



25<sup>th</sup> ABCM International Congress of Mechanical Engineering  
October 20-25, 2019, Uberlândia, MG, Brazil

## COB-2019-0640

# THERMODYNAMIC EVALUATION OF POWER AND HEAT COGENERATION SYSTEMS FUELED BY BIOGAS

Ana Elisa Achiles

Thalyta Stefani Amâncio Santiago

José Vicente Hallak d'Angelo

Department of Chemical Systems Engineering, School of Chemical Engineering, University of Campinas – Av. Albert Einstein, 500, Bloco A, Campinas – São Paulo, CEP: 13083 – 852.

a211805@dac.unicamp.br; t211758@dac.unicamp.br; dangelo@unicamp.br

**Abstract.** *Environmental concerns as global warming and energy depletion have been motivating the use of renewable energy sources to meet energy requirements. Power and heat cogeneration systems consist of one of the alternative technologies in terms of energy saving and power generation high efficiency. In this work, two configurations of a cogeneration system were studied. In the first one, a topping regenerative gas turbine cycle combined with a steam generation and a bottoming basic organic Rankine cycle while the second one used a bottoming organic Rankine cycle with a reheater. These systems were fueled by biogas to generate a constant net output power of 1000 kW and a quantity of 0.194 kg.s<sup>-1</sup> of steam at 506.6 kPa and 433.15 K. Energetic and exergetic performances evaluation were carried out in order to investigate optimum operating conditions. A parametric analysis was also performed varying biogas molar composition; percentage of excess air; compression ratio and air inlet temperature in the combustion chamber. The results showed that the addition of a reheater in bottoming ORC improved both energetic and exergetic system performances, reducing biogas consumption and exergy destruction rate in 4.43 % and 7.28 %, respectively.*

**Keywords:** *Gas Turbine, Organic Rankine Cycle (ORC), Thermodynamic Analysis*

## 1. INTRODUCTION

Environmental concerns as the intensification of greenhouse gases emissions and, consequently, global warming increasing, have made countries all over the world aware to the importance of studying more efficient and sustainable means to produce, supply and use energy.

In this context, biogas usage in power generation is a promising alternative since it is a renewable energy source, with low net CO<sub>2</sub> emission and its potential has gained attention (Barzegaravval *et al.*, 2018). In Brazil, biogas power generation is only responsible for 0.086 % of its total installed power capacity (ANEEL, 2019) and its potential of biogas production is considered the largest in the world, since this country has enough raw material to supply up to 36 % of the Brazilian electricity consumption (Abiogás, 2019). Considering this great potential, researches involving this subject are relevant, seeking for better ways of using this energy source, improving the efficiency of energy systems as well.

Biogas can be produced by anaerobic digestion of many different organic matters, as animal and human wastes. So, its molar composition will depend on the kind of substrate digested. A typical biogas composition is: methane (50 to 75 mol%); carbon dioxide (25 to 50 mol%) and small quantities of other components like water, oxygen and traces of sulfur and hydrogen sulfide (Wellinger *et al.*, 2013). This composition variation affects biogas heating value and, consequently, its flame temperature. Thus, the selection of an adequate power generation technology using biogas as a fuel must consider this characteristic.

Usually, biogas is burned in the combustion chamber of a boiler providing heat to generate a vapor stream that can be used to operate a steam Rankine cycle and generate power (Wellinger *et al.*, 2013). However, there are other alternative cycles to generate energy from biogas. Some researchers, for instance, used biogas as fuel in gas turbines (Somehsaraei *et al.*, 2014; Barzegaravval *et al.*, 2018); others utilized the gases from biogas combustion to drive an organic Rankine cycle (ORC) (Mudasar *et al.*, 2017). Another option is to combine cycles like gas turbine and ORC as the topping and bottoming cycles, respectively, in which ORC is operated by the energy surplus from the gas turbine topping cycle (Hosseini *et al.*, 2016; Sung *et al.*, 2017; Gholizadeh *et al.*, 2019).

In addition to generating power, biogas can also be applied to produce heat. A cycle that can produce, simultaneously, heat and power is named a *Combined Heat and Power (CHP)* cycle. Therefore, the cogeneration of heat and power is a better way to save energy compared to simple systems that only produce power or heat separately.

Considering the use of biogas, there are some authors who have investigated its application in CHP plants: Kang *et al.* (2012) studied the effect of burning biogas on the performance and operating characteristics of simple and recuperative cycle gas turbine combined heat and power systems; Kang *et al.* (2014) investigated the economic feasibility of two systems consisting of a simple gas turbine cycle to produce power and heat or a combined gas turbine topping cycle with a steam Rankine bottoming cycle to generate power only; Zareh *et al.* (2018) evaluated a CHP system, both thermodynamically and economically, composed by a gas turbine cycle, a steam generator, and a heat exchanger for heating the digester; and fueled by a blend of natural gas and biogas.

