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DETERMINATION OF GLOBAL EFFICIENCY WITHOUT/WITH SUPPLEMENTARY BURNING OF A THERMOELECTRIC PLANT WITH COMBINED CYCLE OF NATURAL GAS

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Abstract. *The present work applies the energy analysis with the objective of determining the overall efficiency of the combined cycle thermoelectric plant installed in Panaíba, in the state of Piauí; as well as to determine the benefits and losses of energy by the different equipment that compose the system, which will be represented by a Sankey diagram of the systems without/with supplementary burning. Correction factors for shaft power, exhaust gas flow, relative humidity, among others, will be used in order to adapt the conditions to the operating location of the thermoelectric plant. The work was developed to steady-state operation and under time averaged (yearly) conditions. It is seen that the energy analysis shows the advantages of the use of natural gas in a combined cycle, which presents a better efficiency of 50,81 % when the thermoelectric plant is without supplementary burning, compared to 47,39 % when the plant thermoelectric is with supplementary burning.*

Keywords: *combined cycle, global efficiency, natural gas, supplementary burning, thermoelectric plant.*

1. INTRODUCTION

The liberation of the electric market coupled with the growing commitment to the environment has driven the development of new generation techniques that involve lower investment costs and at the same time help reduce greenhouse gas emissions.

Currently, the need to reduce pollution has led to a strong commitment to the implementation of renewable energy as the main source of supply. These are a clean source of energy, but not enough to meet the growing energy demand.

The storage of energy in large quantities is not currently possible, for this reason, the electrical system is a dynamic system. A system in which it is necessary to produce a constant balance between production and demand. Therefore, due to the variability of renewable energies, it is necessary to have another form of energy production that ensures that we can satisfy the growing demand at all times.

Combined cycle power generation technology is one of the most efficient, with lower environmental impact and greater responsiveness. Such plants are very fast in terms of power refers to other conventional technologies (coal or nuclear), which are often used to meet demand or power failures. They are therefore the best option to meet renewable production plants, through which they may not always have all the power required. Thanks to its characteristics seems to be the best alternative to meet the demand with the least possible environmental impact (Antunes, 1999; Pantalena, 1997; Villela *et al*., 2007b).

Thermoelectric plants, when operating on natural gas and emitting flue gases, are also causing environmental problems on a smaller scale compared to other fuels. Its components resulting from combustion harm human life, animals and plants. The main components generated from the combustion of these fuels are carbon oxides (CO and CO_2), sulfur oxides (SO_2 and SO_3) and nitrous oxides (NO and NO_2, normally denoted by NO_x). There is also the particulate matter (PM) that changes the temperature of the system and influences the growth of the plants. CO_2 and NO_x emissions are directly related to the greenhouse effect, since the negative effects of SO2 can be seen in the formation of acid rain. There are also other harmful components present in the combustion of some fuels, such as heavy metals, dioxins, etc., which in small concentrations are harmful to the environment (Villela, *et al*., 2007a).

The waste from thermoelectric plants with gas turbines associated with condensing steam turbines has a great environmental impact, since they use fuels such as: coal, oil and natural gas. In the analysis, emissions of CO_2, SO_2 and NO_x were considered, and a comparison was made of the concentration of each of these products with the standards in Romania (Cardu *et al*., 1999).

In the broadest sense, a combined cycle plant consists of the integration of two or more thermodynamic energy cycles, in order to obtain a more complete and efficient conversion of the supplied energy into work or power. The exhaust gas heat from the gas turbine is used to produce steam in the recovery boiler to power the steam turbine. Currently, advanced combined cycle plants employing 2 or 3 pressure levels for steam and with a temperature in the 420 to 650 °C range for exhaust gases can increase their thermal efficiency by up to 58 % (Franco *et al*., 2002; Villela *et al*., 2007b).

2. METHODOLOGY

2.1. Parameter corrections for selected gas turbines

The parameters of the selected gas turbines must be corrected according to the local conditions of the plant installation, because the ambient temperature, the altitude and the relative humidity influence the performance of the turbine. Thus, the following conditions were adopted (Antunes, 1999; Pantalena, 1997):

Ambient temperature of 27 °C, altitude of 900 m (local conditions of the plant), and relative humidity 60 %.

According to (Brooks, 1994; Antunes, 1999), the correction factors are used according to the following equations shown in Tab. 1:

Table 1. Parameter corrections for selected gas turbines

Parameter	Correction equation
From the ambient temperature on the shaft power	$fc_{SP}^{T} = -0{,}004 * T_{amb} + 1{,}06$ (1)
From ambient temperature to exhaust gas mass flow rate	$fc_{MF}^{T} = -0{,}006 * T_{amb} + 1{,}07$ (2)
From the ambient temperature on the Heat Rate	$fc_{HR}^{T} = 0{,}002 * T_{amb} + 0{,}97$ (3)
From the altitude (A) on the shaft power	(200 m ≤ A ≤ 2800 m) $fc_{SP}^{A} = -0{,}0001 * A + 0{,}9816$ (4) (A < 200 m ou 2800 m ≤ A ≤ 3000 m) $fc_{SP}^{A} = -0{,}0001 * A + 0{,}9975$ (5)

