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# AMBIENT CONDITIONS EFFECTS ON PERFORMANCE OF GAS TURBINE COGENERATION POWER PLANTS

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## ABSTRACT

In this study, the performances of a simple and an air preheated cogeneration cycles in ambient conditions are compared with each other. A computer program written by the author in FORTRAN codes is used for the calculation of the enthalpy and entropy values of the streams, Exergy analysis is done and compared for the simple and the air preheated cogeneration cycles for different ambient conditions. The two cogeneration cycles are evaluated in terms of heat powers and electric, electrical to heat ratio, heat exergy, and exergetic efficiency with respect to humidity, temperature, and atmospheric pressure. The simple cycle is better in obtaining high heat rate and heat exergy than the air preheated cycle. The effects of all ambient conditions on these cycles are considered and compared with each other in this study.

**KEYWORDS:** Cogeneration, environment, efficiency.

**Nomenclature**

| | |
|---|---|
| c | specific heat (kJ/kgK) |
| $\dot{E}$ | exergy flow rate (kW) |
| e | specific exergy (kJ/kg) |
| h | specific enthalpy (kJ/kg) |
| H | enthalpi (kJ) |
| $\dot{m}$ | mass flow rate (kg/s) |
| LHV | lower heating value (kJ/kg) |
| M | molecular weight (kg/kMol) |
| n | number of moles (kMol) |
| P | pressure (kPa) |
| $\dot{Q}$ | heat flow rate (kW) |
| $\bar{R}$ | universal gas constant |
| s | specific entropy (kJ/kgK) |
| S | entropy (kJ/K) |
| T | temperature (K) |
| W | power (kW) |
| $x_i$ | molar fraction |
| $x_{mi}$ | mass fraction |

**Greek letters**

| | |
|---|---|
| $\eta$ | efficiency |

**Subcripts**

| | |
|---|---|
| *C* | compressor |
| *cc* | combustion chamber |
| Ch | chemical |
| *eg* | exhaust |
| *ex* | exergy |

| | |
|---|---|
| *HRSG* | heat recovery steam generator |
| *is* | isentropic |
| Ph | physical |
| *R* | recuperator |
| *T* | turbine |
| 0 | environment conditions |

## INTRODUCTION

Production of useful thermal energy and electricity in one operation by using fuel efficiently is named cogeneration. Cogeneration cycles have many advantages over the conventional cycles such as safe and reliable operation, dual fuel capability and fast starting time, higher efficiency, compact size, and lower weight per unit power, more economic and less environmental emissions. In gas turbine systems lots of kind of fuel are used such as natural gas or mixed fuels such as biomass, refinery residues, naphtha, alcohols, etc. Improving alternative fuel studies for gas turbines or for diesel motors to cogenerate electric and heat power are very important for industry and environment [1, 2]. Using biodiesel in motors to cogenerate electric and heat power are very popular in our days to improve [3]. Gas turbine cogeneration systems have some applications in buildings, industry and others. Gas turbines are efficient devices that use high temperature heat energy to produce power and heating. On the market there are many gas turbines cogeneration systems; however they differ in efficiency, power output, firing temperature, pressure ratio, exhaust temperature, etc [4]. Three-generation is also possible with gas turbine systems. Producing power, heating and cooling is known technologies. For cooling absorption cycles can be used in three generation and the exhaust gases heat energy is used for absorption cooling. Also for efficient utilization of low temperature heat sources such as geothermal, solar, waste heat sources, etc, ammonia-water power cycles are important and present good economical options. Ammonia-water power cycle can be used as independent cycles to provide power output and cooling. For refrigeration also cascade refrigeration systems can be used. The exergetic analysis of these technologies can be found in literature [5, 6].

For a cogeneration system, there are some methods for improving efficiency at design and operating stages such as air preheating, fuel preheating, intercooling, reducing auxiliary power consumption, using advanced gas turbine cooling, steam injection, hydrogen cooled generators, increasing gas turbine inlet air temperature, serving low compressor inlet air temperature, increasing compressor inlet air pressure, cooling and humidification of the inlet air of the compressor, increasing air excess rates, multiple pressure cycle with reheat and better HRSG design. In those methods steam injection into the combustion chamber is one of the most important methods to use for variable heat and electric demands. The details can be found in literature [4, 7, 8]. .

Exergy analysis is the most effective tool to evaluate thermal systems. The irreversibilities in each component and in overall cycle can be calculated and evaluated by using exergy analysis method. The details of the exergy analysis method can be found in literature [9, 10].

