THERMAL ANALYSIS IN COGENERATION PLANT USING BIOGAS IN SUGAR CANE INDUSTRY

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Abstract. *An alternative that has been shown promising to increase the cogeneration potential in sugarcane sector refers to biogas burning, from the vinasse biodigestion, as fuel in power cycle. Besides being an alternative source of energy to the alcohol plants, the conversion to biogas can minimize the environmental problems of the vinasse, that presents high pollutant potential and that is usually thrown to the soil. In this work, the use of the vinasse biogas as fuel for combined cycle using gas and steam turbines is studied in order to evaluate the energy and exergy efficiencies, and to identify the main component that increases the irreversibilities in the cycle (entropy generation). For that, the First and Second Thermodynamic Laws are applied, using mass, energy and exergy balances and the EES® (Engineering Equation Solver) software to predict the main cycle parameters. Analysing sugarcane plants with capacity to crush 300, 500 and 700 tons of cane per hour (tch), power generation of 9495 kW, 15825 kW and 22155 kW were obtained, respectively, besides of 54% of energy efficiency and 33% of exergy efficiency. It is also verified that the largest exergy destruction of the proposed cogeneration system happens in the combustion chamber, that is responsible for 60% of the irreversibilities. Regarding to the increase of the compression rate between 4 and 25, it is observed the improvement of the exergy efficiency around 10% for the combustion chamber. Besides, for the same compression rate range, the maximum efficiency occurs when the compression rate is approximately 12.*

Keywords*: cogeneration, vinasse, biogas, exergy*

1. INTRODUCTION

With the perspective of shortages of fossil fuels, followed by an increasing consumption of energy in worldwide level, it, there is a need to search new alternatives in terms of energy generation. Within this energy context, the sugarcane sector shows great potential in generation of electric through the cogeneration system. Cogeneration corresponds to the simultaneous production of different forms of useful energy, such as electromechanical and thermal energy, to fulfill the needs of a process unit, be it from the industrial, agricultural, commercial, or an isolated system, starting from a common primary energetic source (Balestieri, 2002). With the current stimulus, especially from the government, it allows Brazilian sugarcane industry to get more interested in increasing its production of exceeding electric energy. This allows the sale of electricity to utilities, thus increasing their revenue, and contributes to the Brazilian energy matrix. According to Franco (2009), the plants gradually acquire a new profile, leaving aside the name of companies "sugarcane" to become “solution companies in matters of renewable energy”.

An alternative that shows promise refers to biogas burning from the vinasse biodigestion in combined cycles, since it has a satisfactory calorific power which results in a good energetic potential. The vinasse from sugarcane is a residual liquid substance resulting from the distillation process during alcohol production. The vinasse contains high levels of organic matter, potassium, calcium and moderate amounts of nitrogen and phosphorus., Although utilized in fertigation, currently stills reveals itself as the biggest villain in industrial wastes in the sector, because it has high volume and pollution potential. It is produced at a rate of 10-14 liters of vinasse per liter of alcohol, being the biggest effluent in alcohol industry (Pinto, 1999)

When treated by anaerobic biodigestion in a suitable digester, vinasse biogas has about 60 to 65% methane and 35 to 40% of carbon dioxide (Pinto, 1999). According to Salomon (2007), biogas potential is demonstrated when treated, because its heat capacity can reach 60% of the calorific value of natural gas. The vinasse, although utilized in fertigation, currently stills reveals itself as the biggest villain in industrial wastes in the sector, because it has high volume and pollution potential. It is produced at a rate of 10-14 liters of vinasse per liter of alcohol, being the biggest effluent in alcohol industry (Pinto, 1999).

2. MATERIALS AND METHODS

For the thermodynamic analysis involved in this work are applied the equations of mass conservation, energy conservation (Thermodynamics First Law), entropy balance (Thermodynamics Second Law) and exergy balance, and this combines the principles of mass and energy conservation, together with the Thermodynamics Second Law, aiming at the design and analysis of thermal systems (Shapiro *et. al.*, 2002). In general, for processes in steady state and in which the variations of kinetic and potential energy are considered negligible, we have those following balance equations of mass, energy and exergy as seen, respectively, according to Eq. (1), Eq. (2) and Eq. (3).