2.2. Determination of turbine parameters for ISO and corrected conditions

ISO - Standard pressure and temperature condition. The equations are shown in Tab. 2 (Pantalena, 1997):

Table 2. Turbine parameters for ISO and corrected conditions

Parameter	Correction equation
Gas turbine efficiency	$n_T(\%) = \left(\frac{3600}{HEAT\ RATE}\right) * 100\%$ (6)
Mass fuel flow	$m_F\left(\frac{kg}{s}\right) = \left(\frac{HEAT\ RATE * E_P}{n_{generador} * PCI * 3600}\right),\ n_{generador} = 0{,}95$ (7)
Total heat of gases	$Q_G = m_G * Cp_G * (T_4 - T_G),\ T_G = 150 + T_{PP}$ (8)
Total heat of steam	$Q_S = m_S * (h_6 - h_9),\ 16\frac{kg}{s} * (3454 - 219{,}6)\frac{kJ}{kg} = 51750{,}4\frac{kJ}{s}$ (9)
Supplemental fuel	$m_{SF} = \frac{\frac{Q_S}{n_{RB}} - Q_G}{PCI},\ n_{RB} = 0{,}80$ (10)
Corrected parameters	$E_P^* = E_P * FC_4 * FC_1$ (11) $m_G^* = m_G * FC_2$ (12) $m_F^* = m_F * FC_4$ (13) $T_4^* = T_4 * FC_2$ (14)
Corrected Heat Rate	$HEAT\ RATE^*\left(\frac{kJ}{kWh}\right) = \frac{PCI * m_F^* * 3600}{E_P^*}$ (15)
Corrected gas turbine efficiency	$n_T^*(\%) = \left(\frac{3600}{HEAT\ RATE^*}\right) * 100\%$ (16)
Corrected total gas heat	$Q_G^* = m_G^* * Cp_{GE} * (T_4^* - T_G)$ (17)
Corrected supplemental fuel	$m_{SF}^* = \frac{\frac{Q_S}{n_{RB}} - Q_G^*}{LHV}$ (18)
Gas mass flow rate as a function of the gas temperature at the outlet of the gas turbine (Antunes, 1999)	$\dot{m}_G = \frac{\dot{m}_S * (h_6 - h_9)}{n_{RB} * Cp_G * (T_4 - T_G)}$ (19)

The formulas presented will be used for the development of each turbine selected from the (Gas Turbine World Handbook, 2012) described in references as (Biasi, 2012).

2.3. First law analysis of thermodynamics

The proposed systems operate on a permanent basis; all or adiabatic components, operate without loss of heat. Two decision variables are used: temperature (T_4) of the exhaust gases in the gas turbine and the pressure ratio of the selected turbine ($PR = P_3/P_4$). These variables were chosen by the thermodynamic influence of the system and also according to the criteria used in the selection of the gas turbine system. Thus, P_1 and P_5 equal to ambient pressure 0,101325 MPa, and ΔP_{RB} and ΔP_{CC} are the gas pressure losses in the recovery boiler and in the combustion chamber respectively (Franco *et al*., 2002; Silveira *et al*., 2003; Villela *et al*., 2007a, 2007b). The equations are presented in Tab. 3:

Table 3. Equations of the analysis of the first law of thermodynamics

Parameter	Equation
Turbine input temperature	$T_3 = \frac{T_4}{\left\{1-n_{ISO\,GT}\left[1-(PR)^{\frac{(1-\lambda_G)}{\lambda_G}}\right]\right\}}$ (20)
Ratio of specific heats	$\lambda_G(T) = \frac{1}{1-\frac{R_G}{Cp_G(T)}},\ \lambda_G\left(\frac{Cp_G}{Cv_G}\right)$ (21)
Turbine output temperature	$T_2 = T_1 * \left\{1 + \frac{1}{n_{ISO\,GT}}\left[\left(\frac{P_2}{P_1}\right)^{\frac{(\lambda_{air}-1)}{\lambda_{air}}} - 1\right]\right\} \lambda_{air}(T) = \frac{1}{1-\frac{R_{air}}{Cp_{air}(T)}}$ (22)
Ratio of specific heats of the air	$\lambda_{air}(T) = \frac{1}{1-\frac{R_{air}}{Cp_{air}(T)}}$ (23)
The specific enthalpies of air and gases in points 2, 3	$h_2 = Cp_{air}(T_2) * (T_2 - T_0) + h_0$ (24) $h_3 = Cp_G(T_3) * (T_3 - T_0) + h_0$ (25)
Isentropic process at the outlet of the steam turbine, with P7 and enthalpies of liquid and vapor.	$s_7 = (1 - x_7) * sl_7 + x_7 * ss_7$ (26) $h_{7\,ideal} = (1 - x_7) * hl + x_7 * hs_7$ (27) $n_{ST} = \frac{(h_6 - h_{7\,real})}{(h_6 - h_{7\,ideal})}$ (28)
Air mass flow	$m_{air} = m_G - m_f$ (29)
Pump and compressor work	$W_P = \frac{m_S*(h_9-h_8)}{n_P}$ (30) $W_C = m_{air}\int_{T_1}^{T_2} Cp_{air}(T)\,dt$ (31)
The shaft power (W_{ST}), electricity (Ep_{ST}) and n_{ST} is 95 %	$W_{ST} = m_S * (h_6 - h_7)$ (32) $Ep_{ST} = W_{ST} * n_{ST}$ (33)
Power in the gas turbine (Ep_{GT}) with a n_{GT} of 98 %	$Ep_{GT} = W_{GT} * n_{GT}$ (34) $Ep_{TOTAL} = Ep_{GT} + Ep_{ST}$ (35)
Power supplied by the fuel	$E_f = m_f * LHV$ (36) $E_f = E_f + E_{sup\,fuel}$ (37)
Overall efficiency of plant without/with supplementary burning	$n_{OP} = \frac{Ep_{GT}+Ep_{ST}-W_P}{E_{fuel}}$ (38) $n_{OP} = \frac{Ep_{GT}+Ep_{ST}-W_P}{E_{fuel}+E_{sup\,fuel}}$ (39)

