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PARAMETRIC ANALYSIS OF AN EXTERNALLY FIRED-GAS TURBINE (EFGT): EFFECT OF MAIN PARAMETERS OVER THE POWER GENERATED, CYCLE EFFICIENCY AND EXERGY DESTRUCTION

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Abstract. *The use of externally fired gas turbines (EFGT) has opened the gates for the use of relative low-quality fuels, such as biomass, to power these prime movers. These fuels require cleaning and suitability processes to be used in a conventional gas turbine, but in EFGT systems these requirements are not preponderant. However, studies on EFGT systems are necessary to achieve a greater understanding of the behavior of these systems and its components under different operating conditions. In this paper an externally fired gas turbines (EFGT) was simulated using the commercial software GateCycleTM 5.51, aiming to attend a power of 99.80 kW (design point), and using wood carbonization residual gas as fuel. The model developed is used to analyze the effects of changes of the most important parameters over the power generated and cycle efficiency, and the exergy destruction in each EFGT component.*

Keywords: *Externally-fired-gas-turbine; exergy destruction; parametric analysis*

1. INTRODUCTION

In the beginnings of the industrial revolution, when dirty solid fuels (e.g. coal) were used, the engines with an external firing system have been widely used due to the lack of refined fuels. However, with the availability of clean fuels, these systems were abandoned, with the exception of steam power plants. At present, systems with external firing, especially the external fired gas turbine (EFGT) has captured the attention again (hence with the global trend moving to the green energy production) due to its flexibility and the potential of using renewable energy, such as biomass (Al-attab and Zainal, 2015). EFGT is a under development technology for the production of power and heat in small and medium scale (Kautz and Hansen, 2007). In an EFGT system the fuel is burned externally at atmospheric pressure (Baina et al., 2015) so the combustion gases do not come into direct contact with the turbine blades which allows to use a wide range of fuels without an intensive gas cleaning process. Thus, the thermal energy of the combustion gases is transferred to the working fluid at the compressor outlet through a high temperature heat exchanger, (HTHE) (Fig. 1a) which is the most critical and costly component of the system (Cocco et al., 2006). The fact that the working fluid does not come into contact with the combustion gases allows this system to work both in an open cycle (where the working fluid is air), or in a closed cycle where a fluid with better thermodynamic properties than air can be used (e.g., helium, carbon dioxide and nitrogen) (Al-attab and Zainal, 2015). In the closed cycle configuration it is necessary to cool the gas before it returns to the compressor, which would allow use this waste heat for example, in a cogeneration system.

According to Al-attab and Zainal (2015), one of the advantages of the closed cycle is the possibility of working at higher-pressure levels while maintaining low-pressure ratios similar to the open cycle. This can considerably reduce the size of system components due to a lower gas specific volume of at higher pressure.

From the thermodynamic point of view, the operation of the external fired turbines (Fig. 1a) has many similarities with the conventional turbines (Fig. 1b) and can therefore be explained by the Brayton cycle. Figure 2 shows an EFGT system where hot air exiting the turbine is used in the external combustor to further take advantage of the heat and thus obtaining a saving in fuel. The EFGT systems are very flexible and can use a wide range of thermal energy sources, such as combustors, furnaces, solar concentrators, and even nuclear reactors (Al-attab and Zainal, 2015), and constitute an interesting option for using the gases produced by biomass gasification, for which, it has been difficult to find an appropriate prime mover that operates properly (Baina et al., 2015).

