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ENVIRONMENTAL AND THERMOECONOMIC ANALYSIS OF A WASTE-TO-ENERGY FACILITY– CASE STUDY OF ZABALGARBI/BILBAO PLANT ADAPTED TO THE BRAZILIAN REALITY

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Abstract. *The final disposal of municipal solid waste in landfills or dumps is a regular problem to the municipalities in Brazil. It involves environmental degradation, high costs and the shortage of available and appropriated areas to build landfills. Therefore, the development of new technologies to properly treat and dispose the urban solid waste is necessary and urgent. In addition, the use of municipal waste in processes of thermal treatment with energy recovery meets the search for alternative and renewable sources of energy production. In that context, waste-to-energy facilities are presented as a solution to both the waste disposal problem and the growing energy demand in the cities. Even though this practice stands out in the world among one of the most coherent alternatives for municipal solid waste management, in Brazil it is still incipient. This work presents a proposal of a waste-to-energy facility that could operate in the city of Rio de Janeiro (Brazil), based on the Zabalgarbi/Bilbao (Spain) plant. It is performed an energetic and exergetic analysis along with a cost estimate influenced by an eco-efficiency indicator that takes into account the air pollution emissions. The conclusion shows the capacity of the proposed facility to treat the municipal solid waste of Rio de Janeiro (Brazil) and supply its electricity demand with a competitive cost.*

Keywords: *Municipal solid waste, biomass energy recovery, incineration, waste treatment, waste-to-energy, exergetic analysis, thermoeconomic analysis, atmospheric emissions.*

1. INTRODUCTION

In Brazil, the potential for energy recovery of waste incineration is largely unexplored. Only 0.03% of the waste collected is destined for incineration (EPE, 2014). As a solution for the management of municipal solid waste, the incineration has many benefits, such as: (HESTER & HARRISON, 1994)

- Reduction of up to 90% of volume and 75% of weight of materials going to landfill;
- Partial or complete destruction of toxic substances, which become inert and have their final disposal facilitated;
- Destruction of the organic component of biodegradable waste, which when landfilled generates methane (greenhouse gas);
- Complementation of other good waste management practices such as recycling and composting.

Besides that, the energetic content of the waste can be used as a thermal source in power plants, which are called waste-to-energy (WTE) facilities. They represent a renewable source of electricity and/or thermal energy and a solution to the waste disposal problem of the urban centers.

This work aims to propose a WTE system based in an existing facility that could operate in a Brazilian city. An evaluation of the state of the art of the WTE technologies in the world showed that the combined cycle power plants with mass burn of municipal solid waste (MSW) are among the most effective ones to treat waste in large scale. Experts point out the Spanish Zabalgarbi facility as a reference plant working with this type of technology, and for that reason it was chosen as the study basis. The methodology includes determining: the thermal efficiency of the system; the losses due to exergy destruction in each component inside the system; the environmental performance of the system through an indicator that considers the atmospheric emissions and the annual estimate cost of acquisition, operation and maintenance of the system.

2. ZABALGARBI WASTE-TO-ENERGY FACILITY

The Zabalgarbi waste-to-energy facility is located on the metropolitan area of Bilbao, in the Biskaia province of the Basque Country. It occupies an area of about 5 hectares and its operation begun in 19/04/2004 (Zabalgarbi, 2015).

According to Ribeiro & Kimberlin (2010), this facility is one of the most important plants combining steam and gas power cycles in the world. Figure 1 shows the diagram of the plant.

The Zabalgarbi facility incinerates MSW to provide heat to a boiler that produces saturated steam at 100 bar. At the same time, a combustor burns natural gas providing heat to a gas turbine system of 43 MW. The heat from the exaust gases of the gas turbine superheats the steam to 538°C at 100 bar in a heat recovery boiler (HRB) with supplementary firing of natural gas. The facility processes about 33 tons of MSW per hour with lower heating value (LHV) of 8000 kJ/kg and consumes 13870 Nm³/h of natural gas with LHV of 38992 kJ/Nm³, generating 94 MW of net electricity to the city grid (CNIM, February 2005). The atmospheric pollutants are controled by several procedures:

- NO_x is abated directly in the furnace by addition of a 23% NH_3 solution;
- SO_x and other acid gases are neutralized by lime slurry injection in two semi-dry scrubbers;
- Dioxins, furans, volatile organic compounds and heavy metals are adsorbed by activated carbon;
- Particles and dust are removed in a bag house filter system;
- CO emissions are minimized by the reuse of the exaust gases as secondary air in waste combustion.