Kehlhofet, et al., have shown that the effects of the compressor inlet air temperature on relative work and cooling inlet air increases electrical power [11]. Al-Fahed et al., 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. They also have found that 1 $^0$C increase in the compressor inlet air temperature leads to a decrease in electricity production around 0.7 %. 15 $^0$C Cooling of the compressor inlet air by absorption method increases the electric efficiency around 6 % [12].The variations in the compressor inlet air temperatures cause changes in the electricity and thermal energy power [13]. Boyce has showed 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 % [2]. Santo and Gallo have found that since decreasing the temperature of the compressor inlet air increases the density of the air for the same mass, less energy is spent [14].

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 [15]. 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 % [16].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. Increasing the specific humidity decreases the molecular weight of air and that affects the temperature and increasing specific humidity increases specific heat and gas constant value at constant pressure. In this study, by using exergy analysis method the effect of the ambient conditions and the evaporative cooling of the compressor inlet air on the performance of the cogeneration cycles are analyzed. The two cogeneration cycles which are analyzed in this study are called simple and air preheating cycles.

## MATERIALS AND METHODS

The schematic diagram of the simple cycle and air preheated cycle are given in Figure 1and Figure 2. The compressed air enters the combustion chamber without preheating in the simple cycle. After the combustion in the chamber, the hot gases are expanded at the gas turbine and from the gas turbine, the hot gases become the source of the heat recovery steam generator. In the air preheated (recuperated) cycle compressed air is heated by hot exhaust gases in the recuperator and after that the hot air enters the combustion chamber for combustion with fuel. The hot exhaust gases that exit from the combustion chamber are then expanded in the gas turbine. From the gas turbine the hot gases are the source of the heat of recuperator and the heat recovery steam generator.

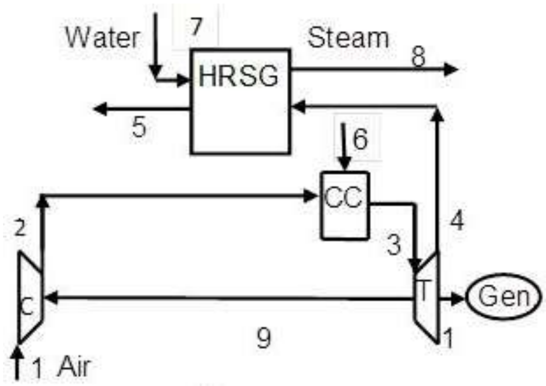


*Figure 1. Simple cycle*

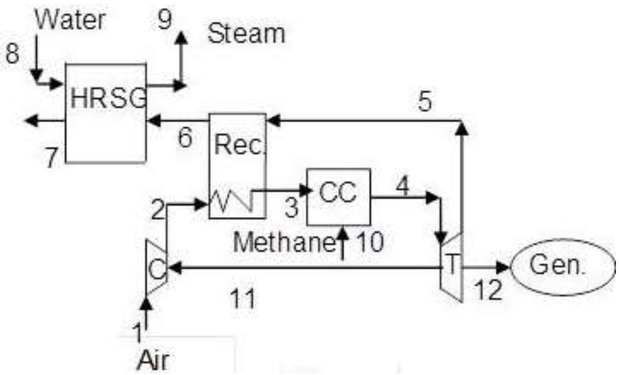


*Figure 2. Air preheated cycle*

These cycles are fueled with natural gas; however it is taken to be methane for the sake of simplicity. The following assumptions are introduced in modeling each cycle: The pressure losses in the combustion chamber, air preheater and HRSG are known as 5 %, the environmental conditions are fixed and defined as $T_0 = 298.15$ K and $P_0 = 1.013$ bars, the gas turbine net electric power 30 MW, the main capacity of the air compressors are $m_1 = 91.4$ kg/s, HRSG $m_s = 14$ kg/s saturated steam at 20 bar, combustion chamber's fuel $m_f = 1.64$ kg/s methane. The mass, the energy and the entropy equations of the components and the exergy and the exergy efficiency equations of the components of the air preheated cycle are given in Table 1 and in Table 2 for the air preheated cycle. Specific enthalpies and specific entropies are calculated for each stream from the equations of the reference [5] and the chemical reaction in the combustion chamber can be written as follows [17].

