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PERFORMANCE ANALYSES OF GAS TURBINE COGENERATION PLANTS

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Abstract: In this study, some improving methods of gas turbine cogeneration cycles are applied on a simple cogeneration cycle. These methods are preheating air, preheating air and fuel, inlet air cooling by using evaporative cooling and absorption cooling. These cogeneration systems are evaluated with respect to energy efficiency (energy utilization factor), exergetic efficiency, electric and heat power, electric-heat energy rate, artificial thermal efficiency and fuel energy saving ratio and are compared with each other. In these analyses, the thermodynamic parameters such as compressing ratio, air and fuel mass ratio and compressor inlet temperatures of the cycles are used. It is concluded that these parameters can be listed from most effective to least effective as air fuel ratio, pressure ratio and compressor inlet temperature. It is also concluded that the most efficient cycle is found to be the air-fuel preheated cycle for obtaining more electric power and less heat power, and the simple cycle is the most suitable one for obtaining more heat power and less electric power.
Keywords: Cogeneration, Performance, Exergy.

GAZ TÜRBİNLİ KOJENERASYON TESİSLERİNİN PERFORMANS ANALİZLERİ

Özet: Bu çalışmada, gaz türbinli kojenerasyon çevrimlerinin geliştirilmesinde kullanılan bazı yöntemler basit bir kojenerasyon çevrimi üzerinde uygulanmıştır. Bu yöntemler havanın ön ısıtılması, hava ve yakıtın ön ısıtılması ve giriş havasının evaporatif ve absorpsiyonlu soğutma ile soğutulmasıdır. Bu kojenerasyon sistemleri enerji verimi (enerji kullanım faktörü), ekserji verimi, elektrik ve ısı gücü, elektrik ısı enerjisi oranı, yapay termal verim ve yakıt enerjisi kazanım oranı yönünden değerlendirilmiş ve birbirleri ile karşılaştırılmışlardır. Bu analizlerde basınç oranı, hava-yakıt kütleleri oranı ve çevrimlerin kompresör giriş sıcaklıkları gibi termodinamik parametreler kullanılmıştır. Bu parametrelerin en çok etkili olanından en az etkili olanına göre, hava-yakıt kütleleri oranı, basınç oranı ve kompresör giriş sıcaklıkları şeklinde sıralandığı anlaşılmıştır. Ayrıca daha çok elektrik ve daha az ısıl güç yönünden en verimli çevrimin hava-yakıt ön ısıtmalı çevrim ve daha çok ısıl güç daha az elektrik gücü için basit çevrimin en uygun çevrim oldukları ortaya çıkarılmıştır.
Anahtar Kelimeler: Kojenerasyon, Performans, Ekserji.

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)
$\bar{R}$	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

η	efficiency

Subcripts

C	compressor
cc	combustion chamber
Ch	chemical
eg	exhaust
ex	exergy
gen	generated
$HRSG$	heat recovery steam generator
is	isentropic
Ph	physical
R	recuperator
T	turbine
U	useful
0	environment conditions

INTRODUCTION

Cogeneration is the concept used to indicate production of electricity and useful thermal energy in one operation by using fuel efficiently. For a given amount of process heat, gas turbines are capable of producing more electric power than the conventional ones. Cogeneration systems have many advantages over the conventional ones such as lower weight per unit power, higher efficiency, dual fuel capability, compact size, safe and reliable operation, fast starting time, more economic and less environmental emissions. In gas turbine systems natural gas or mixed fuels such as biomass, alcohols, refinery residues, naphtha, etc., are used as fuel. The fuel flexibility for gas turbine systems is an important advantage (ASHRAE, 2000; Boyce, 2002; Horlock, 1997). Improving performance of gas turbine cogeneration cycles will be an important objective in the future.

Gas turbine cogeneration systems find applications in buildings, industry and others. The appropriate cogeneration system for a specific purpose is chosen with respect to some criteria such as efficiency, heat to power ratio and the grade of heat (Boyce, 2002). Obtaining high efficiency depends on some factors such as reduced auxiliary power consumption, increased gas turbine inlet temperature, fuel preheating, advanced gas turbine cooling, inter-cooling, hydrogen cooled generators, low compressor inlet air temperature, high compressor inlet air pressure, high compressor inlet air humidity, multiple pressure cycle with reheat and better HRSG design (Jaluria, 2008; Karaali, 2010; Atmaca, 2011). There are many gas turbines cogeneration systems on the market, however they differ in efficiency, power output, pressure ratio, exhaust temperature, firing temperature, etc. (Huang, 1990).

