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EXERGETIC EVALUATION OF A COGENERATION SYSTEM IN A PETROCHEMICAL COMPLEX

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Abstract—The purpose of this paper is the study of a real cogeneration system existing in a petrochemical complex. The total capacity of the plant is 200 MW of electrical power and 2100 t/h of steam for processes. This work presents the mass and energy balances of this cogeneration system. Exergy of each involved stream is calculated and the exergetic balance of each subsystem is presented, as well as of the global system, identifying where and why losses and irreversibility occur. Efficiencies based on the second law of thermodynamics are calculated for each subsystem and compared. Some conclusions regarding operational strategies are presented. © 1998 Elsevier Science Ltd. All rights reserved

Cogeneration Exergy analysis Combined heat and power plants

INTRODUCTION

Petrochemical industries in Brazil are usually situated in complexes which reunite numerous different factories operating in the same field. This conception of topographic proximity aims to optimize raw material distribution between petroleum sub-products producers and consumers. For the same reason, petrochemical complexes are usually located near petroleum refineries and/or to sea terminals.

The leader industry produces petrochemical goods of first generation (ethylene, propylene, butadiene, benzene, toluene, etc.) and supplies them to manufacturers of second or third generations that use such raw materials to produce different kinds of plastics, resins, fibers, fertilizers and various other products.

The complementary relation existing between the complex companies may extend, beyond raw materials exchange, to other objectives: as a matter of fact in some cases it is interesting to centralize the production of utilities (steam, electricity, treated water, compressed air, etc.) and also the collection and treatment of liquid and solid effluents that performed in the complex.

This work presents a study on a central cogeneration system which supplies utilities to a petrochemical complex. In the analysed case, steam for process at three pressure levels and electric power are produced by one basic plant and distributed to the complex industries.

DESCRIPTION OF THE COGENERATION SYSTEM

The cogeneration unit is capable of producing 180 MW of electrical power and 2100 t/h of steam. It consists of five boilers and four turbo-generators, each boiler having nominal capacity of 400 t/h of steam at 12 MPa and 530°C. A gas turbine operates in combined cycle.

At present the boilers operate below nominal conditions, namely four of them operate at partial load and the fifth is under maintenance. They burn liquid fuels (refinery residual fuel oil, fuel oil, industrial resins) and gaseous fuels (natural gas and process by-product gases). Water for the boilers is supplied by a water treatment unit, where it is demineralized and delivered as make-up water to direct contact condensers. The boilers feature natural circulation and pressurized furnace.

The deaerator is supplied with condensate, water and steam and operates at 3.5 bar. Boiler feed pumps take preheated water from the deaerator.

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Maximum capacity of each turbo-alternator is 42 MW when running at 120 bar and 530°C. Turbine extractions are made at 42 bar and 15 bar. Maximum steam flow per machine is 425 t/h, whereas 10 MW is the turbine minimum load.

One gas turbine and its heat recovery steam generator (HRSG) is included in the system. The steam is raised to nominal pressure in the HRSG and sent to a high-pressure steam header. Fuel used in the turbine is natural gas and its exhaust gases are delivered to the heat recovery boiler. Additional fuel is fired in the HRSG unit to keep steam conditions. The HRSG capacity is 100 t/h of superheated stead at 120 bar and 530°C. Feed water is supplied at 144°C and 150 bar.

Other equipment such as expansion valves, water pre-heaters, etc., are included in the system, as can be seen in Fig. 1.

