ISSN: 2277-9655
Impact Factor: 4.116
CODEN: IJESS7

IJESRT

INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

THERMODYNAMIC ANALYSIS OF GAS TURBINE COGENERATION POWER PLANTS

Necmi Ozdemir*

[*] Department of Electrical Engineering, Kocaeli University, Umuttepe Campus 41380, Kocaeli, Turkey

DOI: 10.5281/zenodo.155102

ABSTRACT

Cogeneration cycles have many advantages over the conventional cycles. In this study, the performances of the simple and the air preheated cogeneration cycles are compared with each other. To calculate the enthalpy and entropy values of the streams, a computer program written by the author in FORTRAN codes is used. Exergy analysis is done for the simple and the air preheated cogeneration cycles. The results present that by changing the compression rate from 6 to 16, the electric power increases about 20 %, but the heat power decreases about 11 % for the simple cycle. By changing excess air rate from 1.3 to 3.5, the decrease of the heat power and the heat exergy power are about 20 % for the simple cycle at compression rate 6. Increasing compression ratio increases the exergetic efficiency for the two cycles. Maximum efficiencies are obtained about 2 and 2.5 excess air rates for the simple cycle. For the air preheated cycle increasing excess air rates increases the exergetic efficiency.

KEYWORDS: Power, Cogeneration, Exergy.

Nomenclature

c	specific heat (kJ/kgK)
COP	coefficient of performance
$\dot{E}$	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)
$\overline{R}$	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

η	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

By using fuel efficiently, production of useful thermal energy and electricity in one operation is named cogeneration. Cogeneration cycles have many advantages over the conventional cycles such as higher efficiency, lower weight per unit power, safe and reliable operation, dual fuel capability, compact size, fast starting time, more economic and less environmental emissions. In diesel or gas motor cogeneration systems the heat energy is obtained from the exhaust and electrical energy is obtained from the mechanical energy. 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. The fuel flexibility for gas turbine systems is the most important advantage over other systems. 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]. Gas turbine cogeneration systems have some applications in buildings, industry and others. There are many methods to improve the efficiency of gas turbine cogeneration systems such as fuel preheating, reducing auxiliary power consumption, increasing gas turbine inlet temperature, inter-cooling, steam injection, using hydrogen cooled generators, advanced gas turbine cooling, low compressor inlet air temperature, high compressor inlet air pressure, high compressor inlet air humidity, multiple pressure cycle with reheat and better HRSG design. Steam injection into the combustion chamber is one of the most important methods to use for variable heat and electric demands [3, 4]. On the market there are many gas turbines cogeneration systems; however they differ in power output, firing temperature, efficiency, exhaust temperature, pressure ratio, etc.

Exergy analysis is the most effective tool to evaluate thermal systems. By using exergy analysis method the irreversibilities in each component and in overall cycle can be calculated and evaluated. In thermal design and optimization this method gives the best results. The details of the exergy analysis method can be found in literature [5, 6, 7].

In design and optimization of thermal systems the demand is very important and should be taken into consideration. In producing power by using fuel, if there is no need of the heat energy, the heat energy of the exhaust can be used to produce cooling [8, 9]. In addition, for some kind of demands, for using low temperature heat energy, or to decrease power consumption in cooling cascade refrigeration systems can be taken into consideration [10, 11]. In this study, the performances of the simple and the air preheated cycles are compared with each other.