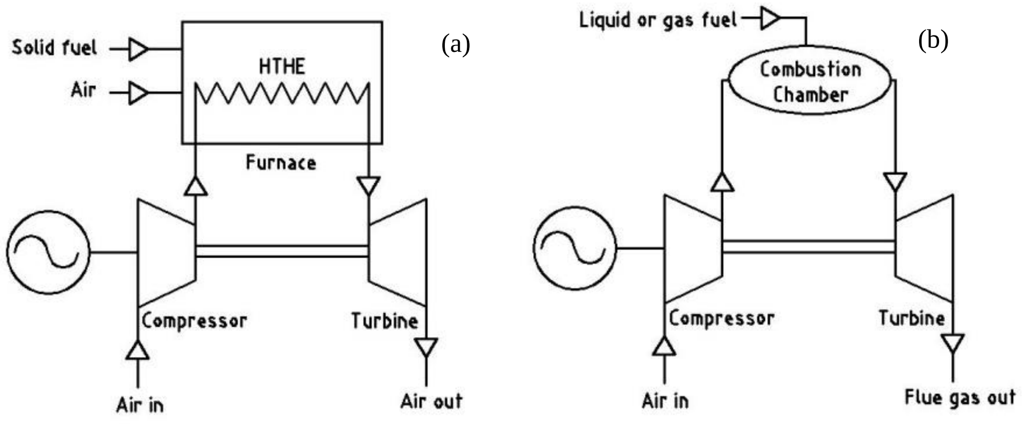


Figure 1. a) Simple EFGT cycle and b) directly fired gas turbine cycle (Al-attab and Zainal, 2015).

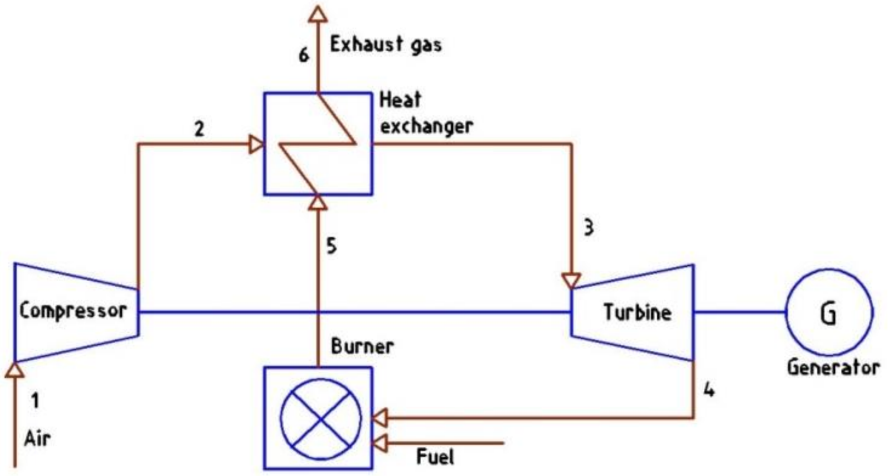


Figure 2. Externally fired-gas turbine (Kautz and Hansen, 2007)

2. DESCRIPTION OF THE EFGT CYCLE

It was simulated an externally fired-gas turbine (EFGT) using the commercial software GateCycleTM 5.51(Gate Cycle, 2003). For the design point, it was used data presented by Kautz and Hansen (2007) with some modification to meet a net power of 99.80 kW, using wood carbonization residual gas as fuel with the composition reported by Vilela et al. (2014). The model developed is used to analyze the influence of the most important parameters over the power generated and cycle efficiency, and the exergy destruction in each EFGT component.

Figure 3 shows the scheme of the EFGT cycle simulated in GateCycle. The cycle is formed by compressor (C), gas turbine (EX), heat exchanger (HX) and combustor (CMB) where the fuel is combusted externally.

Air at ambient conditions (1) enters the compressor where it is compressed to a pressure 4.5 times the atmospheric pressure reaching a temperature around 214 ° C. The air (2) immediately enters the heat exchanger (HX) where it is heated by the combustion gases from the combustor (5). The hot air leaving the exchanger (5) is expanded in the turbine (EX). The air exiting the turbine (4) is used in the combustor to thereby take advantage of the heat that posed still searching for an improvement in cycle efficiency. Finally, the combustion gases are thrown to the environment (6) after providing heat to the air in the heat exchanger. The fuel input in Figure 3 corresponds to stream 7.