Feng et al., in their study introduced a new performance criterion for cogeneration systems called cogeneration efficiency. In the numerator of the definition of this efficiency the sum of work and exergy in the useful heat and the inevitable exergy loss which has to be paid when the useful heat has been supplied takes place while the fuel energy is in the denominator (Feng et al., 1998). Huang has studied the performance evaluation of three kinds of gas turbine cogeneration systems (Huang, 1990). The effect of pinch point on the system parameters and the effect of pressure of process steam on the system performance are analyzed. It was found that the first law analysis is not adequate and the second law analysis is needed (Huang, 1990). Khaliq and Kaushik in their study have analyzed thermodynamic performance evaluation of three selected gas turbine cogeneration systems with reheat, and found that the pinch point temperature has an effect on the fuel utilization efficiency, on the power to heat ratio and on the second law efficiency. They also found out that the process steam pressure affects the fuel utilization efficiency, the power to heat ratio and the second law efficiency (Khaliq and Kaushik, 2004). Malinowska and Malinowski in their parametric study of exergetic efficiency of a small scale cogeneration plant incorporating a heat pump have found that exergetic efficiency and power to heat ratio are better than conventional ones (Malinowska and Malinowski, 2003). Santo and Gallo in their study have evaluated a cogeneration system with inlet air cooling producing electricity, steam and chilled water by using the first law of thermodynamics and the economic analysis methods (Santo and Gallo, 2000). Wang and Chiou have studied the performance improvement of a simple cycle gas turbine GENSET- a retrofitting example, and found out that there is effect of pressure ratio on power output, ambient temperature on generation efficiency and power output, and steam injection ratio on efficiency (Wang and Chiou, 2002).

The maximum temperature of the cycle should be kept under a certain temperature because of metallurgical reasons. That can be achieved by using approximately between two to four times of the air that is theoretically required for complete combustion of the cycles. The air properties have major impacts on the exhaust gas properties, and thus the temperatures decrease. Accordingly, when the pressure ratio (P_2/P_1) increases, the compressor outlet temperature (T_2) increases and so does the efficiency. Because of the metallurgical reasons the higher temperature is limited. Therefore, adding a recuperator rises the outlet temperature of the air of the compressor and that increases the efficiency of the cycle (Najjar, 2000; Sue and Chuang, 2004).

The majority of the work produced by the turbine (work produced by the turbine to the compressor is called back work rate and is around 50-60 %) is spent by the compressor so that the pressure ratio (compressor work) are very effective on the cycle efficiency (Najjar, 2001; Atmaca et al., 2016). Many publications are based on finding better evaluation criteria and the most effective parameters on efficiency for gas turbine cogeneration cycle (Atmaca et al., 2009; Atmaca, 2011). These studies generally contain fewer criterions, parameters and cycles therefore they are not satisfactory as a rule of thumb. For better design and optimization process there is a need of detailed knowledge of the factors that affect the performance of cogeneration systems. That is why, in this paper as many as competitive cycles, evaluation criteria and parameter are taken into account and they are compared with each other.

In this study, evaluation criteria for cogeneration cycles such as energy efficiency (energy utilization factor), electric and heat powers, heat exergy power, electric production efficiency, electric and heat energy rate, exergy efficiency, artificial thermal energy efficiency and fuel energy saving ratio (nine criteria) are studied with three parameters that are the pressure ratio, the excess air rate and the inlet air temperatures for four different cogeneration cycles. Results are compared and discussed.

DESCRIPTION OF THE CYCLES

In simple cycle and inlet air cooling cycle compressed air (figure 1-2) enter the combustion chamber and in recuperated cycles (figure 3-4) compressed air is heated by hot exhaust gases in the recuperator and then enter the combustion chamber. The hot gases that exit from the combustion chamber are then expanded at the gas turbine and from the gas turbine hot gases are the source of the heat of recuperator and the heat recovery steam generator

