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Analysis of Steam Injection into Combustion Chamber of Gas Turbine Cogeneration Cycles

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ABSTRACT

Cogeneration is known as the generation of heat energy and electricity at same time by using the fuel's energy. There are various cogeneration systems, and steam injection is made into combustion chamber to increase the efficiency of the cycle and to reduce nitrogen oxide emissions. The most fundamental thermodynamic, operational, economical and thermo-economic factors must be considered when choosing the appropriate cogeneration system and designing the system. For the thermodynamic factors, such as the amount of fuel to be consumed, the electric heat rate, the artificial thermal efficiency, the fuel energy gain rate must be found for the unit amount of electric power to be obtained. The cost and availability of the fuel to be used must also be estimated by considering the problems will be affected by repair maintenance and economic fluctuations. In economic factors, the annual cash flow of the system and the amortization itself are calculated. In the thermo-economic factors, the investment costs depend on the exergy efficiency and the exergy of the products of the devices and the fuel required to operate are calculated. In this study, the analysis of steam injection into cogeneration systems according to performance and evaluation criteria was done using energy, exergy and economic methods. To calculate the energy and exergy values of the flows, a program was written by the authors in the FORTRAN programming language and the results obtained by running them were used. The results obtained were compared with the literature values and correctness was observed.

Keywords:
Cogeneration; Performance; Evaluation Criteria

INTRODUCTION

As is known, the usage of electric energy continues to increase in our country and in the world. Industrial establishments set up cogeneration facilities to meet both electricity and heat energy needs, and provide a more efficient use of fuels. Thus, energy consumption reduces their costs and they are also avoided of electricity interruptions. Small-scale cogeneration facilities, however, are also widely used in small businesses, university campuses, hotels, and district heating systems.

In the cogeneration plants, electrical energy and heat energy are produced at the same time so that higher efficiencies can be obtained. The energy efficiency of such plants is approaching 90%. If these plants are used especially for heating of the houses, cooling water can be produced by operating an absorption cooling group with the help of exhaust heat even outside the heating season. In this regard, the production is directed to three purposes, so three generation is the issue and the annual usage period of the cogeneration plant is increasing [1, 15, 19]. When these advantages are evaluated, more economical energy production can be saved. In addition, less pollution is created and the amount of CO_2 released to the atmosphere is lower. Different cogeneration systems are available. These can be classified as steam turbines, gas turbines, internal combustion diesel or gas engines and fuel cells. Different methods are proposed and used in the design stage and usage stage to increase the efficiency of the selected cogeneration plants according to the usage purposes.

Steam injection into the cycle's combustion chamber has continued to be implemented since in the 1950s with the aim of reducing the outlet temperature and increasing the work force achieved, and now reducing NOx and its compounds to minimum levels. Studies of exergy analysis in this area have been started by many researchers since the 1980s [24].

The concept of thermoeconomics was first introduced in the 1960s and later developed by C. Frangopoulos, G.Tsatsaronis, A. Valero and M.Spakovsky, especially in the 1990s [17, 20, 22]. By describing a recuperative gas turbine cogeneration system called the CGAM problem (consisting of the initials of their names), as a simple and defined optimization problem, they propose their own optimization methods by comparing their respective solution paths. This system has a steam capacity of 30 MW electricity and saturated steam at 14 kg/s, 2 MPa, and each researcher's optimization method gave similar results.

Lazzeratto and Tsatsaronis developed cost equations, that are named SPECO / AVCO-specific cost / avarege cost approach [12, 13, 21, 23]. Kim et al., proposed Modified productive structure analysis (MOPSA), and Rosen and Dincer, proposed Exergy cost energy mass analysis (EXCEM) methods [16]. According to El-Sayed and Gaggioli, thermoeconomics implements two basic methods [6]. These are integral calculation methods and algebraic method. The algebraic method always uses the cost equations of the devices, and gives information on the average cost. In the integral calculation method, flow costs are calculated for each flow and device using differential equations and marginal costs are found. The Lagrange multipliers method is used most frequently [23]. The definition of cost equations for devices in the algebraic method is not objective.