$$\sum_{i=1}^{n} \dot{m}_e - \sum_{i=1}^{n} \dot{m}_s = 0 \quad (1)$$

$$\sum_{i=1}^{n} \dot{Q} - \dot{W}_{vc} + \sum_{i=1}^{n} \dot{m}_e h_e - \sum_{i=1}^{n} \dot{m}_s h_s = 0 \quad (2)$$

$$\sum_{i=1}^{n} \dot{Q}_i (1 - \frac{T_0}{T_i}) - \dot{W}_{vc} + \sum_{i=1}^{n} \dot{m}_e ex_e - \sum_{i=1}^{n} \dot{m}_s ex_s = \dot{I}_{vc} \quad (3)$$

where:

$\dot{m}_e$ - Mass flow entering the control volume (kg/s);

$\dot{m}_s$ - Mass flow leaving the control volume (kg/s);

$\dot{I}_{vc}$ - Irreversibility rate in the control volume (kW);

$\dot{Q}_i$ - Heat flux in the control volume (kW);

$\dot{W}_{vc}$ - Power related to control volume (kW);

h_e - Specific enthalpy at the entrance of the control volume (kJ/kg);

h_s - Specific enthalpy at the outlet of the control volume (kJ/kg);

ex_e - Specific exergy at the entrance of the control volume (kJ/kg);

ex_s - Specific exergy at the outlet of the control volume (kJ/kg);

T_i - Surface temperature of the control volume (K);

T_0 - Fluid temperature in the reference state (K);

When using only Thermodynamics First Law it is not possible to see clearly where is occurring the irreversibility of the process. However, exergy analysis accounts not only the amount of energy used but also the irreversibility (entropy generation) occurring on each component of the thermal system proposed.

According to Kotas (1985), the exergy is the quality standard of energy equal to the maximum useful work that can be obtained from a given form of energy using the environmental parameters (temperature and pressure) as reference. The maximum reversible work that can be extracted from a given system at a certain thermodynamic state will be given when the matter of that system to reach the dead state, in other words when the amount of mass is in mechanical and thermal equilibrium with the environment.

In this work it is considered that the thermal exergy is the sum of physical and chemical exergy as shown on Eq. (4).

$$ex_{total} = ex_{física} + ex_{química} \quad (4)$$

where:

$ex_{física}$ - Physical exergy;

$ex_{química}$ - Chemical exergy;

The specific physical exergy for a given flow in a control volume is shown as on Eq. (5).

$$ex_{física} = (h_i - h_0) - T_0 (s_i - s_0) \quad (5)$$

where:

h - Specific enthalpy to a given state i (kJ/kg);

h_0 - Specific enthalpy to a given reference state (kJ/kg);

s - Specific entropy to a given state i (kJ/kg.K);

s_0 - Specific entropy of the reference state (kJ/kg.K);

The chemical exergy, on the other hand, should be analyzed related to the fluid that makes up the air and combustible fluids separately. According to Shapiro *et. al.* (2002), the exergy for an ideal gas mixture, the temperature and pressure in the reference state, consisting of the substances present in the environment, is obtained by summing each component. That is, the result, per mole of mixture, is given according to Eq. (6).

$$\bar{e}\bar{x}_{química} = \bar{R}T_0 \sum_i y_i \ln\left(\frac{y_i}{yi^e}\right) \tag{6}$$

where:
$\bar{R}$ - Universal gas constant (kJ/kmol.K);
y_i - Molar fraction of component i in the mixture temperature and pressure of the reference state;
yi^e - Molar fraction of component i in the environment;

An alternative way to express the Eq. (7) is written as:

$$\bar{e}\bar{x}_{química} = \sum_i y_i \bar{e}\bar{x}^0{}_{iquímica} + \bar{R}T_0 \sum_i y_i \ln y_i \tag{7}$$

where:
$\bar{e}\bar{x}^0{}_{iquímica}$ - Specific standard molar exergy of the component i (kJ/kmol.K);

It was used the Eq. (6) to make the calculations of chemical exergy for the fuel gas and the Eq. (7) for the states that constitute the combined cycle.