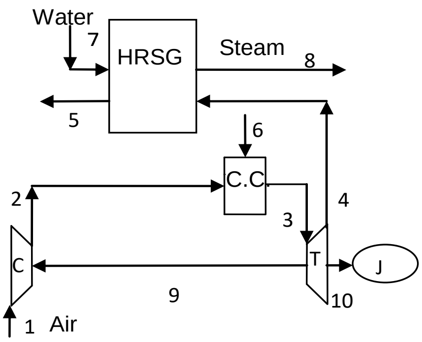

Figure 1..Simple cycle

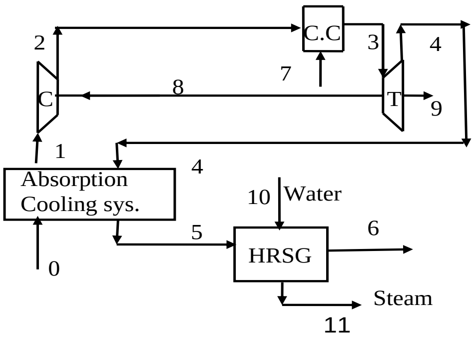

Figure 2..Inlet air cooling cycle

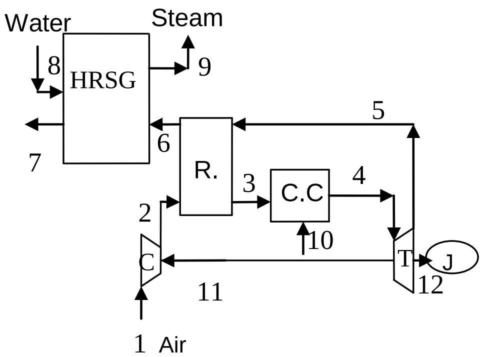

Figure 3. Air preheated cycle

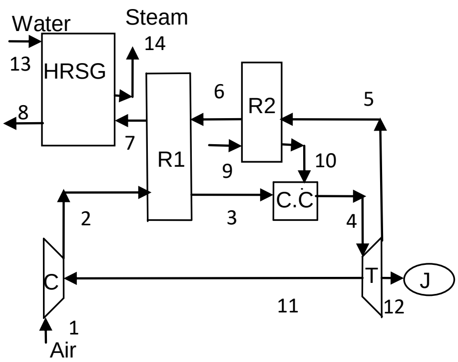

Figure 4. Air-fuel preheated cycle

ANALYSES OF THE CYCLES

The thermodynamic analysis of the cycles and their components introduced in the previous section will be done and the mathematical modeling 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 pressure losses in the combustion chamber, air preheater and HRSG are known as 5 %. The environmental conditions are taken as T_0 = 298.15 K and P_0 = 1.013 bar. 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. The gas turbine net electric power is 30 MW (net electric power is equal to the mechanic power obtained from the gas turbine minus mechanic power used by compressor), and the combustion chamber's inlet fuel is m_f = 1.64 kg/s methane. Methane LHV is taken as 802361.0 kJ/kMol.

The working fluid assumed as ideal gas, cogeneration systems operates at steady state, natural gas is taken as methane modeled as an ideal gas, the combustion is complete and N_2 is inert and heat loss for the combustion chamber is 2 % of the fuel's LHV and all other components operates without heat loss (Bejan et al., 1996; Moran and Tsatsaronis, 2000). Kinetic and potential energy effects are ignored. The outlet temperature of the heat recovery steam generator is taken as 400 K to avoid corrosive sulfuric acid formation in the exhaust. The turbine and the compressor operate adiabatically. By inserting specific entropy expressions for N_2, O_2, CO_2 and H_2O from the reference (Moran and Tsatsaronis, 2000), the combustion chamber outlet, the compressor outlet, the recuperator exhaust side outlet, the heat recovery steam generator inlet exhaust side and the gas turbine isentropic temperatures are calculated. Also the entropies of the streams are calculated from the same reference. The thermodynamic model and the calculation procedure are as follows for the CGAM cycle (air preheated cycle) (Moran and Tsatsaronis, 2000; Rosen and Dincer, 2003). Specific enthalpies and specific entropies are calculated for each stream from the equations of the reference (Bejan et al., 1996).

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

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

$$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch} \quad (3)$$

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

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

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

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

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

$$X_{N2} = \frac{0.7748}{1+\bar{\lambda}} \quad (7)$$

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

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

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

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,

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

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

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

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

Artificial thermal efficiency is the energy in the fuel supply to the cogeneration plant is supposed to be reduced by that which would be required to produce the heat load in a separate ‘heat only’ boiler of efficiency. However, the artificial thermal efficiency gives equal weight to the useful heat at different temperatures so that is not a very reasonable criterion which should be used carefully (Horlock, 1997; Feng et al., 1998).

$$ATE = \frac{\dot{W}}{\dot{Q}_f-(\frac{\dot{Q}_U}{\eta_B})} = \frac{(\eta_0)_{CG}}{1-(\frac{\dot{Q}_U}{\dot{Q}_f\eta_B})} \tag{16}$$

Fuel energy saving ratio is the comparison between the fuel required to meet the given loads of electricity and heat in the cogeneration plant with that required in a ‘reference system’ (conventional plants that meet the same load demands). Fuel energy saving ratio is also defined as the ratio of the savings to the fuel energy required in the conventional plants. Fuel energy saving ratio directly measures the extent of fuel savings which the extent of energy utilization in a cogeneration plant. Increase in the rate of the fuel energy saving ratio provides information about the electric energy increases of the cogeneration system according to the first law. For the conventional system boiler efficiency $\eta_B = 0.9$ and the electrical efficiency $\eta_{el} = 0.4$ are taken (Horlock, 1997; Feng et al., 1998).

$$FESR = (\frac{\dot{Q}}{\eta_B} + \frac{\dot{W}}{\eta_{el}} - \dot{Q}_{fuel})/(\frac{\dot{Q}}{\eta_B} + \frac{\dot{W}}{\eta_{el}}) \tag{17}$$

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	$\dot{m}_1 = \dot{m}_2$	$\dot{m}_1h_1 + \dot{W}_C = \dot{m}_2h_2$	$\dot{m}_1s_1 - \dot{m}_1s_2 + \dot{S}_{gen,C} = 0$
Combustion Chamber	$\dot{m}_3 + \dot{m}_{10} = \dot{m}_4$	$\dot{m}_3h_3 + \dot{m}_{10}h_{10} = \dot{m}_4h_4 + 0.02\dot{m}_{10}LHV$	$\dot{m}_3s_3 + \dot{m}_{10}s_{10} - \dot{m}_4s_4 + \dot{S}_{gen,CC} = 0$
Recuperator	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_5 = \dot{m}_6$	$\dot{m}_2h_2 + \dot{m}_5h_5 = \dot{m}_3h_3 + \dot{m}_6h_6$	$\dot{m}_2s_2 + \dot{m}_5s_5 - \dot{m}_3s_3 - \dot{m}_6s_6 + \dot{S}_{gen,R} = 0$
Turbine	$\dot{m}_4 = \dot{m}_5$	$\dot{m}_4h_4 = \dot{W}_T + \dot{W}_C + \dot{m}_5h_5$	$\dot{m}_4s_4 - \dot{m}_5s_5 + \dot{S}_{gen,T} = 0$
HRSG	$\dot{m}_6 = \dot{m}_7$ $\dot{m}_8 = \dot{m}_9$	$\dot{m}_6h_6 + \dot{m}_8h_8 = \dot{m}_7h_7 + \dot{m}_9h_9$	$\dot{m}_6s_6 + \dot{m}_8s_8 - \dot{m}_7s_7 - \dot{m}_9s_9 + \dot{S}_{gen,HRSG} = 0$

