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EXERGETIC ANALYSIS OF THE COGENERATION BY USING SUGARCANE BAGASSE IN SUGAR PLANT

Luciana Faria Saint-Martin Pereira
Gabriel Costa Guerra Pereira
Cristiane Aparecida Martins
Instituto Tecnológico de Aeronáutica, Praça Marechal Eduardo Gomes, 50, São José dos Campos, SP
lpereira@ita.br, gpereira@ita.br, cmartins@ita.br

Abstract. *The exergetic analysis was employed to evaluate the sugar and alcohol plants performance. Three literature plants and one real were analyzed, to results validate. The real plant found results were: energetic and exergetic cogeneration efficiency (84.5% and 28.7%), energetic and exergetic boiler efficiency (85.0% and 32.2%), boiler and turbo-generator exergy destruction (95.1% and 3.8%). The thermodynamics first law alone does not evaluate the performance correctly. Many exergy is destroyed in the boiler due the combustion irreversibility, explaining the low exergetic efficiency. To improve the system performance, a greater attention must be give mainly to the boiler, and next, to the turbo-generator turbine.*

Keywords: *exergy, availability, cogeneration, bagasse, sugarcane plant*

1. INTRODUCTION

The cogeneration - defined as a process of simultaneous production of mechanical energy (or electric) and thermal energy, allows the optimization and the increase of the performance in the systems of conversion and utilization of energy - it is a process that allows the sugar and alcohol sector to have one more product to sell beyond the sugar and alcohol: the electrical energy produced that exceeds the electrical energy consumed by the plant itself. In this way, it is possible to increase further more the profits of the sector. The cogeneration is also advantageous for the environment because the burning of the bagasse configure a nobler final disposition of this waste and reduces the emission of air pollutants.

The thermodynamic analysis proposed is a method that allows to identify and quantify the inefficiencies of a processes, making possible, to determine where investments should be made (if economically viable) in order to improve the efficiency of the conversion of a fuel (which is in this case, sugar cane bagasse) in electric energy.

The thermodynamic analysis of sugarcane plants is not a common practice in Brazil, where the majority of published studies is based only on the first law of thermodynamics, as presented by Coelho (1999) and by Fórum de Co-Geração (2002). The main disadvantage of this method is that it is based on a quantitative approach, in terms of energy, and does not consider the loss of the quality of the energy in the processes. Nevertheless, some authors have proposed a method of analysis involving the first and the second law of thermodynamics combined - known as exergetic analysis or availability analysis. Among these authors, we can cite: Moran (1982), Szargut (1988) and Li (1995). Using this method of analysis, it was developed a methodology for carrying out the thermodynamic analysis of cogeneration in sugarcane plants.

The exergy (or available energy) is an orderly form of energy with unvarying quality and totally capable of being converted, through the interaction with work, in other forms of energy. The exergy can be understood as the maximum theoretical work obtained by interactions of the system with the environment. And the exergetic analysis is a tool used to identify and quantify the destruction of exergy in a transformation. Thus, it is possible to study ways to reduce them, where they are most significant, and thus, make the process more efficient.

1.1 Sugarcane

According to Eletrobrás (1993), up to 1974, the cane industry used to import part of the energy used in the processes in the form of oil fuel, wood and electricity. With the impacts caused by abrupt increases in the oil price, in 1973 and 1979, which reached harshly the economy of the country, the agriculture industry of cane sought to replace the fuels for sugar cane bagasse, providing the vapor and the electricity required in the production process of sugar and alcohol. The bagasse was sold to produce cattle feed, but currently there are opportunities extremely good for its use in the sale of electric energy exceeding.

Brazil is the largest producer of sugarcane of the world. The newspaper Estado de Minas (2006) showed that the Brazilian harvest of sugarcane in 2005/2006 was closed in 436.8 million tons, the highest up to that year. The production continued to grow, according to Brazilian Energy Balance (2011), the sugarcane production in the year of 2010 reached 627.3 million tons. This was 0.8% higher than the production of the previous year when the harvest was 622.6 million tons. Figure 1 shows, still according to Brazilian Energy Balance (2011), that the Brazilian production of

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sugar cane bagasse increased in the last 10 years, as well as the amount of bagasse used as input for the generation of electricity also increased.

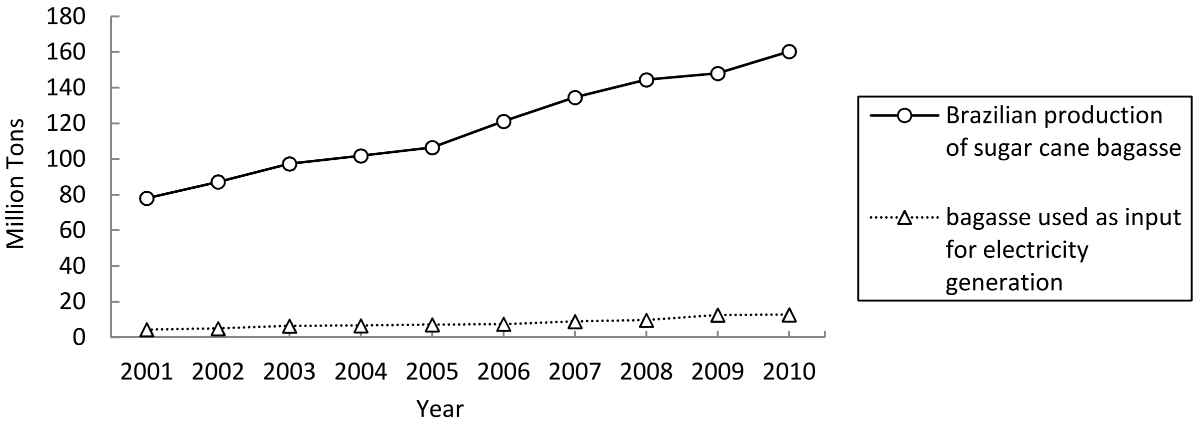

Figure 1. Brazilian production of sugar cane bagasse in the last 10 years.

The residues of sugarcane (bagasse and straw, in particular) have a very large potential of energy, which can be better exploited. The energy content present in one ton of sugarcane is about 7,560MJ and the bagasse has a very high energy potential of 2,630 MJ (as far as of the straw with 2,630MJ as well) and, surprisingly slightly larger than the alcohol (with 2,300MJ), according to Unica (2002).