3. RESULTS AND DISCUSSIONS

3.1. Description of thermoelectric plant

The scheme of a gas turbine combined cycle plant associated with the recovery boiler without supplementary burning and steam turbine is shown in Fig. 1.

The proposed scheme presents seven components of the thermal plant: compressor, combustion chamber, gas turbine, recovery boiler, steam turbine, condenser and pump. In atmospheric conditions the air enters the compressor and is compressed to the combustion pressure. Subsequently, it is sent to the combustion chamber where the fuel is continuously burned under constant pressure and the combustion gases expand in the gas turbine producing electricity. Exhaust gases are then directed to the recovery boiler, where superheated steam is produced and sent to the steam turbine, producing even more electrical energy (Villela *et al*., 2007a, 2007b).

Figure 2 shows the gas turbine system associated with the recovery boiler with supplementary burning and steam turbine. It is verified that the process is identical to that of Fig. 1, but in the recovery boiler a value of 30 % of the fuel (natural gas) consumed in the combustion chamber of the gas turbine system is set for the additional firing.

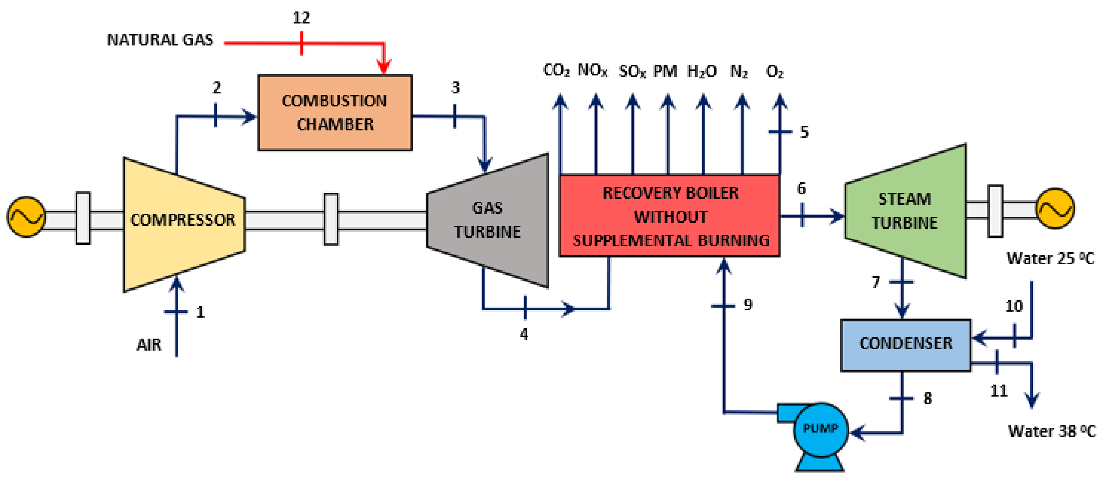

Figure 1. Gas turbine system associated with the recovery boiler without supplementary burning and steam turbine.