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	$\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9$	$\eta_{ex,HRSG} = \frac{\dot{E}_{steam,HRSG} - \dot{E}_{water,HRSG}}{\dot{E}_{in,exhaust,HRSG} - \dot{E}_{out,exhaust,HRSG}}$

RESULTS AND DISCUSSIONS

All the analysis results are presented in Figs. 5 to 15. In Figure 5 variation of energy and exergy efficiencies with pressure ratio for constant combustion temperature are given. In the same way, outlet temperature of the combustion chamber is kept constant and recuperator outlet temperature is taken 7-15 K below the turbine outlet temperature. Adding second recuperator for fuel decreases the energy efficiency however increases the exergy efficiency of the cycles. Increasing the pressure ratio of these two cycles increases the energy efficiency, but decreases the exergy efficiency for constant outlet temperature of the combustion chamber. The energy efficiency increases about 12 % but the exergy efficiency decreases about 7 % by pressure ratio range of 6 to 16.

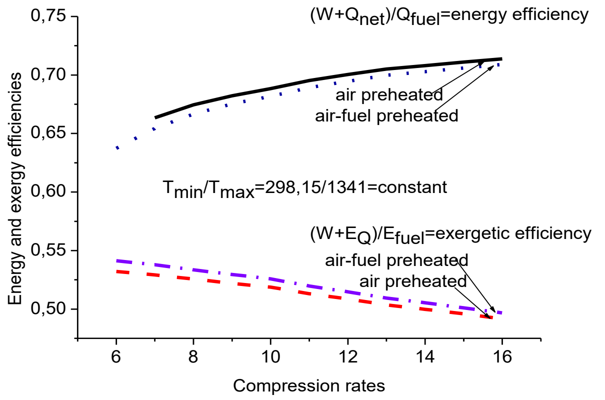

Figure 5. Variation of energy and exergy efficiencies with pressure ratio for constant combustion temperature.

In Figure 6 variation of electric and heat power with pressure ratio for variable combustion temperature and the four cogeneration cycles are given. Increasing pressure ratio increases the electric power but decreases the heat power, because increasing the pressure ratio increases combustion chamber outlet temperature which increases the turbine work but decreases the amount of heat obtained from HRSG.

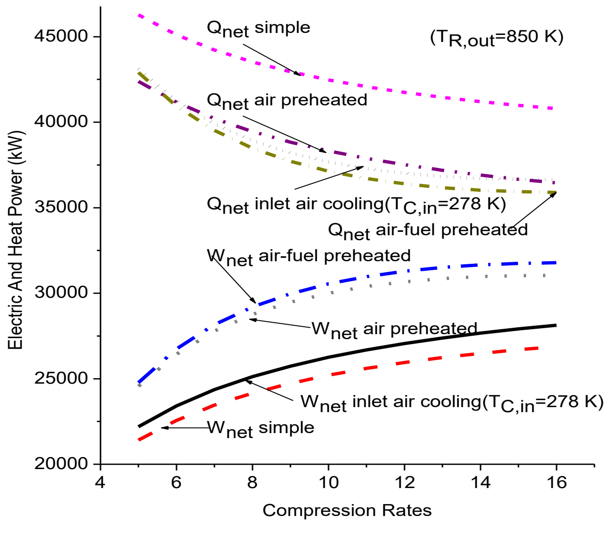

Figure 6. Variation of electric and heat power with pressure ratio for different cogeneration cycles and variable combustion temperature where m_{fuel} = 1,64 kg/s, m_{air} = 91,3 kg/s, excess air rate = 2,5, $\eta_{is,C}$ = $\eta_{is,T}$ = 0,86, $T_{R,out}$ = 850 K, T_{steam} = 485,57 K, T_{ex} = 426 K.

For simple cycle heat power is higher but electric power is lower than air and air-fuel preheated cycles. Air-fuel preheated cycle has the highest electric power but the lowest heat power among the four cycles. Electric power increases about 22 % but heat power decreases about 28 % for air-fuel preheated cycle and electric power increases 20 % but heat power decreases 11 % for simple cycle by pressure ratio range 6 to 16.

In Figure 7 variation of electric power with excess air rates for different pressure ratio are given. The inlet air cooling and 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 and the air-fuel preheated cycles increases electric power. All the cycles have higher electric power output at higher pressure ratio. However air–fuel preheated cycle works at 10 to 16 pressure ratio until at excess air rate 3 to 2.75. Electric power of air preheated cycle increase about 30 % by excess air rate range 1.3 to 3.5 at pressure ratio 10.

