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THERMODYNAMIC ANALYSIS OF RECUPERATED GAS TURBINE COGENERATION CYCLES

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ABSTRACT

Cogeneration cycles have better advantages over the conventional cycles. In this study, the performances of the recuperated cogeneration cycles are analyzed. For the calculation of the enthalpy and the entropy values of the streams, a computer program written by the author in FORTRAN codes is used. Exergy analysis is done for the air-fuel preheated cogeneration cycle. The results present that by changing the compression rate from 6 to 16, the electric power increases about 20-40 %, but the heat power decreases about 10-30 % for the air-fuel preheated cycle. The decrease of the heat power is about 20-40 % for the air-fuel preheated cycle at compression rate 6 by changing excess air rate from 1.3 to 3.5. Increasing compression ratio increases the exergetic efficiency for the cycle. For the air-fuel preheated cycle increasing excess air rates increases the exergetic efficiency.

KEYWORDS: Recuperated, 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

I. INTRODUCTION

Cogeneration is the production of useful thermal energy and electricity in one operation by using fuel efficiently. Cogeneration cycles have better advantages over the conventional cycles such as lower weight per unit power, higher efficiency, compact size, safe and reliable operation, fast starting time, dual fuel capability, more economic and less environmental emissions. In cogeneration systems electrical energy is obtained from the mechanical energy and the heat energy is obtained from the exhaust. In gas turbine systems several kind of fuel are used such as natural gas or mixed fuels such as naphtha, biomass, alcohols, refinery residues, etc. The fuel flexibility for gas turbine systems is a very important advantage over all other systems. Improving alternative fuel for gas turbines or for diesel motors are very important for industry and environment [1, 2, 3]. Gas turbine cogeneration systems have lots of applications in, industry, buildings and others. There are a lot of methods to improve the efficiency of gas turbine cogeneration cycles such as increasing gas turbine inlet temperature, inter-cooling, fuel preheating, reducing auxiliary power consumption, advanced gas turbine cooling, steam injection, using hydrogen cooled generators, low compressor inlet air temperature, high compressor inlet air pressure, better HRSG design, multiple pressure cycle with reheat and high compressor inlet air humidity. Steam injection into the combustion chamber is also an important method to use for variable electric and heat demands [3, 4, 5]. On the market there are many gas turbines cogeneration systems for sale; however they differ in efficiency, power output, pressure ratio, firing temperature, exhaust temperature, etc.

By using exergy analysis method which is the most effective tool to evaluate thermal systems, 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]. The demand is very important in design and optimization of thermal systems and should be taken into consideration. If there is no need of the heat energy in producing power by using fuel, the heat energy of the exhaust can be used to produce cooling [8, 9]. Also, 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, 12]. In this study, the performances of the air-fuel preheated cycles are analyzed.

II. MATERIALS AND METHODS

The schematic diagram of the recuperated (air-fuel preheated) cycle is given in Figure 1. In the air-fuel preheated (recuperated) cycle the compressed air and fuel are heated by hot exhaust gases in two different recuperators. After that the hot air enters the combustion chamber for combustion with the heated fuel. After the combustion in the chamber, the hot gases are expanded at the gas turbine to obtain work and from the gas turbine; the hot gases become the source of the heat recovery steam generator and air fuel heating.