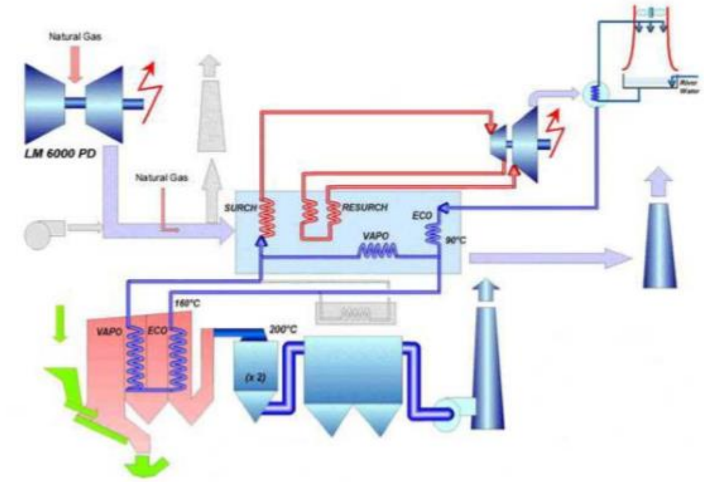

Figure 1. Zabalgarbi waste-to-energy facility diagram

3. CARACTERIZATION OF THE SYSTEM

The scope of the analysis is the system showed in Figure 2. It is based on the Zabalgarbi facility and includes the following elements:

I. Gas power cycle containing:
   a. 1 compressor;
   b. 1 natural gas combustor;
   c. 1 gas turbine of 43 MW;
   d. 1 supplementary natural gas combustor;

II. Steam power cycle containing:
   a. 1 furnace for MSW mass burn on a firing grate;
   b. 1 steam boiler with all exposed furnace tubes coated with inconel 625 to reduce corrosion rates;
   c. atmospheric pollution control equipment: 2 semi-dry scrubber and 1 baghouse filter;
   d. 1 steam turbine;
   e. 1 condenser;
   f. 1 pump;
   g. 1 heat recovery steam generator at 100 bar.

The first part of the study aimed at determining the thermodynamic conditions (pressure, temperature and physical state) of the fluids at each point numbered in Fig. 2. The assumptions of the calculations are:

1. The system operates in steady state;
2. Each component of the system is analysed as a control volume;
3. The working fluid of the gas cycle and the exaust gases from the waste furnace are modeled as common air;
4. The fluids kinetic and potential energy variation is negligible.

3.1 Gas power cycle

Table 1 shows the input variables obtained from literature, Tab. 2 shows the variables hypothetically assumed and Tab. 3 presents the properties evaluated at each point of the gas power cycle. Figure 3 shows the diagram of temperature versus specific entropy of the gas power cycle.