MATERIALS AND METHODS

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

ISSN: 2277-9655
Impact Factor: 4.116
CODEN: IJESS7

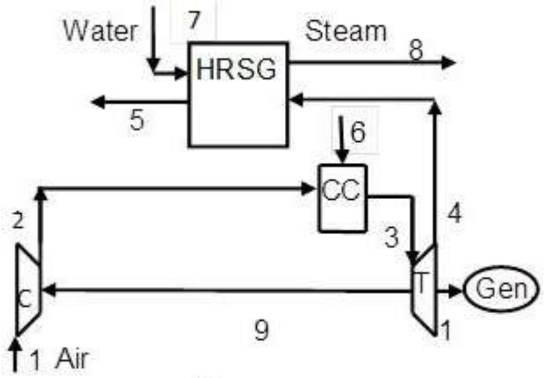

Figure 1. Simple cycle

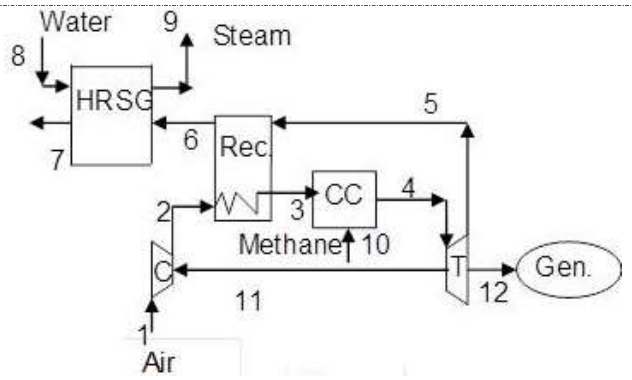

Figure 2. Air preheated cycle

The mathematical modeling and the thermodynamic analysis of the cycles and their components will be explained in this section. 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 environmental conditions are fixed and defined as T_0 = 298.15 K and P_0 = 1.013 bars. The pressure losses in the combustion chamber, air preheater and HRSG are known as 5 %. 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, gas turbine net electric power 30 MW, combustion chamber's fuel m_f = 1.64 kg/s methane. The thermodynamic model and the calculation procedure 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]. The chemical reaction in the combustion chamber can be written as follows [5].

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

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	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_5 = \dot{m}_6$	$\dot{m}_2 h_2 + \dot{m}_5 h_5 = \dot{m}_3 h_3 + \dot{m}_6 h_6$	$\dot{m}_2 s_2 + \dot{m}_5 s_5 - \dot{m}_3 s_3 - \dot{m}_6 s_6 + \dot{S}_{gen,R} = 0$
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	$\dot{m}_4 = \dot{m}_5$	$\dot{m}_4 h_4 = \dot{W}_T + \dot{W}_C + \dot{m}_5 h_5$	$\dot{m}_4 s_4 - \dot{m}_5 s_5 + \dot{S}_{gen,T} = 0$
HRSG	$\dot{m}_6 = \dot{m}_7$ $\dot{m}_8 = \dot{m}_9$	$\dot{m}_6 h_6 + \dot{m}_8 h_8 = \dot{m}_7 h_7 + \dot{m}_9 h_9$	$\dot{m}_6 s_6 + \dot{m}_8 s_8 - \dot{m}_7 s_7 - \dot{m}_9 s_9 + \dot{S}_{gen,HRSG} = 0$
Overall Cycle	$\bar{h}_i = f(T_i)$ $\bar{s}_i = f(T_i, P_i)$ $\dot{m}_{air} h_{air} + \dot{m}_{fuel} LHV_{CH4} - \dot{Q}_{Loss,CC} - \dot{m}_{eg.,out} h_{eg.,out} - \dot{W}_T - \dot{m}_{steam}(h_{water,in} - h_{steam,out}) = 0$ $\dot{Q}_{Loss,CC} = 0.02\dot{m}_{fuel} LHV_{CH4}$		

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	$\dot{E}_{D,R} = \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6$	$\eta_{ex,R} = \frac{\dot{E}_{out,air,R} - \dot{E}_{in,air,R}}{\dot{E}_{out,exhaust,R} - \dot{E}_{in,exhaust,R}}$
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	$\dot{E}_{D,T} = \dot{E}_4 - \dot{E}_5 - \dot{W}_C - \dot{W}_T$	$\eta_{ex,T} = \frac{\dot{W}_{net,T} + \dot{W}_C}{\dot{E}_{in,T} - \dot{E}_{out,T}}$
HRSG	$\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9$	$\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$
Overall Cycle	$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch}$ $\dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0))$ $\dot{E}_{ch} = \frac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k \ln x_k\right\}$ $\eta_{ex} = \frac{\dot{W}_{net,T} + (\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG})}{\dot{E}_{fuel}}$	

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. All the analysis results are presented in Figure 3 to 6.