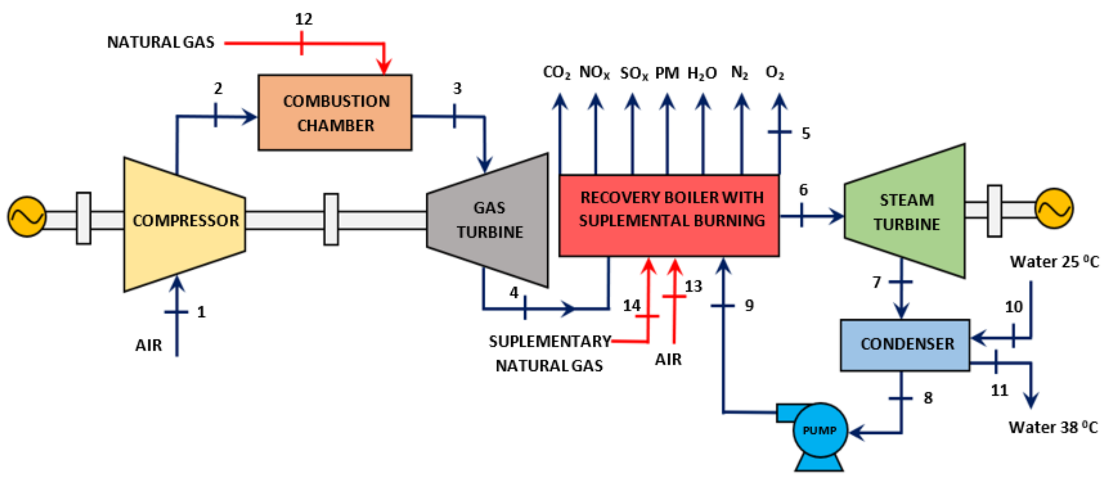

Figure 2. Gas turbine system associated with the recovery boiler with supplementary burning and steam turbine.

3.2. Selection of steam turbine

Among the parameters necessary for the comparison of the thermoelectric plants, using natural gas, are important mainly those related to the selected turbine. The choice of steam turbine used was based on the manufacturer's catalog (Siemens, 2018) on the market. Table 4 shows the selected steam turbine. The machine was chosen in view of the three temperature and pressure levels of the steam at the steam turbine inlet.

Table 4. Selected steam turbine (Siemens, 2018)

Parameters	Values
Temperature (°C)	540
Pressure (MPa)	12
Velocity (rpm)	14600
Model	SST-200
Manufacturer	Siemens
Power (kW)	20000

3.3. Selection of gas turbine

In this section, the gas turbine is selected according to the parameters of Tab. 5 and then we continue with the procedure.

Table 5. Input data for the selection of the gas turbine

Physical parameters	Symbology	Specification	Units
Exhaust gas temperature with subsequent correction by the Pinch Point method.	T_G	150	°C
Enthalpy at point 6 superheated steam determined as a function of the temperature and inlet pressure of the steam turbine.	h_6	-	kJ/kg
Enthalpy at point 9 for a feed water temperature entering the boiler at 50 °C and pressure of 12 MPa, compressed liquid.	h_9	-	kJ/kg
Enthalpy at point 8 for a temperature of 29,8 °C and pressure of 0,005 MPa: compressed liquid.	h_8	-	kJ/kg
Pressure in point 7 (Siemens, 2018)	P_7	0,005	MPa
Recovery boiler efficiency without supplementary burning (Silveira, 2003).	H_{RC}	70	%
Recovery boiler efficiency without supplementary burning (Silveira, 2003).	H_{RC}	90	%
Steam turbine steam flow (project data)	m_{ST}	16	kg/s
Specific heat exhaust gas	Cp_G	1,14	kJ/kg K
Specific heat at constant pressure of natural gas	Cp_{NG}	1,209	kJ/kg K
Lower Heating Value	LHV_{NG}	47966 37000	kJ/kg kJ/Nm^3

Table 6 it shows the enthalpy values to determine the solution of Eq. (19), and to obtain the T_4 as a function of the mass flow of the exhaust gases.

Table 6. Pressure enthalpies and operating temperatures

P (MPa)	T (°C)	h (kJ/kg)
12	540	h_6 = 3454
0,005	X=0,9	h_7 = 2319
0,005	X=0,0	h_8 = 137,8
12	50	h_9 = 219,6

$$\dot{m}_G = \frac{16\frac{kg}{s}*(3454-219{,}6)\frac{kJ}{kg}}{0{,}9*1{,}14\frac{kJ}{kgK}*(T_4-150)°\mathrm{C}} \tag{40}$$

$$\dot{m}_G = \frac{50438{,}9}{(T_4-150)}\frac{kg}{s} \tag{41}$$

According to Eq. (41), the best gas turbine is determined by varying the temperature T_4 in a range of 400 to 655 °C. The data used are ISO (T = 15 ° C, altitude = 0 m and 60 % relative humidity) and are available in the “Gas Turbine Handbook” (Biasi, 2012). The results are shown in Tab. 7.

Table 7. Mass gas flow in function of the maximum temperature in the gas turbine

T_4 (°C)	$\dot{m}_G$ (kg/s)
400	201,8
450	168,1
500	144,1
550	126,1
600	112,1
650	100,9

Table 8, shows the gas turbine systems selected from the “Gas Turbine World Handbook” (Biasi, 2012)., under the ISO conditions.

The tables and figures below show the gas flow ($\dot{m}_G$) as a function of the temperature at the outlet of the selected turbines, considering the natural gas.

Table 8. Gas turbine systems selected from Gas Turbine World Handbook, 2012

GT	Model	Year	$W_{liq}=E_P$ (kW)	HR (kJ/kWh)	m_G (kg/s)	T_G (°C)	P_r
A	GE Energy Oil & Gas: LM6000PG	2008	51204	8590	143,9	470	30
B	IHI Power Systems: LM6000PH Sprint	2010	51342	8774	139,9	470	35
C	Pratt & Whitney Power Systems: SWIFTPAC 50 DLN	2003	51235	9395	169,6	458	19,5
D	Rolls-Royce: Trent 60 DLE	1996	51685	8586	154,6	440	34
E	Siemens Energy: SGT-900	1982	49500	11025	175,1	514	15,3

The values shown will be plotted on the thermal parity curve of Fig. 3.