The objective of this work is to investigate and compare, theoretically, the energetic and exergetic performances of two different cogeneration power and heat systems fueled by biogas. Systems' configurations will consist of a regenerative gas turbine topping cycle followed by a heat recovery steam generator (HRSG) and a conventional organic Rankine bottoming cycle (base case) or an organic Rankine bottoming cycle with a reheater (modified case). These choices are based on three main reasons:

1. Gas turbine is considered flexible in terms of fuel consumption in comparison to other heat engines and, depending on the operating conditions, it can produce higher net power as well as heat supply (Boyce, 2012).
2. According to Boyce (2012), the use of regenerators combined with gas turbines increases cycle's efficiency and can reduce fuel consumption up to 30 %.
3. The combination of the gas turbine with steam turbines is an attractive alternative, especially for process industries where water steam is employed. This system provides higher efficiencies in comparison to gas turbines and steam cycles operating separately due to the waste heat recovery of the gas turbine exhaust gases.
4. Organic Rankine cycle is characterized by its low maintenance requirements; its high safety and great flexibility; and especially, its great feasibility and efficiency in recovering low grade waste heat (Macchi and Astolfi, 2017).
5. It was not found in the open literature a similar thermodynamic analysis of a CHP system using biogas as a fuel with these proposed configurations, so this may be a major contribution of the present study.

## 2. METHODOLOGY

To perform and simulate the proposed CHP systems, Aspen Hysys<sup>®</sup> v. 10 (from AspenTech) process simulator was used and the thermodynamic property package chosen was the Peng-Robinson (PR) equation of state. In order to ensure data reliability, the gas turbine topping cycle was validated with the results obtained by Kang *et al.* (2012), and the results authenticity of the ORC bottoming cycle was confirmed through the work performed by Chacartegui *et al.* (2009), both under the same operation parameters.

### 2.1 System Description

Schematics of the proposed power and heat cogeneration systems are shown in Fig. 1, in which Fig. 1 (a) and (b) represent the base case and modified case configurations, respectively.

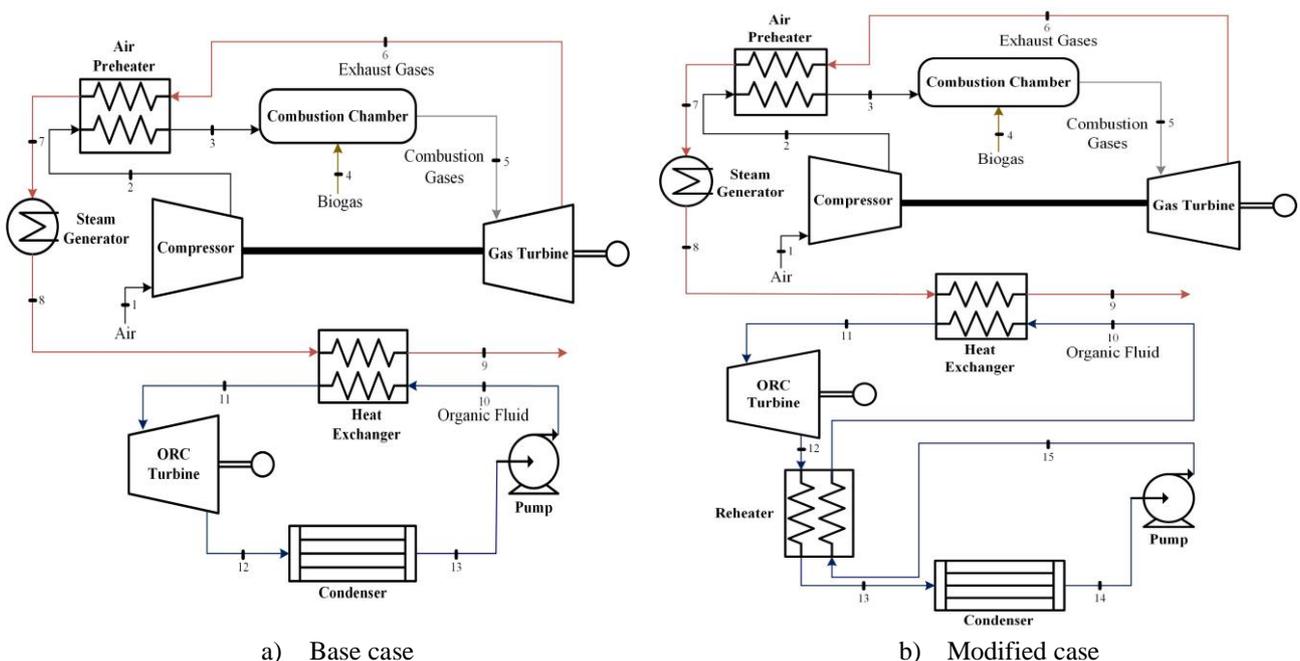


Figure 1. Schematics of CHP systems configurations.

Base case configuration includes a top cycle of regenerative gas turbine, a HRSG (heat exchanger) and a basic bottoming organic Rankine cycle, while the modified one is composed by a bottoming organic Rankine cycle with a reheater.