ANALYSIS OF THE SUBSYSTEMS

In Fig. 1, mass flows, power and heat flows are identified (numbered streams), as well as sub-systems composing the global system (capital letters). These are: boilers (sub-system A), steam turbo-generators (sub-system B), deaerator (sub-system C), expansion valve (sub-system D), boiler feed pump (sub-system E), low-pressure water heater (sub-system F), high-pressure water heater (sub-system G), steam trap (sub-system H), gas turbine with alternator (sub-system I) and HRSG (sub-system J). Mass, energy and exerby balances for any control volume at steady-state, with negligible kinetic and potential energy variations, can be expressed respectively by the equations:

$$\sum_{\text{in}} \dot{m}_{\text{in}} = \sum_{\text{out}} \dot{m}_{\text{out}} \tag{1}$$

$$\sum_{j} \dot{Q}_j + \sum_{\text{in}} \dot{m}_{\text{in}} h_{\text{in}} = \sum_{\text{out}} \dot{m}_{\text{out}} h_{\text{out}} + \sum_{j} \dot{W}_j \tag{2}$$

$$\sum_{j} \dot{E}_j^Q + \sum_{\text{in}} \dot{m}_{\text{in}} e_{\text{in}} = \sum_{\text{out}} \dot{m}_{\text{out}} e_{\text{out}} + \sum_{j} \dot{W}_j + \dot{I} \tag{3}$$

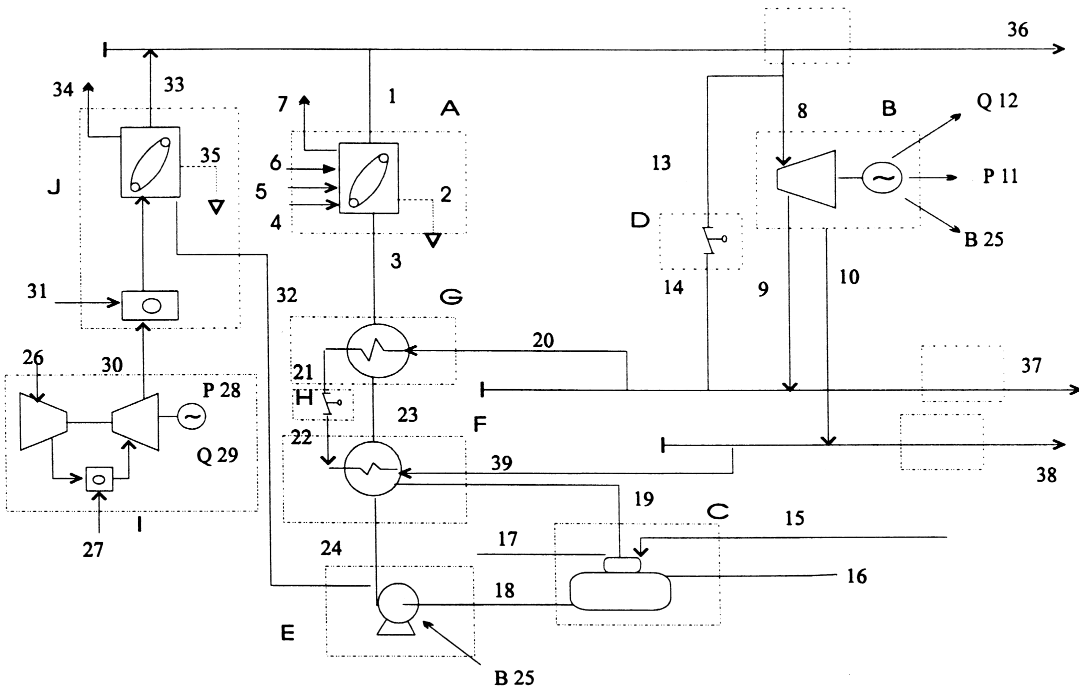

Fig. 1. Flow sheet and identification of adopted control volumes.

where I is the irreversibility rate (exergy destruction rate) of the control volume under analysis and E_i^Q is the exergy associated to a heat flow available at a temperature T_j, given by

$$\dot{E}_j^Q = \dot{Q}_j\left(1 - \frac{T_0}{T_j}\right). \tag{4}$$

The specific flow exergy (e) is given by

$$e = (h - h_0) - T_0(s - s_0) + e^{ch} \tag{5}$$

where the subscripts (**0**) stand for the restricted dead-state (ambient pressure and temperature) and superscript (**ch**) stands for the standard chemical component.