The efficiency calculation given by the first and second thermodynamics laws applied to the appropriate equipment, as well as for the analysis of the cycle as a whole, is given respectively by Eq. (8) and Eq. (9).

$$\eta = \frac{\dot{W}}{\dot{m}\,\Delta h_{iso}} \tag{8}$$

$$\psi = \frac{\Delta h_{real}}{ex_e - ex_s} \tag{9}$$

where:
η - Greek symbol that represents the efficiency given by the thermodynamics first law;
ψ - Greek symbol that represents the efficiency given by the thermodynamics second law;
$\dot{m}$ - Mass flow (kg/s);
Δh_{iso} - Difference between the inlet and outlet enthalpies, for isentropic process (kJ/kg);
Δh_{real} - Difference between the inlet and outlet enthalpies of the device (kJ/kg);

An important parameter to be considered in the exergy analysis is the amount of exergy destroyed in each device, in other words, the contribution of each device to the overall irreversibility of the cycle. An equation to quantify the percentage of the irreversibility of each one in relation to whole plant can be defined, according to Eq. (10).

$$\delta_i = \frac{\dot{I}_i}{\sum_{i=1}^{n} \dot{I}_i} \tag{10}$$

where:
$\dot{I}_i$ - Irreversibility to a given device;
$\sum_{i=1}^{n} \dot{I}_i$ - Whole irreversibility of the cycle;

To perform the analysis mentioned, it was used the software EES® (*Engineering Equation Solver*), developed by Klein & Alvarado (1995). This software allows the determination of thermodynamic properties of the system, such as enthalpy and entropy, allowing the execution of calculations in a simple and efficient way, without the need to look for thermodynamic tables. Besides that, a computer code was implemented on the EES® platform to perform the exergy balance and also to perform simulations from parameters variation most relevant cogeneration cycle studied.

For the thermodynamic analysis performed in this work it was done a modeling of the thermodynamic equating involved in the devices that makes up the combined cycle, as was granted some assumptions similar to those used by Branco (2005), these being:

- Operation in steady state, in the operation condition of the facility;
- Atmospheric air, fuel and combustion products are treated as a mixture of ideal gases;
- Kinetic and potential energies of gases are neglected;

• Adiabatic processes in all equipment forming part of the cycles, except for the heat exchangers.

3. COGENERATION SYSTEM

The plant studied refers to a simple combined cycle, as seen in Fig. 1. The gas cycle is composed by the compressor, the combustion chamber and the gas turbine, while the steam cycle is composed by the heat recovery steam generator (HRSG), the steam turbine, the condenser and a pump. The air entering the compressor is on environment conditions.

According to Fiomari (2004), the term "combined cycle" is used for facilities which aim to use energy from the exiting gases of the steam turbine in order to generate pressured steam in the HRSG and, later, gain more electric power in a steam turbine, like shown in Fig. 1, thus being this a combination of gas and steam cycles.

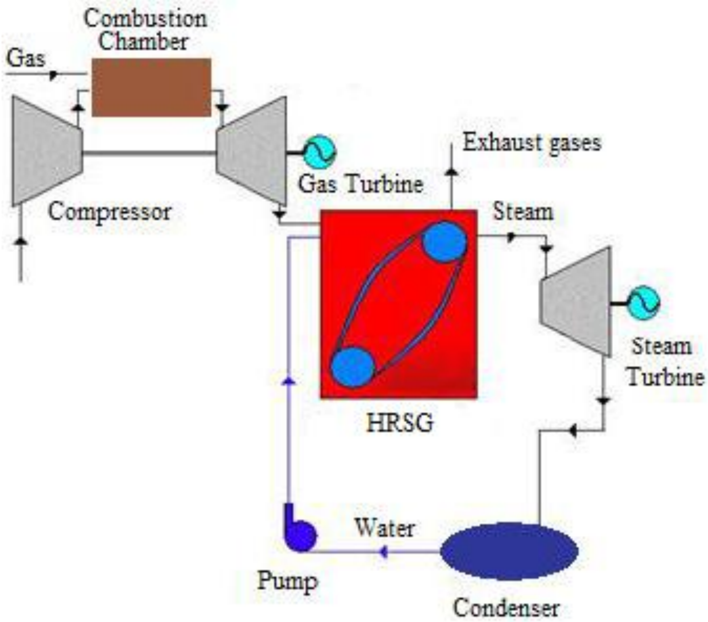

Figure 1. Combined cycle in study - Adapted from Fiomari (2004)

In the sugarcane sector, the cogeneration system occurs basically by using steam turbines. However, the technological development of components of steam turbines, which lead it to higher efficiencies, added to the fact that scape gases save between 60 and 80% of the energy input in the turbine (Fiomari, 2004), helps implementing cogeneration systems with gas turbines. The global efficiencies of conversion of energy in cogeneration systems are very high, from 70 to 90% according to Walter *et al.* (1997), much higher than the efficiencies achieved through independent systems of heat and power.