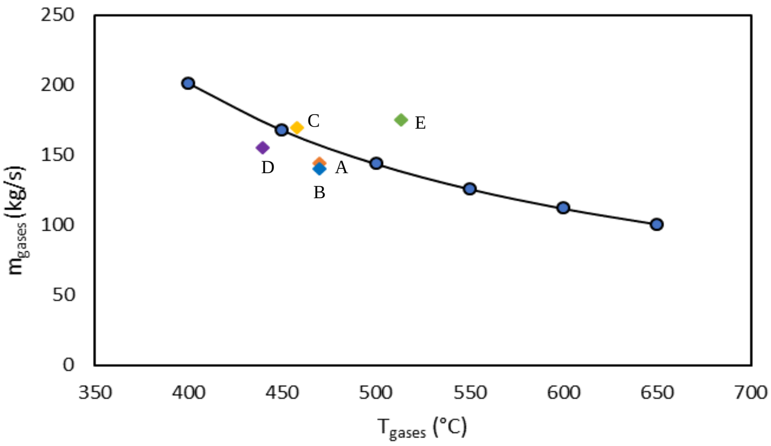

Figure. 3. Thermal parity curve with turbines selected from the Gas Turbine World Handbook, 2012.

3.4. Correction Factors for selected gas turbine parameters

The combined cycle thermoelectric plant will have an installed capacity of 60 MW and will be installed in Panaíba, in the state of Piauí. Which has the following climatic conditions: Ambient temperature 27 °C, 1 atmosphere pressure, 900 m altitude and 60 % relative humidity. The work was developed to steady-state operation and under time averaged (yearly) conditions. Using the equations of Tab. 1. The values of the correction factors of the local conditions (ambient temperature, altitude and relative humidity) of the plant installation are presented in Tab. 9:

Table 9. Results of correction factors for local conditions

CF	Correction Factors	Values
CF_1	From the ambient temperature on the power axis	0,952
CF_2	From ambient temperature to exhaust gas mass flow rate	0,908
CF_3	From the ambient temperature on the Heat Rate	1,024
CF_4	From the altitude (A) on the shaft power	0,892

We apply the values of the correction factors to the gas turbines selected from the Gas Turbine World Handbook (Biasi, 2012), shown in the Tab. 10.

Table 10. Gas Turbine Corrected Values Selected from the Gas Turbine World Handbook, 2012

GT	Model	Year	$W_{liq}=E_P$* (kW) CF_1 e CF_4	HR* (kJ/kWh) CF_3	m_G* (kg/s) CF_2	T_G* (°C) CF_2	P_r
A	GE Energy Oil & Gas: LM6000PG	2008	43482	8796	130,7	427	30
B	IHI Power Systems: LM6000PH Sprint	2010	43599	8985	127,1	427	35
C	Pratt & Whitney Power Systems:50 DLN	2003	43508	9620	153,9	416	19,5
D	Rolls-Royce: Trent 60 DLE	1996	43890	8792	140,4	400	34
E	Siemens Energy: SGT-900	1982	42035	11290	158,9	467	15,3

The values shown will be plotted on the new thermal parity curve of Fig. 4.

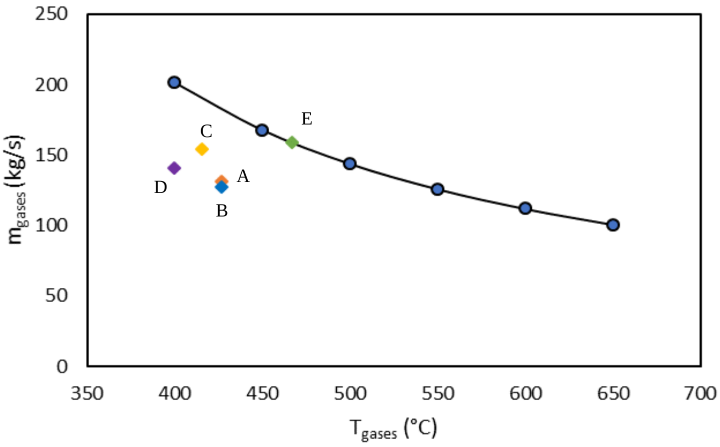

Figure 4. Thermal parity curve - corrected with turbines selected from the Gas Turbine World Handbook, 2012.

According to Fig. 4, the points below the curve are more likely to reach the thermal parity with the use of a supplementary burner and they are subjected to evaluation.

3.5. Determination of Pinch-Point

The temperature of the gases at the exit of the recovery boiler may not be less than a minimum value, which depends on the fuel and the combustion process, since acids may form when the products of combustion are cooled in the boiler, damaging the equipment due to the composition of the fuel (in the case of natural gas the formation of acid hardly exists). Therefore, the temperature (T_5) of the exhaust gases in an already selected gas turbine model should be corrected by the *pinch-point* according to the following criteria, obtained from Fig. 5 (Balestieri, 1994; Barclay, 1995).