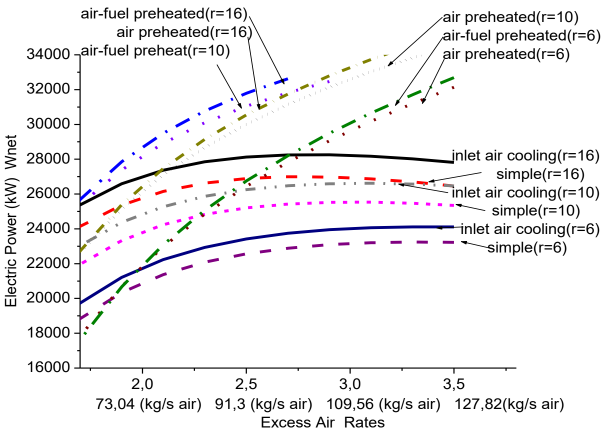

Figure 7. Variation of electric power with excess air rates for different pressure ratio.

In Figure 8 variation of heat power with excess air rates for different pressure ratio are given. Increasing the pressure ratio of the cycles decreases the heat power (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 pressure ratio and adding a recuperator increases the electric power but decreases the heat power of the cycles. This decrease is about 20 % for simple cycle and 40 % for air fuel preheated cycle at pressure ratio 6 by excess air range from 1.3 to 3.5.

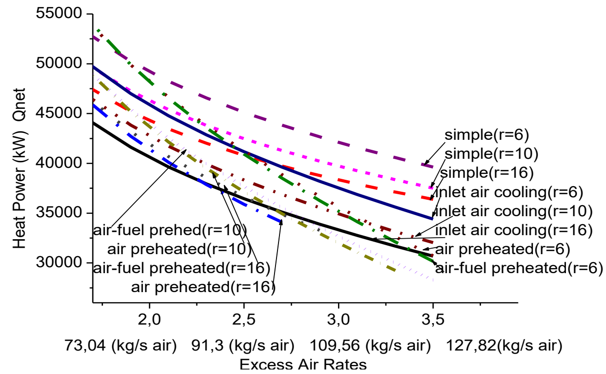

Figure 8. Variation of heat power with excess air rates for different pressure ratio.

In Figure 9 variation of electric to heat energy rate with excess air rates for different pressure ratio are given. Recuperated cycles by preheating air use some of the exhaust energy that decreases the energy of the heat recovery steam generator and that decreases the heat power. Increasing pressure ratio increases electric to heat energy rate of the four cycles, however this increase is greater than the others for the air and the air-fuel cycles.

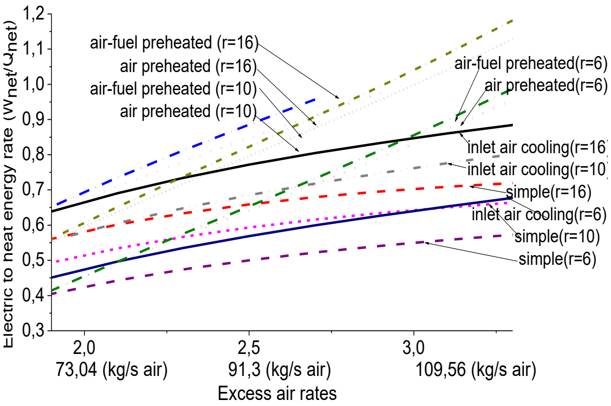

Figure 9. Variation of electric to heat energy rate with excess air rates for different pressure ratio.

Increasing excess air rates increases electric to heat energy rate for the four cycles, however this increase is greater than the others for the air and the air-fuel cycles. Electric to heat energy rate of cycles increase about 66 % for air-fuel preheat cycle (r = 6), about 61 % for air preheat cycle (r = 10) and 30 % for simple cycle (r = 10) by excess air rate range from 1.3 to 3.5.

In Figure 10 variation of heat exergy power with excess air rates for different pressure ratio are given. It shows that increasing pressure ratio increases the electric power and that decrease the heat exergy power and as explained before, increasing excess air rates decreases heat exergy power. The simple cycle has the maximum heat exergy power and the air-preheated cycle has the minimum heat exergy power among the four cycles. Decreases of heat exergy power is about 20 % for simple cycle and 43 % for air-fuel preheated cycle at pressure ratio 6 by execs air range from 1,3 to 3,5.

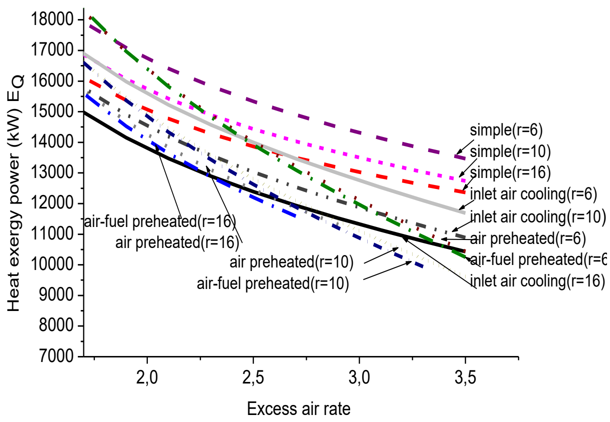

Figure 10. Variation of heat exergy power with excess air rates for different pressure ratio

In Figure 11 variation of exergetic efficiency with excess air rates for different pressure ratio are given. As can be seen in this figure that increasing pressure ratio increases the exergetic efficiency for the four cycles. The reason for this is that increasing pressure ratio increases the outlet temperature of the combustion chambers which means that increasing the inlet temperature of the turbine which increases the exergetic efficiency. The most exergetic efficient cycle is found as air-fuel preheated cycle. The exergetic efficiencies of the air-fuel preheated and air preheated cycles are continuing increasing with increasing excess air rate. Maximum efficiencies are obtained about 2 and 2.5 excess air rates for the simple and the inlet air cooling cycles. For the air and the air-fuel preheated cycles increasing excess air rates increases the exergetic efficiency. Some of the curves are cut because of the unsuitable working conditions of the systems. Exergy efficiency increases about 16 % for air fuel preheated cycle by excess air rate range 1.3 to 3.5 at 6 pressure ratio.