In both configurations presented in Fig. 1, ambient air stream [1], with a pressure of 1 atm and a temperature of 298.15 K, is compressed. Then, compressed air [2] flows into air preheater and leaves it at a temperature of 773.15 K [3]. In the combustion chamber, pressurized biogas [4] is also injected in the combustion chamber where it is burned. The combustion gases stream [5] resulted from the complete combustion is then expanded in the gas turbine producing power. In order to recover the energy content of exhaust gases [6] and generate steam, the hot gases [7], after passing through the air preheater, are fed in the steam generator to produce low pressure water steam at 506.6 kPa and 433.15 K, that can be employed in another part of the plant for heating. Since stream [8] still contains residual energy content, then it is transferred to the heat exchanger (HRSG) to provide its remained thermal energy to ORC bottoming cycle. Process stream [11], is an organic fluid that leaves the heat exchanger as saturated vapor with a pressure of 2000 kPa. Then it expands in the ORC turbine also generating power. In the base case configuration (a), the organic fluid [12] is then condensed until achieving a saturated liquid state [13] and pumped into the heat exchanger [10]. In the modified case configuration (b), after passing through the ORC turbine, part of the energy content of organic fluid stream [12] is recovered in the reheater in order to preheat stream process [15] and, consequently, increase its temperature before this being fed in the heat exchanger [10]. So, stream [13] leaves the reheat with a lower temperature and then it is condensed. Finally, the organic fluid exits the condenser as saturated liquid [14] and is transferred to the pump.

The selection of an efficient and safe working fluid is a relevant in order to operate the ORC system efficiently. Compared to a conventional steam Rankine cycle, the working pressures and temperatures in an ORC are lower, ensuring better safety (Macchi and Astolfi, 2017). The organic fluid chosen to drive the ORC was cyclohexane, since it presented greater efficiencies in comparison with other fluids as pointing out in the study done by Chacartegui *et al.* (2009), who assessed a system of commercial gas turbines combined with an ORC in two different cycles by applying several organic fluids.

The assumptions used in the simulations performed in this present work are as follows:

1. All control volumes of the cycle operate at steady state;
2. All devices are adiabatic, except the combustion chamber;
3. Pressure drops are neglected in heat exchangers, combustion chamber and pipes;
4. The organic fluid feeds the ORC turbine as saturated vapor and leaves the condenser as saturated liquid.

The basic input parameters in the proposed systems are listed in Tab. 1.

Table 1. Input parameters assumed in the simulations.

Parameter	Value
Total net power produced [kW]	1000
Water steam generated mass flow [kg.s <sup>-1</sup> ]	0.194
Pressure [kPa] and temperature [K] of water steam generated	506.6 and 433.15, respectively
<b>Gas Turbine Cycle</b>	
Biogas molar composition [mol%]	60 CH <sub>4</sub> + 40 CO <sub>2</sub>
Air molar composition [mol%]	79 N <sub>2</sub> + 21 O <sub>2</sub>
Excess air [%]	200
Inlet pressure [kPa] and temperature [K] of air in the compressor	101.3 and 298.15, respectively
Ratio between outlet and inlet pressures of the air in the compressor (compression ratio)	10
Inlet pressure [kPa] and temperature [K] of biogas in the combustion chamber	1013 and 308.15, respectively
Air inlet temperature [K] in the combustion chamber	773.15
Combustion thermal efficiency [%]	97
Gas turbine inlet pressure [kPa]	1013
Gas turbine outlet pressure [kPa]	110
Compressor and gas turbine isentropic efficiencies [%]	80
<b>ORC</b>	
ORC turbine inlet pressure [kPa]	2000
ORC turbine outlet pressure [kPa]	40
Pump and ORC turbine isentropic efficiencies [%]	88

## 2.2 Energy analysis

The energy performance evaluation was computed by the overall energetic (thermal) efficiency of the power and heat cogeneration system, given by Eq. (1).

$$\eta_{th} = \frac{\dot{W}_{net} + \dot{Q}_h}{\dot{m}_{biogas} LHV_{biogas}} \quad (1)$$

where  $\dot{Q}_h$  is the heat produced [kW];  $\dot{m}_{biogas}$  is the biogas mass flow rate [kg.s<sup>-1</sup>];  $LHV_{biogas}$  biogas low heating value [kJ.kg<sup>-1</sup>]; and  $\dot{W}_{net}$  is the net power rate generated [kW].

The heat and net power rates generated can be calculated by Eq. (2) and (3), respectively.

$$\dot{Q}_h = \dot{m}_{steam}(\bar{h}_8 - \bar{h}_7) \quad (2)$$

in which  $\dot{m}_{steam}$  is the mass flow rate of the water steam generated (0.194 kg.s<sup>-1</sup>) in the steam generator and  $\bar{h}$  refers to the specific mass enthalpy [kJ.kg<sup>-1</sup>].