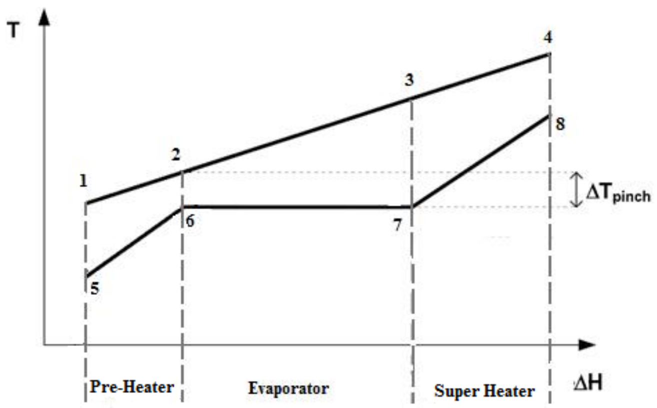

Figure 2. Temperature (T) vs. Enthalpy (ΔH) diagram of the HRSG - Adapted from Pelegrini (2009)

For making the thermodynamic analysis, some parameters were estimated, such as the turbines, pump and compressor efficiencies, from the compression ratio (π), which is the relation between the pressure of the inlet and outlet air, and the *pinch point* (ΔT_{pp}). This is defined, according to Begazo (2008), by being the point in which it establishes the minimum temperature difference in the zone limited by the curves (one hot and one cold), as seen in Fig. 2, existing in the heat transfer in the HRSG.

In Fig. 2, the curve made out from the points 1 to 4 describes the decrease of temperature and enthalpy suffered from the combustion gases during the heat transfer, nevertheless the other curve, describes the enthalpy increase of the water. According to the same figure, the *pinch point* is the difference between the steam saturation temperature (T_7) and the heat gases (T_3).

4. RESULTS AND DISCUSSIONS

The results obtained were from a combined cycle where some states were pre-established, as seen on Table 1. They were defined according the pressure (P) and temperature (T).

In the plant studied, the high and low pressure of the steam cycle are invariable, as noted in Table 1, and the temperature of each state depends on their enthalpy, which is directly associated with the enthalpy available from the flue gases exhausted from the gas cycle.

In the gas cycle, air is admitted into the compressor at natural conditions, and the temperature of the flue gases entering the gas turbine are pre-determined, and the work generated in the turbine is associated, for example, the amount of required air in stoichiometric combustion equation, as the flow and amount of methane of the gas fuel.

Table 1. Initial conditions established for combined cycle.

Combined Cycle						
Gas cycle				**Gas cycle**		
State	**P (kPa)**	**T (K)**	**Fluid**	**State**	**P (kPa)**	**Fluid**
Air entering the compressor	100	300	Air	Steam leaving the HRSG	6.000	Overheated steam
Air leaving the compressor	1.330	-	Air	Steam leaving the steam turbine	20	Biphasic mixture of water
Gases leaving the combustion chamber	1.330	1.400	Flue gas	Water leaving the condenser	20	Saturated liquid water
Gases leaving the gas turbine	100	-	Flue Gases	Water leaving the pump	6.000	Under cooled liquid water
Gases leaving the HRSG	100	-	Flue Gases			

The reference state, defined by temperature (T_0) and pressure (P_0), necessary for the exergy calculations are defined as seen in Table 2.

Table 2. Values for the reference state.

Reference state	
T_0 (K)	**P_0 (kPa)**
298	100

Based in three different configurations of sugarcane plants according to their crushing capacity, small (300 tch), medium (500 tch) and big (700 tch), we obtained the power (Table 3) from the biogas burning.

Table 3. Electric power obtained from the burning of biogas, for different plants configurations.