The method of integral calculus is also subjective. Because, it is based on the Lagrange multipliers technique and is based on the definition of the mathematical function of each device, and when the isolation of the devices is not successful, there would be major errors in the iterative steps. For the solution of this problem, C. Frangopoulos proposed the thermoeconomic functional approach in his doctoral dissertation in 1983. Accordingly, a function and a product are defined for each device, to eliminate the need of a cost equation [7].

Cerqueira and Nebra compared the CGAM cycle by using the four thermoeconomic analyzes. According to this, the thermoeconomic functional approach gives a result in the middle among others (7.1 $ / GJ), and the exergetic costs method gives the most expensive power cost (8.2 $ / GJ). The exhaust gas cost is taken as zero at the exergy cost. The result is that all the irreversibilities are found in the heat exchanger [3]. Kwak et al. applied MOPSA and SPECO / AVCO methods to solve the CGAM problem, and compared these findings with the findings of Torres et al. [10]. According to Torres et al., for per the unit cost, $ 7.42 $ / GJ for their method, 8.46 $ / GJ for the MOPSA and 7.80 $ / GJ for the SPECO / AVCO methods.

According to Boyce, steam injected into the combustion chamber injects about 2-3% of the air mass for reducing and controlling NOx, which increases the electric power obtained by about 3-5%. When steam mass injected into the combustion chamber about 5% of the air mass, the electricity efficiency increases in the sample cycle 8.3% and if this steam is produced from the exhaust of the turbine, the electricity efficiency is increased by 19%. According to the same article, if the amount of water or steam is about 12% of the amount of air, the electric power obtained increases by about 25% [2]. Kehlhofet et al., draws curves showing the effect of the amount of injected steam on relative work and relative efficiency, where the amount of work obtained is 14% when the steam fuel ratio is 1.5 [9]. According to Wang and Chiou, the amount of steam to be injected can be up to 20% of the mass of the compressor inlet air [24]. The energy efficiency does not change when the compression ratio in the steam injection cycle increases from 5 to 20 in the study. In addition, the same researchers analyzed the use of regeneration and steam injection methods to increase the efficiency of a simple gas turbine power generation system based in Taiwan. All the findings of Wang and Chiou are consistent with the findings in this study.

In this study, the sample, the air preheated and the air fuel preheated cogeneration systems are analyzed by using the first and second laws of thermodynamics, and the required income methods. The cost of the main product is calculated in four steps in the economic analysis with the required income method. These are cost accounting and forecasting of the total investment, determination of economic, financial, labor and market input parameters for the detailed cost account, calculation of total income needed and calculation of product cost with these values.

In the calculations the compression ratios, the compressor and turbine isentropic efficiencies the combustion chamber outlet temperatures, the change in air flow, fuel flow and the recuperator outlet temperatures are taken into consideratin. Thus, the effects of steam injection into these three different cycles, design differences and the effect of each added device on optimum values are also investigated. The thermoeconomic analysis of the sample, air, and fuel air preheated cycles, for the injection of steam into their combustion chamber, for the different air fuel and compression ratios are done, and the performance curves obtained and compared. For this, two separate studies have been carried out, namely the performance of all the systems and the performance of each device that constitutes the systems. The

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effects of various air fuel and compressor compression ratios on the power, the efficiency, the cost of obtained products, the artificial thermal efficiency, the fuel energy gain, the electric heat ratios and fuel consumption have been drawn and related curves are plotted.

Many advanced computer programs exist in the market to be used in performing the analyze described in this study and can be grouped into two groups, one approaching solving sequential modules and equations systems. In sequential module approach programs, the devices are combined by selecting from the menu, the input values are given and the program is executed and results are obtained. These are ready-made visual programs, such as ASPEN PLUS, PROCESS, CHEMCAD programs are such programs. The SPEED UP and the EESP programs are solving the equations of the systems of by establishing a mathematical model of each device so that the mathematical model of the system is revealed as hundreds of equations (or set of equations) and solved for the common variables. Here, as a two-tuple synthesis, mathematical and economic models of systems with separate sequential modules are developed in the FORTRAN programming language, executed, and the results obtained are discussed.

MATERIALS AND METHODS

In conventional systems, heat is generated at two separate sites to produce power and heat. In the cogeneration system, heat is generated by a single heat generation system. Electric energy is generated by the energy carrier fluid and the remaining energy is used for the production of heat (steam or hot water). The obtained total energy of the conventional system is around 50%, while the efficiency of the cogeneration plant is around 80-90%. In addition, by installing a cogeneration plant, the operating and initial investment costs of the system can be significantly reduced [25]. In gas turbine cogeneration plants, the main machine is the gas turbine.