The thermodynamic properties of water and steam were obtained from steam tables [1]. The properties of gases (air, gas fuels and combustion products) were calculated employing the methodology proposed by Reid *et al.* [2], with polynomial regressions for specific heat.

The ideal gas model was applied for air and combustion products. The chemical exergies of different substances (e^{ch}) were determined according to Kotas [3]. Temperature, pressure, mass flow, enthalpy, entropy and exergy for each mass flow are shown in Table 1, according to the nomenclature presented in Fig. 1. In the same table, power and heat flows were also identified.

For each sub-system, second law efficiency ε, also known as "rational efficiency" [3], and ε', also called "degree of thermodynamic perfection" [4], have been determined according to the

Table 1. Flow identification and properties

Flow identification	Flow ♯	T (°C)	P (bar)	m (kg/h)	h (kJ/kg)	s (kJ/kgK)	e (kJ/kg)
Steam 120 bar	1	530	120	1,075,000	3428.0	6.5880	1642.7
Boiler blow-down	2	324	120	15,000	1486.2	3.4960	622.3
Boiler feed water	3	250	150	1,090,000	1085.5	2.7670	438.8
Combustion air: boiler	4	25	1	939,440	0.0	0.0000	0.0
Fuel (refinery residual oil)	5	210	—	60,453	40,130.5	—	42,847.6
Fuel (natural gas)	6	25	—	5107	47,612.6	—	52,062.0
Exhaust gases	7	140	1	1,005,000	118.0	0.3348	127.0
Steam to turbine	8	520	120	825,000	3402.0	6.5550	1626.5
Turbine outlet (42 bar)	9	385	42	372,000	3174.0	6.6880	1358.9
Turbine outlet (15 bar)	10	285	15	453,000	3004.0	6.8580	1138.2
Turbo-alternator power	11	—	—	—	—	—	—
Heat loss to environment	12	—	—	—	—	—	—
Steam to reducing valve	13	520	120	80,000	3402.0	6.5550	1626.5
Steam from reducing valve	14	482	42	80,000	3402.0	7.0120	1490.3
Steam to deaerator	15	190	3,5	158,400	2842.0	7.1920	876.7
Make-up water	16	25	1	747,600	104.9	0.3674	173.3
Condensate return	17	110	9	31,000	461.8	1.4178	217.2
Deaerator exit water	18	143,6	3,4	1,171,000	607.1	1.7726	256.8
Condensate from LP heater	19	170	15	234,000	719.2	2.0410	288.9
Steam to preheater (HP)	20	385	42	147,700	3174.0	6.6880	1358.9
Condensate (from HP)	21	200	42	147,700	853.5	2.3270	337.9
Recycle to LP heater	22	sat	15	147,700	853.5	2.3330	336.2
Water to HP heater	23	190	150	1,090,000	814.2	2.2160	331.7
Water to LP heater	24	144,5	150	1,090,000	617.7	1.7699	268.2
Pump power	25	—	—	—	—	—	—
Gas turbine inlet air	26	25	1	435,571	—	—	—
Gas turbine NG fuel	27	25	1	6400	47,612.6	—	52,062.0
Gas turbine power	28	—	—	—	—	—	—
Heat loss to environment	29	—	—	—	—	—	—
Turbine exhaust gases	30	550	1,05	441,971	563.0	1.0893	347.1
Natural gas to HRSG	31	25	—	1,050	47,612.6	—	52,062.0
HRSG feed water	32	144,5	150	81,000	617.7	1.7699	268.2
Steam 120 bar	33	530	120	80,000	3428.0	6.5880	1642.7
HRSG exhaust gases	34	160	1	443,021	138.5	0.3833	133.1
HRSG blow-down	35	sat.	120	1000	1491.2	3.4960	627.3
Steam export (120 bar)	36	530	120	250,000	3428.0	6.5880	1642.7
Steam export (42 bar)	37	385	42	304,300	3174.0	6.6880	1358.9
Steam export (15 bar)	38	285	15	366,700	3004.0	6.8580	1138.2
Steam (15 bar)	39	285	15	86,300	3004.0	6.8580	1138.2