$$\dot{W}_{net} = \dot{W}_{Gas\ turbine} + \dot{W}_{ORC\ turbine} - \dot{W}_{Compressor} - \dot{W}_{Pump} \quad (3)$$

where  $\dot{W}_{Gas\ turbine}$  and  $\dot{W}_{ORC\ turbine}$  are the power rates generated by the gas and ORC turbines; and  $\dot{W}_{Compressor}$  and  $\dot{W}_{Pump}$  correspond to the power rates required by the compressor and pump, respectively. These terms can be calculated using the following equations:

$$\dot{W}_{Gas\ turbine} = \dot{m}_{gas}(\bar{h}_6 - \bar{h}_5) \quad (4)$$

$$\dot{W}_{ORC\ turbine} = \dot{m}_{ORC}(\bar{h}_{12} - \bar{h}_{11}) \quad (5)$$

$$\dot{W}_{Compressor} = \dot{m}_{air}(\bar{h}_2 - \bar{h}_1) \quad (6)$$

$$\dot{W}_{Pump} = \dot{m}_{ORC}(\bar{h}_{10} - \bar{h}_{13}) \text{ for base case or } \dot{W}_{Pump} = \dot{m}_{ORC}(\bar{h}_{15} - \bar{h}_{14}) \text{ for modified case} \quad (7)$$

being  $\dot{m}_{gas}$ ;  $\dot{m}_{ORC}$ ; and  $\dot{m}_{air}$  the mass flow rates [kg.s<sup>-1</sup>] of the combustion exhaust gases; organic fluid in the ORC; and air fed in the compressor, respectively.

## 2.3 Exergy analysis

The exergy performance evaluation was accomplished through two analyses based on exergetic efficiency and exergy destruction rate, according to Szargut *et al.* (1988).

The exergy efficiency of a system may be defined as:

$$\eta_{ex} = \frac{\text{Product exergy}}{\text{Total supplied exergy}} = \frac{\left(1 - \frac{T_0}{T_h}\right)\dot{Q}_h + \dot{W}_{net}}{\dot{m}_{biogas}\bar{e}x_{biogas}} \quad (8)$$

in which,  $T_0$  and  $T_h$  are the reference state and steam generated temperatures; and  $\bar{e}x_{biogas}$  is the specific exergy rate for biogas [kJ.kg<sup>-1</sup>].

The specific exergy for a substance  $k$  in the system can be determined by in Eq. (9).

$$\bar{e}x_k = \bar{e}x_k^{ph} + \bar{e}x_k^{ch} \quad (9)$$

where  $\bar{e}x_k^{ph}$  refers to the specific physical exergy of process stream  $k$ , defined by in Eq. (10), and  $\bar{e}x_k^{ch}$  represents the specific chemical exergy, calculated using Eq. (11).

$$\bar{e}x_k^{ph} = (\bar{h}_k - \bar{h}_0) - T_0(\bar{s}_k - \bar{s}_0) \quad (10)$$

$$\bar{e}x_k^{ch} = \sum y_i \bar{e}x_i^{ch,0} + RT_0 \sum y_i \ln y_i \quad (11)$$

in which,  $\bar{h}_k$  and  $\bar{s}_k$  are the specific mass enthalpy and entropy of the substance, respectively, at the operating conditions, and  $\bar{h}_0$  and  $\bar{s}_0$  stands for properties taken in a reference state;  $\bar{e}x_i^{ch,0}$  is the standard mass chemical exergy for each component  $i$  presented in the substance;  $y_i$  is the mass fraction of component  $i$ ; and  $R$  is the universal constant of gases.

Exergy destruction rate for a control volume ( $cv$ ) can be obtained using Eq. (12).

$$\dot{E}x_d = \left(1 - \frac{T_0}{T_{cv}}\right) \dot{Q}_{cv} - \dot{W}_{cv} + \sum_{j=1}^{j=n} \dot{m}_j \bar{e}x_j - \sum_{s=1}^{s=n} \dot{m}_s \bar{e}x_s \quad (12)$$

where the subscripts  $j$  and  $s$  are the input and output streams in the control volume, respectively. The reference state adopted for all calculations was 1 atm and 298.15 K.

The standard chemical exergies for each component of the systems are listed in Tab. 2.

Table 2. Standard chemical exergy.

Component ( $i$ )	$\bar{e}x_i^{ch,0}$ [kJ.kg <sup>-1</sup> ]
CH <sub>4</sub>	51841.85
CO <sub>2</sub>	451.50
H <sub>2</sub> O	527.35
O <sub>2</sub>	311.58
N <sub>2</sub>	25.70

Source: Szargut *et al.* (1988).

## 2.4 Parametric Analysis

In order to understand how some operating conditions impact on energetic and exergetic efficiencies, a parametric analysis was also conducted varying the following input parameters presented in Tab. 3.

Table 3. Input parameters range.

Input parameter	Range
Biogas composition [mol% CH <sub>4</sub> ]	50 - 75
Excess air percentage used in biogas combustion [%]	100 - 250
Compression ratio* [-]	6 - 12
Preheated air temperature - $T_3$ [K]	630.75 – 873.15

\*Ratio between outlet and inlet pressures of the air in the compressor.

## 3. RESULTS AND DISCUSSION

The comparison of the thermodynamic performance between the base and modified configurations was done under the conditions presented in Tab. 1. The main results obtained for each system are shown in Tab. 4.

Table 4. Thermodynamic performance of base and modified cases under the conditions defined at Tab. 1.