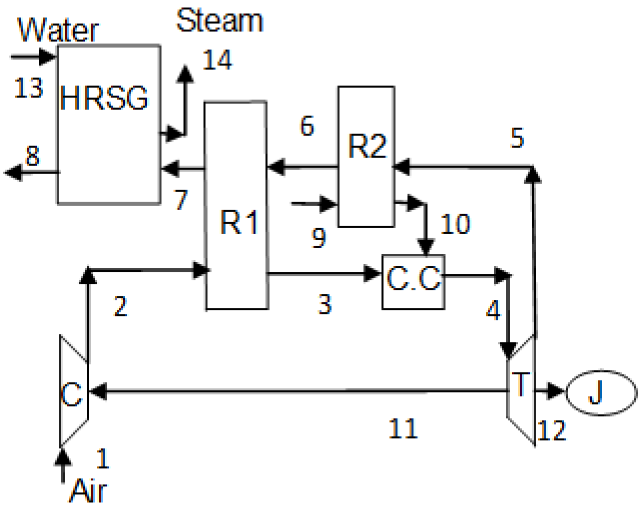

Figure 1. Air-fuel (recuperated) preheated cycle

The following assumptions are introduced in modeling each cycle: These cycles are fueled with natural gas; however it is taken to be methane for the sake of simplicity. 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 main capacity of the air compressors are m_1 = 91.4 kg/s, gas turbine net electric power 30 MW, combustion chamber's fuel m_f = 1.64 kg/s methane, HRSG m_s = 14 kg/s saturated steam at 20 bar. The thermodynamic model and the calculation procedure are given in Table 1 and in Table 2. 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	$\dot{m}_1 = \dot{m}_2$	$\dot{m}_1 h_1 + \dot{W}_C = \dot{m}_2 h_2$	$\dot{m}_1 s_1 - \dot{m}_1 s_2 + \dot{S}_{gen,C} = 0$
Recuperator1	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_6 = \dot{m}_7$	$\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 + \dot{m}_7 h_7$	$\dot{m}_2 s_2 + \dot{m}_6 s_6 - \dot{m}_3 s_3 - \dot{m}_7 s_7 + \dot{S}_{gen,R1} = 0$
Recuperator2	$\dot{m}_5 = \dot{m}_6$ $\dot{m}_9 = \dot{m}_{10}$	$\dot{m}_5 h_5 + \dot{m}_9 h_9 = \dot{m}_6 h_6 + \dot{m}_{10} h_{10}$	$\dot{m}_5 s_5 + \dot{m}_9 s_9 - \dot{m}_6 s_6 - \dot{m}_{10} s_{10} + \dot{S}_{gen,R2} = 0$
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	$\dot{m}_7 = \dot{m}_8$ $\dot{m}_{13} = \dot{m}_{14}$	$\dot{m}_7 h_7 + \dot{m}_{13} h_{13} = \dot{m}_8 h_8 + \dot{m}_{14} h_{14}$	$\dot{m}_7 s_7 + \dot{m}_{13} s_{13} - \dot{m}_8 s_8 - \dot{m}_{14} s_{14} + \dot{S}_{gen,HRSG} = 0$
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	$\dot{E}_{D,C} = \dot{E}_1 + \dot{W}_C - \dot{E}_2$	$\eta_{ex,C} = \frac{\dot{E}_{out,C} - \dot{E}_{in,C}}{\dot{W}_C}$
Recuperator1	$\dot{E}_{D,R1} = \dot{E}_2 + \dot{E}_6 - \dot{E}_3 - \dot{E}_7$	$\eta_{ex,R1} = \frac{\dot{E}_{out,air,R1} - \dot{E}_{in,air,R1}}{\dot{E}_{out,exhaust,R1} - \dot{E}_{in,exhaust,R1}}$
Recuperator2	$\dot{E}_{D,R2} = \dot{E}_5 + \dot{E}_9 - \dot{E}_6 - \dot{E}_{10}$	$\eta_{ex,R2} = \frac{\dot{E}_{out,air,R2} - \dot{E}_{in,air,R2}}{\dot{E}_{out,exhaust,R2} - \dot{E}_{in,exhaust,R2}}$
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	$\dot{E}_{D,HRSG} = \dot{E}_7 - \dot{E}_8 + \dot{E}_{13} - \dot{E}_{14}$	$\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$
Overall Cycle	$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch}$ $\dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0))$ $\dot{E}_{ch} = \frac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k \ln x_k\right\}$ $\eta_{ex} = \frac{\dot{W}_{net,T} + (\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG})}{\dot{E}_{fuel}}$	

III. 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). Increasing the compression ratio of the cycle 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.