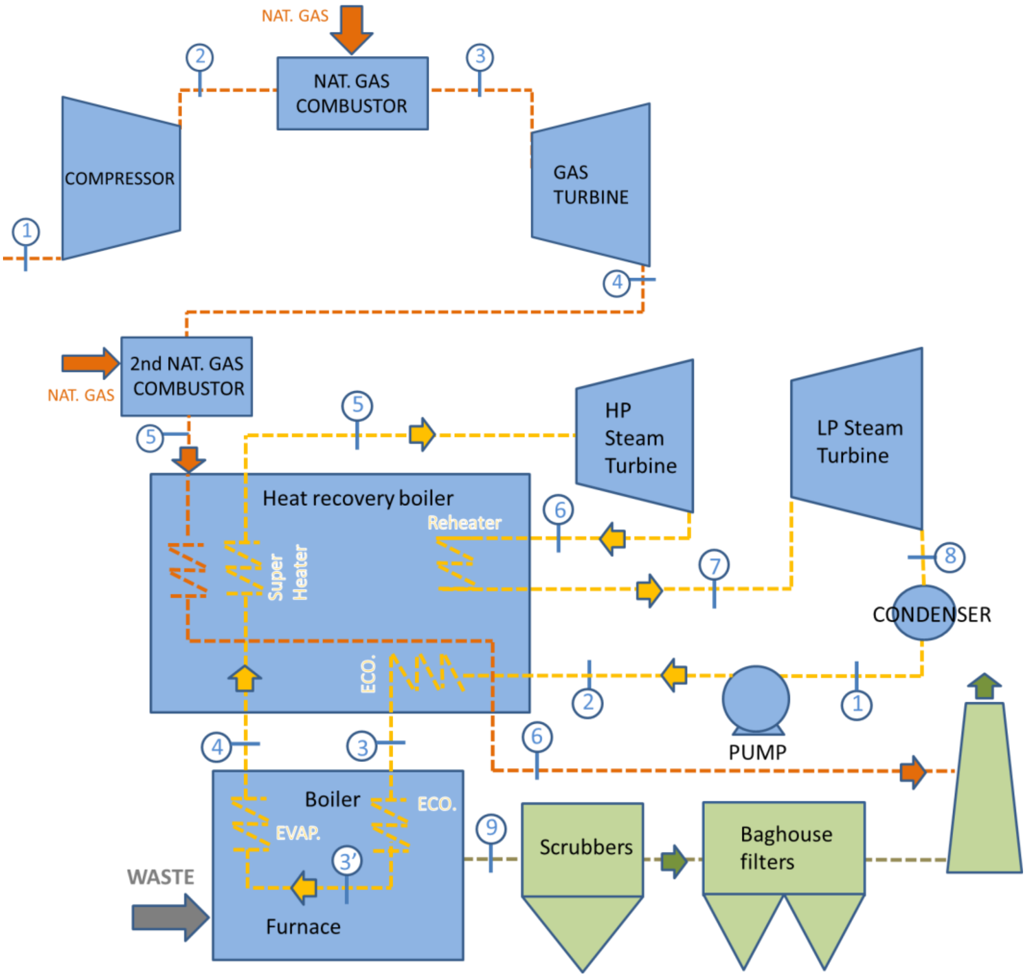

Figure 2. Diagram of the proposed system

Table 1. Input variables of the gas power cycle obtained from literature

Symbol	Variable	Value	Source
LHV_{gn}	Lower heating value of natural gas per unit mass	47730 kJ/kg (1)	(Lourenço, 2003)
ro_{gn}	Natural gas density	0.768 kg/Nm³(1)	(Lourenço, 2003)
m_{tg}	Mass flow rate at the exit of the gas turbine	125 kg/s (2)	(GE, 2013)
T_4	Temperature at the exit of the gas turbine	455°C (2)	(GE, 2013)
P_2/P_1	Compression rate	29.1 (2)	(GE, 2013)
R_{ar}	Air constant	0.2870 kJ/kg.K	(Moran & al., 2013)
W_{netTG}	Net power (grid connection) generated by the gas turbine	43000 kW (2)	(GE, 2013)
Heat_rate	Gas turbine heat rate	8630 kJ/kWh(2)	(GE, 2013)

(1) Value from COMBRAS for Brazilian natural gas

(2) Corresponds to General Electric LM-6000 PD gas turbine

Table 2. Assumed input variables of the gas power cycle

Symbol	Variable	Value	Unit
P_1	Pressure at the compressor inlet	0.101	MPa
T_1	Temperature at the compressor inlet	20[1]	°C
N_{cp}	Compressor isentropic efficiency	80	%
D_{p23}	Pressure drop inside the combustion chamber	0.015	MPa
P_4	Pressure at the exit of the gas turbine	P1+0.03 =0.131	MPa
D_{pQA}	Pressure drop in the second combustion chamber	0	MPa
D_{pCR}	Pressure drop in the heat recovery boiler (HRB)	0.03	MPa
T_5	Temperature at the inlet of the HRB	600	°C
T_6	Temperature at the exit of HRB	150	°C
P_6	Pressure at the exit of HRB	0.101	MPa

(1) The average temperature of Brazil (Sua Pesquisa.com, 2015)

Table 3. Calculated variables of the gas power cycle

Point	T (°C)	P (bar)	h (kJ/kg)	s (kJ/kg.k)
1	20.00	1.01	293.60	6.84
2	580.70	29.39	881.73	6.99
3	1252.00	29.24	1665.86	7.66
4	455.00	1.31	743.90	7.71
5	600.00	1.31	903.10	7.90
6	150.00	1.01	424.80	7.21

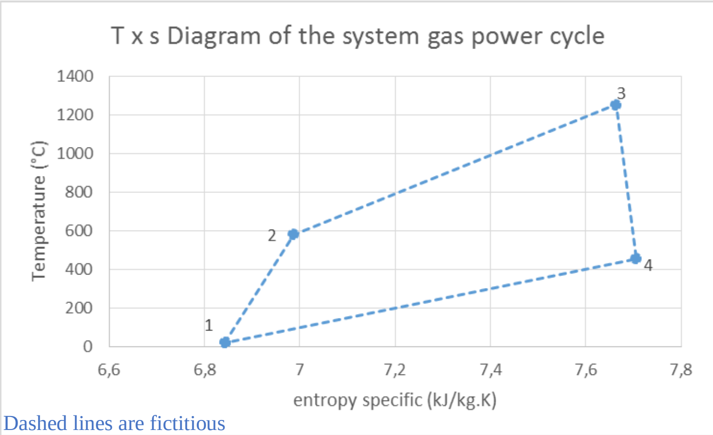

Figure 3. Temperature versus entropy diagram of the gas power cycle

3.2 Steam power cycle

Table 4 presents the input variables obtained in the literature, Tab. 5 shows the variables hypothetically assumed and Tab. 6 presents the properties evaluated at each point of the steam cycle. Figure 4 shows the diagram of temperature versus entropy per unit mass of the steam power cycle.