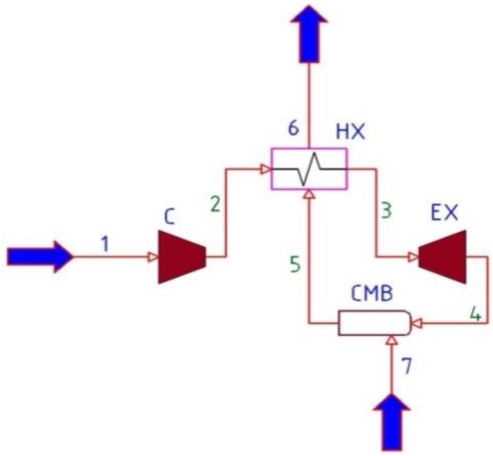


Figure 3. Scheme of the EFGT cycle simulated in GateCycle™ 5.51

Table 1 shows the chemical composition (on dry basis mass fraction) of the fuel gas used in the EFGT cycle. All calculations for the design point were performed using ISO standard conditions (15 ° C / 101.32 kPa / 60% humidity). Table 2 presents the main parameters used to configure the design point to the EFGT cycle in study.

Table 1. Wood carbonization residual gas composition

Component	% dry basis mass fraction
H_2	0.63
CO	34
CO_2	62
Methane - CH_4	2.43
Ethane – C_2H_6	0.13
Others	0.81

Table 2. Design data for EFGT system

Net power	99.8	kW
Turbine power	268	kW
Compressor power	162	kW
Net efficiency, ISO	25.68	%
Fuel inlet temperature	60	°C
Wood carbonization residual gas PCI	5470	kJ/kg
Air temperature at the compressor outlet (T2)	214	°C
Air temperature at the turbine inlet (T3)	850	°C
Air temperature at the turbine outlet (T4)	557	°C
Air mass flow (m1)	0.8	kg/s
Fuel mass flow (m7)	0.071	kg/s
Combustion gases mass flow (m5)	0.87	kg/s
Exhaust gases temperature (T6)	328	°C
Combustion gas temperature (T5)	900	°C
Pressure ratio	4.5	
Compressor Efficiency	0.768	
Turbine Efficiency	0.826	
Heat exchanger area	164	m²

3. RESULT OF THE SIMULATIONS

According to Arrieta (2006), the purpose of the parametric study is to visualize and understand the effect caused by the change of different parameters (the mains) on t power plant performance indicators. The parametric study provides valuable information for monitoring systems and thermodynamic diagnosis, enabling to establish which are the high-priority equipment in programing maintenance tasks (Rúa et al., 2014). The parameters chosen for this work are pressure ratio (PR), combustion gas temperature (T5), temperature entering the expander (T3), pressure drop in the heat exchanger cold side and heat exchanger hot side. The following are the effects caused by the variation of these parameters on the electric power, the cycle efficiency and the destruction of exergy.

3.1 Effects of the selected parameters over the power generated and cycle efficiency

Variations of several main parameters were performed to observe how these parameters affect the performance of the cycle. Figure 4 shows the effects caused by the variation of pressure ratio over net power generated and cycle efficiency. It is observed that the greatest power is obtained for pressure ratios of 5.5, which exceeds 101 kW, but the maximum efficiency (27.63%) is achieved at a pressure ratio of 2.5. It can also be noted that varying the pressure ratio in a range from 2.0 to 8.0 the value of the pressure ratio it is possible to obtain power ranging from 64% to 101% of the design point power (RP = 4.5). The efficiency varies in a range of 79% to 107% the design point efficiency.