Figure 5 shows the gas cooling profile up to the intersection with the T axis denoted by T_{PP}. The temperature T_4 is the already corrected gas temperature at the outlet of the gas turbine and T_S is the saturation temperature, under the saturation pressure condition.

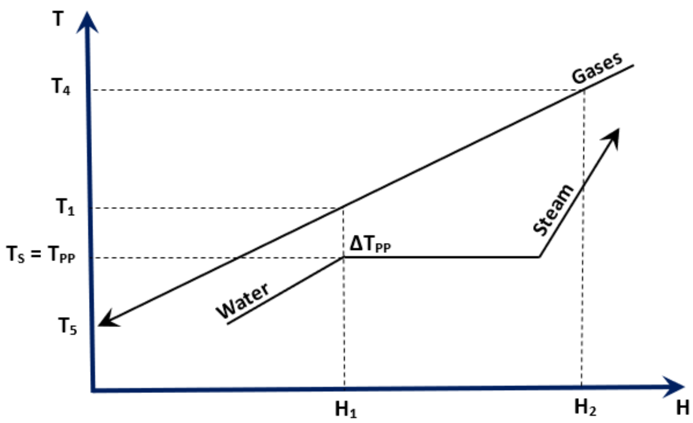

Figure 5. Determination of Pinch Point.

In order not to incur a thermodynamic impropriety, a minimum delta T (ΔT_{PP}) must be obtained for the gas cooling profile from the turbine. According to (Sue and Chuang, 2004) this value can be between 10 and 30 °C. A delta value of 25 °C was used for this delta, determining the dew point temperature (T_1):

$$T_1 = T_S + \Delta T_{PP} \tag{42}$$

The temperature of the boiler exhaust gas is determined by (Antunes, 1999; Barclay, 1995; Balestieri, 1994):

$$T_G = T_4 - \frac{T_4 - T_1}{H_2 - H_1} * H_2 \tag{43}$$

Where: the rates of heat transfer received (H_2) and assigned (H_1) are respectively given by:

$$H_2 = \dot{m}_G * Cp_{G\,(T4)} * T_4 \tag{44}$$

$$H_1 = \dot{m}_S * h_l \tag{45}$$

Where: h_l is the specific enthalpy of the saturated liquid, under the saturation pressure condition.

3.6. Turbine parameters for ISO and corrected conditions

The formulas presented in Tab. 2 will be used for the development of each turbine selected from the Gas Turbine World Handbook, 2012.

Table 11. Turbine data at ISO and corrected (*) conditions for local parameters

Turbines	HR (kJ/kWh)	E_P (kW)	n_T (%)	T_4 (°C)	Q_G (kJ/s)	m_G (kg/s)	m_F (kg/s)	m_{SF} (kg/s)	m_{TF} (kg/s)
A	8590	51204	41,9	470	48393,6	143,9	2,681	0,339	3,020
A*	9495	43482	37,9	427	37547,5	130,7	2,391	0,566	2,957
B	8774	51342	41,0	470	47048,4	139,9	2,746	0,218	2,964
B*	9699	43599	37,1	427	36513,3	127,1	2,449	0,587	3,036
C	9395	51235	38,3	458	54813,1	169,6	2,934	0,056	2,990
C*	10387	43508	34,7	416	42282,5	153,9	2,617	0,467	3,084
D	8586	51685	41,9	440	46704,7	154,6	2,705	0,225	2,930
D*	9494	43890	37,9	400	36012,6	140,4	2,413	0,389	2,802
E	11025	49500	32,7	514	67669,1	175,1	3,327	0,000	3,327
E*	12192	42035	29,5	467	52894,6	158,9	2,968	0,246	3,214

From Tab. 11, the cycle with the turbine D spends less fuel (2,802 kg/s), has a better efficiency (37,9 %) and still meets the electric demand of 40000 kW.

3.7. Analyzing the first law of thermodynamics

Table 12 shows the fixed parameters for the following analysis.

Table 12. Fixed parameters for energy analysis

Physical parameters	Symbology	Specification	Units
Compressor efficiency	n_C	86	%
Pump Efficiency	n_P	90	%
Ratio of natural gas	R_{NG}	0,29	-
Ratio of air	R_{air}	0,298	-
Lower calorific value of natural gas	LHV_{GN}	47966	kJ/kg
Loss of gas pressure in the recovery boiler	ΔP_{RB}	0,05	-
Loss of gas pressure in the combustion chamber	ΔP_{CC}	0,05	-
Gas turbine ISO efficiency	$n_{ISO\ GT}$	89	%
Compressor ISO efficiency	$n_{ISO\ C}$	80	%
Steam turbine ISO efficiency	$n_{ISO\ ST}$	86	%
Recovery Boiler Efficiency	n_{RB}	80	%
Efficiency of the combustion chamber	n_{CC}	99	%
Steam turbine efficiency	n_{ST}	98	%
Gas turbine efficiency	n_{GT}	87	%

Table 13, shows the results of the energy analysis using the equations in Tab. 3 and the fixed parameters in Tab. 12, considering the system without/with supplementary burning, respectively.

Figures 6 and 7 show the Sankey diagrams with system efficiencies, both for a thermal cycle without supplementary and supplementary burning, respectively.