Table 4. Input variables of the steam power cycle obtained from the literature

Symbol	Variable	Value	Source
LHV_{lixo}	Lower heating value of the municipal solid waste per unit mass	16000 kJ/kg [1]	(Soares, 2011) and (CEMPRE, 2010)
m_{lixo}	Waste mass flow rate at the furnace inlet	48 t/h [2]	(CNIM, February 2005)
P_4	Steam pressure at pt. 4	100 bar	(CNIM, February 2005)
T_5	Steam temperature at pt. 5	538 °C	(CNIM, February 2005)

(1) It was assumed an average value between those presented by (Soares, 2011) and (CEMPRE, 2010)

(2) CNIM assumes 33 t/h. It was assumed 48 t/h in order to increase the capacity of the system.

Table 5. Assumed input variables of the steam power cycle

Symbol	Variable	Value	Unit
P_6	Pressure at pt. 6	0.3	MPa
T_1	Temperature at point 1	45	°C
x_1	Quality at pt. 1	0	-
x_4	Quality at pt. 4	0	-
N_{tv}	Steam turbines isentropic efficiency (HP and LP)	0.9	-
D_{p23}	Pressure drop between pts. 2 and 3	0.2	MPa
D_{p33}	Pressure drop between pts. 3 and 3')	0.2	MPa
D_{p34}	Pressure drop between pts. 3' and 4)	0.3	MPa
D_{p45}	Pressure drop between pts 4 and 5	0.3	MPa
D_{p67}	Pressure drop between pts 6 and 7	0.1	MPa
D_{p81}	Steam pressure drop in the condenser tubes	0.005	MPa
N_b	Pump isentropic efficiency	0.8	-
N_{cr}	Heat recovery boiler thermal efficiency	0.8	-
T_3	Temperature at pt. 3	90	°C
$x_{3'}$	Quality at pt. 3'	0	-
T_{8s}	Isentropic temperature at LP steam turbine exit (pt. 8s) (1)	45	°C
x_{8s}	Isentropic quality at LP steam turbine exit (pt. 8s) (1)	x_{6s}=0.95	-

(1) Point 8s is a fictitious point representing an isentropic expansion at LP steam turbine.

Table 6. Calculated variables of the steam power cycle

Point	P (bar)	T (°C)	h (kJ/kg)	s (kJ/kg.K)
1	0.10	45.00	188.40	0.64
2	107.00	46.00	201.90	0.65
3	105.00	90.00	385.00	1.19
3'	103.00	313.20	1420.00	3.38
4	100.00	311.10	2725.00	5.61
5	97.00	538.00	3474.00	6.74
6	3.00	133.50	2706.30	6.95
7	2.00	239.60	2950.00	7.67
8	0.15	53.41	2530.60	7.81
9(1)	1.01	200.00	475.80	7.33

(1) Point 9: waste combustion gases at the furnace outlet (T9 from Zabalgarbi facility - CNIM, February 2005)

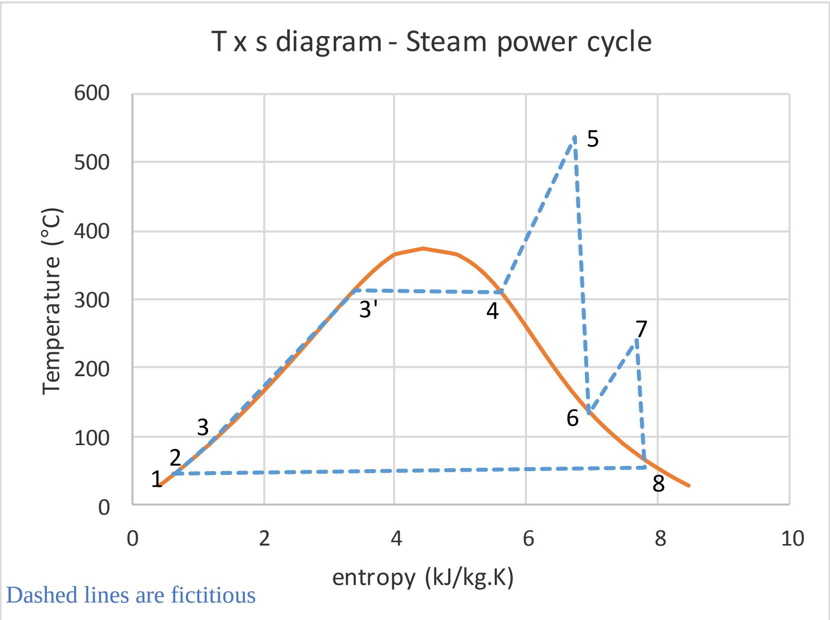

Figure 4. Temperature versus entropy diagram of the gas power cycle

4. ENERGETIC ANALYSIS

The energy analysis aims at evaluating the power generated by the system, its thermal efficiency and fuel consumption. The mass flow rates, thermal efficiencies, electrical powers and heat transfer rates were calculated based on the Energy and Mass Conservation Principles to both gas and steam power cycles.