Plant Capacity	Gas cycle	Steam cycle	Combined cycle
Small	7.599 kW	1.896 kW	9.495 kW
Medium	12.665 kW	3.160 kW	15.825 kW
Big	17.731 kW	4.424 kW	22.155 kW

Were obtained an energetic efficiency of 54%, according to the First Law of Thermodynamics, and an exergetic efficiency of 33%, using the Second Law of Thermodynamics, as seen from Table 4.

Table 4. Energetic and exergetic efficiencies obtained.

Biogas of Vinasse					
Efficiencies (%)					
η_{gas}	ψ_{gas}	η_{steam}	ψ_{steam}	$\eta_{combined}$	$\psi_{combined}$
43,18	27,65	30,85	60,20	53,95	33,10

where:

$\eta_{combined}$ - Combined cycle efficiency;

η_{steam} - Steam cycle efficiency;

η_{gas} - Gas cycle efficiency;

To investigate the irreversibilities that occurred in each equipment, according to the sum of the irreversibilities of the complete system, presented in Figure 3. Is noted that the combustion chamber is the responsible for the largest irreversibilities with the equivalent of 60% of all exergy destruction of the combined cycle and its low exergetic efficiency. According to the Table 4, the vapor power cycle presented an exergetic efficiency larger than energy efficiency. This happens because the largest irreversibilities of this cycle type are usually associated the combustion in the boiler. As the combined cycle uses the heat recovery steam generator (HRSG) the irreversiblities are drastically reduced for the smallest temperature difference between the exhausted gases from the turbine and the generated steam. Besides, the gases still have considerable amount of energy after the exit of the HRSG, but not much exergy due to the low temperature.

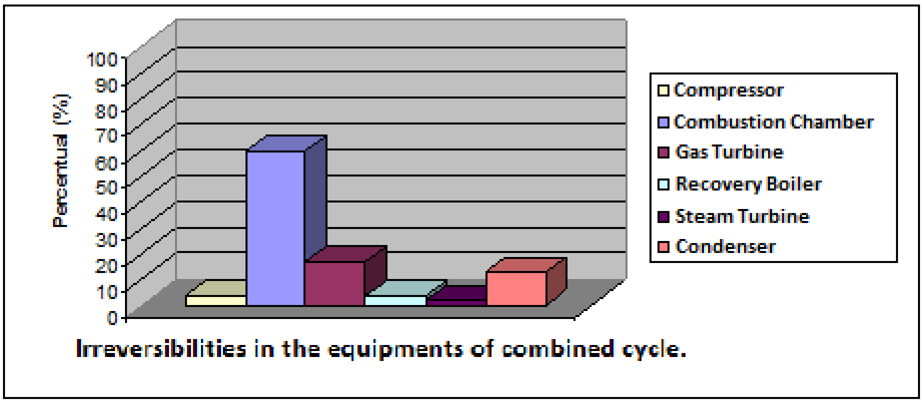

Figure 3. Percentage of the irreversibility of each equipment according to the total of the combined cycle

In this work, without being considered the economic factor related to the assembly cost of each cycle which consisted the cogeneration system, there is still, according to Fig. 3 that the irreversibilities associated with the cycle gas is higher in relation to the steam cycle. The first shows 81% of the total exergy destruction while the second shows 19%. This discrepancy is directly associated with the combustion chamber which is responsible for 74% of the irreversibility of the cycle gas.

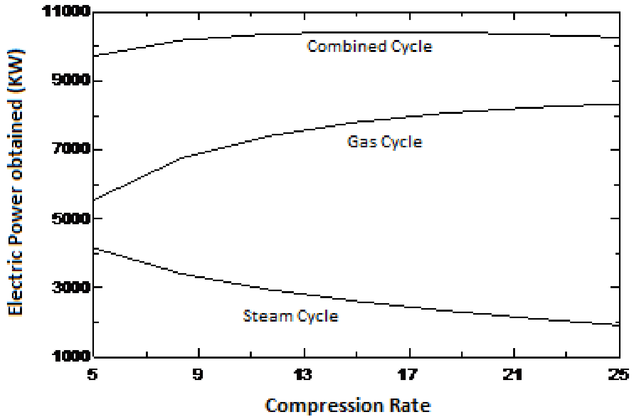

Figure 4. Electric power obtained from the variation of the rate

Using the software EES®, was made a change of the main relevant parameters on the combined cycle, such as compression ratio of the compressor and the pinch point. The case of a plant with crushing capacity of 300 tch is shown in Fig. 4, with the behavior of the observed results for the remaining mills have been similar.