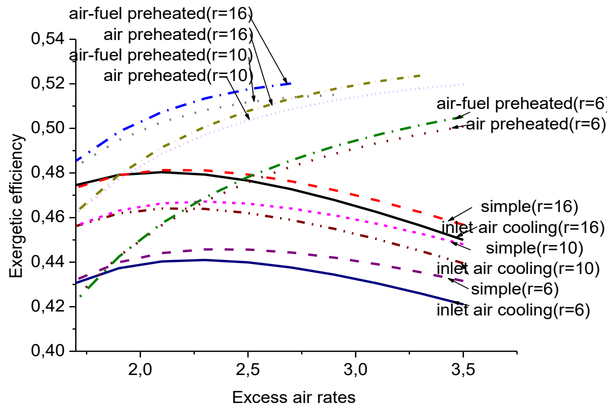

Figure 11. Variation of exergetic efficiency with excess air rates for different pressure ratio.

In Figure 12 variation of artificial thermal efficiency with excess air rates for different pressure ratio are given. It can be seen in this figure that increasing the pressure ratio increases the work obtained from systems and thus increases the artificial thermal efficiency. By the view of the artificial thermal efficiency air-fuel preheated cycle is the best cycle and the inlet air cooling cycle is the worst one among the four cycles analyzed. Increasing excess air rates decreases the artificial thermal efficiency for the four cycles. Increasing pressure ratio increases the artificial thermal efficiency for the four cycles. These decreases are about 13 % for simple cycle and about 10 % for air-fuel preheated cycle at 6 pressure ratio by excess air rate range 1.3 to 3.5.

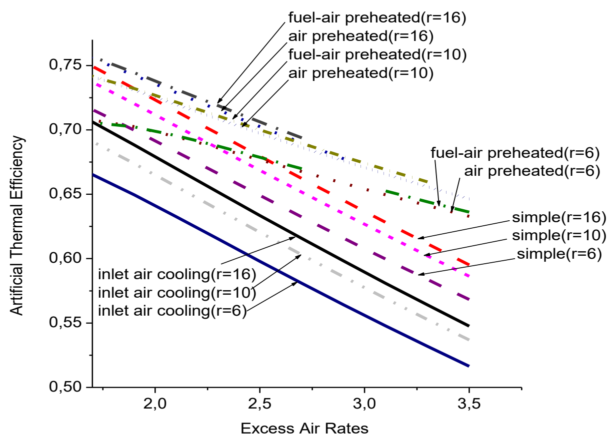

Figure 12. Variation of artificial thermal efficiency with excess air rates for different pressure ratio

In Figure 13 variations of the artificial thermal efficiency with excess air rates for different compressor inlet air temperatures are given. As can be seen that increasing the compressor inlet air temperatures increase the artificial thermal efficiency. Increasing excess air rates decreases the artificial thermal efficiency of the three cycles but for the simple cycle the decrease is greater than others. These decreases are about 17 % for simple and about 12 % for air-fuel preheated cycles at 308 K compressor air inlet temperature by excess air rate range 1.3 to 3.5.

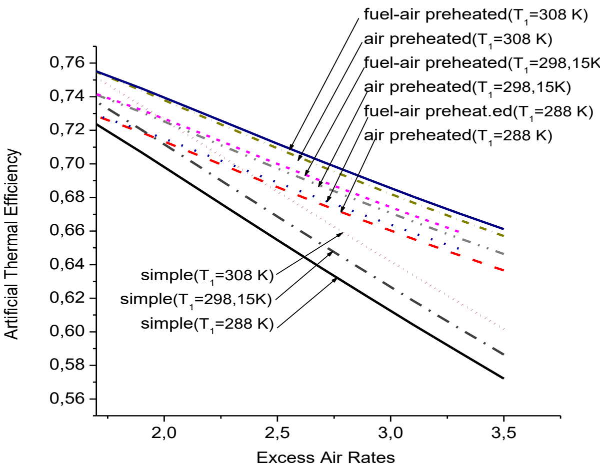

Figure 13. Variations of artificial thermal efficiency with excess air rates for different compressor inlet air temperatures

In Figure 14 variation of the fuel energy saving ratio with excess air rates for different pressure ratio are given. As can be seen here that increasing pressure ratio increase the fuel energy saving. The maximum values of the fuel energy saving ratio for the simple and the absorption cooling cycles are obtained in the excess air ratios of 2-2.5. These ratios are about 2.5-3.5 for the air and the air-fuel preheated cycles. Fuel energy saving ratio increases about 16 % for air-fuel preheated and decreases about 25 % for simple cycles at 6 pressure ratio by excess air rate 1.3 to 3.5 at pressure ratio 6.