As can be seen in Figure 1, after the high-pressure air from the compressor is burned with methane gas in the combustion chamber, and after that some of the energy of the exhaust gases are converted to electrical energy in the gas turbine. After that the resulting high-temperature exhaust gases are released by leaving a large portion of the remaining energy in the heat exchanger to the water [1, 18]. The obtained hot water is used for steam heating, drying, meeting the process requirement, generating steam by using steam turbine, absorbing cooling and similar processes. Different cycles are obtained by adding other devices to the main machine such as recuperator, steam injector, heat exchanger, absorption cooling, and steam turbine [1].

In Figure 1 steam injection of a) air preheated, b) air-fuel preheated, and c) sample cycles are shown. Cogeneration plants consist of different devices, in which temperature, pressure, chemical composition change. There is also a chemical reaction in the combustion chamber. The assumptions made in the analysis of the systems in this thesis are as follows [1].

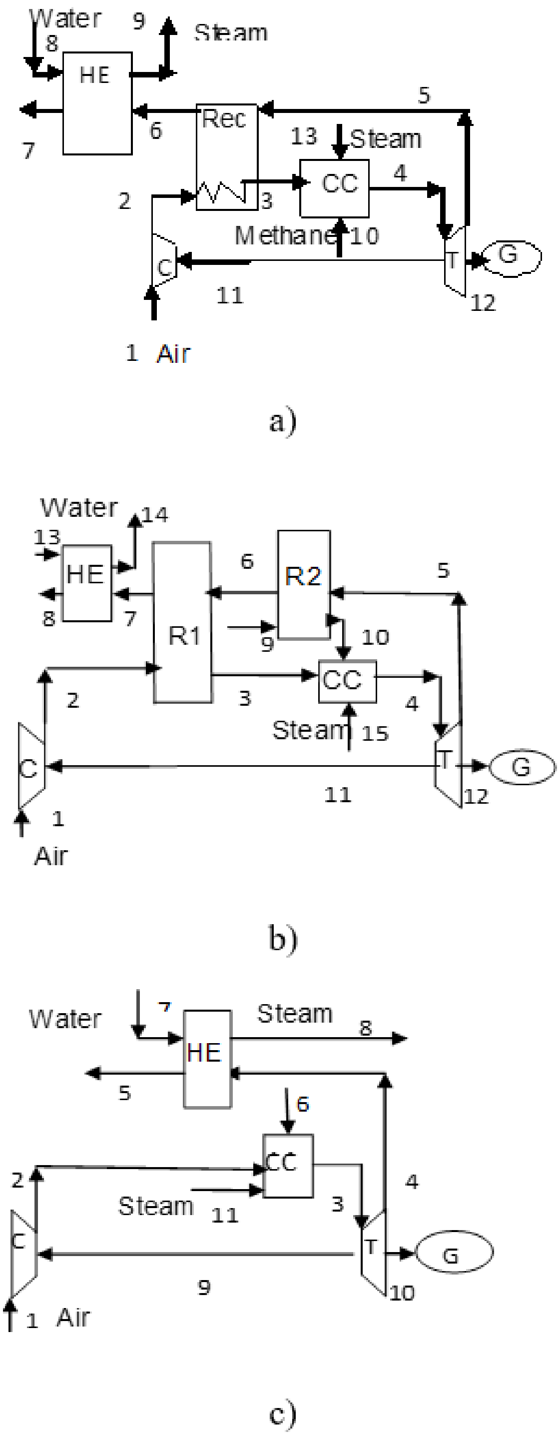

Figure 1. Steam injection of **a.** air preheated, **b.** air-fuel preheated, and **c.** sample cycles.

Table 1. Mass, energy and enropy equations of each devices of the air preheated cycle [4, 5].