From the variation of compression rate, it is possible to verify that even a raise of it, doesn't result in a variation of the power in the combined cycle, as seen in Fig. 4. This fact occurs because there is an increase in obtained power of the gas cycle, at the same proportion that the steam cycle is reduced. It is observed that an increase of the work, done by the gas turbine occurs a reduction of the thermal energy of combustion gases in the exhaust, causing also a reduction in the work of the steam turbine.

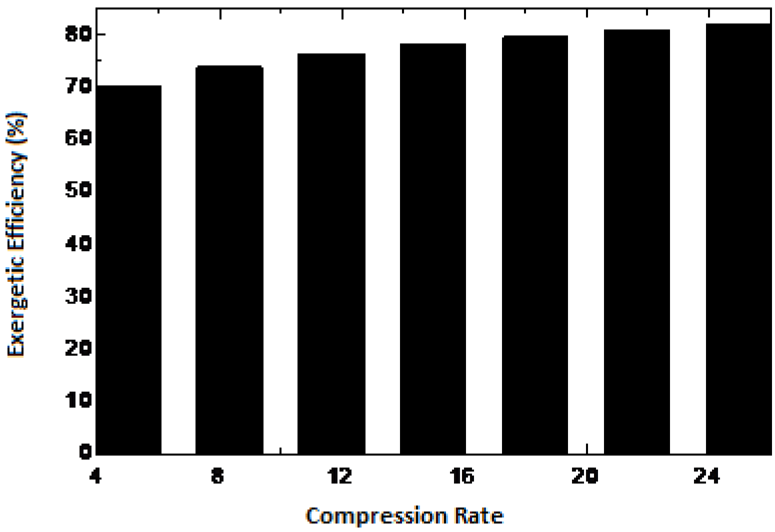

Figure 5. Variation of exergetic efficiency of the combustion chamber in a cycle

To analyse the effect of varying the compression rate according to the exergy efficiency of the combustion chamber, component of the system that showed the higher irreversibility, it is able to notice that, as seen in Fig. 5, which improves its efficiency by about 10% to a variation of compression rate from 4 to 25.

The variation of energy efficiency of the cycle according to the compression rate can be seen in Tab. 5. Note a reduction of the steam cycle efficiency caused by the decrease of the power generated by it, as well as a contrary behavior in case of a gas cycle. It is observed that the efficiency of the combined cycle has a maximum value for a compression rate of about 12 for an analyzed interval of 4 to 25. This effect is caused by the variations of the gas and steam cycles, which determinates the net power of the total cycle, as the compression rate is modified. For this maximum efficiency, it is able to see that at this point the power of combined cycle is maximum.

Tabela 5. Power variation of cycles according to the compression rate.

Cycle using the biogas of Vinasse: Plant 300tch						
$\dot{W}_{COMBINED}$ (kW)	$\dot{W}_{STEAM}$ (kW)	$\dot{W}_{GAS}$ (kW)	α	η_{STEAM} (%)	η_{GAS} (%)	$\eta_{COMBINED}$ (%)
9.275	3.743	5.532	5,00	46,29	31,43	52,70
9.485	2.746	6.739	8,33	45,43	38,29	53,90
9.515	2.131	7.383	11,67	45,09	41,95	54,06
9.459	1.680	7.779	15,00	44,99	44,20	53,75
9.351	1.315	8.036	18,33	45,03	45,66	53,14
9.209	1.003	8.205	21,67	45,17	46,62	52,32
9.038	725,2	8.313	25,00	45,38	47,24	51,36

where:

$\dot{W}_{COMBINED}$ - Power referent to the combined cycle;
$\dot{W}_{STEAM}$ - Power referent to the steam cycler;
$\dot{W}_{GAS}$ - Power referent to the gas cycle;
α - Compression rate;

In Fig. 6, is noticed a variation of produced power at a steam cycle and combined cycle according to the temperature of the pinch point.

According to Fig. 6, an increase in pinch point leads to a reduction of generated power from steam cycle and the consequence of combined cycle. What happens is that a higher interval of temperature at pinch point results in a lower quantity of transmitted energy to steam cycle, i.e., a higher irreversibility associated to the process.