The use of biomass in the sugar and alcohol sector, in substitute of oil, coal and other fossil fuels, reduces the emission of atmospheric pollutants, mainly by the absence of sulfur, according to Leal (2003).

2. METHODOLOGY

The method of exergetic analysis of cogeneration in a sugar and alcohol plant was developed and applied to a typical plant described in the literature, with three different configurations of cogeneration, being them: Case 1, 2 and 3. To test it and to verify its appropriateness, the method was applied to a real case, the Coruripe Plant, subsidiary of Campo Florido, located on Triangulo Mineiro, in Minas Gerais, Brazil. The energetic and exergetic analyzes were performed for the current configuration of the cogeneration system on this plant using the data obtained from its operation.

2.1 Selection of Cases

To perform the energetic and exergetic analyzes, it was take as reference a sugarcane plant of mid size, typical, with national technology (that is, Brazilian technology), milling 490 tons of cane per hour (tch), which corresponds to the production of approximately 41,000 liters of alcohol per day[1]. It was studied three typical plants described in the literature with different configurations of cogeneration (Cases 1, 2 and 3). These three configurations were based on data from Fórum de Co-Geração (2002) and there they appear with the following names: "Balanço Atual", "Uso do Bagaço Sobrante" and "Alta Pressão", respectively. In Fórum document are given the mass flow of vapor and fuel, pressure in the boiler, mechanical power utilized by the machines of the process of production of alcohol, electric energy generated and consumed by the plant itself. The cases that were chosen to be analyzed are described below:

Case 1: The plant contains a pressure reducing valve which meets the demands of low pressure vapor for the process of alcohol production. In this case, are produced 6,373 kW of electrical power, using a multi-stage turbine (MS) associated with a generator installed, with more than 8,039 kW of mechanical power to actuate the machines of the process, with turbines of single-stage (SS). The machines of the process are the machines used to extract the cane juice and are actuaded when the turbine axis coupled to the machines rotates. The machines are: picker, defibrator and mills. In this case, all the electric energy produced by the plant is also consumed by the plant itself and, therefore, there is no surplus of electricity to be sold. A portion of the bagasse is burned in the boiler. The other part, corresponding to 19.7% of the bagasse produced, is not burned in the boiler. Currently, many Brazilian sugarcane plants still have this configuration. The Case 1 is shown in the Fig. 2 with the values of mass flow rate of vapor and bagasse consumption in kg/h.

By the Fig. 2 it can be observed a harvest of 1,800,000 tons of cane. As the plant has 3,672 hours of generation (value provided by the Fórum de Co-Geração, 2002), the milling of cane per hour will be approximately 1,800,000 t divided by 3,672 h is equal to 490 tons of cane per hour. The figure shows that 490 tch enters into the mills and produce

[1] According to CTC (Copersucar, 2003), 1 ton of cane produces approximately 83 liters of alcohol.

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132,353 kg/h of bagasse. Of this total, 106,280 kg/h of bagasse is burned in the boiler and 26,073 kg/h of bagasse will remain (corresponding to 19.7% of the bagasse produced). The boiler of 2.1 MPa produces 255,073 kg/h of vapor. The vapor line is divided in two, the first one, is directed to the "other consumers", to the turbines of the machines of the process and to the turbine of the generator of electric energy installed, and the second line of vapor is directed to the valve of reducing pressure. The vapor at the exit of the turbine is joined to the vapor that passes through the pressure reducing valve and the resulting vapor is directed to the production process of sugar and alcohol, passing before by a desuperheater.

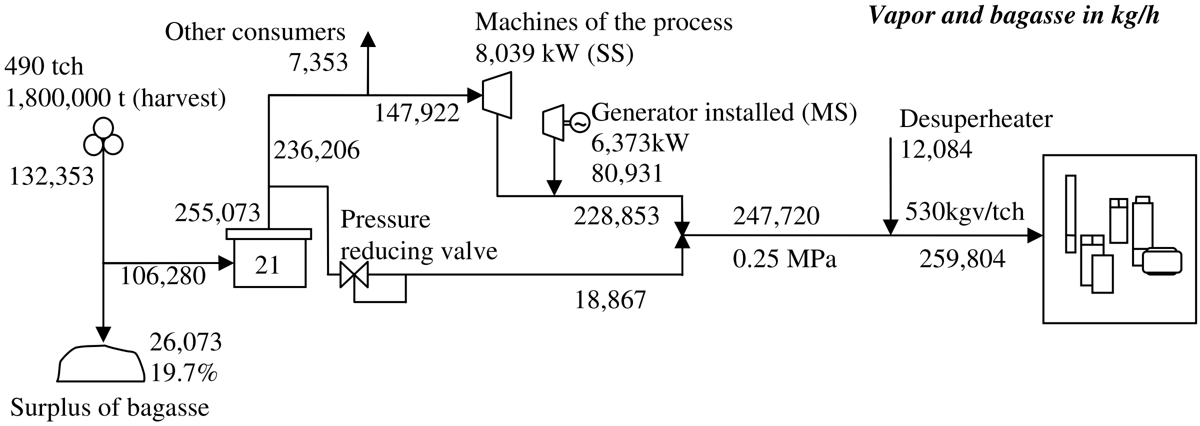

Figure 2. Flowchart of the plant in Case 1 (Fórum de Co-Geração, 2002).

Case 2: the pressure reducing valve of the Case 1 is substituted by a turbine of controlled extraction and condensation, coupled to a new generator. Moreover, all the bagasse is burned and thus, there is no surplus of bagasse. With the substitution of the turbine, it was obtained the reduction of the vapor pressure necessary to the production process of alcohol, and it becomes possible to generate electric energy in excess to be sold. Thus, it was generated the 6,373 kW of electrical power as shown in Case 1, and it was possible to generate even more 15,412 kW of electric energy with the new generator of controlled extraction and condensation (CD). The substitution of the pressure reducing valve by the turbine of new generator, with technology available in the national market, that is, Brazilian market, associated with the burning of the surplus of bagasse, can make that the electrical power produced by the plant triple, that is, going from 6.4 MW to 21.8 MW. The Case 2 is shown in Fig. 3.