4.1 Gas power cycle

Table 7 shows the calculations of the gas power cycle, which generates 43 MW of electricity to the grid with a thermal efficiency of 42% (LM-6000 PD gas turbine model efficiency - Company, 2013). The rate of natural gas consumed by the system is about 18000 Nm³/h. For comparison, in Zabalgarbi the total natural gas rate consumed is 13870 Nm³/h. (CNIM, February 2005)

Table 7. Power and efficiency of the gas power cycle

Symbol	Variable	Equation	Value
Q_{cc}	Heat transfer rate into the combustor	$Q_{cc} = m_{gn1}\ LHV_{gn}$	103 MW
Q_{QA}	Heat transfer rate into the 2nd combustor	$Q_{QA} = m_{gn2}\ LHV_{gn}$	78 MW
W_{tg}	Power generated by the gas turbine	$W_{tg} = m_{tg}\ w_{tg}$	115 MW
N_{tg}	Gas turbine isentropic efficiency	$N_{tg}=(h_3-h_4)/(h_3-h_{4s})$	97%
W_{cp}	Electrical power consumed by the compressor	$W_{cp} = m_{ar1}\ w_{cp}$	72 MW
W_{tgnet}	Net electrical power generated by the gas cycle	$W_{tgnet} = W_{tg} - W_{cp}$	43 MW
N_{cc}	Combustor thermal efficiency	$N_{cc}= Q_{23}/ Q_{cc}$	97%
N_{cg}	Thermal efficiency of the gas power cycle	$N_{cg} = (W_{tg} - W_{cp})/ Q_{cc}$	42%
V_{gn} (1)	Volumetric flow rate of natural gas consumed (1)	$V_{gn}=(m_{gn1}+m_{gn2})/ro_{gn}$	4.9 Nm³/s

(1) consumed in the main and 2nd combustor, not considering eventual consume in the furnace.

4.2 Steam power cycle

Table 8 shows the energy calculations of the steam power cycle. It generates 84 MW of net electricity to the grid with a thermal efficiency of 26%. The expected range of thermal efficiencies of WTE facilities is 22-40% (Afval Energie Bedrijf, 2007; CEMPRE, 2010).

Table 8. Mass flow rate, power and efficiency of the steam power cycle

Symbol	Variable	Equation	Value
m_v	Steam mass flow rate	$m_v= N_{cr}\ m_5\ q_{56}^{(1)} /(h_3-h_2+h_5-h_4+h_7-h_6)$	71.5 kg/s
Q_{lixo}	Heat transfer rate from MSW in the furnace	$Q_{lixo} = LHV_{lixo}\ m_{lixo}$	212 MW
N_{inc}	Incinerator (Furnace + Boiler) efficiency	$N_{inc}=m_v\ (h_4-h_3)/ Q_{lixo}$	79%
W_{TVA}	Electric power generated at HP steam turbine	$W_{TVA} = m_v\ (h_5-h_6)$	55 MW
W_{TVB}	Electric power generated at LP steam turbine	$W_{TVB} = m_v\ (h_7-h_8)$	30 MW
W_b	Pump electric power	$W_b = m_v\ (h_2-h_1)$	0.9 MW
W_{netTV}	Net electric power generated by the steam cycle	$W_{netTV} = W_{TVA} + W_{TVB} - W_b$	84 MW
N_{CV}	Steam power cycle thermal efficiency	$N_{CV} = W_{netTV} / (m_5\ (h_5-h_6) + Q_{lixo})$	26%

(1) q_{56}: heat exchanged per unit mass by the hot gases at the HRB

4.3 Conclusion of the energetic analysis

The net electrical power generated by the system is 127 MW with a thermal efficiency of 32%. For comparison, the thermal efficiency of the Zabalgarbi facility is 42%. If this system was used in a single facility in Rio de Janeiro, Brazil, it would be able to treat 13% of the amount of waste disposed in landfills in 2013 and could supply 6% of the city's 2014 electricity consumption. The system generates 9.6 MJ of energy per kg of waste with a consumption of 18000 Nm³/h of natural gas.

5. EXERGETIC ANALYSIS

The specific exergy (*e* - kJ/kg) associated to mass flow is described in Eq. (5.1):

$$e = h - h_0 - T_0(s - s_0) \tag{5.1}$$

Where *h* is the specific enthalpy, *s* the specific entropy and h_0 and s_0 are those properties evaluated at T_0 and P_0, which are the temperature and pressure in the reference environment.