In Figure 3 variation of electric and heat power with compression rates for variable combustion temperatures are given (where m_{air} = 91,3 kg/s, m_{fuel} = 1,64 kg/s, excess air rate = 2,5, $T_{rec.out}$ = 850 K, T_{steam} = 485,57 K, $T_{eg.}$ = 426 K, $\eta_{is,C} = \eta_{is,T}$ = 0,86). Adding a recuperator for air preheating decreases the heat energy, however increases the electrical power. Increasing the compression ratio of these two cycles increases the electrical power, but decreases the heat energy. Increasing the compression ratio increases the combustion chamber outlet temperature which increases the turbine work, but decreases the amount of heat obtained from HRSG. For the simple cycle heat power is higher but electric power is lower than the air preheated cycle.

In Figure 4 variation of electric power with excess air rates for different compression rates are given. The simple cycles have the maximum electric power around 2.3 and 3.0 excess air rates. However increasing excess air rates of the air preheated cycles increases the electric power more than the simple one. Electric power of air preheated cycle increase about 30 % by excess air rate range 1.3 to 3.5 at compression rate 10.

In Figure 5 variation of exergetic efficiency with excess air rates for different compression rates are given. As can be seen in this figure that increasing compression ratio increases the exergetic efficiency for the two cycles. The reason for this is that; increasing compression ratio increases the outlet temperature of the combustion chambers which means that increasing the inlet temperature of the turbine increases the exergetic efficiency. The better exergetic efficient cycle is found as air preheated cycle. The exergetic efficiencies of the air preheated cycle are continuing increasing with increasing excess air rate. Maximum efficiencies are obtained about 2 and 2.5 excess air rates for the simple cycle. For the air preheated cycle increasing excess air rates increases the exergetic efficiency.

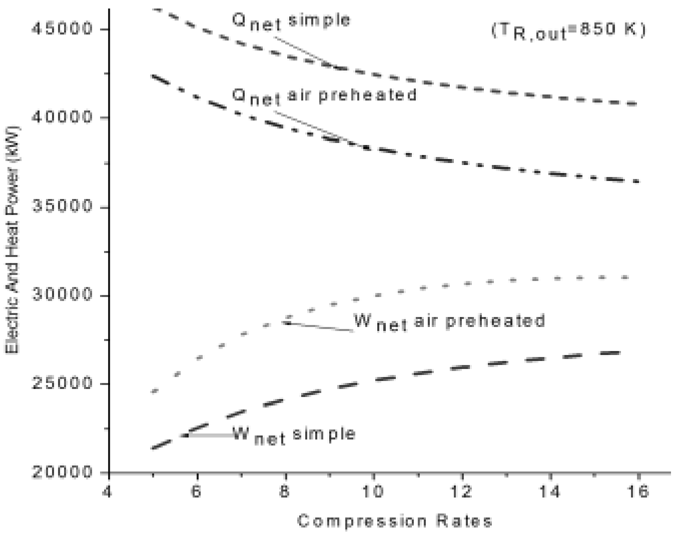

Figure 3.