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

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

| | |
|---|---|
| Overall Cycle | $\bar{h}_i = f(T_i)$<br>$\bar{s}_i = f(T_i, P_i)$<br>$\dot{m}_{air}h_{air} + \dot{m}_{fuel}LHV_{CH4} - \dot{Q}_{Loss,CC} - \dot{m}_{eg.,out}h_{eg.,out} - \dot{W}_T - \dot{m}_{steam}(h_{water,in} - h_{steam,out}) = 0$<br>$\dot{Q}_{Loss,CC} = 0.02\dot{m}_{fuel}LHV_{CH4}$ |

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

| Component | Exergy Equation | Exergy Efficiency |
|---|---|---|
| Compressor | $\dot{E}_{D,C} = \dot{E}_1 + \dot{W}_C - \dot{E}_2$ | $\eta_{ex,C} = \frac{\dot{E}_{out,C} - \dot{E}_{in,C}}{\dot{W}_C}$ |
| Recuperator | $\dot{E}_{D,R} = \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6$ | $\eta_{ex,R} = \frac{\dot{E}_{out,air,R} - \dot{E}_{in,air,R}}{\dot{E}_{out,exhaust,R} - \dot{E}_{in,exhaust,R}}$ |
| HRSG | $\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9$ | $\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$ |
| Combustion Chamber | $\dot{E}_{D,CC} = \dot{E}_3 + \dot{E}_{10} - \dot{E}_4$ | $\eta_{ex,CC} = \frac{\dot{E}_{out,CC}}{\dot{E}_{in,CC} + \dot{E}_{fuel}}$ |
| Turbine | $\dot{E}_{D,T} = \dot{E}_4 - \dot{E}_5 - \dot{W}_C - \dot{W}_T$ | $\eta_{ex,T} = \frac{\dot{W}_{net,T} + \dot{W}_C}{\dot{E}_{in,T} - \dot{E}_{out,T}}$ |
| Overall Cycle | $\dot{E} = \dot{E}_{ph} + \dot{E}_{ch}$<br>$\dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0))$<br>$\dot{E}_{ch} = \frac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k \ln x_k\right\}$<br>$\eta_{ex} = \frac{\dot{W}_{net,T} + (\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG})}{\dot{E}_{fuel}}$ | |

## RESULTS AND DISCUSSION

To calculate the enthalpy and entropy values of the streams, a computer program written by the author in FORTRAN codes is used. In Figure 2, variations of exergy efficiencies with injected steam mass for different compression rates are given. It can be seen that, increasing the compression ratio increases the exergetic efficiency for each cycle. The exergetic efficiency of the air preheated cycle is higher than the simple one. Injection steam mass into the combustion chamber increases the exergetic efficiency for lower compression rates. But, this method decreases the exergetic efficiency for higher compression rates.

In Figure 3, variations of exergetic efficiency with different compressor inlet air pressures are given. Decreasing in atmospheric pressure which that means increasing the height of the installation decreases the exergetic efficiency for the same exhaust pressure.

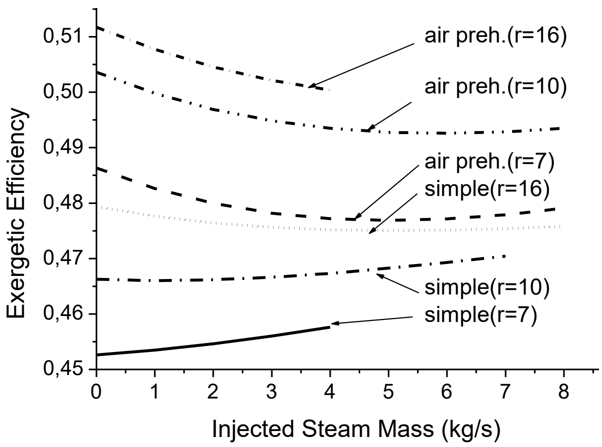


***Figure 2: Variations of exergy efficiencies with injected steam mass for different compression rates.***

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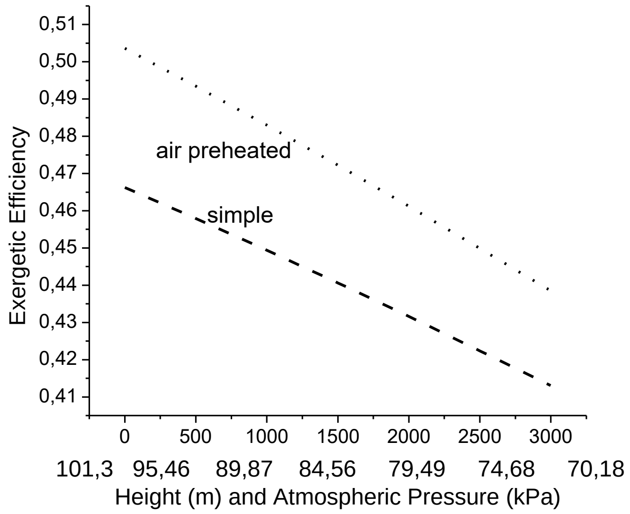