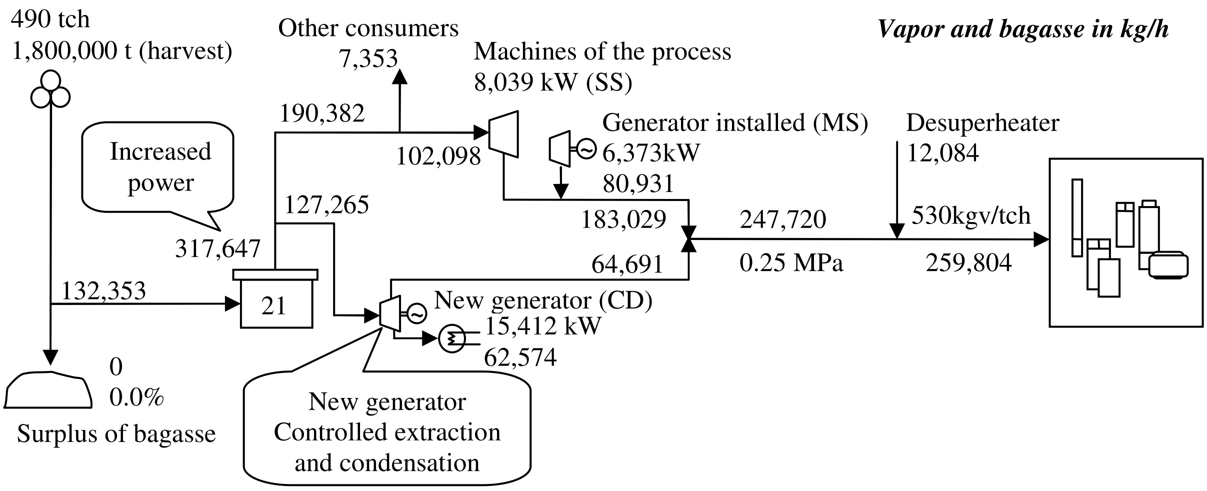

Figure 3. Flowchart of the plant in the Case 2 (Fórum de Co-Geração, 2002).

By the Fig. 3, it can be observed that all the bagasse produced, at a consumption rate of 132,353 kg/h originating from the milling of 490 tch are directly driven to the boiler and, therefore, there are no surplus of bagasse. The boiler of 2.1 MPa passes, then, to produce 317,647 kg/h of vapor, increasing the power. That is, this mass flow of vapor is 24.5% higher than the flow of vapor produced by the boiler in the Case 1, which was 255,073 kg/h. The vapor line is divided in two: the first line is directed to the "other consumers," for the turbine of the machines of the process,

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and for the turbine of the generator installed, and the second line is directed to the turbine to of the new generator. The turbine of the new generator has an intermediate extraction, which goes to the production process of sugar and alcohol, and has another extraction of vapor which goes to the condenser. The vapor lines originating from the exits of the turbine of the generator installed and of the turbine of the controlled extraction are united and are directed to the production process of sugar and alcohol, passing before for a desuperheater. The turbine of the new generator is a turbine of controlled extraction and condensation, which replaced the pressure reducing valve on the Case 1.

Case 3: the boiler of low pressure (2.1 MPa) is replaced by a higher pressure boiler of 8.1 MPa. The high pressure vapor at the exit of the boiler goes directly to a new generator. The turbine of the generator installed is previously disabled and the production of electric energy will be entirely performed only by the turbine of the new generator. With these attitudes, the total electric power produced by the plant may reach 47.7 MW, that is, it would increase approximately seven times, would rise from 6.4 MW in Case 1, to 47.7 MW in Case 3. The Case 3 is shown in Fig. 4 with the values of mass flow of vapor and bagasse consumption in kg/h. The turbine of the new generator has two controlled extraction. In the first extraction, the vapor of high pressure (2.1 MPa), according to Fórum de Co-Geração (2002), goes directly to the turbine of the machines of the process. In the second extraction, the vapor of intermediate pressure (0.25 MPa) is mixed with the vapor that exits from the turbine of the machines of the process in the mixing chamber. This mixture is directed to the production process of sugar and alcohol, passing before for a desuperheater. In the exit of the turbine of new generator, the vapor at low pressure (0.01 MPa), enters into a condenser. Thus, the turbine of new generator, which in the Case 2 generated 15,412 kW, now passes to generate 47,737 kW, that is, the electrical power of this turbine tripled.

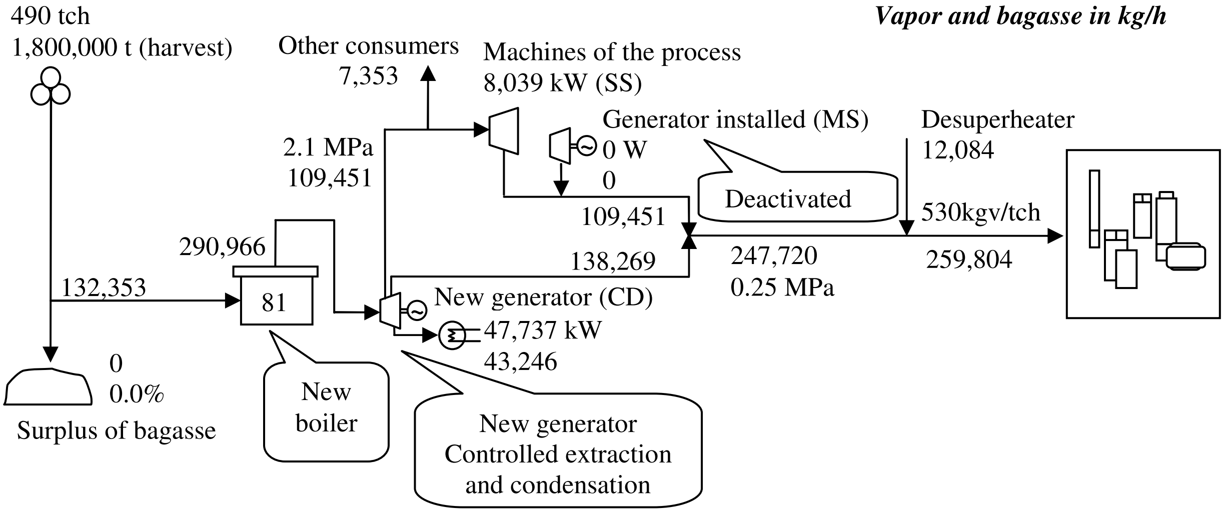

Figure 4. Flowchart of the plant in Case 3 (Fórum de Co-Geração, 2002).