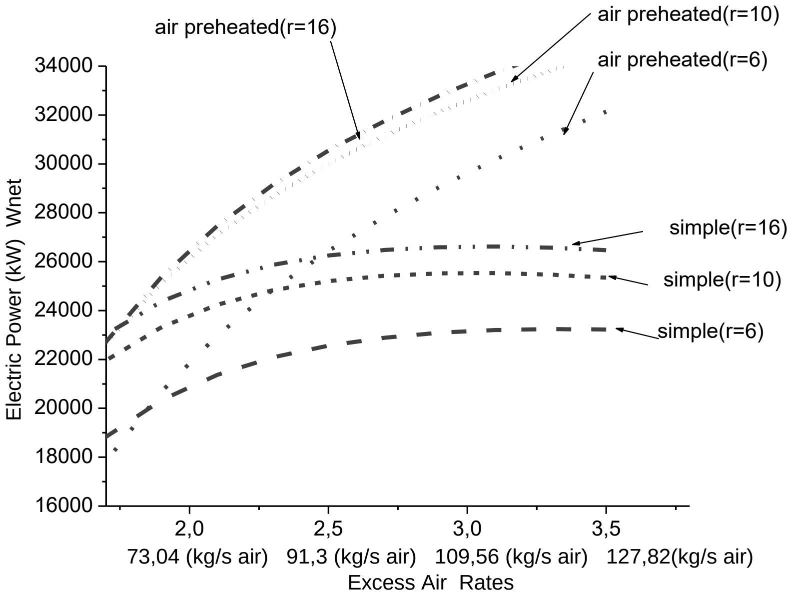

Figure 4.

In Figure 6 variation of heat power with excess air rates for different compression rates are given. Increasing the compression ratio of the two cycles decreases the heat power and more electrical power is obtained. By increasing excess air rates combustion chamber outlet temperature decreases. And that increases the turbine work but decreases heat power. Increasing the compression ratio and adding a recuperator increases the electric power, but decreases the heat power of the cycles. For the simple cycle at compression rate 6 by excess air range from 1.3 to 3.5, this decrease is about 20 %.

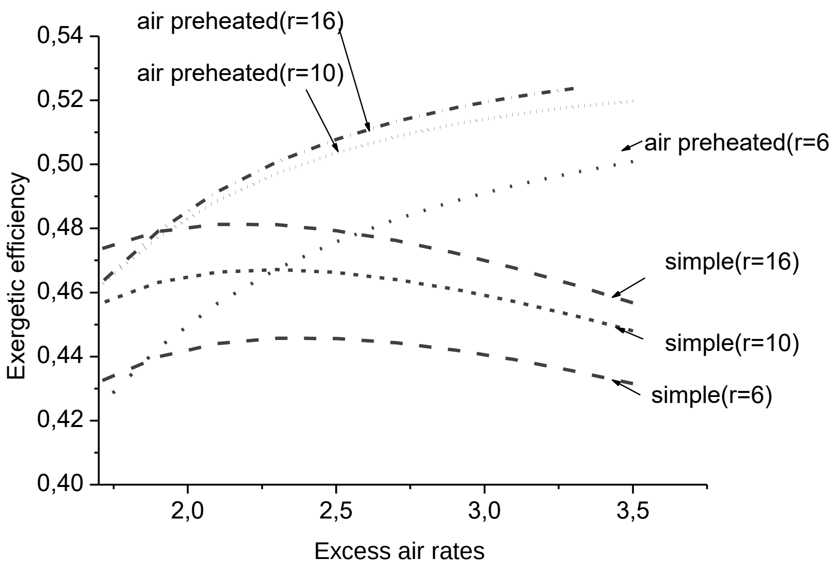

Figure 5.