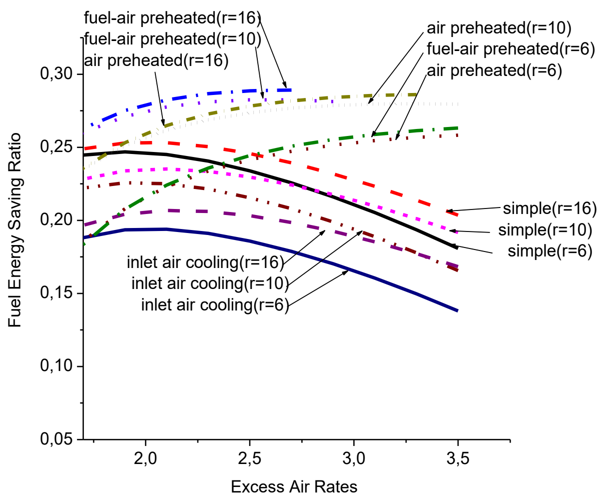

Figure 14. Variation of fuel energy saving ratio with excess air rates for different pressure ratio

In Figure 15 variations of the fuel energy saving ratio with excess air rate for different cycles and the compressor inlet air temperatures are given. Increasing the compressor inlet air temperature increases the fuel energy saving ratio of the simple cycle, however decreases the fuel energy saving ratio of the air and the air-fuel cycles but this increases and decreases are less than 4 % for maximum cases. The maximum values of the fuel energy saving ratio for the simple cycle is obtained about 2 for the excess air ratios then the fuel energy saving ratio decreases by increasing the excess air ratio.

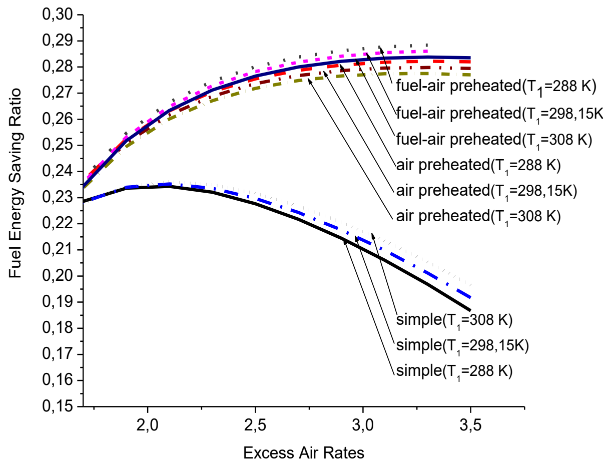

Figure 15. Variations of fuel energy saving ratio with excess air rate for different cycles and compressor inlet air temperatures.

For the air and the air-fuel cycles increasing excess air ratios increases the fuel energy saving ratio. The maximum values of the fuel energy saving ratio for the air and the air-fuel cycles are obtained between 3 and 3.5 for excess air ratios. For the low excess air rates the compressor inlet temperature is not as effective as the high excess air rates on the fuel energy saving ratio.

As can be seen in the figures given above increases in the pressure ratio, results in higher electrical power that means slight increase in the fuel causes more electricity production. It appears that the air and the air-fuel preheated cycles have much better performance than the simple and the absorption cooling cycles. Increases in the excess air rate increase the amount of the fuel for per unit electricity, which these increase are more for the absorption cooling and the simple cycles.

Adding a recuperator decreases the energy efficiency however increases the exergy efficiency of the cycles. Increasing the pressure ratio of the two cycles increases the energy efficiency, but decreases the exergy efficiency for the constant outlet temperature of the combustion chamber. Increasing the excess air rates increases the net work, the compressor work and the exhaust energy loss but decreases the heat energy, the combustion chamber outlet temperature and the energy efficiency. Increasing the pressure ratio increases the exergetic efficiency for the four cycles. The reason is that increasing the pressure ratio increases the outlet temperature of the combustion chambers which that means increasing the inlet temperature of the turbine which increases the exergetic efficiency.

CONCLUSION

In this study, most of the improving methods of the gas turbine cogeneration systems are applied step by step on a simple cogeneration system such as preheating air, preheating air and fuel, inlet air cooling by using evaporative cooling and absorption cooling then analyzed by using evaluation criteria such as the energy efficiency (the energy utilization factor), the exergetic efficiency, the electric and heat power, the electric to heat energy rate, the artificial thermal efficiency and the fuel energy saving ratio for the different air and the fuel mass ratio, and the compressing ratio and compressor inlet temperature. The performances of the cycles are compared with each other. The results presents that by changing pressure ratio from 6 to 16, the energy efficiency and electric power increase about 12 % and 22 % but exergy efficiency and heat power decrease about 7 % and 28 % respectively for air-fuel preheated cycle, electric power increases about 20 % but heat power decreases about 11 % for simple cycle.