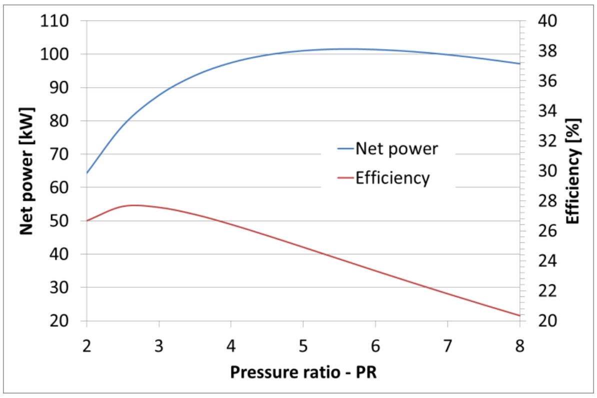


Figure 4. Power and efficiency as function of the pressure ratio.

Thus, three intervals are identified: the first one goes from a pressure ratio of 2.0 to 2.5. In this range both efficiency and power increase with an increase in pressure ratio. The second range is from 2.5 to 5.5 where here the power increases an increase in pressure ratio, whereas efficiency decreases. In the range of 5.5 to 8.0 of pressure ratio both efficiency and power decrease, so that a system with characteristics similar to the system presented in this work should be designed to work at a point located in the range of pressure ratios comprising the first two intervals.

It was simulated the combined variation of the temperature of the combustion gases entering the heat exchanger (T5) and the variation of the air temperature entering the turbine (T3). The effects of these variations on the cycle efficiency are presented in Figure 5. It is observed that 30% of efficiency is attained for T5 = 900 ° C and T3 = 890 ° C. To achieve this condition is necessary a heat exchanger with an area 5.7 times larger than the heat exchanger of the design point.

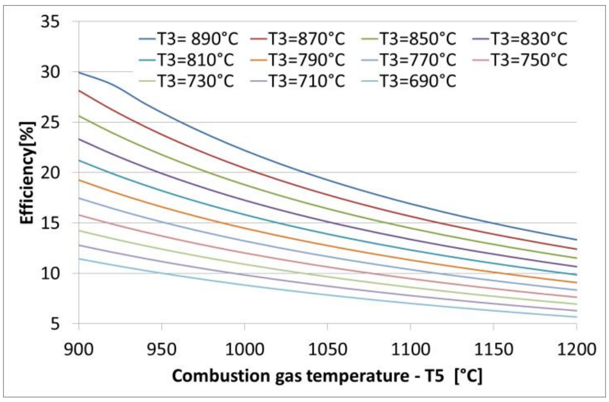
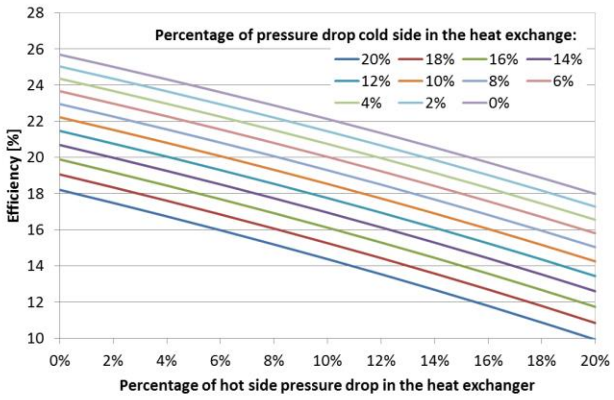


Figure 5. Efficiency as function of temperature of the combustion gases (T5) and temperature of the air entering the gas turbine (T3).

Figures 6a and 6b show the effects caused by pressure drops in heat exchanger and the combustor over the power generated and efficiency of the cycle.