The exergy analysis aims at evaluating the total amount of exergy lost by irreversibilities (internally and externally). The exergy destruction (E_d) inside the system (Eq. 5.2) is due to friction that causes internal heating. The external irreversibilities are due to heat transfer to the environment (cooling water leaving the condenser and emission of exhaust gases). The irreversibilities are summarized in Tab. 9 in terms of exergy magnitudes on a rate basis (MW).

$$E_{d(inside)} = \Sigma \text{ Exergy rates entering} - \Sigma \text{ Exergy rates leaving} \quad (5.2)$$

Table 9. Balance sheet of exergetic power of the system

	Exergy transfer	MW	%
Exergy in (fuel)	Fuel:	393	100%
	Natural gas at the combustor	103	26%
	Natural gas at the 2nd combustor	78	20%
	Municipal solid waste	212	54%
Exergy out as useful power	Net power to the grid:	127	32%
	High pressure steam turbine	54	43%
	Low pressure steam turbine	30	24%
	Gas turbine	43	34%
Exergy out as loss (not useful)	External irreversibility:	20	5%
	HRB exhaust gases	5	25%
	Furnace exhaust gases	9	44%
	Cooling water from the condenser	6	31%
	Internal irreversibility	246	63%
	Combustor	29	12%
	Second combustor	37	15%
	Pump	<1	<1%
	HP steam turbine	4	2%
	LP steam turbine	3	1%
	Condenser	11	4%
	Compressor	5	2%
	Gas turbine	2	1%
	HRB	26	11%
	Furnace	129	52%

In the proposed system 32% of the exergy is transformed in useful power and 68% is lost through irreversibilities. Most of the irreversibilities occur internally, mainly in the combustion equipment (combustors and furnace). The majority of the fuel thermal power comes from the waste.

To improve the performance of a system it is useful to know its exergetic efficiency, which can be described as the ratio between the exergy leaving the system (useful product) and the exergy entering the system (fuel). Table 10 shows that the lowest efficiencies are from the furnace and heat recovery boiler.

Table 10. Exergetic efficiencies of the system components

Symbol	Variable	Equation	Value
ε_{cp}	Compressor exergetic efficiency	$\varepsilon_{cp} = (E_2 - E_1)/W_{cp}$	93%
ε_{cc}	Combustor exergetic efficiency da	$\varepsilon_{cc} = E_3/(E_2+Q_{23})$	85%
ε_{tg}	Gas turbine exergetic efficiency	$\varepsilon_{tg} = W_{tg}/(E_3 - E_4)$	99%
ε_{qa}	2nd combustor exergetic efficiency	$\varepsilon_{qa} = E_5/(E_4+Q_{qa})$	64%
ε_{cr}	HRB Exergetic efficiency	$\varepsilon_{cr} = m_v*(e_3+e_5+e_7-e_2-e_4-e_6)/E_5$	52%
ε_{inc}	Furnace exergetic efficiency	$\varepsilon_{inc} = m_v*(e_4-e_3)/Q_{lixo}$	35%
ε_{tva}	HP ST exergetic efficiency	$\varepsilon_{tva} = W_{tva}/(m_v*(e_5-e_6))$	93%
ε_{tvb}	LP ST exergetic efficiency	$\varepsilon_{tvb} = W_{tvb}/(m_v*(e_7-e_8))$	91%
ε_{bomba}	Pump exergetic efficiency	$\varepsilon_{bomba} = m_v*(e_2-e_1)/W_b$	82%

E : exergy [kJ]
e : specific exergy [kJ/kg]

6. ECOLOGICAL EFFICIENCY

The ecological efficiency (EE) is an indicator of the environmental performance of a power cycle proposed by (Villela, 2007) and described by the following expression:

$$EE = \{0.204\ N_{sys}\ ln\ (135 - IP) / (N_{sys} + IP)\ \}^{0.5} \quad (6)$$

Where EE is the ecological efficiency (adimensional), IP is the pollution indicator (kg/MJ) and N_{sys} is the thermal efficiency of the system (adimensional). IP is given by Cardu & Baica (1999a) and (Villela, 2007):

$$IP = f_{CO2e} / LHV \quad (6.1)$$

Where f_{CO2e} is the emission factor of equivalent carbon dioxide (kg of equivalent CO_2 per kg of fuel) and LHV is the lower heating value of the fuel (MJ/kg). The emission factor of equivalent CO_2 is estimated by Cardu & Baica (1999a):

$$f_{CO2e} = f_{CO2} + 80\ f_{SO2} + 50\ f_{NOx} + 67\ f_{MP} \quad (6.2)$$

Where f_{CO2}, f_{SO2}, f_{NOx} and f_{MP} are the emission factors of the pollutants (mass of pollutant per mass of fuel).