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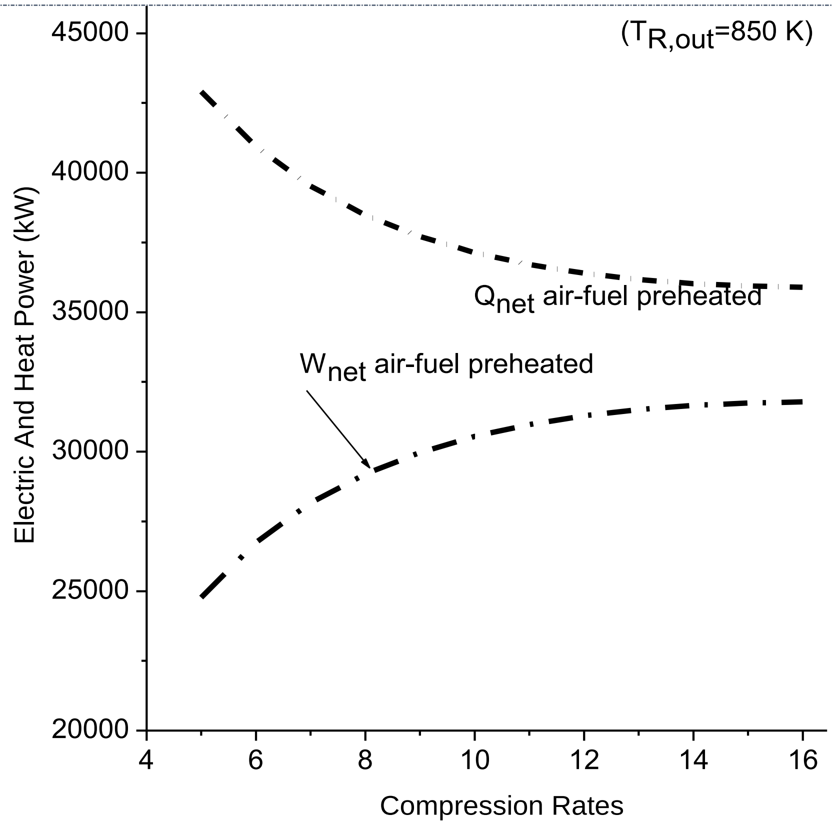

Figure 3

In Figure 4 variation of electric power with excess air rates for different compression rates are given. Increasing excess air rates of the air-fuel preheated cycle increases the electric power. Electric power of air-fuel preheated cycle increase about 60 % by excess air rate range 1.3 to 3.5 at compression rate 10.

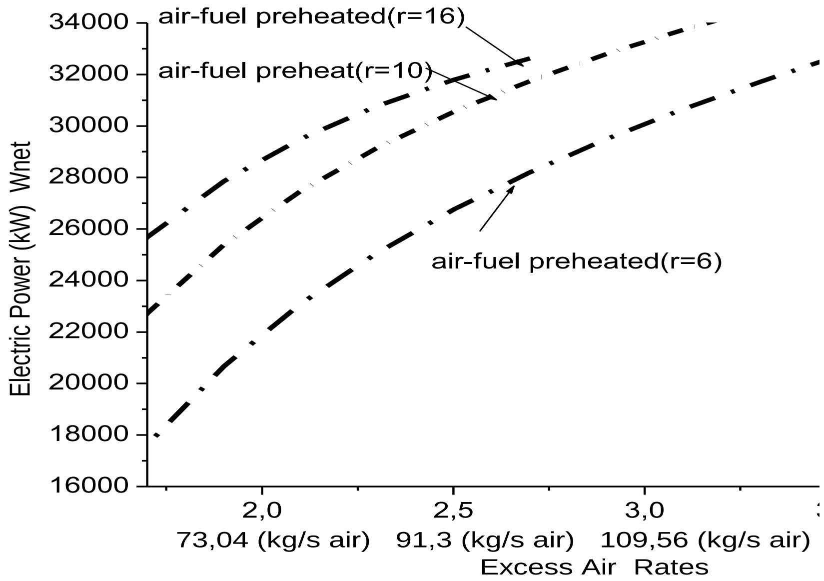

Figure 4

In Figure 5 variation of electric to heat energy rate with excess air rates for different compression rates are given. Increasing the compression ratio of the 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.