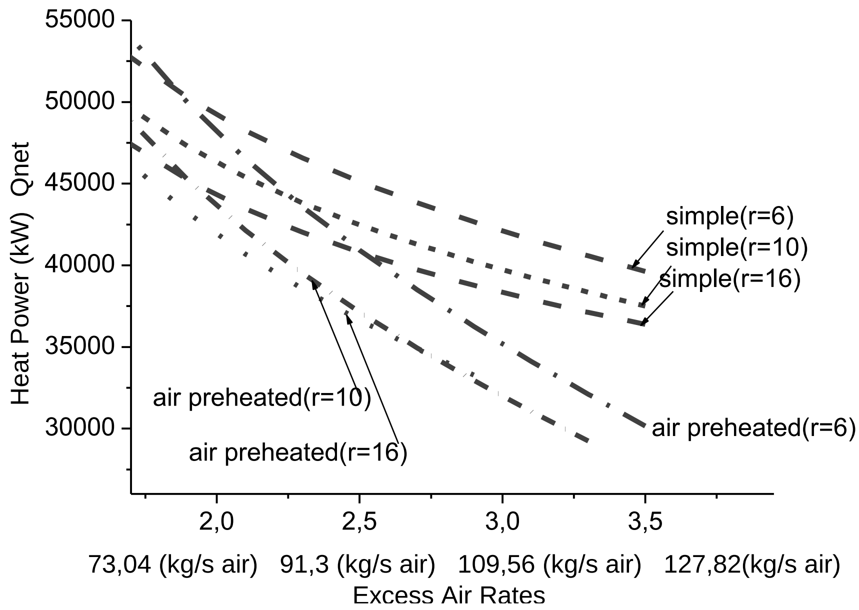

Figure 6.

CONCLUSION

In this study, the performances of the simple and the air preheated cycles are compared with each other. The results present that by changing the compression rate from 6 to 16, the electric power increases about 20 %, but the heat power decreases about 11 % for the simple cycle. By changing excess air rate from 1.3 to 3.5, the decrease of the heat power and the heat exergy power are about 20 % for the simple cycle at compression rate 6. Adding a recuperator for air preheating decreases the heat energy, however increases the electrical power. Increasing the compression ratio of these two cycles increases the electrical power, but decreases the heat energy. Increasing the compression ratio increases the combustion chamber outlet temperature which increases the turbine work, but decreases the amount of heat obtained from HRSG. For the simple cycle heat power is higher but electric power is lower than the air preheated cycle. The simple cycles have the maximum electric power around 2.3 and 3.0 excess air rates. However increasing excess air rates of the air preheated cycle increases the electric power more than the simple one. Electric power of air preheated cycle increases about 30 % by excess air rate range 1.3 to 3.5 at compression rate 10. Increasing compression ratio increases the exergetic efficiency for the two cycles. The reason for this is that; increasing compression ratio increases the outlet temperature of the combustion chambers which means that increasing the inlet temperature of the turbine increases the exergetic efficiency. Maximum efficiencies are obtained about 2 and 2.5 excess air rates for the simple cycle. For the air preheated cycle increasing excess air rates increases the exergetic efficiency.

REFERENCES

[1] ASHRAE, Cogeneration Systems and Engine and Turbine Drives, ASHRAE Systems And Equipment Handbook (SI), Chapter 7, American society of Heating, Refrigerating and air conditioning Engineers New York, 2000.
[2] A. Keven, and R. Karaali, "Investigation of an alternative fuel for diesel engines", ACTA Physica Polonica A, VOL: 128, No:2B, p:B282-B285, 2015. doi: 10.12693/APhysPolA.128.B-282
[3] J.H. Horlock, Cogeneration-Combined Heat and Power (CHP), CRIEGER Pub., Florida 1997.
[4] R. Karaali, and I.T. Ozturk,"Thermoeconomic analyses of steam injected gas turbine cogeneration cycles" ACTA Physica Polonica A, VOL:128, No:2B, p:B279-B281, 2015. doi:10.12693/APhysPolA.128.B-279
[5] A. Bejan, G. Tsatsaronis, and M. Moran, Thermal design and optimization, Wiley Pub, New York, 1996.
[6] K. Annamalai, and I.K. Puri, Advanced thermodynamics engineering, CRC Press LLC, 2002.
[7] I. Dincer, and M.A. Rosen, EXERGY, energy, environment and sustainable development, 1st ed., Elsevier Ltd., 2007.
[8] R. Karaali, "Exergy analysis of a combined power and cooling cycle", ACTA Physica Polonica A, VOL: 130, No:1, p:209-213, 2016. DOI: 10.12693/APhysPolA.130.209

[9] Y. Jaluria, Design and optimization of thermal systems, Second ed. CRC Press, New York, 2008.
[10] R. Karaali, "Thermodynamic analysis of a cascade refrigeration system", ACTA Physica Polonica A, VOL: 130 No:1, p:101-106, 2016. DOI: 10.12693/APhysPolA.130.101
[11] J.M. Moran, and G. Tsatsaronis, The CRC Handbook of Thermal Engineering, CRC Press LLC., 15-109, Florida, 2000.