Devices	*Mass equations*	*Energy equations*	*Enropy equations*
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	$\dot{m}_3 + \dot{m}_{10} + \dot{m}_{13} = \dot{m}_4$	$\dot{m}_3 h_3 + \dot{m}_{10} h_{10} + \dot{m}_{13} h_{13} - 0.02\dot{m}_{10} LHV = \dot{m}_4 h_4$	$\dot{m}_3 s_3 + \dot{m}_{10} s_{10} + \dot{m}_{13} s_{13} - \dot{m}_4 s_4 + \dot{S}_{gen,CC} = 0$
Turbine	$\dot{m}_4 = \dot{m}_5$	$\dot{m}_4 h_4 = \dot{W}_T + \dot{W}_C + \dot{m}_5 h_5$	$\dot{m}_4 s_4 - \dot{m}_5 s_5 + \dot{S}_{gen,T} = 0$
Heat exchanger	$\dot{m}_6 = \dot{m}_7$ $\dot{m}_8 = \dot{m}_9$	$\dot{m}_6 h_6 + \dot{m}_8 h_8 = \dot{m}_7 h_7 + \dot{m}_9 h_9$	$\dot{m}_6 s_6 + \dot{m}_8 s_8 - \dot{m}_7 s_7 - \dot{m}_9 s_9 + \dot{S}_{gen,HE} = 0$
All cycle	$\bar{h}_i = f(T_i)$ $\bar{s}_i = f(T_i, P_i)$ $\dot{m}_{air} h_{air} + \dot{m}_f LHV_{CH4} - \dot{Q}_{Loss,CC} - \dot{m}_{eg.,out} h_{eg.out} - \dot{W}_T - \dot{m}_{steam}\left(h_{W,in} - h_{steam,out}\right) = 0$ $\dot{Q}_{Loss,CC} = 0.02\dot{m}_f LHV_{CH4}$		

The cogeneration system operates in a steady state regime, the ideal gas mixture laws are valid for air and exhaust, methane is chosen as fuel and it is accepted as ideal gas, the combustion is complete, there is no NOx formation and the heat loss in the combustion chamber is 2% of the upper heat value of the fuel.

There is no heat loss in other devices and kinetic and potential energy effects are not considered. In addition, the environmental conditions are taken as follows; T_0 = 298.15 K and P_0 = 1.013 bar, pressure loss for combustion chamber, recuperator and heat exchanger 5% and capacity for compressor m_1 = 91.4 kg / s, heat exchanger m_v = 14 kg / s saturated vapor pressure 20 bar, gas turbine net electric power 30 MW, combustion chamber fuel mass m_f = 1.64 kg / s methane. The thermodynamic model and calculation procedure

Table 2. Exergy and exergy efficiency equations of each devices of the air preheated cycle [11, 14].

Devices	*Exergy equations*	*Exergy efficiency*
Compressor	$\dot{E}_{D,C} = \dot{E}_1 + \dot{W}_C - \dot{E}_2$	$\eta_{ex,C} = \dfrac{\dot{E}_{O,C} - \dot{E}_{I,C}}{\dot{W}_K}$
Recuperator	$\dot{E}_{D,R} = \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6$	$\eta_{ex,R} = \dfrac{\dot{E}_{O,air,R} - \dot{E}_{I,air,R}}{\dot{E}_{O,eg,R} - \dot{E}_{I,eg,R}}$
Combustion chamber	$\dot{E}_{D,CC} = \dot{E}_3 + \dot{E}_{10} + \dot{E}_{13} - \dot{E}_4$	$\eta_{ex,CC} = \dfrac{\dot{E}_{O,CC}}{\dot{E}_{I,CC} + \dot{E}_{Fuel}}$
Turbine	$\dot{E}_{D,T} = \dot{E}_4 - \dot{E}_5 - \dot{W}_C - \dot{W}_T$	$\eta_{ex,T} = \dfrac{\dot{W}_{net,T} + \dot{W}_C}{\dot{E}_{I,T} - \dot{E}_{O,T}}$
Heat exchanger	$\dot{E}_{D,HE} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9$	$\eta_{ex,HE} = \dfrac{\dot{E}_{Steam,HE} - \dot{E}_{Water,HE}}{\dot{E}_{I,eg,HE} - \dot{E}_{O,eg.,HE}}$
All cycle	$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch}$ $\dot{E}_{ph} = \dot{m}\left(h - h_0 - T_0\left(s - s_0\right)\right)$ $\dot{E}_{ch} = \dfrac{\dot{m}}{M}\left\{\sum x_k \bar{e}_k^{ch} + \bar{R}T_0 \sum x_k ln x_k\right\}$ $\eta_{ex} = \dfrac{\dot{W}_{net,T} + (\dot{E}_{Steam,HE} - \dot{E}_{Water,HE})}{\dot{E}_{Fuel}}$	