definitions given in equations (6) and (7):

$$\varepsilon = \frac{\text{desired exergetic effect}}{\text{exergy used to drive the process}} = \frac{\text{``product''}}{\text{``fuel''}} \tag{6}$$

$$\varepsilon' = \frac{\text{exergy outlet}}{\text{exergy inlet}} = \frac{\sum_{\text{out}} \dot{E}_{\text{out}}}{\sum_{\text{in}} \dot{E}_{\text{in}}}. \tag{7}$$

equation (7) simply considers all leaving exergetic flows and compares them to the entering exergetic flows. This thermodynamic performance parameter measures only the internal irreversibility, without considering the function the sub-system may perform. Although it can be calculated for any process, it should be adopted in processes where a desired exergetic effect cannot be defined (purely dissipative processes).

Although the first definition—equation (6)—needs to be defined before its use for each sub-system, it brings more information than the second one, since one can identify "generalized products" at numerator and "generalized fuels" at denominator, having an immediate link to economical concepts and giving rise to a new methodology called thermoeconomy [5–8]. On the other hand, equation (6) cannot be applied to the expansion valve and steam trap (for they are purely dissipative processes). The expressions for the rational efficiency of boiler, steam turbo-generator, water pump, low and high pressure water preheaters, deaerator, gas turbine and HRSG are given respectively by:

$$\varepsilon_C = \frac{\dot{E}_1 - \dot{E}_3}{\dot{E}_5 + \dot{E}_6} \tag{8}$$

$$\varepsilon_{TV} = \frac{\dot{P}_{11} + \dot{B}_{25}}{\dot{E}_8 - \dot{E}_9 - \dot{E}_{10}} \tag{9}$$

$$\varepsilon_B = \frac{\dot{E}_{24} + \dot{E}_{32} - \dot{E}_{18}}{B_{25}} \tag{10}$$

$$\varepsilon_{PB} = \frac{\dot{E}_{23} - \dot{E}_{24}}{\dot{E}_{22} + \dot{E}_{39} - \dot{E}_{19}} \tag{11}$$

$$\varepsilon_{PA} = \frac{\dot{E}_3 - \dot{E}_{23}}{\dot{E}_{20} - \dot{E}_{21}} \tag{12}$$

$$\varepsilon_{DA} = \frac{\dot{m}_{16}(e_{18} - e_{16}) + \dot{m}_{17}(e_{18} - e_{17})}{\dot{m}_{15}(e_{15} - e_{18}) + \dot{m}_{19}(e_{19} - e_{18})} \tag{13}$$

$$\varepsilon_{TG} = \frac{\dot{P}_{28}}{\dot{E}_{26} + \dot{E}_{27} - \dot{E}_{30}} \tag{14}$$

$$\varepsilon_{CR} = \frac{\dot{E}_{33} - \dot{E}_{32}}{\dot{E}_{31} + \dot{E}_{30}}. \tag{15}$$

Boilers sub-system (A in Fig. 1) is composed of five boilers. For simulation purposes they were considered as one equivalent boiler. The steam flow and blow-down losses are known from field data. Boiler heat losses to the environment were estimated by energy balance.

The boiler burns a mixture of natural gas and refinery residual fuel oil. The use of other fuels (process gas, liquid effluents, etc.) is only occasional and it was not considered in this work. To

Table 2. Mass and energy balances for the sub-systems

Subsystem	Mass balance	Heat losses (kW)	Power (kW)
Steam generator	0.00	−7327	—-
Steam turbine	0.00	−865	67,577
Deaerator	0.00	−81.1	—-
Expansion valve	0.00	—-	—-
Water pump	0.00	−1,752	5200
Preheater LT	0.00	−786	—-
Preheater HT	0.00	−13,061	—-
Steam trap	0.00	—-	—-
Gas turbine	0.00	−2325	13,200
HRSG	0.00	−3269	—-
Header 120 bar	0.00	−6536	—-
Header 42 bar	0.00	−4802	—-
Header 15 bar	0.00	—-	—-

determine the rational efficiency, exergies of water losses and of the exhaust gases have been considered external irreversibilities (emissions to the environment).