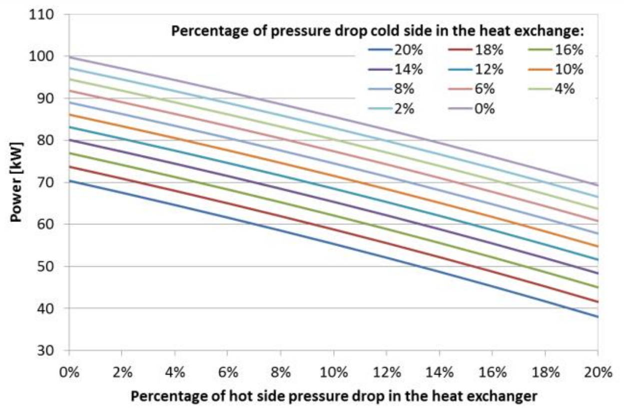


Figure 6. a) Efficiency as function of hot side pressure drop and cold side pressure drop for the heat exchanger. b) Power as function of hot side pressure drop and cold side pressure drop for the heat exchanger.

It is observed in Figure 6a that a pressure drop of 1% in both the cold side and the hot side of the heat exchanger causes a decrease in efficiency of approximately 0.4%. A pressure drop in the cold side of the exchanger means that air to be expanded from a low pressure, which leads to a lower power produced per kg/s of fuel entering. Similarly, a pressure drop on the hot side of the heat exchanger means that air cannot be expanded in the turbine to the atmospheric pressure also represents a loss of produced power. The effects on potency are shown in Figure 6b. It can be seen that for 1% pressure drop of both the hot side and the cold side of the heat exchanger, the power output is decreased by an average of 1.6 kW, so it is the heat exchanger the equipment that requires special attention from the viewpoint of maintenance.

3.2 Exergy destruction in the main components of the cycle.

The effects of changes in parameters in the exergy destruction in each cycle component were also analyzed. The results are shown below. Figure 7 shows the exergy destroyed as a function of the pressure ratio (PR). The combustor has the highest exergy destruction of the system. The compressor and turbine have similar values. All equipment show an increment of exergy destruction with increasing pressure ratio except the heat exchanger where the destroyed exergy decreases, but in general the destroyed exergy increases in overall EFGT cycle with the pressure ratio.

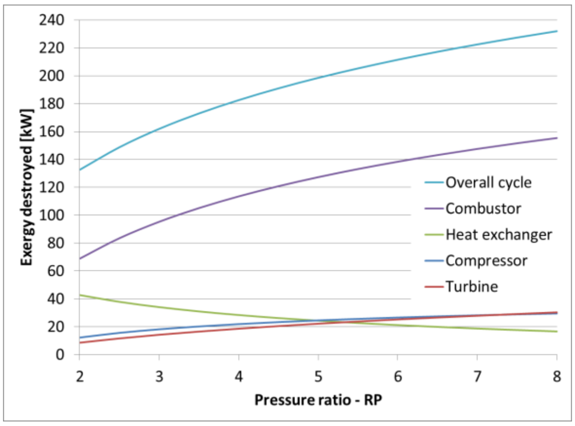


Figure 7. Exergy destroyed as a pressure ratio function.

The effects caused by variations in temperature of the combustion gases and the temperature of the air leaving the heat exchanger (cold side), which enters the gas turbine are different for each device. For the compressor, under prescribed conditions, are not presented changes in the destroyed exergy to variations in these temperatures. In the gas turbine the two temperatures were varied, only the change in turbine inlet temperature affects, though minimally, the exergy destroyed in the equipment as shown in Fig. 8.

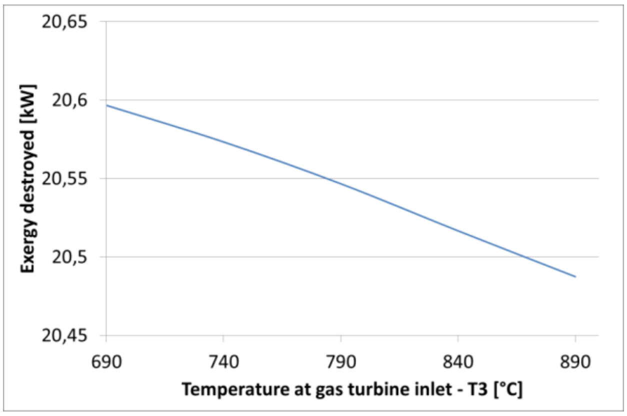


Figure 8. Exergy destroyed in the turbine with inlet temperature variation.