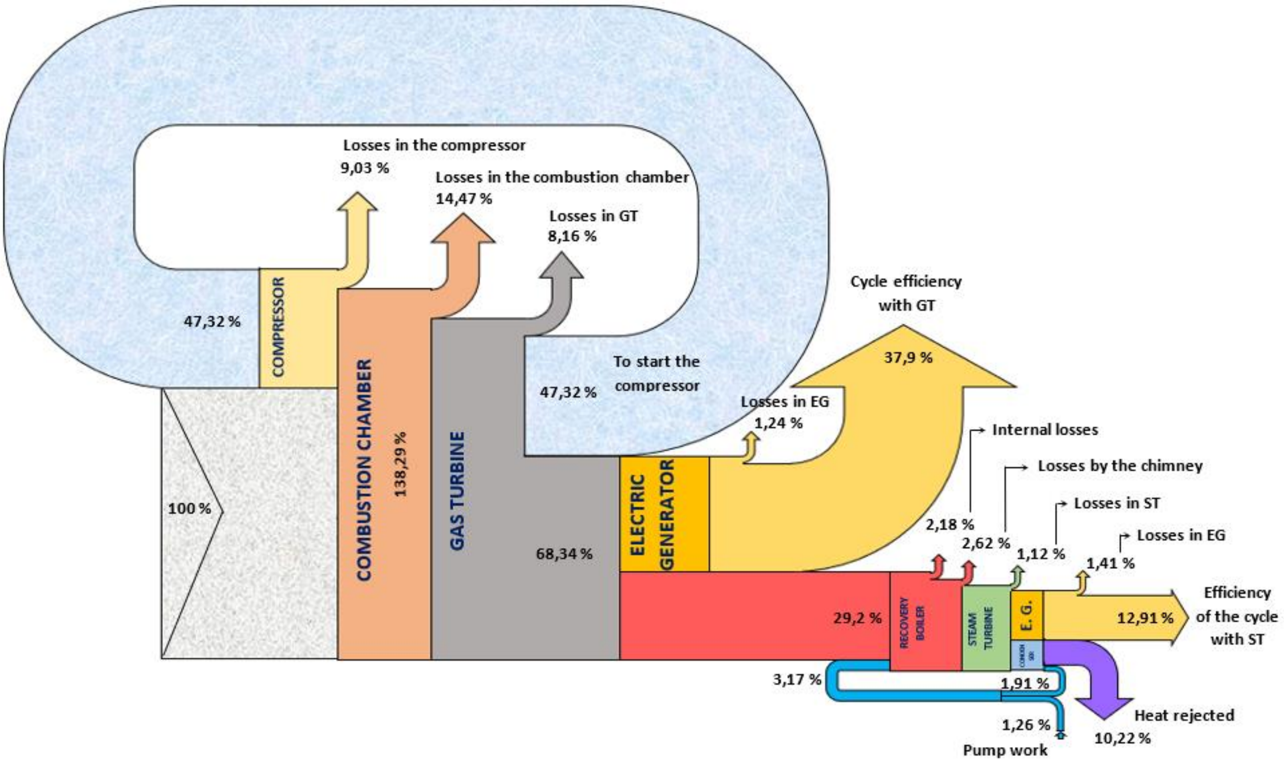

Figure 6. Sankey diagram of the combined cycle thermal plant without supplemental burning.