***Figure 3: variations of exergetic efficiency with different compressor inlet air pressures.***

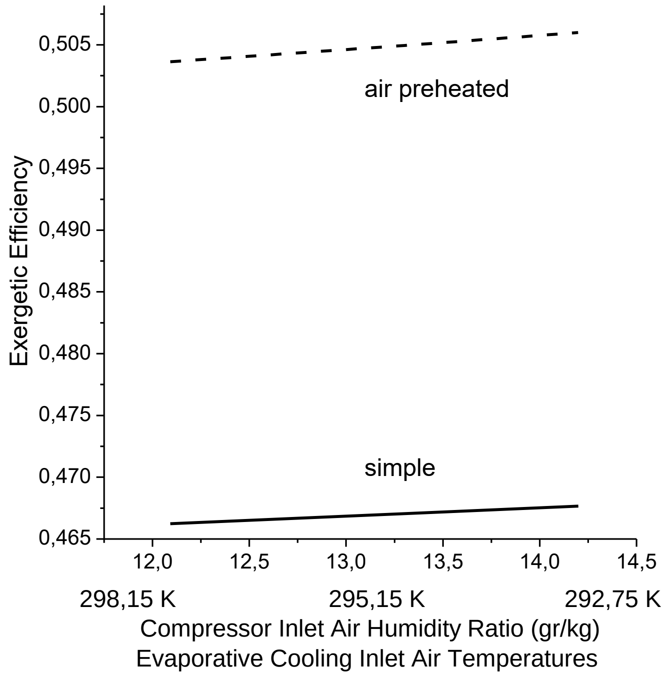


***Figure 4: variations of exergetic efficiency with different compressor inlet air humidity ratios.***

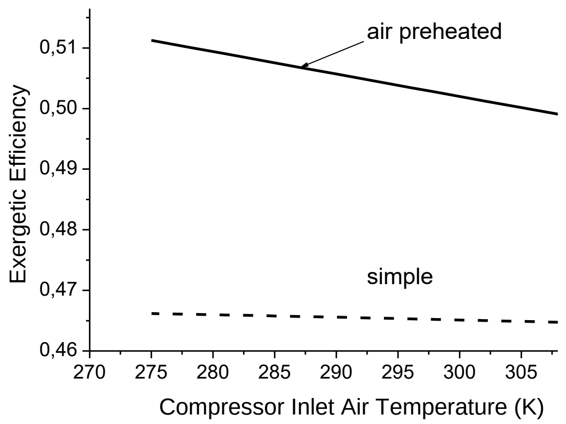


***Figure 5: variations of exergetic efficiency with different compressor inlet air temperatures.***

In Figure 4, variations of exergetic efficiency with different compressor inlet air humidity ratios are given. Increasing the inlet air humidity ratios increases the exergetic efficiency about 0.5 %. By injection water spray into the compressor inlet air that can be achieved.

In Figure 5, variations of exergetic efficiency with different compressor inlet air temperatures are given. Increasing the temperature of the compressor inlet air decreases the exergetic efficiency about 1 % for the air preheated cycle and about 0.5 % for the simple cycle.

## CONCLUSION

The two cogeneration cycles are evaluated in terms of heat powers and electric, electrical to heat ratio, heat exergy, and exergetic efficiency with respect to humidity, temperature, and atmospheric pressure. It is concluded that, increasing the compression ratio increases the exergetic efficiency for each cycle. The exergetic efficiency of the air preheated cycle is higher than the simple one. Injection steam mass into the combustion chamber increases the exergetic efficiency for lower compression rates. But, this method decreases the exergetic efficiency for higher compression rates. Decreasing in atmospheric pressure which that means increasing the height of the installation decreases the exergetic efficiency for the same exhaust pressure. Increasing the inlet air humidity ratios increases the exergetic efficiency about 0.5 %. By injection water spray into the compressor inlet air that can be achieved. Increasing the temperature of the compressor inlet air decreases the exergetic efficiency about 1 % for the air preheated cycle and about 0.5 % for the simple cycle.

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