In the heat exchanger, as expected, the variations of these two temperatures have important effects on the exergy destroyed (Fig. 9a). It is observed in Figure 9a that for the heat exchanger larger exergy destroyed are given for higher inlet temperatures of the hot side and low exit temperature on the cold side. A similar behavior shows the combustor (Fig. 9b). This effect is propagated through the gas turbine, since the temperature change of the air entering the gas turbine also change the temperature of the air exiting the turbine. This air then goes to the combustor.

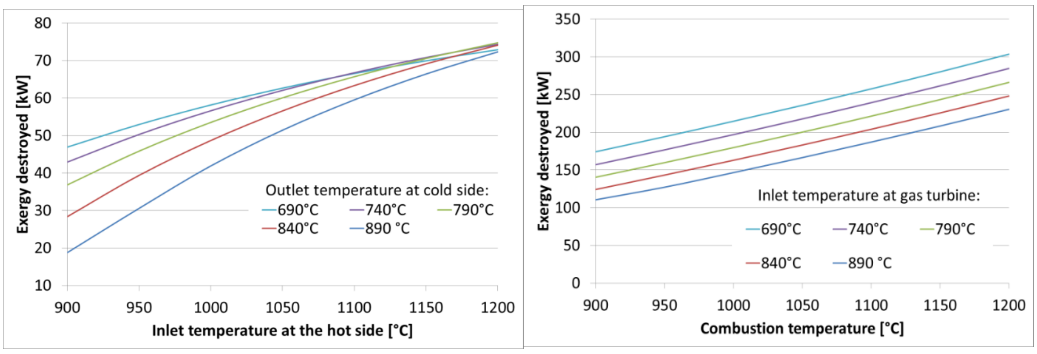


Figure 9. a) Exergy destroyed in the heat exchanger as a function of the inlet temperature of the hot side (combustor outlet) and the outlet temperature of the cold side (gas turbine inlet). b) Exergy destroyed the combustor as a function of temperature of the combustion gases and the inlet temperature of the gas turbine.

The effect caused by the pressure drop over the destroyed exergy is also shown, for each component of the cycle, in Figures 10 and 11. In these Figures, the pressure drop for both the cold side and the hot side of the system is represented as a percentage of pressure loss.

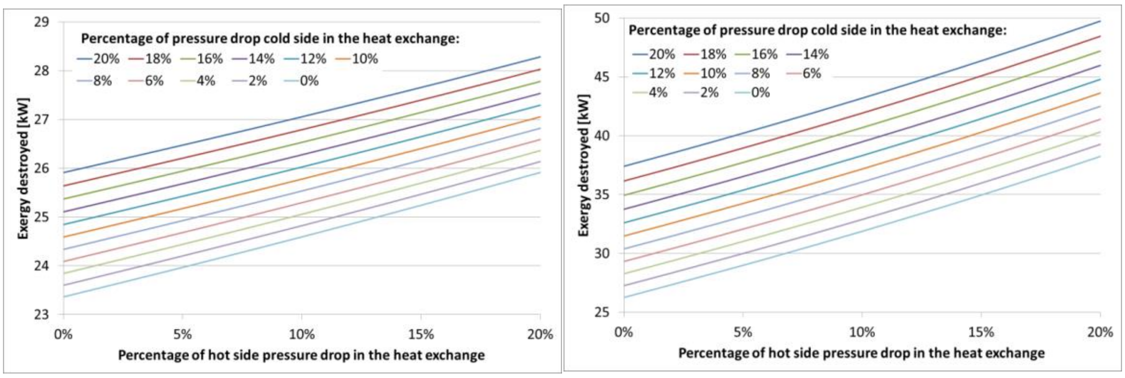


Figure 10. a)Exergy destroyed in the compressor as a function of pressure drops in the system. b) Exergy destroyed in the heat exchanger as a function of pressure drops in the system.