By changing excess air rate from 1.3 to 3.5, decrease of heat power and heat exergy power are about 20 % and 20 % for simple cycle and about 40 % and 43 % for air fuel preheated cycle respectively at pressure ratio 6, electric to heat energy rate of cycles increase about 66 % for air-fuel preheat cycle (r = 6), about 61 % for air preheat cycle (r = 10) and 30 % for simple cycle (r = 10) and exergy efficiency increases about 16 % for air fuel preheated cycle at 6 pressure ratio. For excess air rate range 1.3 to 3.5, artificial thermal efficiency decreases are about 13 % for simple cycle and about 10 % for air-fuel preheated cycle at constant 6 pressure ratio and these decreases are about 17 % for simple and about 12 % for air-fuel preheated cycles at constant (308 K) compressor air inlet temperature. Fuel energy saving ratio increases about 16 % for air-fuel preheated and decreases about 25 % for simple cycles at pressure ratio 6. The effectiveness of parameters on cogeneration cycle cold be ordered as air fuel ratio, pressure ratio and compressor inlet temperature. According all the evaluating criteria the most efficient cycle is found as the air-fuel preheated cycle for obtaining more electric power and less heat power. The simple cycle is suitable for obtaining more heat power and less electric power.

***Highlights*:**

Four cogeneration cycles are analyzed by nine evaluation criteria with three parameters.

The three parameters can be listed from most effective to least effective as air fuel ratio, pressure ratio and compressor inlet temperature.

The most efficient cycle is found as the air-fuel preheated cycle for obtaining more electric power.

The simple cycle is found to be the most suitable one for obtaining more heat power and less electric power.

REFERENCES

ASHRAE, 2000, 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.

Atmaca, M., 2010, Efficiency Analysis of Combined Cogeneration Systems with Steam and Gas Turbines, *Energy Sources, Part A: Recovery, Utilization, and Environmental Effects,* 33:4, 360-369, DOI: 10.1080/15567031003741434

Atmaca, M., Yilmaz, E., Kurtluş AB., 2016, Application of Cogeneration on a Housing Complex, *Journal of Clean Energy Technologies,* volume 4,No.2, pp 129-135, DOI: 10.7763/JOCET.2016.V4.266

Atmaca, M., Gumus, M., Inan, A.T., Yilmaz, T., 2009, Optimization of Irreversible Cogeneration Systems under Alternative Performance Criteria, *International Journal of Thermophysics,* Vol:30, issue:5, pp: 1724–1732, DOI:10.1007/s10765-009-0621-3

Bejan, A., Tsatsaronis, G., and Moran, M., 1996, *Thermal Design and Optimization*, Wiley Pub, New York.

Boyce, M.P., 2002, Handbook for Cogeneration and Combined Cycle Power Plants, *Asme Pres*s, 42, 220, New York.

Feng, X., Cai, Y.I., and Qian, L.L., 1998, A New Performance Criterion For Cogeneration System, *Energy Convers. Mgmt.,* 39, 1607-1609.

Horlock, J.H., 1997, Cogeneration-Combined Heat and Power (CHP), *CRİEGER Pub.,Florida*.

Huang, F.F., 1990, Performance Evaluation of Selected Combustion Gas Turbine Cogeneration Systems Based on First and Second-Law Analysis, *Journal of Engineering for Gas Turbines and Power*, 112, 117-121.

Jaluria, Y., 2008, *Design and Optimization of Thermal Systems,* Second ed. CRC Press, New York.

Karaali, R., 2010, Thermoeconomic Optimization of Cogeneration Power Plants, PhD Thesis, Kocaeli Univ.

Karaali, R., and Ozturk, I.T., 2015, Thermoeconomic optimization of gas turbine cogeneration plants. *Energy* 80, 474-485.

Khaliq, A., and Kaushik, S.C., 2004, Thermodynamic Performance Evaluation of Combustion Gas Turbine Cogeneration System with Reheat, *Applied Thermal Engineering* 24, 1785-1795.

Lazzaretto, A., and Tsatsaronis, G., 2006, SPECO: A Systematic and General Methodology for Calculating Efficiencies and Costs in Thermal Systems, *Energy* 31, 1257-1289.

Malinowska, W., and Malinowski, L., 2003, Parametric Study of Exergetic Efficiency of A Small-Scale Cogeneration Plant Incorporating A Heat Pump, *Applied Thermal Engineering* 23, 459-472.

Moran, J.M., and Tsatsaronis, G., 2000, *The CRC Handbook of Thermal Engineering*, CRC Press LLC., 15-109, Florida.

Najjar, Y.S.H., 2000, Gas Turbine Cogeneration Systems: A Review of Some Novel Cycles, *Applied Thermal Engineering* 20, 179-197.

Najjar, Y.S.H., 2001, Efficient Use of Energy by Utilizing Gas Turbine Combined Systems, *Applied Thermal Engineering* 21, 407-438.

Rosen, M.A., and Dincer, I., 2003, Exergy-Cost-Energy-Mass Analysis of Thermal Systems and Processes, *Energy Convers. Mgmt.*, 44, 1633-1651.

Santo, D.B.E., and Gallo, W.L.R., 2000, Predicting Performance of a Gas Turbine Cogeneration System with Inlet Air Cooling, Ecos2000 Proceedings, Universiteit Twente, Nederland.

Sue, D.C., and Chuang, C.C., 2004, Engineering Design and Exergy Analyses for Combustion Gas Turbine Based Power Generation System, *Energy* 29, 1183-1205.

Wang, F.J., and Chiou, JS., 2002, Performance Improvement for a Simple Cycle Gas Turbine GENSET-a Retrofitting Example, *Applied Thermal Engineering*, 22, 1105-1115.

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