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are given in Table 1 and Table 2 for the air preheated cycle. The combustion equation is taken as follows.

$$\bar{\lambda}CH_4 + \begin{bmatrix} 0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + \\ 0.019H_2O \end{bmatrix} \rightarrow$$
$$\left(1+\bar{\lambda}\right)\left[X_{N_2}N_2 + X_{O_2}O_2 + X_{CO_2}CO_2 + X_{H_2O}H_2O\right]$$

The cost of the main product is calculated in four steps in the economic analysis with the required income method. These are cost accounting and forecasting of the total investment, determination of economic, financial, labor and market input parameters for the detailed cost account, calculation of the total income needed and calculation of the product cost with these values. The CEPCI equipment index (CHEMICAL ENGINEERING PLANT COST INDEX) was used to find the current values of past device prices.

$$C_{EQ} = C_{ref}\left(2016CEPCI\ EQ.IN./1994CEPCI\ EQ.IN.\right) \quad (1)$$

There are three methods used for cost estimation of the purchased equipment: cost indices, cost estimating charts, and calculation effect of size on equipment. The last one is used in this study. For the overall system operating at steady state the cost balance is given as

$$C_{ref-year} = C_{ref}\left(\dot{E}_{net}/\dot{E}_{ref}\right)^{a} \quad (2)$$

$$\dot{C}_{P,tot} = \dot{C}_{f,tot} + \dot{Z}_{tot}^{CI} + \dot{Z}_{tot}^{OM} \quad (3)$$

In this equation, Z is non exergy related cost rate, C is cost rate, f is fuel, CI is capital investment, P is product, OM is operating and maintenance and tot is total [1, 26, 27]. The details of the calculation can be found in literatur [1, 8].

RESULTS AND DISCUSSION

Table 3 shows the net work, net heat energy, loss of energy, compressor work, air and exhaust mole numbers, combustion chamber outlet temperature, for the air preheated cogeneration cycles. Also energy efficiency and energy balances are given.

Accordingly, as the amount of steam injected increases, the net work, the energy withdrawn from the boiler, the net heat energy, energy efficiency and combustion chamber outlet temperature are decreases for the sample, the air and the fuel air preheated cogeneration cycles.

In Figure 2, it can be seen that increasing the injected steam flow increases the electrical power of the systems. The increases the injected steam increase the flow rate entering the turbine, and the work obtained from the turbine are increased.

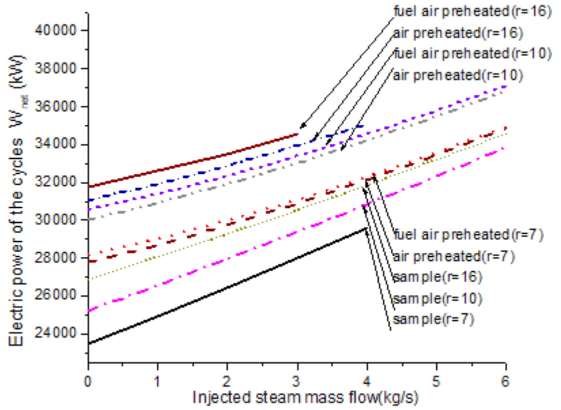

Figure 2. Variations of electric power of the cycles with steam injected mass flow. (m_{fuel}=1,64 kg/s, m_{air}=91,3 kg/s, $\eta_{isC}=\eta_{isT}$=0,86, T_1=298,15 K, $T_{rec,out}$=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K).

In steam injected cycles, increasing the compressor compression ratio increases, the amount of the electricity. In some of the cycles, the curves are cut off at certain mass flow, since the heat energy required for the operation is not provided.

Figure 3 shows the change in the electric heat energy ratio of the systems with the injected steam flow at different compressor compression ratios. As the injected steam flow increases, the heat power decreases rapidly and the electric power increases, so the electric heat energy rate increases rapidly. It is understood that the compression ratio is very effective in the ratio of electric heat energy to the air and the fuel air preheated cogeneration cycles.