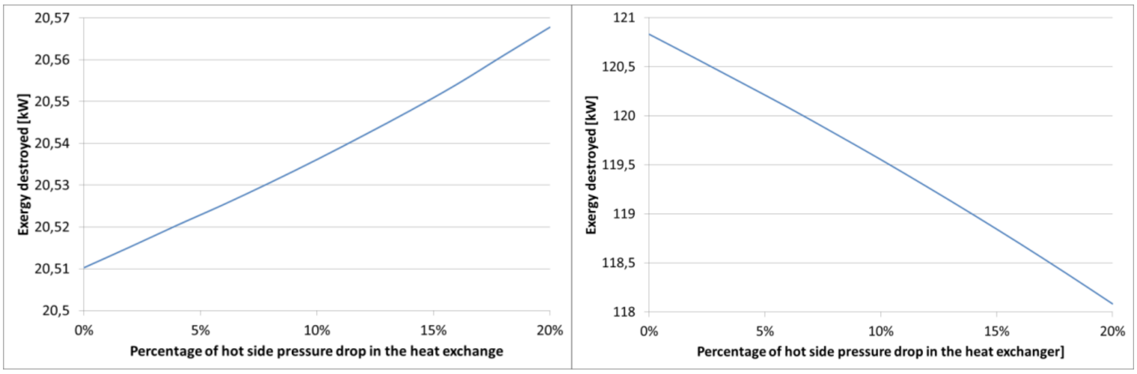


Figure 11. a) Exergy destroyed in the gas turbine as a function of the pressure drops in the system. b) Exergy destroyed in the combustor as a function of pressure drops in the system.

It is observed that for the compressor (Fig. 10a) as well as the heat exchanger (Fig. 10b) destroyed exergy is increased with increased pressure drop in the system. In turn the gas turbine is only influenced by the pressure drop of the hot side (Fig. 11a). However, the combustor experiences a reduction of its exergy destroyed by an increase in pressure drop of the hot side (Fig. 11b). Similarly as with the gas turbine, the pressure drop on the cold side does not affect considerably the exergy destroyed in the combustor.

4. CONCLUSIONS

In this study, the effects of various operational and design parameters over the performance of an EFGT cycle were presented. It was demonstrated that the pressure ratio parameter has a large influence on power produced, on the cycle efficiency and on destroyed exergy. For the simulated EFGT cycle it is not justified to work with a pressure ratio greater than 5.5 because both efficiency and power decrease. It is up to the designers of the EFGT system to choose the proper pressure ratio, if they intend on to prioritize the power or efficiency.

Parameters: (i) the temperature of the combustion gases and (ii) inlet temperature of the gas turbine has relevance in system performance. The first depends on the type of fuel used, the air excess and the amount of fuel entering the system. The second parameter depends on the first parameter and heat transfer process in the heat exchanger. The turbine inlet temperature would be increased if improved heat transfer coefficient and increased the area of heat exchange. In the latter case it would require a larger heat exchanger which implies higher investment cost. Furthermore the heat exchanger also influences the cycle performance since the pressure drop occurring in this equipment has important effects in the cycle as previously shown.

The EFGT technology (such as developing technology) still requires further study leading to a greater understanding of these systems allowing the optimization of the same and the use of energy resources not yet used. Thus, the EFGT systems appears today as a tangible alternative for applications involving solid fuels such as biomass or wood carbonization residual gas as in conventional gas turbines and internal combustion engines require complex and expensive cleaning systems. Furthermore, they can be used to work with a wide range of other energy sources being untapped in many industrial and agricultural processor.

The values of the parameters presented in the results of this paper are specific to EFGT cycle study and based on the conditions set for the simulations, so they cannot be generalized to other plants in magnitude, but in behavior, they will certainly show the same trend.

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