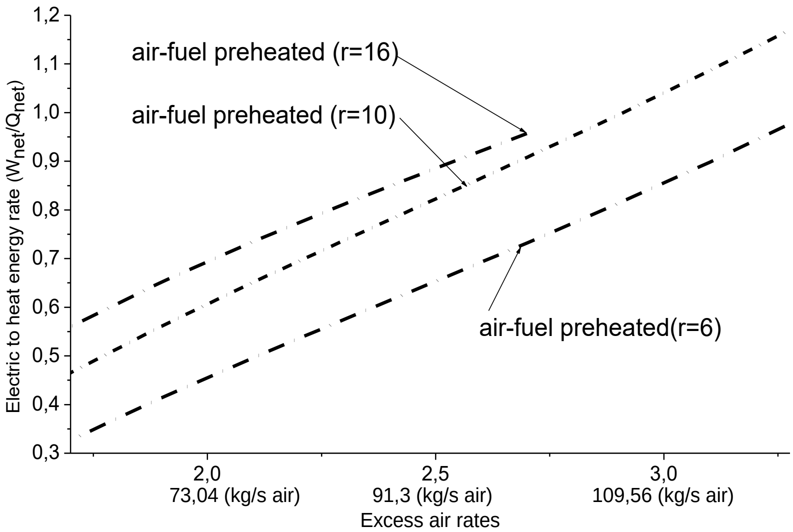

Figure 5

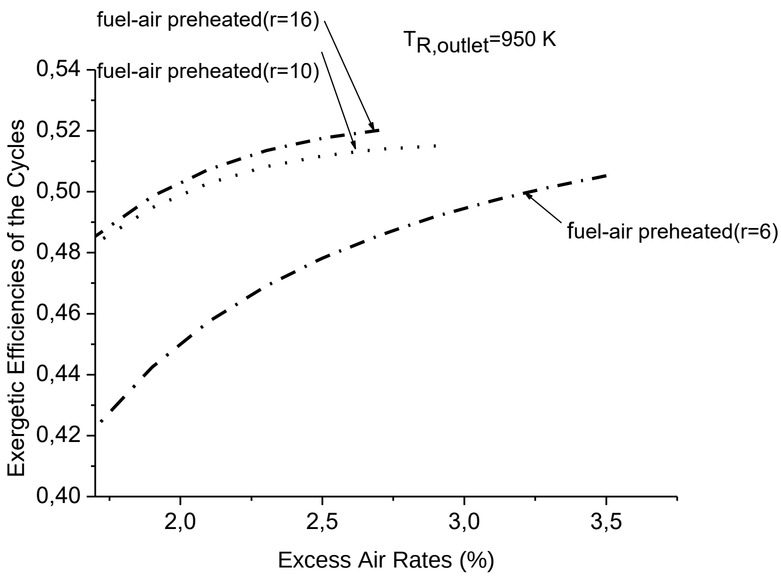

Figure 6

In Figure 6 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 cycle. 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 exergetic efficiencies of the air-fuel preheated cycle are continuing increasing with increasing excess air rate.

IV. CONCLUSION

In this study, the performances of the air-fuel preheated cycle are obtained. The results present that by changing the compression rate from 6 to 16, the electric power increases about 25 %, but the heat power decreases about 15 %. By changing excess air rate from 1.3 to 3.5, the decrease of the heat power is about 25 % for the air-fuel preheated at compression rate 6. Increasing the compression ratio of the cycle 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. Electric power of air-fuel preheated cycle increases about 35 % by excess air rate range 1.3 to 3.5 at compression rate 10. Increasing compression ratio increases the exergetic efficiency of the cycle. 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.

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