The assumed efficiency of boiler feed pump electric motor is 75%. The pump is electrically driven (B25), making use of a small part of the power produced by the turbo-alternators.

ANALYSIS OF RESULTS

Based on field data (mass flow, pressure and temperature) the consistency of data related to mass conservation has been verified. By means of energy balance the energy losses to the environment have been determined. Table 2 shows mass and energy balance, for each sub-system. From this, it can be seen that the mass balance for all sub-systems has been checked (second column). The energy balance allowed the evaluation of the residual heat lost to the environment through the equipment's carcasses (third column). In the case of the steam and gas turbines, these losses were estimated as a percentile of the converted thermal energy. The power values (fourth column) do not constitute the nominal conditions of the cogeneration system but part load condition given by operating parameter presented at the Table 1.

The total net power generation is 72.77 MW, a load of 40% of the nominal value. The total steam generation amounts 1155 t/h, or 52% of nominal rating. The steam turboalternators and the gas turbine both operate under part load in this case. The power production efficiency of the gas turbine (first law) is of only 15.6%.

Table 3 shows the exergy destruction for each sub-system as well as its percentile contribution to the plant total irreversibility. In this table it can be observed that the largest exergy destruction occurs in the steam generator. The exergy destruction in any boiler is due to the irreversibility of the combustion process itself and also due to heat transfer with large temperature differences. Exergy destruction associated with exhaust gases dispersion into the environment and the irreversibilities associated with blow-down water loss were also considered, although

Table 3. Irreversibilities (exergy destruction) at each sub-system

Subsystem	Irreversibility (exergy destruction) (kW)	As a percentile from total plant irreversibility
Steam generator	435,704	81.51
Steam turbine	16,320	3.05
Deaerator	11,648	2.18
Expansion valve	3027	0.57
Water pump (with motor)	1492	0.29
Preheater LT	3075	0.57
Preheater HT	9462	1.77
Steam trap	70	0.01
Gas turbine	36,741	6.87
JRSG	10.774	2.02
Steam Header 120 bas	3301	0.62
Steam Header 42 bar	2921	0.54
Steam Header 15 bar	0	0.00
Total	534,535	100.00

they are usually viewed as exergy losses, going out of the control volume. In this case, they were viewed as external irreversibilities. Although very important, combustion irreversibility is called intrinsic, since it is inherent to the combustion process. This part of boiler irreversibility can be somewhat reduced by preheating the combustion air.

The gas turbine irreversibility amount is substantially smaller than that of the boilers. This fact is not related to any qualitative difference between the two combustion processes, but only to the smaller amount of fuel burned in the gas turbine when compared with that burned in the boilers (the boiler converts much more thermal energy than the gas turbine). For the HRSG associated with the gas turbine, apart from processing lower amounts of steam, there is a proportional reduction on the exergy destruction, motivated by the more sophisticated design (from thermodynamic point of view) of this kind of equipment, and by the smaller contribution of combustion process (the supplementary firing) to HRSG irreversibility.

The exergetic efficiencies of each sub-system (defined by the equations (6) and (7) respectively) are presented in Table 4. As expected, rational efficiency shows values lower than those inherent to the "degree of thermodynamic perfection" for each sub-system analysed. The rational efficiencies of processes involving combustion (boilers and gas turbines) are low. Heat transfer equipment (preheaters and deaerator) are more efficient. In the case of HRSG, its rational efficiency stands between the two groups above mentioned, due to the supplementary fuel burning.

Rational efficiency of the gas turbine is close to the first-law thermal efficiency for the cycle, since fuel LHV and chemical exergy are numerically similar (usually chemical exergy stands between LHV and HHV).

In the steam turbine sub-system, exergetic efficiency has a value in the same order as the isentropic efficiency of the turbine.