Parameter	Base case	Modified case
Energetic efficiency [%]	53.02	55.49
Exergetic efficiency [%]	38.43	40.22
Biogas consumption [kg.s <sup>-1</sup> ]	0.1618	0.1546
Power of gas turbine cycle [kW]	771.53	737.33
Power of ORC cycle [kW]	228.47	262.67

The simulations were carried out for a constant net output power and water steam generation rate of 1000 kW and 0.194 kg.s<sup>-1</sup>, respectively, for both cogeneration systems. According to the results, the addition of the reheater in the ORC bottoming cycle provided an increment in both energetic and exergetic efficiencies from 53.02 % to 55.49 % and 38.43 % to 40.22 %, respectively. Beside this, the modified case configuration could save 4.43 % of biogas consumption. Thus, in a general way, the modified system has a better performance in comparison to the base case from thermodynamic point of view.

Figure 2 shows the exergy destruction rate for each equipment of the base and modified systems. It can be observed that, in both cases, the combustion chamber presented the greatest exergy destruction rate, being responsible for more than 50 % of all exergy destruction rate in the systems. In addition, the destruction exergy rate was slightly higher for the base case, especially in the heat exchanger, 48.25 % greater than that one from the modified case. Regarding the total exergy destruction rate, it was reduced in 7.28 % in the modified case configuration.

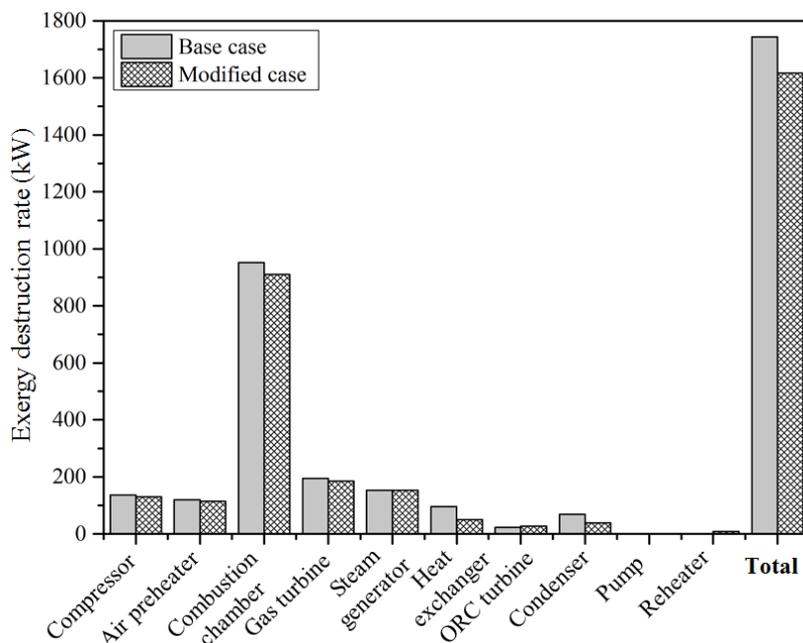


Figure 2. Exergy destruction rate for base and modified case configurations.

### 3.1 Effect of the biogas composition over system performance

The influence of biogas composition on the system thermodynamic performance was evaluated by varying the molar fractions of methane from 50 % to 75 %, as demonstrated in Fig. 3. For both systems, as the CH<sub>4</sub> content increases, the low heating value of the biogas also increases, requiring less biogas supply to the combustion chamber to achieve the same net output power (see Tab. 5). This lower fuel supply is responsible to improve the energetic and exergetic performances. Varying biogas molar composition from 50 % to 75 % of CH<sub>4</sub> resulted in a reduction of, approximately, 52 % in biogas consumption for both systems, and increased the energetic efficiency from 51.08 % up to 54.87 % for base case configuration and from 53.49 % up to 57.42 % for the modified one. Regarding the exergetic efficiency, it raised from 37.19 % to 39.57 % for base case and from 38.94 % to 41.41 % for the modified one.

Table 5. Variation of biogas low heating value and mass flow rate with CH<sub>4</sub> molar fraction.

Biogas molar composition [%]	LHV [kJ.kg <sup>-1</sup> ]	Biogas mass flow [kg.s <sup>-1</sup> ]	
		Base case	Modified case
50	13366.67	0.2222	0.2123
55	15421.52	0.1887	0.1803
60	17687.42	0.1618	0.1546
65	20198.65	0.1398	0.1336
70	22997.32	0.1214	0.1160
75	26135.78	0.1058	0.1011