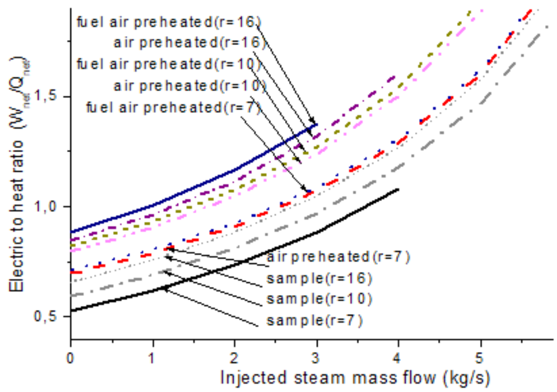

Figure 3. Variations of electric to heat ratio of the cycles with steam injected mass flow. (m_{fuel}=1,64 kg/s, m_{air}=91,3 kg/s, $\eta_{isC}=\eta_{isT}$=0,86, T_1=298,15 K, $T_{rec,out}$=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K).

Table 3. Variations of steam injection mass flow with some parameters and the energy balance of the steam injected sample, air preheated and fuel air preheated cycles.

	HHV	Sample cycle (m_{air}=91.3 kg/s)						
Steam(Kg/s)	Fuel En.(kW)	W_{net}(kW)	Q_{net}(kW)	Q_{egz}(kW)	CC Loss**(kW)	WC(kW)	CC outlet temp.(K)	I.Law Efficiency
0	91011	25166	42339	24369	1820	29675	1303	0,7415
1	91011	26513	38325	27046	1820	29675	1287	0,71216
2	91011	27908	34256	29727	1820	29675	1271	0,68281
3	91011	29342	30148	32408	1820	29675	1256	0,65347
4	91011	30809	26008	35090	1820	29675	1242	0,62412
5	91011	32300	21844	37771	1820	29675	1228	0,59477
6	91011	33809	17662	40453	1820	29675	1214	0,56542
	Inlet En.=Outlet En. - Air En.(2663 kW)				Egzhaust Mol N.		Air Mol N.	
0	91011=91031				3,2891		3,1869	
1	91011=91040				3,34463		3,1869	
2	91011=91048				3,40014		3,1869	
3	91011=91056				3,45565		3,1869	
4	91011=91064				3,51116		3,1869	
5	91011=91072				3,56667		3,1869	
6	91011=91080				3,62218		3,1869	
	HHV	Air preheated cycle (m_{air}=91.3 kg/s)						
Steam(Kg/s)	Fuel En.(kW)	W_{net}(kW)	Q_{net}(kW)	Q_{egz}(kW)	CC Loss**(kW)	WC(kW)	CC outlet temp.(K)	I.Law Efficiency
0	91011	29977	37526	24369	1820	29675	1519	0,74145
1	91011	30863	33967	27051	1820	29675	1498	0,7121
2	91011	31870	30286	29732	1820	29675	1479	0,6828
3	91011	32982	26501	32414	1820	29675	1460	0,6534
4	91011	34186	22624	35095	1820	29675	1442	0,6241
5	91011	35469	18667	37777	1820	29675	1424	0,5947
6	91011	36821	14643	40458	1820	29675	1407	0,5654
	Inlet En.=Outlet En. - Air En.(2663 kW)				Egzhaust Mol N.		Air Mol N.	
0	91011=91029				3,28912		3,186897	
1	91011=91038				3,34463		3,186897	
2	91011=91046				3,40014		3,186897	
3	91011=91054				3,45565		3,186897	
4	91011=91062				3,51116		3,186897	
5	91011=91070				3,56667		3,186897	
6	91011=91078				3,62218		3,186897	
	HHV	Fuel air preheated cycle (m_{air}=91.3 kg/s)						
Steam(Kg/s)	Fuel En.(kW)	W_{net}(kW)	Q_{net}(kW)	Q_{egz}(kW)	CC Loss**(kW)	WC(kW)	CC outlet temp.(K)	I.Law Efficiency
0	91011	30519	36984	24370	1820	29675	1542	0,7414
1	91011	31342	33487	27051	1820	29675	1521	0,7121
2	91011	32295	29861	29732	1820	29675	1501	0,68276
3	91011	33361	26122	32414	1820	29675	1481	0,65341
4	91011	34526	22283	35095	1820	29675	1463	0,62406
5	91011	35777	18359	37777	1820	29675	1445	0,59471
6	91011	37103	14359	40458	1820	29675	1428	0,56536
	Inlet En.=Outlet En. - Air En.(2663 kW)				Egzhaust Mol N.		Air Mol N.	
0	91011=91030				3,2891		3,1869	
1	91011=91037				3,34463		3,1869	
2	91011=91046				3,40014		3,1869	
3	91011=91054				3,45565		3,1869	
4	91011=91062				3,51116		3,1869	
5	91011=91070				3,56667		3,1869	
6	91011=91078				3,62218		3,1869	