The Coruripe plant located in the city of Campo Florido in MG, Brazil, operates with two boilers, represented by only one in the diagram of Fig. 5. For this plant, the term "other consumers" is used for the vapor that is injected, for example, to dry sugar and yeast, to boil soda in the boxes of evaporation and to clean the cookers, and does not return to the system. The production process of sugar and alcohol operates with a pressure of 0.25 MPa. The bagasse consumption in the boiler of Coruripe plant (34.56 kg/s), results from part of the feeding of 490 tons of cane per hour in the mills, because not all the bagasse generated in the plant is burned. Figure 5 shows the flowchart of Coruripe plant used to perform the energetic and exergetic analysis, based on the data collected during the visit to the Plant. It was not possible to obtain complete data about the functioning of the desuperheaters.

The method consists in making energetic analysis (using the first law of thermodynamics) and exergetic (using a combination of first and second law) for each component individually, assuming that they operate in steady state (hypothesis something distant from the actual operation, but that simplifies the analysis), and then to make energetic and exergetic analysis of the cogeneration system as a whole. The energetic analysis, although unnecessary, serves to check the values obtained by calculations (for example, of energetic efficiencies).

From the data provided by the real plant (Coruripe), it was possible to perform the energetic and exergetic analyzes. That is, to perform the calculations of the energetic and exergetic efficiencies, for each equipments as well as for the cogeneration system as a whole. More details about this work can be extracted from Saint-Martin (2005). It is also found in Saint-Martin (2005), the complete energetic and exergetic analyzes of the three configurations of cogeneration systems for the three literature plants and for the real plant (Coruripe).

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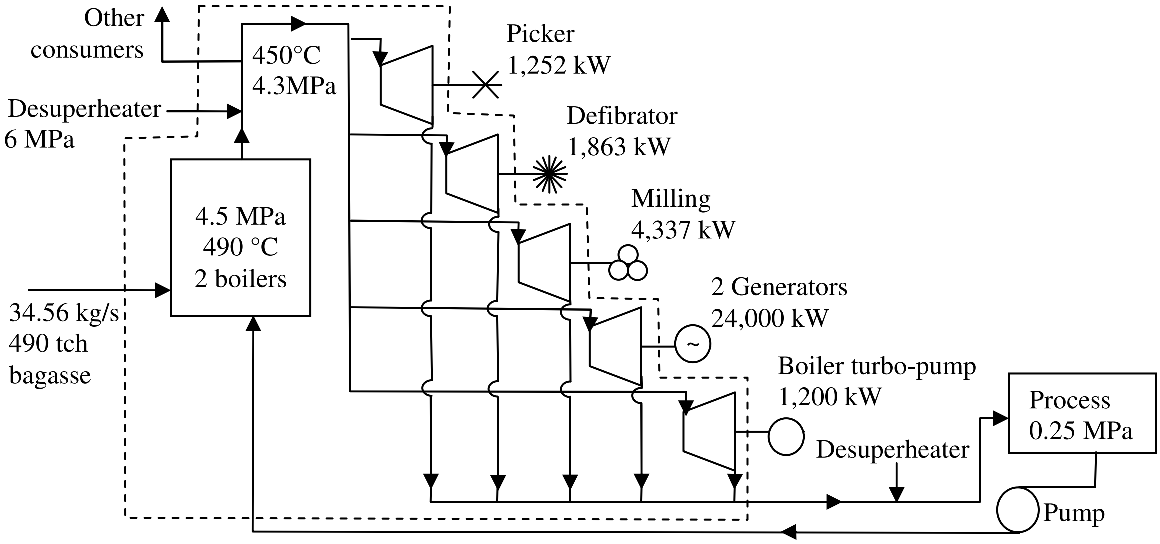

Figure 5. Schematic diagram of Coruripe plant in Campo Florido (courtesy of plant).

2.2 Thermodynamics Theoretical Foundations

The theoretical foundations for performing the calculations of energetic and exergetic analysis comes from thermodynamics. The following mass rate balance and energy rate balance are based on the writings of Moran and Shapiro (2000).

Energetic Analysis

For a control volume, that is, a system on which the mass flows through its control volume boundary, in steady state, the rate of mass flow does not vary with time, that is, $\mathrm{dm_{cv}/dt} = 0$. The subscript (cv) means control volume. Hence, the mass flow rate that enters into the control volume is equal to the mass flow rate that exits the control volume. Thus, the mass rate balance is:

$$\sum_i \dot{m}_i = \sum_e \dot{m}_e \tag{1}$$

where ($\dot{m}_i$) and ($\dot{m}_e$) are the mass flow rate at the inlet and exit, respectively. The summation indicates that there may be several inlets and exits. The energy rate balance for a control volume can be expressed as:

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) \tag{2}$$

where ($\dot{Q}$) is the heat transfer rate, ($\dot{W}$) accounts for flow work (or power), (h) the specific entalpy, (V) velocity, (g) acceleration of gravity and (z) height.

Mass rate balance and energy rate balance of the boiler

The mass rate balance and the energy rate balance of the boiler can be deduced with the aid of Fig. 6. The volume control defined for the mass rate balance on the boiler includes only the mass flow rate of water. As the boiler has only one input and one exit for the mass flow rate of water, the mass rate balance shown in Eq. (1) becomes:

$$\dot{m}_i = \dot{m}_e \tag{3}$$

It is considered that the boiler does not perform work, since the amount of work done due to the dilation of the boiler is negligible in relation to the amount of heat produced. Hence, the Eq. (2) becomes:

$$\dot{Q}_{boi} = \dot{m}_i (h_e - h_i) \tag{4}$$

where ($\dot{Q}_{boi}$) is the heat transfer rate received by the mass flow of water in the boiler, to perform a change of phase from liquid to vapor, (h_e) is the especific entalpy of the exiting vapor and (h_i) is the especific entalpy of the inlet vapor.

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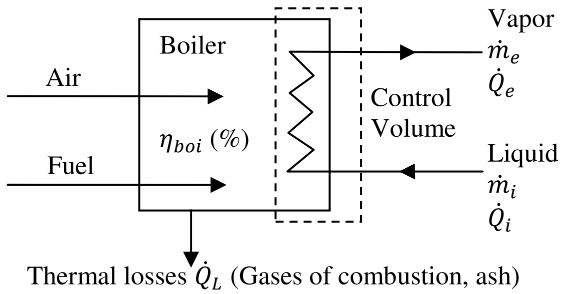

Figure 6. Mass rate balance and energy rate balance of the boiler.

