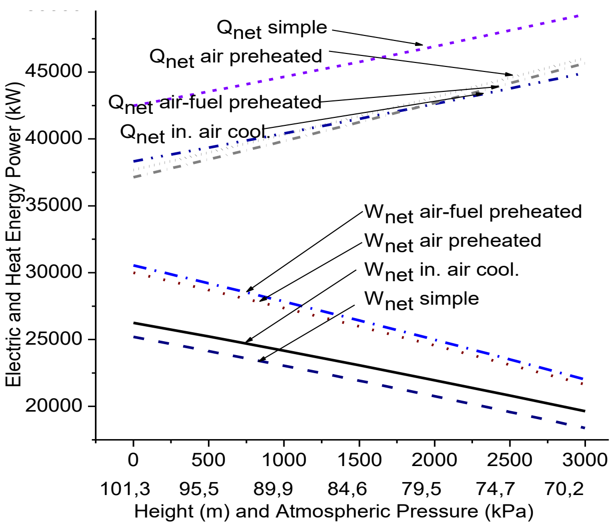


**Figure 15.** Variations of electric and heat energy power with altitude (atmospheric pressure) for different cycles.

In Figure 16 it is seen that increasing the altitude decreases the turbine power and increases the heat energy so that the electric to heat power ratio decreases. The electric to heat power rate decreases about 42 % for all cycles by changing the altitude from 0 m to 3000 m. It is better to built cogeneration plants as air-fuel preheated cycle plant at sea level for obtaining high electric to heat energy ratio.

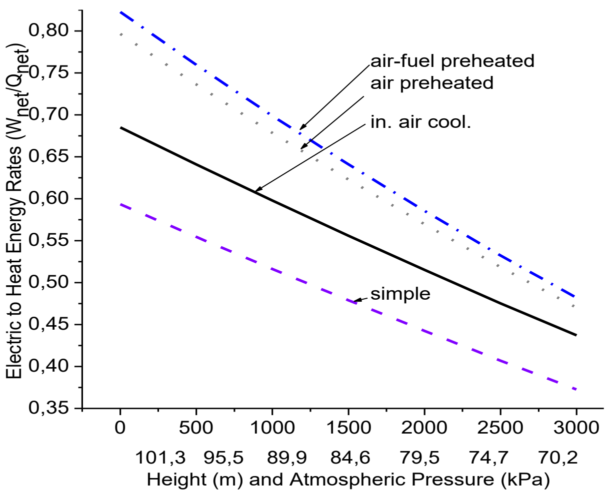


**Figure 16.** Variations of electric to heat ratio with altitude (atmospheric pressure) for different cycles.

Also Figure 17 indicates that increasing the altitude or decreasing the compressor inlet air pressure increases the heat exergy. The heat exergy power increases about 14 % for the simple cycle. It is better to built cogeneration plants to obtain high heat exergy power at places with high elevations.

In Figure 18 it is seen that increasing the altitude decreases the exergy efficiency because of less power obtained from the gas turbine. The exergy efficiency decreases about 13 % for the air-fuel preheated cycle and 11 % for the simple cycle by increasing the altitude from 0 m to 3000 m. The higher exergy efficiencies are obtained for the air-fuel preheated cycle than the other cycles.

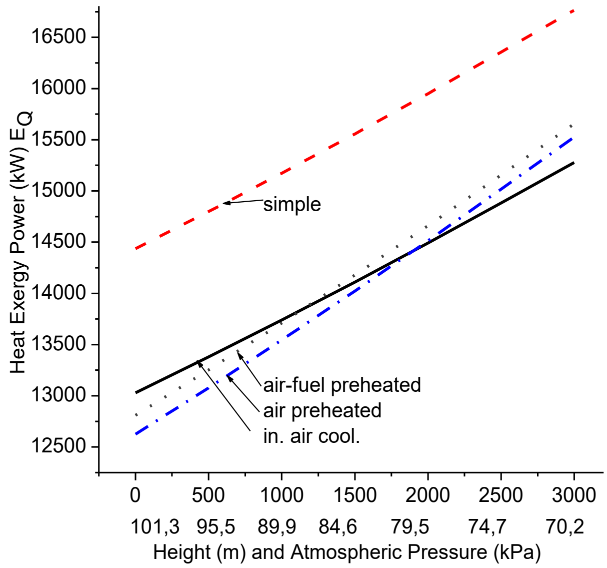


**Figure 17.** Variations of heat exergy power with height (atmospheric pressure) for different cycles.

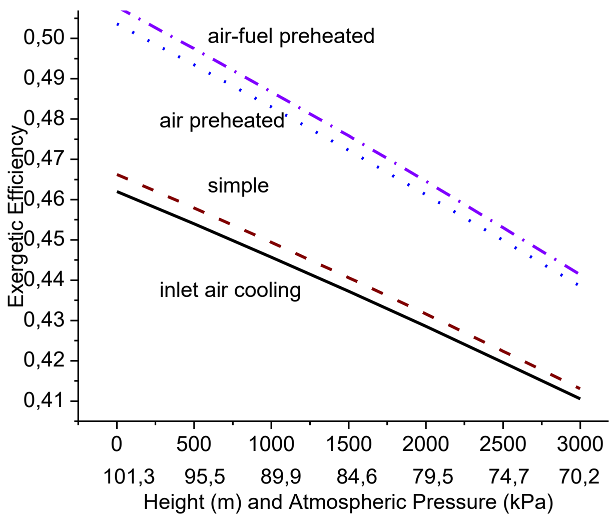


**Figure 18.** Variations of exergy efficiency with altitude (atmospheric pressure) for different cycles.

## CONCLUSION

The effects of all ambient conditions on the cycles analyzed above are considered simultaneously, analyzed by using exergetic method and compared with each other in this study. The four cogeneration cycles are evaluated in terms of electric and heat powers, heat exergy, electrical to heat ratio and exergetic efficiency with respect to temperature, humidity and atmospheric pressure. Decreasing the compressor's inlet air temperature from 25 $^0$C to 0 $^0$C the electrical power generation increases about 10-15 % and the exergetic efficiency increases about 1-1.5 %. However the thermal power decreases about 10 % with the decreasing inlet air temperature. The evaporative cooling methods for cogeneration systems increase the exergetic efficiency about 1-1.5 %. During the summer, the electric power can be easily increased by 2-7 % by spraying water into the inlet air of the compressor to increase the specific humidity and to decrease the inlet air temperature. However, this method decreases the heat energy power about 2-4 %. Decreasing the compressor inlet air pressure from 101.3 kPa to 70.18 kPa or equivalently increasing the altitude from 0 m to 3000 m the exergetic efficiency of the cycles is decreased by 11-13 %. But this change decreases the electric power generation about 25 %, and increases the heat power about 10 %. The highest exergy efficiency is obtained for the air-fuel preheated cycle. The simple cycle is the best among the four cycles to obtain the highest heat power and heat exergy.

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