*%1,9 of the inlet air is accepted as steam and the steam in the egzhaust is accepted as condensed (HHV) so that the condensation energy of the steam in the air is taken into consideration.

**Combustion chambers heat loss is taken as % 2 of the HHV of the fuel.

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In Figure 4, variations of the exergy efficiency of the cycles with injected steam rate are given. For the air and the fuel air preheated cogeneration cycles, as the injected steam flow increases, the exergy efficiency is decreases. In the case of the sample cycle, higher efficiency is obtained because higher temperature is reached at high compression ratios and efficiency decreases as the amount of injected steam increases.

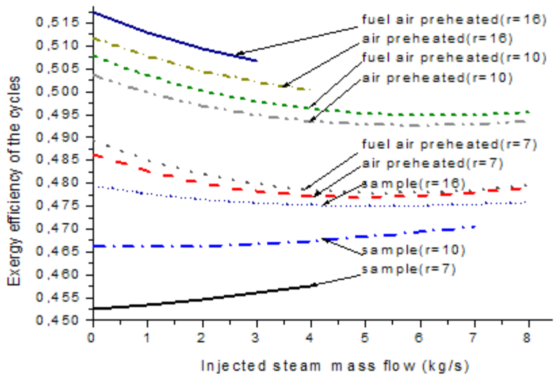

Figure 4. Variations of the exergy efficiency of the cycles with injected steam rate. (m_{fuel}=1,64 kg/s, m_{air}=91,3 kg/s, η_{isC}=ηisT=0,86, T_1=298,15 K, $T_{rec,out}$=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K).

As can be seen in Figure 5, as the injected steam flow increases, the obtained work increases, but reduces the artificial thermal efficiency of the systems.

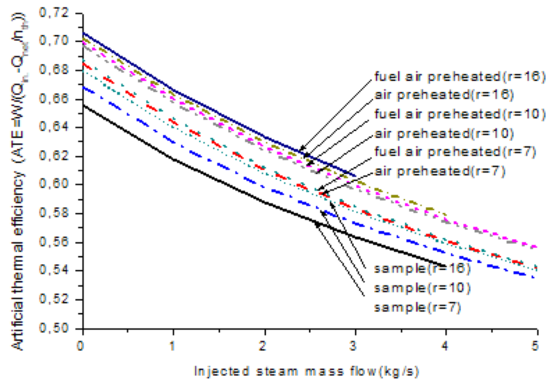

Figure 5. Variations of artificial thermal efficiency with injected steam rate. (ATE=$W/(Q_{in}-Q_{net}/\eta_{th})$) ($m_{fuel}$=1,64 kg/s, η_{isC}= η_{isT}=0,86, T_{recout}=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K, T_0=298,15 K).

Figure 6 shows that as the injected steam flow increases at different compression ratios, the fuel energy gain rate of the systems decreases. That is why the heat energy obtained is thrown into the combustion chamber in the form of vapor. Decreasing the compression ratios also decreases the fuel-to-energy ratio.