The heat transfer rate supplied to the water of the boiler ($\dot{Q}_{boi}$) is originated from the thermal energy of the fuel combustion, and according to Prasad (1995), a part of this heat transfer rate is lost through the exhaust gases (which represents the most part), lost by radiation and convection, lost due to incomplete combustion and lost with the thermal energy from the ash. This heat transfer rate loss, called ($\dot{Q}_L$), can be calculated by the difference between the heat transfer rate supplied by the fuel ($\dot{Q}_{fue}$), and the heat transfer rate delivered to the water in the boiler:

$$\dot{Q}_L = \dot{Q}_{fue} - \dot{Q}_{boi} \quad (5)$$

The heat transfer rate from the combustion is given by:

$$\dot{Q}_{fue} = \dot{m}_{fue} LHV_{fue} \quad (6)$$

where (LHV_{fue}) is the lower heating value of the fuel. It was adopted the LHV of the bagasse with 50% of humidity of $LHV_{bag} = 7{,}536\ kJ/kg$. The energetic efficiency of the boiler (η_{boi}) can be defined as the ratio between the amount of thermal energy received by the water to vaporize and the amount of thermal energy supplied by the fuel:

$$\eta_{boi} = \frac{\dot{Q}_{boi}}{\dot{Q}_{fue}} = \frac{\dot{m}_i(h_e - h_i)}{\dot{m}_{fue} LHV_{fue}} \quad (7)$$

Mass rate balance and energy rate balance of the turbine

The mass flow rate on the inlet ($\dot{m}_i$) of the control volume is equal to the mass flow rate on the exit ($\dot{m}_e$) of the control volume. Therefore, as the turbine has only one input and one output, the mass rate balance presented in the Eq. (1) becomes:

$$\dot{m}_i = \dot{m}_e \quad (8)$$

The turbine produces a power ($\dot{W}$) which can be converted into electric power (if the turbine is coupled to an electrical generator) or be used as a mechanical power (if the turbine is coupled to the process machine). The turbine may be considered adiabatic because the amount of heat transferred to the surroundings is negligible in relation to the amount of work done. The energy rate balance of the turbine can be calculated as:

$$\dot{W} = \dot{m}_i(h_i - h_e) \quad (9)$$

The vapor can be expanded in a turbine in one or more passages, called stages. The turbine with the extraction of vapor is composed of more than one stage. Figure 7 shows the process of expansion on the diagram for a turbine of a single stage. The process (ab) is an isentropic process, that is, is a process where the entropy does not vary between the inlet and the exit of the turbine. The isentropic process is an ideal reversible process. The concept of reversible process will be presented later, when describing the second law of thermodynamics. The process (ab') is the actual process. The pressure at the inlet is represented by the isobaric curve (p_a), the pressure at the exit is represented by the isobaric curve (p_b).

The isentropic efficiency of the turbine (η_{tb}) is the ratio between the actual work and the reversibly work (or ideal). In this manner, the isentropic efficiency can be considered as a distance measure that the actual work is from the ideal work. Thus, the closer the actual work is from ideal work, the better is the isentropic efficiency of the turbine. The isentropic efficiency of the turbine can be calculated by the Eq. (10).

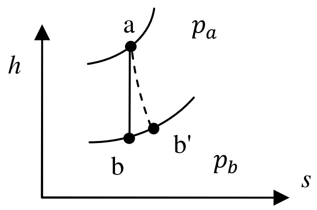

Figure 7. Expansion of the vapor in a single stage turbine (Sears, 1969).

$$\eta_{tb} = \frac{W_{act}}{W_{rev}} = \frac{h_i - h_{eR}}{h_i - h_{eI}} \tag{10}$$

where (R) means the real process (or actual process) and (I) ideal process. The turbine of the controlled extraction can be represented as shown in Fig. 8, according to Li (1995). In this case the turbine will have three extractions. If one of the extractions is coupled to a condenser, then the turbine may be referred as turbine of controlled extraction and condensation.

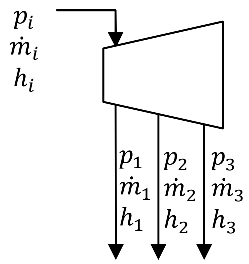

Figure 8. Model of a controlled extraction turbine.

The description of the process performed by the turbine in Fig. 8 can be performed with the aid of Fig. 9:

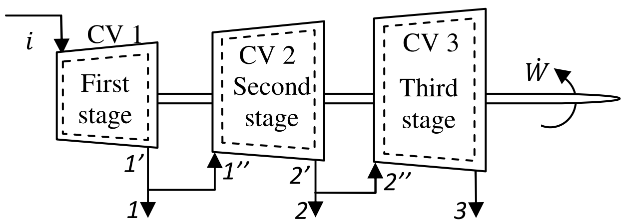

Figure 9. Expansion process in a turbine of multiple (three) stages.

The mass rate balance for this turbine are:

$$\dot{m}_i = \dot{m}_{1'} \tag{11}$$
$$\dot{m}_{1'} = \dot{m}_1 + \dot{m}_{1''} \tag{12}$$
$$\dot{m}_{1''} = \dot{m}_{2'} \tag{13}$$
$$\dot{m}_{2'} = \dot{m}_2 + \dot{m}_{2''} \tag{14}$$
$$\dot{m}_{2''} = \dot{m}_3 \tag{15}$$

And the energy rate balances are:

$$\dot{Q}_i = \dot{Q}_{1'} + \dot{W}_1 \tag{16}$$
$$\dot{Q}_{1'} = \dot{Q}_1 + \dot{Q}_{1''} \tag{17}$$
$$\dot{Q}_{1''} = \dot{Q}_{2'} + \dot{W}_2 \tag{18}$$
$$\dot{Q}_{2'} = \dot{Q}_2 + \dot{Q}_{2''} \tag{19}$$
$$\dot{Q}_{2''} = \dot{Q}_3 + \dot{W}_3 \tag{20}$$
$$\dot{W}_{tb} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 \tag{21}$$

where ($\dot{W}_{tb}$) correspond to the potencie developed by the whole turbine. According to Sears (1969), the majority of the turbines has more than one stage. Nevertheless, the relationship developed for the turbine of a single-stage are equally valid for any turbine considered as a whole. Thus, the expansion process of the steam in a turbine of many stages can be represented on the diagram of the Fig. 10. Assuming that the expansion process follow the path $abb'c'c''d'$, the efficiency of one stage is the ratio of the actual work performed by the stage and the ideal work of this stage. Thus, the sum of the actual work performed by the stages must be equal to the actual work performed by the turbine. Assuming a unique value (η_{tb}) for the efficiency of each stage, it follows that:

$$\eta_{tb} = \frac{h_a - h_{b'}}{h_a - h_b} = \frac{h_{b'} - h_{c''}}{h_{b'} - h_{c'}} = \frac{h_{c''} - h_{d''}}{h_{c''} - h_{d'}} \tag{22}$$