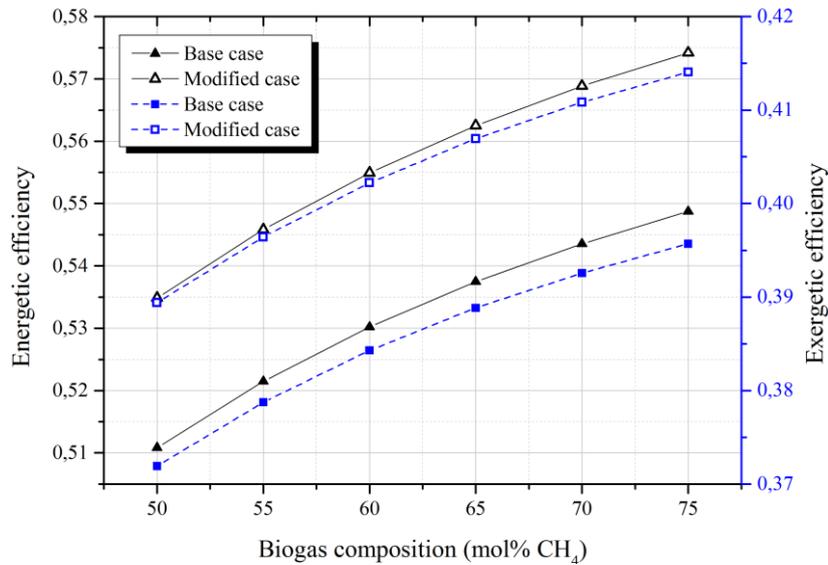


Figure 3. Effect of biogas composition over system performance.

### 3.2 Effect of the excess air percentage over system performance

Figure 4 illustrates the behavior of energetic and exergetic efficiencies against the variation of excess air percentage in the biogas combustion chamber, which shows that an increase in the excess air percentage fed into the combustion chamber promotes a decrease in the thermodynamic performance of both systems. This trend was expected and can be justified by the fact that net output power of gas turbine topping cycle is reduced since it is required a greater power to drive the compressor. So, to compensate this reduction and keep the total net power output of the system constant and equal to 1000 kW, more power is generated by ORC bottoming cycle. For this purpose, more biogas had to be supplied in the combustor, because, increasing excess air, the exhaust gases temperature available to drive ORC was reduced. Quantitatively, the thermal efficiencies decreased from 55.92 % and 58.52 % to 51.42 % and 53.84 %, for base and modified cases, respectively. With respect to exergetic efficiencies, they were reduced from 40.53 % and 42.42 % until 37.27 % and 39.02 %, for base and modified configurations, respectively.

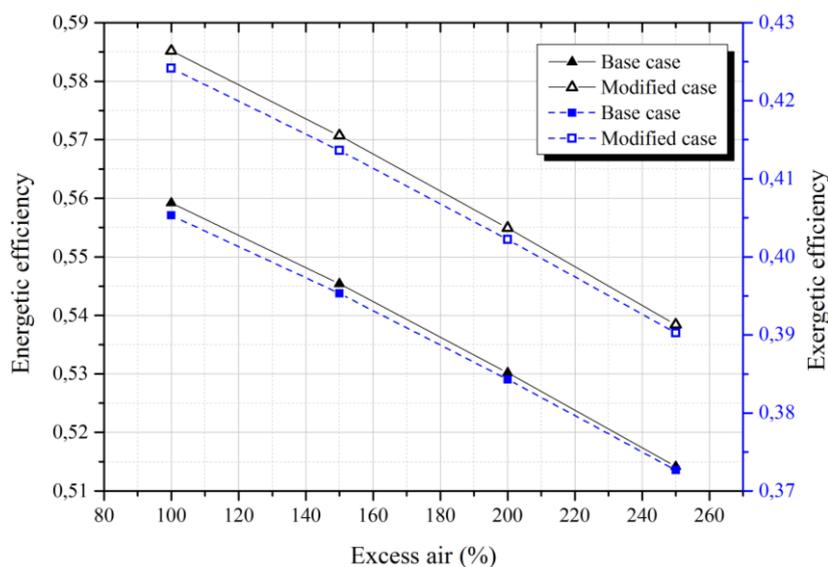


Figure 4. Effect of excess air percentage over system performance.

### 3.3 Effect of air compression ratio over system performance

The thermodynamic performance of the systems as a function of air compression ratio is given by Fig. 5, showing that both energetic and exergetic efficiencies increase with the raise of air compression ratio reaching a maximum value and, from this point, they begin to decrease. This occurs because increasing compression ratio up to a certain value

causes an increment in the net output power of gas turbine topping cycle and, consequently, decreases the biogas consumption and the net power of ORC bottoming cycle. Nonetheless, for compression ratios value greater than 8, the thermodynamic performance of both systems begins to decrease. This happens because the power required by the compressor increases so much that the net output power of gas turbine topping cycle starts to decrease. Thus, to balance this decrement while maintaining the total net power output of the combined cycle constant and equal to 1000 kW, more power begins to be produced by ORC bottoming cycle, increasing the quantity of biogas supplied and, therefore, decreasing systems thermodynamic performance.