Table 11. Ecologic efficiency calculations

Symbol	Variable	Equation	Value
C_{MP}	Particulates emitted concentration	From (Zabalgarbi, 2015)	1 mg/Nm³
C_{NOx}	NOx emitted concentration	From (Zabalgarbi, 2015)	148 mg/Nm³
C_{SO2}	SO_2 emitted concentration	From (Zabalgarbi, 2015)	5.44 mg/Nm³
f_{CO2e}^{GN}	Equivalent CO_2 emission factor for natural gas combustion	Average from (Lourenço, 2003) and (Villela, 2007)	3 kg/kg
f_{CO2}^{MSW}	CO_2 emission factor for MSW combustion	$f_{CO2}^{MSW} = 985\ 10^{-3}\ LHV_{lixo}/10466$ (1)	1.5 kg /kg
f_{MP}	Particulate emission factor for MSW combustion	$f_{MP} = (C_{MP}\ 10^{-6})\ V_{vg} / m_{lixo}$ (2)	1.4 10^{-8} kg/kg
f_{NOx}	NO_x emission factor for MSW combustion	$f_{NOx} = (C_{NOx}\ 10^{-6})\ V_{vg} / m_{lixo}$ (2)	2.0 10^{-6} kg/kg
f_{SO2}	SO_2 emission factor for MSW combustion	$f_{SO2} = (C_{SOx}\ 10^{-6})\ V_{vg} / m_{lixo}$ (2)	7.5 10^{-8} kg/kg
f_{CO2e}^{MSW}	Equivalent CO_2 emission factor for MSW combustion	Eq. (6.3): $f_{CO2e}^{MSW} = f_{CO2}^{MSW} + 50 f_{NOx} + 80\ f_{SO2} + 67\ f_{MP}$	1.5 kg/kg
IP_{GN}	Pollution indicator for nat. gas combustion	$IP_{GN} = f_{CO2e}^{GN} / (LHV_{gn}/1000)$	0.06 kg/MJ
IP_{MSW}	Pollution indicator for MSW combustion	$IP_{MSW} = f_{CO2e}^{MSW} / (LHV_{lixo}/1000)$	0.09 kg/MJ
IP_{sys}	Pollution indicator for the system	$IP_{sys} = \{IP_{GN}\ (m_{gn1}+m_{gn2}) + IP_{MSW}(m_{lixo})\ \} / (m_{gn1}+m_{gn2}+ m_{lixo})$	0.09 kg/MJ
EE	Ecological efficiency of the system	Eq. (6): $EE=\{0.204\ N_{sys}\ ln\ (135 - IP) / (N_{sys} + IP)\ \}^{0.5}$	89%

(1) From (EPA, 2015): f_{CO2} is 985 kg/ton and LHV adjustment coefficient is 10466 kJ/kg.
(2) V_{vg} = volumetric flow (in Nm³/s) of exhaust gases from the furnace, calculated using the Ideal Gases Equation.

The ecological efficiency (*EE*) may vary from 0-100%. A system with *EE*=0% indicates an undesirable situation (maximum pollution) and a system with *EE*=100% indicates the ideal situation (zero pollution). The proposed system presents an ecological efficiency of 89%.

7. THERMOECONOMIC ANALYSIS

The electricity production cost (C_{ele}) of the proposed system, considering its environmental performance (ecological efficiency indicator), operating and maintenance taxes is determined by Eq. (7), according to (Villela, 2007).

$$C_{ele} = f\ .\ Z_{in} / (H_r\ .\ W_{sys}\ .\ EE) + C_{fuel}.\ Q_{fuel} / (W_{sys}) + C_{op} / (H_r\ .\ W_{sys}) + C_{man}/(H_r\ .\ W_{sys}\ EE) \quad (7)$$

Where: C_{ele} is the eletricity production cost (US$/kWh); f is the capital recovery factor (dimensionless, determined by Eq. 7.1); Z_{in} is the initial capital expenditure (US$); H_r is the number of operating hours in one year (h); W_{sys} is the net eletrical power generated by the system (kW); *EE* is the ecological efficiency (dimensionless); C_{fuel} is the cost of the fuel (US$/kWh); Q_{fuel} is the thermal power of the fuel (kW); C_{op} is the operating cost (US$) and C_{man} is the maintenance cost (US$) of the system.

$$f = i\ (1+i)^n / [(1+i)^n-1] \quad (7.1)$$

Where *i* is the annual interest rate (dimensionless); *n* is the payback period (years)

The calculations to obtain the electricity production cost of the system are detailed in Tab. 12. The capital expenditure (initial investment) is estimated based on the acquisition costs of the main equipment of the system, which were updated to 2015 dollars. The costs update was based on the American Consumer Price Index (US Inflation Calculator, 2015).