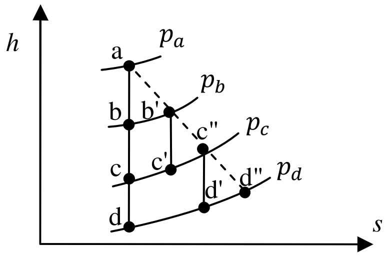

Figure 10. Expansion of the vapor in a multiple stage turbine (Sears, 1969).

Mass rate balance and energy rate balance of the mixing chamber and of the desuperheater

Both desuperheater and mixing chamber are heat exchangers where the streams are mixed. The mass rate balance and the energy rate balance for the mixing chamber are identical to the desuperheater. In both cases there are mixture of two streams of fluid that enter into the device, and a single stream that exits. Both devices are considered adiabatic and do not perform work. The mixing chamber has only the function of mix the streams of the fluids which enters on it. The desuperheater mixture two miscible fluids, such as hot water and cold water, and can be used when it is desired to reduce and control the temperature of the superheated vapor (Baptista e Martins, 2002). The control is made by injection of cooling water in the pipe of superheated vapor. It is basically composed of a control valve of spraying (Celulose On Line, 2004).

Energetic efficiency of the cogeneration

The global energetic efficiency of a conventional thermal power plant can be calculated by dividing the net work produced and the heat supplied by the boiler:

$$\eta = \frac{\dot{W}_{net}}{\dot{m}_{fue} LHV_{fue}} \tag{26}$$

where ($\dot{W}_{net}$) is the net power, ($\dot{m}_{fue}$) and (LHV_{fue}) are the mass flow rate and lower heating value of the fuel, respectively. However, these systems are designed to generate electrical power and/or useful work. In a cogeneration system, in addition to generating useful work, the goal is also to generate the heat used in the process (in the case of sugar and alcohol plants, the useful heat is used in the production process of the sugar and alcohol). Thus, for the cogeneration systems, the global thermal efficiency should take into account not only the net work, but also the useful heat. Thus, the global thermal efficiency can be defined as follows (Li, 1995):

$$\eta_{cg} = \frac{\dot{Q}_{use} + \dot{W}_{eo}}{\dot{m}_{fue} LHV_{fue}} \tag{27}$$

where (η_{cg}) mean energetic efficiency of the cogeneration, ($\dot{Q}_{use}$) the rate of heat transfer used in the production process and ($\dot{W}_{eo}$) the electrical power.

Exergetic analysis

When a system is in equilibrium with the environment, it will not occur spontaneously change of state and the system will not be able to do work, the pressure and the temperature of the system will be equal to the environment, there is no existence of chemical reaction, the system has zero velocity and has a minimum potential energy (Wylen *et al.*, 2001). But, if there is an imbalance between the system and the environment, exists the opportunity to develop a

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work (Moran e Shapiro, 2000). The heat flows spontaneously from hot to cold body. Work can be completely converted into heat, but heat can not be totally converted into work. Hence arises the concept of irreversibility of real processes. What impede the realization of the maximum theoretical work are the irreversibilities of the system. In an irreversible process, the system and all its sorroundings can not be exactly restored to their initial states after occurs a process. If a system pass through a process completely reversible until it reaches the state of equilibrium with the environment, the system would have performed the maximum theoretical work. Reversible process is when a system and its whole neighborhood can have their initial state restored after the occurrence of a process. There is no real process reversible. The exergy is the maximum theoretical work obtained by the interactions of the system with the environment. The exergy depends on an environment of reference; the further the system is from this reference, the greater its exergy. When the system enters in equilibrium with the environment the entropy is zero. The value of exergy can not be negative. The exergy is not conserved, but is destroyed because of the irreversibilities. The exergy rate ($\dot{B}$) can be calculated as (Li, 1995):

$$\dot{B} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{28}$$

The destruction of exergy ($\dot{B}_d$) can be written, in function of the exergy rate that accompany the transfers of energy by heat ($\dot{B}_q$) and by work ($\dot{B}_w$), as (Li, 1995):

$$\dot{B}_d = \sum \dot{B}_i - \sum \dot{B}_e - \dot{B}_q - \dot{B}_w \tag{29}$$

Exergy rate balance in the boiler

The exergy destruction rate in the boiler can be deduced by Fig. 11 and Eq. (29). The exergy rate that accompanies the work ($\dot{B}_w$), is null. Thus,

$$\dot{B}_d = (\dot{B}_i + \dot{B}_{fue}) - (\dot{B}_e) - \dot{B}_q \tag{30}$$

$$\dot{B}_q = \left(1 - \frac{T_0}{T_L}\right) \dot{Q}_L \tag{31}$$

where ($\dot{Q}_L$) is the heat transfer rate where the temperature is (T_L). The fuel exergy rate ($\dot{B}_{fue}$) is given by Eq. (30), where ($\dot{m}_{fue}$) is the fuel mass flow rate and (b_{fue}) is the fuel specific exergy whilst the exergetic efficiency of the boiler (ε_{boi}) is given by Eq. (31):

$$\dot{B}_{fue} = \dot{m}_{fue} b_{fue} \tag{30}$$

$$\varepsilon_{boi} = \frac{\dot{B}_e - \dot{B}_i}{\dot{B}_{fue}} \tag{31}$$

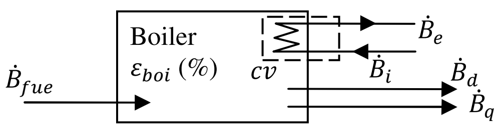

Figure 11. Exergy rate through the boiler.