According to Branco (2005), on his studies about thermoeletric plant using natural gas, the lower pinch point, the grater the efficiency of the HRSG, and consequently there is an increased produced power.

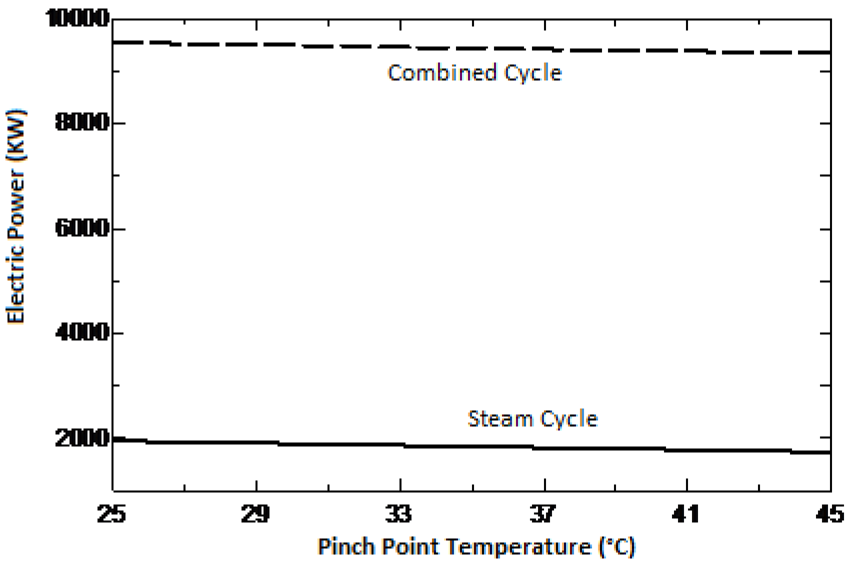

Figure 6. Variation of electrical power in the steam and combined cycle according to the temperature of the pinch point

Another parameter analyzed was the amount of existing air in the combustion reaction, which focuses on the effect of energy efficiency of the combined cycle, as well as the exergy efficiency of the combustion chamber, from the variation of the quantity of existing air in the combustion reaction. The combustion chamber is the equipment responsible for most of the irreversibility of the system, and because of that has greater importance to analyze it . As a result, it was found that an increase in the amount of air involved in the combustion reaction in an interval between ideal combustion and 300% of excess air, increased the efficiency of the equipment with the largest amount of irreversibility. This increase in efficiency occurred almost linear with a high angular coefficient, i.e. the value of efficiency showed a high variation in this interval.

It was also observed an increase in the efficiency of combined cycle, which also showed an almost linear variation, but with a smaller angular coefficient. The improvement in efficiency is due to the increase of enthalpy of combustion gases that occurs with increasing air involved in combustion, resulting in greater power generation, this fact being reflected in the efficiency of the involved cycles.

5. CONCLUSION

By seeing the results from the research, it can be seen that there is the possibility of generating electrical energy related to the use of biogas, for small, medium and large power plants. Besides being an alternative source for sugarcane power plants, the conversion in biogas can decrease the pollution from the vinasse, which is usually thrown on the soil. According to Prada *et al.*(1998) apud Lamonica (2006), the anaerobic biodigestion of the vinasse brings lots of advantages, such as the drastic reduction of its organic charge, increase of pH, which becomes around 7, keeping the same contents of NPK (Nitrogen, phosphorus and potassium) of the vinasse in natura, conservating its fertilizing power and eliminating its unpleasant smell.

Through the simulations it could be verified more clearly how the main parameters of the cycle affect it. In order to improve the cycle, the main element to be studied is the combustion chamber, since it is responsible for the greatest exergy destruction of the system, according to the energy balance.

Even though it was verified the benefits of using the biogas in combined cycles, they will only become a reality in the sugarcane plants when exists incentives to its application, through a higher sale price of the energy produced that may justify the large amount of money spent in the modifications of the plant, such as the installation of the gas cycle and the equipment to the anaerobic biodigestion of the vinasse. Added to this it is necessary to implement development policies which would define clear and encouraging conditions. As quoted by Graciano (2007), technically, the technology is already mature, due to the many experiences in demonstrative scale. It would be important to implement some test units, where resources could be applied to evaluate and solve some uncertainties, such as biogas technique cleaning and digestion stability and it could reduce the financial risks by installing a combined cycle.

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