Table 12. Electricity production cost calculations

Symbol	Variable	Equation	Value
Q_{fuel}	Natural gas heat transfer rate [(1)]	$Q_{fuel} = Q_{QA} + Q_{cc}$	181 MW
Z_{tg}	Gas turbine system investment (compressor, combustor and GT)[(2)]	$Z_{tg}=500\ (W_{tg}-W_{cp})$	20 mi US$
Z_{cr}	Heat recovery boiler investment (Manesh et. al., 2013)	$Z_{cr}=$ [(3)] $Z_{qa}+5.805-0.1653*30+0.0153*m_5$	5 mi US$
Z_{inc}	Furnace investment (Silveira, 1990)	$Z_{inc}= 2567.65\ (3600\ m_v)^{0.67}$	19 mi US$
Z_{tv}	Steam turbine investment (Silveira, 1990)	$Z_{tv}= 57.76\ (3600\ W_{tv}/4.187)^{0.68} + +0.0085(3600W_{tv}/4.187)^{0.95}$	24 mi US$
Z_{cond}	Condensing unit investment (Bohem, 1987) e Bartlett (1958)	$Z_{cond}= 1.9*3000\ (m_v\ (h_8-h_1))^{0.55}$	4 mi US$
Z_{pump}	Pump investment (Villela, 2007)	$Z_{pump}=3540\ W_b^{\ 0.71}$	0.5 mi US$
Z_{in}	Initial capital investiment [(4)]	$Z_{in}=2*(Z_{tg}+Z_{cr}+Z_{inc}+Z_{tv}+Z_{cond}+Z_{pump})$	145 mi US$
C_{man}	Maintenance cost of the system	$C_{man}=8\%\ Z_{in}$	12 mi US$
C_{op}	Cost w/ personnel (operating procedures)	$C_{op}= 1\%\ Z_{in}$	1.5 mi US$
C_{fuel}	Cost of the fuel [(1)]	C_{fuel} = 2.8 (US$/mi BTU) From (Infomine.com, 2015)	0.01 US$/kWh
W_{sys}	Net electrical power generated to the grid	$W_{sys}=W_{tv}+W_{tg}-W_{cp}-W_b$	127 MW
H_r	Number of operating hours	From Zabalgarbi (CNIM, February 2005)	8270 h
i	Annual interest rate	Above SELIC tax, which is 14% in Jun/15	15%
n	Number of years to payback	From Zabalgarbi (CNIM, February 2005)	11 years
C_{ele}	Eletricity production cost of the system	Eq. (7)	0.05US$/kWh

(1) It is assumed that MSW is costless; hence, the only costing thermal power is from the natural gas.
(2) From (Boyce, 2002), to aeroderivative, 40 MW and 39% efficiency gas turbine model.
(3) Cost of the 2nd combustor (Foster-Pegg, 1986): $Z_{qa}=1000*(PCS_{gn}*m_{gn2}/1390+30+20)$, w/ PCS_{gn}=47391 Btu/kg.
(4) Other expenditures such as eletrical, automation, tubing installation, etc., are considered 2 times the equipment investment. (Villela, 2007)

The estimated cost of electricity production in the proposed system is 5 cents of US$/ kWh. If the system operated only with natural gas, the estimated cost of electricity production would be 7-8 cents of US$/kWh, based on similar combined cycles studied by Villela (2007) and Branco (2005).

8. CONCLUSION

The proposed system is a power plant operating in combined cycle fueled by natural gas (1 gas turbine system and supplementary firing in the heat recovery steam generator) and municipal solid waste (vapor power system with superheat and reheat). The system's incinerating capacity is 1147 tons of waste per day, generating 127 MW of net electricity with a thermal efficiency of 32%. The energetic analysis showed that 54% of the fuel thermal power comes from the waste, which is a renewable source.

The exergetic analysis showed that the majority of the losses is due to irreversibilities inside the system, mainly in the combustion devices. It also showed that the furnace is the equipment with best potential for improvement. The environmental performance analysis showed that the system is 89% effective in terms of atmospheric pollution abatement. The economic evaluation predicted an initial investment around US$ 145 millions. It also showed that the electricity production cost (including the environmental control systems) is about 52 US$/MWh, with a payback period of 11 years.

In conclusion, the proposed system showed to be environmentally and economically feasible. If it was implemented in a single waste-to-energy facility in Rio de Janeiro, Brazil, it could supply 6% of the municipal electricity consumption and treat 13% of the waste currently landfilled in the city. In terms of power generation costs, the system showed to be competitive with average biomass and natural gas thermoelectric production costs, which are about 70-85 US$/MWh (Light SESA, 2015).

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10. RESPONSIBILITY NOTICE

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