Exergy rate balance in the turbine

Reversible work ($\dot{W}_{rev}$) is when the entropy of the control volume does not vary, the turbine is considered isentropic. So, (h_{sI}) is the fluid specific enthalpy at the exit of the turbine in the ideal state.

$$\dot{W}_{rev} = \dot{m}_i(h_i - h_{sI}) \tag{32}$$

Real work ($\dot{W}_{act}$) is affected only by the irreversibility at the passage of the water through the turbine. The external losses incurred due friction in the bearings, gears, accessories, etc., and can not influence on the state of the vapor (Sears e Lee, 1969):

$$\dot{W}_{act} = \dot{m}_i(h_i - h_e) \tag{33}$$

The efficiency of the turbine allows to measure how close the actual work is from the ideal work:

$$\eta_{tb} = \frac{\dot{W}_{act}}{\dot{W}_{rev}} \tag{34}$$

The available work ($\dot{W}_{disp}$) is the most important type of work because it represents the maximum amount of work that could be produced between two states. ($\dot{B}_{eR}$) is the exergy rate at the exit of the turbine with the vapor in the real process:

$$\dot{W}_{disp} = \dot{B}_i - \dot{B}_{eR} \tag{35}$$

The lost work ($\dot{W}_L$) on the turbine can be calculated by Eq (36), the exergy destruction rate on the turbine, is given by Eq (37) and the exergetic efficiency of the turbine (ε_{tb}) is the ratio between the actual and available work (Eq 38):

$$\dot{W}_L = \dot{W}_{rev} - \dot{W}_{act} \tag{36}$$

$$\dot{B}_d = \dot{W}_{disp} - \dot{W}_{act} \tag{37}$$

$$\varepsilon_{tb} = \frac{\dot{W}_{act}}{\dot{W}_{disp}} \tag{38}$$

Exergetic efficiency of the cogeneration

The energetic efficiency of the cogeneration does not take into account the quality of the energy and its degradation in the real processe, whilst the exergetic efficiency (ε_{cg}) does take. For example, the electrical energy has much more exergy than the thermal energy, especially thermal energy with low temperature and low pressure (Li, 1995). The exergetic efficiency of the cogeneration can be defined as the ratio between the useful exergy rate of the cogeneration system and the exergy consumption rate of the fuel, and can be deduced with the aid of Fig. 12, resulting in:

$$\varepsilon_{cg} = \frac{\dot{B}_{eo}+\dot{B}_{mec}+\dot{B}_{proc}+\dot{B}_{oc}}{\dot{B}_1+\dot{B}_{bag}+\dot{B}_{des}} \tag{39}$$

where ($\dot{B}_{eo}$) is the exergy rate of the electrical and of the mechanical ($\dot{B}_{mec}$) enegy generated, and (*proc*) is production process of sugar and alcohol. Similarly, it was calculated the energetic efficiency of cogeneration (η_{cg}) can be deduced by:

$$\eta_{cg} = \frac{\dot{W}_{eo}+\dot{W}_{mec}+\dot{Q}_{proc}+\dot{Q}_{oc}}{\dot{Q}_1+\dot{Q}_{bag}+\dot{Q}_{des}} \tag{40}$$

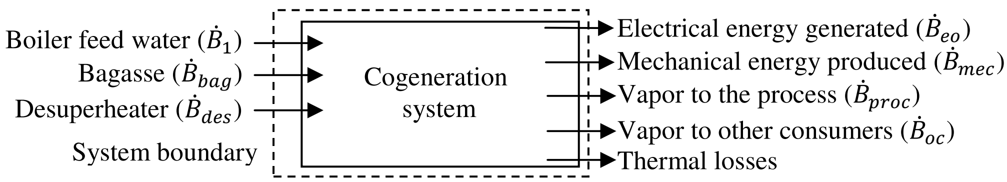

Figure 12. Flowchart of the cogeneration system for the analysis of the exergetic efficiency.

The especific exergy of the bagasse (b_{bag}), with 50% of humidity, is $b_{bag} = 1.15\ LHV_{bag}$, (Esteves, 1996 and Szargut, 1998). Thus, $b_{bag} = 8{,}666\ kJ/kg$. For calculation of the exergy rate, it was considered the ambient temperature $T_0 = 25\ ºC$ and the ambient pressure $po = 101.325\ kPa$.

3. RESULTS

The mean values of the efficiencies obtained for the literature plants (Cases 1, 2 and 3) and for the real plant (Coruripe), appear in the Tab. 1 and Tab. 2, respectively:

Table 1. Efficiencies of cogeneration for Cases 1, 2 and 3 (mean values).

Cases 1, 2 and 3	Energetic efficiencies (%)	Exergetic efficiencies (%)
Boiler	82.9	29.2
Turbines of picker, defibrator and milling	57.5	64.5
Turbo-generator	70.3	75.5
Cogeneration	72.7	24.1

Table 2. Efficiencies of cogeneration for Coruripe Plant (mean values).

Coruripe plant	Energetic efficiencies (%)	Exergetic efficiencies (%)
Boiler	85.0	32.2
Turbines of picker, defibrator and milling	72.1	78.0
Turbo-generator	82.2	73.0
Cogeneration	84.5	28.7

As can be observed, the energetic efficiencies of the boiler, of turbines and of turbo-generator of the real plant (Tab.2) are within the range of the efficiencies found in the literature (Tab. 1). In the case of turbine, the energetic efficiency is the isentropic efficiency. What is observed is that the turbines used for mechanical work (that is, turbine of piker, defibrator and milling) of the real plant (Tab. 2) presented an energetic efficiency lower than that of the electric generator coupled to the turbine (that is, turbo-generator), and that the opposite of this is obtained for the exergetic efficiencies of them. This shows that, there are still improvements to be made in the turbo-generator system, if compared to the turbo-generator of the literature (Tab. 3). Although the energetic efficiency of the cogeneration of the real plant (Tab. 2) is high from the energetic point of view (84.5%), it is found that exergetic efficiency of the cogeneration is very low (28.7%), indicating a potential for improvements in the system. Table 3 and Table 4 shows the exergy destruction rate ($\dot{\mathrm{B}}_{\mathrm{d}}$), that is, the rate of irreversible loss of the potential for performance of work, by equipment and for the entire cogeneration system modeled, being for literature plants (Cases 1, 2 and 3) and real plant (Coruripe), respectively:

Table 3. Exergy destruction rate by equipment for Cases 1, 2 and 3 (mean values).