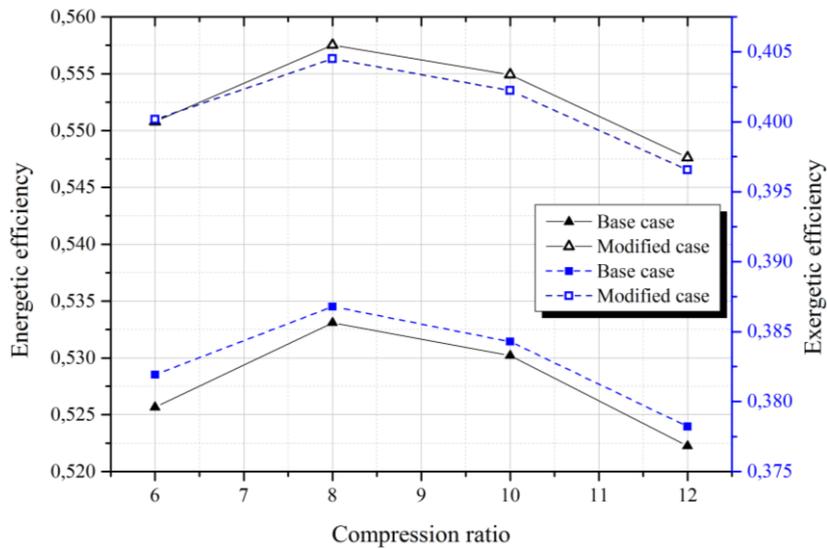


Figure 5. Effect of air compression ratio over system performance.

### 3.4 Effect of preheated air temperature over system performance

Figure 6 presents the effect of air inlet temperature in the combustion chamber ( $T_3$ ) on the energetic and exergetic efficiencies for both system configurations.

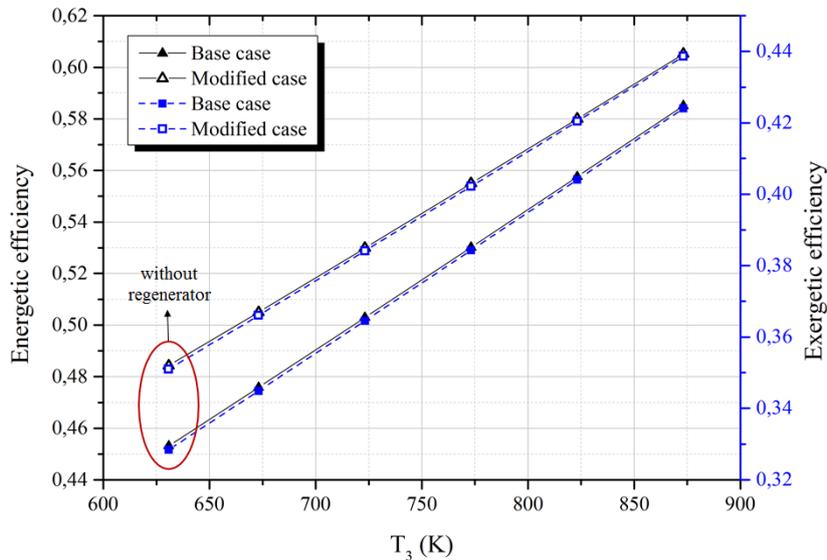


Figure 6. Effect of preheated air temperature over system performance.

This variation was given from the air inlet temperature with no regenerator (630.75 K) up to preheated air temperature of 873.15 K, as showed in Tab. 3. Increasing the preheated air temperature raises the air specific enthalpy and, consequently, net output power generated by the gas turbine topping cycle. As a result, the consumption of biogas supplied is reduced, improving both energetic and exergetic efficiencies. In this study, preheating the air until 873.15 K decreased the biogas consumption in, approximately, 20 % for both systems in comparison to the case in which no

regenerator to preheat the air is employed. The energetic efficiencies increased from 45.32 % to 58.51 % and 48.43 % to 60.52 % for base and modified configurations, respectively. Regarding the exergetic efficiency, it was improved from 32.84 % to 42.44 % for base configuration and 35.10 % to 43.86 % for the modified one.

### 3.5 Final considerations

From all thermodynamic investigation performed, the application of the modified configuration to generate power and heat using biogas as a fuel may be indicated since it could achieve better energetic and exergetic efficiencies in all tested conditions. Also, the operation conditions recommended are the ones that provided higher thermodynamic performances like biogas molar composition rich in methane; lower quantities of excess air percentage in the biogas combustion; compression ratio value next to 8; and higher temperatures for the preheated air. Nonetheless, a more detailed study of optimization should be performed in order to understand how these operational conditions will affect the system performance together, and determine their exact values. Besides, despite of the modified configuration operating by the mentioned conditions have presented greater thermodynamic performances, it is also necessary an economic assessment with the purpose to know if they are in fact more viable, because both the modified configuration and increasing the preheated air temperature require additional heat exchangers, and it is necessary a biogas generation process more advanced to obtain a biogas rich in methane. All these modifications will imply a supplementary cost.

## 4. CONCLUSION

A thermodynamic evaluation in considering both energetic and exergetic efficiencies of a base and modified cogeneration systems fueled by biogas as a heat source was carried out in this study. Both process simulator and thermodynamic packages presented reliable results and can be indicated to perform these analyzes.