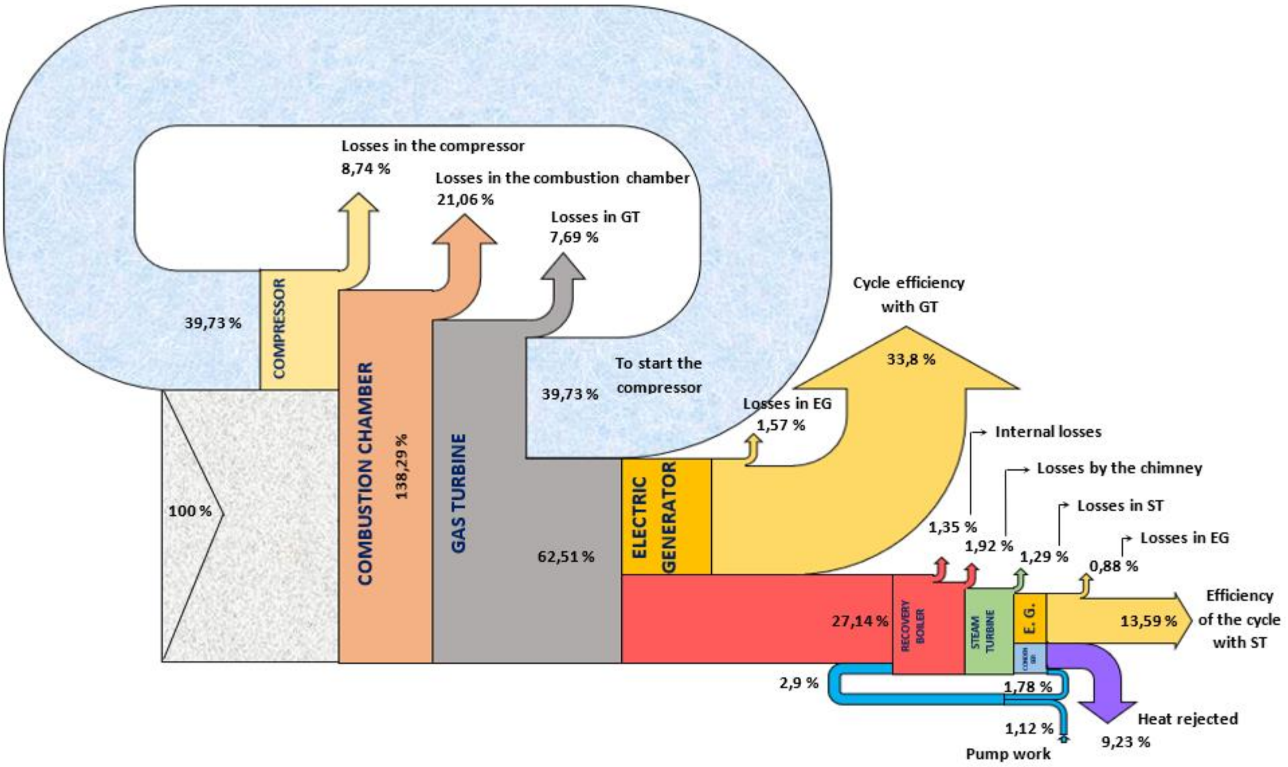

Figure 7. Sankey diagram of the combined cycle thermal plant with supplemental burning.

Table 13. Result of energy analysis the system without/with supplementary burning

Energy analysis	Without supplementary burning	With supplementary burning
E_{fuel} (kW)	115741,9	134400,7
$E_{P\ GT}$ (kW)	43012,2	45451,1
$E_{P\ ST}$ (kW)	17252,0	19796,8
W_P (kW)	1454,2	1499,7
$E_{P\ TOTAL}$ (kW)	60264,2	65247,9
n_{GL} (%)	50,81	47,39

4. CONCLUSIONS

As expected, through energy analysis it is possible to verify that the best overall efficiency is obtained when using natural gas in the case (Rolls-Royce gas turbine: Trent 60 DLE associated with the Siemens SST-200 steam turbine) presenting a higher efficiency (50,81 %) compared to the system with supplementary burning (47,39 %). The two systems were represented in a Sankey diagram.

For future work, a simulation of the process could be carried out, taking into account the time, the annual meteorological variations for an estimate of production and efficiency closer to each season of the year.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Antunes, J. S. 1999. "Código computacional para análise de sistemas de cogeração com turbinas a Gás". Faculdade de Engenharia do Campus de Guaratinguetá, *Universidade Estadual Paulista*, Brasil.

Balestieri, J. A. P. 1994. "Planejamento de Centrais de Cogeração: Uma abordagem Multiobjectivo". Faculdade de Engenharia da Campinas, *Universidade de Campinas*, Brasil.

Barclay, F. J. 1995. "Combined Power and Process and Exergy Approach". *Editora Mechanical Engineering Publication Limited*. ISBN 0-85298-943-3, London, England.

Biasi, V. 2012. "Gas turbine World 2012 Handbook. For project planning, Desing, Construction and operation, Southport". *Pequot Publication*, Vol. 23, pp. 152.

Brooks, F. J. 1994. "GE Gas Turbine Performance Characteristics". *38th GE Turbine State of the art Technology Seminar*. GER 3567E.

Cardu, M.; Baica, M. 1999. "Regarding a new variant methodology to estimative globally the ecologic impact of thermopower plants". *Energy Conversion and Management*, Vol. 40, No. 1, pp. 1569-1575.

Franco, A., Casarosa, C. 2002. "Thermoeconomic evaluation of the feasibility of highly efficient combined cycle power plants". *Energy*, Vol. 29, No. 1, pp. 1963-1982.

Pantalena, A. 1997. "Cogeração para Laticínios de pequeno e médio portes". Faculdade de Engenharia do Campus de Guaratinguetá, *Universidade Estadual Paulista*, Brasil.

SIEMENS. Available in: http://www.siemens.com.br/ Access on Jan 07. 2019.

Silveira, J. L.; Tuna, C. E. 2003. "Thermoeconomic Analysis Method for Optimization of Combined Heat and Power Systems. Part I". *Progress in Energy and Combustion Science*, Vol. 29, No. l, pp. 479-485.

Sue, D. C.; Chuang, C. C. 2004. "Engineering design and exergy analyses for combustion gas turbine based power generation system", *Energy*, Vol. 29, pp.1183-2005.

Villela, I. A. C., Silveira, J. L. 2007a. "Energetic and Exergetic Analysis of a Gas Turbine Combined Cycle Power Plant Operating with Natural Gas or Diesel". *COBEM 2007 – 19th International Congress of Mechanical Engineering*. Brasilia, Brasil.

Villela, I. A. C., Silveira, J. L. 2007b. "Desenvolvimento de um modelo termoeconômico que considera os impactos ambientais". Faculdade de Engenharia do Campus de Guaratinguetá, *Universidade Estadual Paulista*, Brasil.

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