Cases 1, 2 and 3	$\dot{B}_d$ (MJ/s)	%
Boiler	197.2	95.6
Turbines of picker, defibrator and milling	4.6	2.2
Turbo-generator	4.5	2.2
Total	206.3	100.0

Table 4. Exergy destruction rate by equipment for the Coruripe plant of Campo Florido.

Coruripe Plant, subsidiary of Campo Florido	$\dot{B}_d$ (MJ/s)	%
Boiler	163.9	95.1
Turbines of picker, defibrator and milling	2.1	1.1
Turbo-generator	6.4	3.8
Total	172.4	100.0

The boiler contributes to the largest fraction of the total destruction of exergy of the system of cogeneration studied (95.1%) in the real plant on Tab. 4, confirming the men value found in the literature (95.6%). It was expected due to the irreversibilities of the process of the bagasse combustion in the boiler. According to Cortez and Gomez (1998), approximately 70% of the exergy of the fuel (bagasse) is destroyed in the combustion process. And next, in the list of the equipment considered in the study, comes the turbo-generator with a fraction of the total destruction of exergy well below the boiler (3.8%).

4. CONCLUSION

The energetic analysis (using the first law of thermodynamics) and the exergetic analysis (analysis combining the first and second law of thermodynamics) were used for evaluating the performance of sugarcane plants. A typical plant described in the literature was studied for three configurations of cogeneration (Cases 1, 2 and 3). The mean value found of the energy efficiency of the cogeneration to these plants was 72.7% while the average value of the exergetic efficiency was 24.1%, thus very low. A real plant, the Plant Coruripe, subsidiary of Campo Florido, in Minas Gerais, Brazil, was used to evaluate the adequacy of the developed method. The energetic efficiency of the cogeneration of Coruripe plant was 84.5%, while the exergetic efficiency of the cogeneration was 28.7%. Thus, it was confirmed, that the evaluation of the performance of the entire cogeneration system using only the first law of thermodynamics is insufficient to evaluate properly its performance. The values found for the three cases of the literature plants are compatible with the values of the real plant, which demonstrates the adequacy of the method developed for the energetic and exergetic analysis. The exergy destruction is mainly in the boiler, as was expected, due to the irreversibility of the combustion process, and in much lower proportion, in the turbine coupled to the electric generator. Therefore, the boiler is a device that also has the lowest exergetic efficiency (32.2% for the boiler of Coruripe plant and 29.2% on average for the boiler of the plants of Cases 1, 2 and 3), although its efficiency is high and the largest among all the equipment considered (85.0% and 82.9% for Coruripe and for Cases 1, 2 and 3 on average, respectively). The results of exergetic analysis also showed that there is a potential for improvements in the system. As the boiler and the turbine are the equipments that showed the highest destruction of exergy, it is about them that it should be given greater attention.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Baptista, B.S. and Martins, B.F., 2002. "Trocadores de Calor". UFMG, Belo Horizonte, Brazil.

Brazilian Energy Balance, 2011. Year Base 2010. 5 Jul. 2012 <https://ben.epe.gov.br>.

Celulose On Line, 2004. "Dessuperaquecedores". 23 Feb. 2005. <http://www.celuloseonline.com.br>.

Coelho, S.T., 1999. *Mecanismos para implementação da co-geração de eletricidade a partir de biomassa. Um modelo para o estado de São Paulo*. PhD thesis, Universidade de São Paulo.

Centro de Tecnologia Copersucar - CTC, 2003. "Potencial de co-geração com resíduos da cana-de-açúcar, sua compatibilidade com o modelo atual". 14 Oct. 2004 <http://www.unica.com.br>.

Cortez, L.A.G., 1998. "A method for exergetic analysis of sugarcane bagasse boilers". *Brazilian Journal of Chemical Engineering*, Vol. 15, n. 1, São Paulo, Brazil.

Eletrobrás, 1993. "Centrais Elétricas Brasileiras. Plano Nacional de Energia Elétrica 1993-2015. Plano 2015, oferta de energia elétrica, resíduos da cana-de-açúcar."

Estado de Minas, 2006. "Mais de 1 milhão de litros/dia". Caderno de Economia, 5 Feb.

Esteves, O.A., 1996. *Análise exergética da produção do etanol da cana-de-açúcar*. Master dissertation, Universidade Federal de Minas Gerais, Belo Horizonte.

Fórum de Co-Geração, 2002. "Geração com resíduos da cana-de-açúcar". 4 Jul. 2012 <http://www.inee.org.br>.

Leal, M.R., 2003. "Processos de co-geração; equipamentos, custos e potenciais". Centro de Tecnologia Copersucar, São Paulo.

Kotas, T.J., 1995. *The exergy method of thermal plant analysis*. Krieger Publishing Company, Florida.

Koblitz, 2001. "Encontro CEMIG e setor sucroalcooleiro de Minas Gerais". Belo Horizonte, Brazil.

Li, W.K., 1995. "Applied thermodynamics: availability method and energy conversion". New York.

Moran, M.J. and Shapiro, H.N., 2000. *Fundamentals of Engineering Thermodynamics*. John Willen Sans, 4th edtion.

Saint-Martin, L.F., 2005. *Análise exergética da co-geração a partir dos resíduos da cana-de-açúcar em usinas sucroalcooleiras*. Master dissertation, Universidade Federal de Minas Gerais, Belo Horizonte.

Moran, M.J., 1982. *Availability Analysis: a Guide to Efficient Energy Use*. Prentice Hall, New Jersey.

Prasad, S.B., 1995. *Biomass-Fired Steam Power Cogeneration System: a Theoretical Study*. Elsevier Science Ltd. Great Britain, Vol. 36, n. 1, p. 65-77.

Sears, F.W. and Lee, J.F., 1969. *Termodinâmica*. Universidade de São Paulo Editora, Rio de Janeiro, 2nd edition.

Szargut, J., 1988. *Exergy Analysis of Thermal, Chemical and Metallurgical Processes*. Hemisphere Publishing Corporation, New York.

União da Agroindústria Canavieira de São Paulo, 2002. "Potencial de co-geração com resíduos da cana de açúcar sua compatibilidade com o modelo atual".

Wylen, V., 2001. *Fundamentos da Termodinâmica*. Edgard Blücher Ltda, 5th edition.

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