The systems were simulated to generate a constant net output power of 1000 kW and a quantity of 0.194 kg.s<sup>-1</sup> of low pressure water steam at 506.6 kPa and 433.15 K. The exergy destruction rate for both systems was investigated as well. The relevant findings of the current analysis are as follows:

- The addition of a reheater in the ORC bottoming cycle provided an increment in both energetic and exergetic efficiencies from 53.02 % to 55.49 % and 38.43 % to 40.22 %, respectively, able to save 4.43 % of biogas consumption, and to reduce in 7.28 % the exergy destruction rate in comparison to the one from the base case. So, its application to generate power and heat using biogas as a fuel may be indicated.
- In summary, the operation conditions recommended to operate the systems are the ones that provided higher thermodynamic performances like biogas molar composition rich in methane; lower quantities of excess air percentage in the biogas combustion; compression ratio value next to 8; and higher temperatures for the preheated air.
- A more detailed study of optimization should be performed in order to understand how the operational conditions will affect the systems performance together and to determine their exact values. It is also required an economic assessment with the purpose to know if these conditions are in fact more viable.

## 5. ACKNOWLEDGEMENTS

This work was supported by Conselho Nacional de Desenvolvimento Científico e Tecnológico (CNPq) under the Project number 870379/1997-6.

## 6. REFERENCES

- ABiogás, 2019. “Potencial do biogás cresce e fonte pode substituir grande parte da demanda nacional por diesel ou energia elétrica”. Associação Brasileira de Biogás e de Biometano. 30 Jan. 2019 <<https://www.abiogas.org.br/copia-autoridades-sao-confirmadas-nl>>.
- ANEEL, 2019. “Capacidade total instalada no brasil”. Agência Nacional de Energia Elétrica. 30 Jan. 2019 <<http://www2.aneel.gov.br/aplicacoes/capacidadebrasil/OperacaoCapacidadeBrasil.cfm>>.
- Barzegaravval, H., Hosseini, S.E., Wahid, M.A. and Saat, A., 2018. “Effects of fuel composition on the economic performance of biogas-based power generation systems”. *Applied Thermal Engineering*, Vol. 128, pp. 1543 – 1554.
- Boyce, M. P., 2012. *Gas Turbine Engineering Handbook*. Fourth edition. Oxford: Butterworth-Heinemann, 2012. ISBN 978-0-12-383842-1.
- Chacartegui, R., Sánchez, D., Muñoz, J. M., Sánchez, T., 2009. “Alternative ORC bottoming cycles FOR combined cycle power plants”. *Applied Energy*, Vol. 86, pp. 2162 – 2170.

- Gholizadeh, T., Vajdi, M. and Mohammadkhani, F., 2019. "Thermodynamic and thermoeconomic analysis of basic and modified power generation systems fueled by biogas". *Energy Conversion and Management*, Vol. 181, pp. 463 – 475.
- Hosseini, S.E., Barzegaravval, H., Wahid, M.A., Ganjehkaviri, A. and Sies, M.M., 2016. "Thermodynamic assessment of integrated biogas-based micro-power generation system". *Energy Conversion and Management*, Vol. 128, pp. 104 – 119.
- Kang, D.W., Kim, T.S., Hur, K.B. and Park, J.K., 2012. "The effect of firing biogas on the performance and operating characteristics of simple and recuperative cycle gas turbine combined heat and power systems". *Applied Energy*, Vol. 93, pp. 215 – 228. (1) Green Energy; (2) Special Section from papers presented at the 2nd International Energy 2030 Conf.
- Kang, J.Y., Kang, D.W., Kim, T.S. and Hur, K.B., 2014. "Comparative economic analysis of gas turbine-based power generation and combined heat and power systems using biogas fuel". *Energy*, Vol. 67, pp. 309 – 318.
- Macchi, E. and Astolfi, M., 2017. *Organic Rankine Cycle (ORC) Power Systems*. [S.l.]: Woodhead Publishing, 2017. ISBN 978-0-08-100510-1.
- Mudasar, R., Aziz, F. and Kim, M.H., 2017. "Thermodynamic analysis of organic rankine cycle used for flue gases from biogas combustion". *Energy Conversion and Management*, Vol. 153, pp. 627 – 640.
- Somehsaraei, H.N., Majoumerd, M.M., Breuhaus, P. and Assadi, M., 2014. "Performance analysis of a biogas-fueled micro gas turbine using a validated thermodynamic model". *Applied Thermal Engineering*, Vol. 66, No. 1, pp. 181 – 190.
- Sung, T., Kim, S. and Kim, K.C., 2017. "Thermoeconomic analysis of a biogas-fueled micro-gas turbine with a bottoming organic rankine cycle for a sewage sludge and food waste treatment plant in the republic of korea". *Applied Thermal Engineering*, Vol. 127, pp. 963 – 974.
- Szargut, J., Morris, D.R. and Steward, F.R., 1988. *Exergy analysis of thermal, chemical, and metallurgical process*. Hemisphere Publishing Corporation, 1st edition. ISBN 0891165746.
- Zareh, A.D., Saray, R.K., Mirmasoumi, S. and Bahlouli, K., 2018. "Extensive thermodynamic and economic analysis of the cogeneration of heat and power system fueled by the blend of natural gas and biogas". *Energy Conversion and Management*, Vol. 164, pp. 329 – 343.

## 7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.