In Figure 7, the variations of the cost of electricity produced by the systems at different compression ratios with

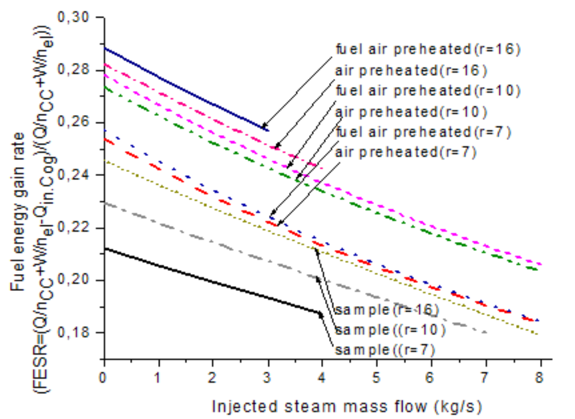

Figure 6. Variations fuel energy gain rate with injected steam rate of the compression rates. (FESR=$(Q/\eta_{CC}+W/\eta_{el}-Q_{in,Cog})/(Q/\eta_{CC}+W/\eta_{el})$) ($m_{fuel}$=1,64 kg/s, m_{air}=91,3 kg/s, η_{isC}=η_{isT}=0,86, T_1=298,15 K, $T_{rec,out}$=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K).

the injected steam flow are given. Here too, curves are obtained for the operating conditions in which the cycles can run, and curves are cut off if they cannot. The cost of electricity generated by the cycles increases as the amount of injected steam increases. The amount of increase is similar in character.

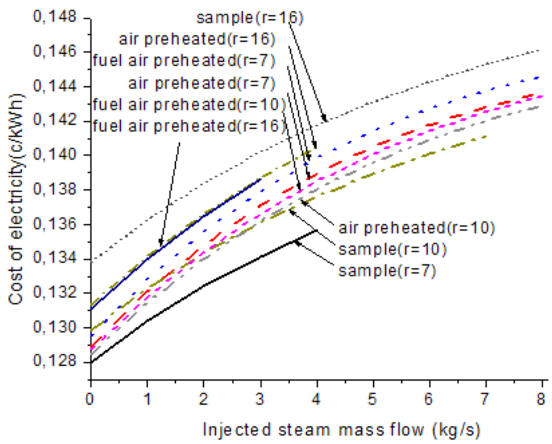

Figure 7. Variations of the cost of electricity with injected steam rate. (m_{fuel}=1,64 kg/s, m_{air}=91,3 kg/s, η_{isC}=η_{isT}=0,86, T_1=298,15 K, $T_{rec,out}$=850 K, T_{steam}=485,57 K, $T_{egzhaust}$=426 K).

CONCLUSION

Various cogeneration systems exist and in gas turbine cogeneration plants steam injection is made in the combustion chamber to increase the efficiency of the cycle and reduce nitrogen oxide emissions. As the amount of steam injected into the the sample, the air and the fuel air preheated cogeneration cycles increases, the net work, the energy withdrawn from the boiler and the number of moles exhausted increases, and the net heat energy, energy consumption and combustion chamber output temperature are decreases. Only a fraction of the heat energy obtained from the exhaust gas emitted from the injected steam turbine is produced in the waste heat recovery device and then exhausted to the surroundings at 426 K. This reduces the amount of heat generated while increasing the amount of electricity generated.

At different compression ratios, it is found for all the cycles that increasing the amount of injected steam increases the electricity obtained. Since the injection of the steam obtained in the waste heat recovery device into the combustion chamber increases the flow rate entering the turbine, the work obtained from the turbine is increased. In some of the cycles, the curves are cut off at certain debts, since the heat energy required for the operation of the waste heat recovery apparatus is not provided.

As the injected steam flow increases, the heat power decreases rapidly and the electric power increases, so the electric heat energy rate increases rapidly. It is understood that the compression ratio is very effective in the ratio of electric heat energy to air and fuel air preheated cogeneration cycles.

In the air and the fuel air preheated cogeneration cycles, as the injected steam flow increases, the exergy efficiency is reduced. In the case of the sample cycle, higher efficiency is obtained because higher temperature is reached at high compression ratios and efficiency decreases as the amount of injected steam increases. But the sample cycle is worse than the other ones in all conditions.

As the amount of injected steam increases, the amount of fuel consumed for the work increases rapidly, which reduces the artificial thermal efficiency of the systems. As the injected steam flow increases at different compression ratios, the fuel energy gain rate of the systems decreases. The reason of that is the heat energy obtained is thrown into the combustion chamber in the form of vapor. Reduced compression ratios also reduce the fuel-to-energy ratio. Curves are obtained for the operating conditions in which the cycles can run, and curves are cut off if they cannot. The cost of electricity generated by cycles increases as the amount of injected steam increases. The amount of increase is similar in character.

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