The processes where a useful exergetic effect cannot be defined, do not allow the calculation of the rational efficiency. On the other hand, the "degree of thermodynamic perfection" (ε') can always be obtained for control volumes. This emphasizes the relevance of both concepts as tools for the measurement of thermodynamic performance.

For an overall plant evaluation, the boundaries of the control volume incorporate all analysed sub-systems. Taking again Fig. 1 as reference, exiting flows are steam export (flows ♯ 36–38), boilers blow-down (flow ♯ 2 and 35), exhaust gases (flow ♯ 7 and 34), electrical power (flow ♯ 11 and 28) and all heat losses to the environment shown in Table 2. The inlet flows are the air (flow ♯ 4 and 26), natural gas (flow ♯ 6, 27 and 31), refinery residual fuel oil (flow ♯ 5), condensate return from processes (flow ♯ 17), low pressure steam (flow ♯ 15) and make-up water (flow ♯ 16).

The overall first-law efficiency for the power generation alone can be written as

$$\eta = \frac{\dot{W}_{ST} + \dot{W}_{GT}}{m_0 LHV_0 + m_{NG} LHV_{NG}}. \tag{16}$$

The second-law efficiency for power generation alone uses the chemical exergy of the fuels instead of the LHV:

Table 4. Second law efficiency of the sub-systems

Subsystem	ε (%)	ε' (%)
Steam generator	45.08	52.96
Steam turbine	81.69	95.62
Deaerator	60.19	87.73
Expansion valve	0.00	91.63
Water pump (with motor)	71.31	98.32
Preheater LP	86.22	97.49
Preheater HP	82.19	93.94
Steam trap	0.00	99.50
Gas turbine	26.43	60.30
HRSG	52.70	83.12

$$\varepsilon = \frac{\dot{W}_{ST} + \dot{W}_{GT}}{m_0 e_0^{\text{ch}} + m_{NG} e_{NG}^{\text{ch}}}. \tag{17}$$

The first-law efficiency, rational efficiency and the "degree of thermodynamic perfection" of the global cogeneration system are, respectively:

$$\eta = \frac{\dot{W}_{ST} + \dot{W}_{GT} + H_{36} + H_{37} + H_{38} - H_{16} - H_{17} - H_{15}}{m_0 LHV_0 + m_{NG} LHV_{NG}} \tag{18}$$

$$\varepsilon = \frac{\dot{W}_{ST} + \dot{W}_{GT} + E_{36} + E_{37} + E_{38} - E_{16} - E_{17} - E_{15}}{m_0 e_0^{\text{ch}} + m_{NG} e_{NG}^{\text{ch}}} \tag{19}$$

$$\varepsilon' = \frac{\dot{W}_{ST} + \dot{W}_{GT} + E_{36} + E_{37} + E_{38} + E_{34} + E_{35} + E_7 + E_2}{m_0 e_0^{\text{ch}} + m_{NG} e_{NG}^{\text{ch}} + E_{15} + E_{16} + E_{17}} \tag{20}$$

where H_i stands for enthalpy flows and E_i stands for exergy flows.

Table 5 presents the above mentioned three types of efficiencies for the overall analysis of the plant. The power cycle generation regards only the electrical power generated by the steam and gas turbo-alternators as products. The resulting low values are due to the exclusive use of back-pressure steam turbines and the low contribution of the gas turbine to the total power. For the complete cogenerative system, two types of useful products may be identified: electrical energy and steam for the processes.

If only power production is considered, it can be seen that first law efficiency and rational efficiency yield very similar values. On the other hand, considering the combined production of industrial steam and electrical power, substantially different values emerge (88.40% versus 38.75%).

This apparent discrepancy occurs because the first law efficiency considers the steam flows on enthalpic basis; hence, 1 kW of steam (enthalpic basis) is equalized to 1 kW of electrical power. In the case of the second-law efficiency, only the available part of the steam energy (exergy) is taken into account. As a matter of fact, the first-law efficiency applied to cogeneration systems tends to overrate the steam enthalpic flow in relation to the electrical power. The second-law efficiency is much closer to the economic concepts of cost attribution to steam and electrical power.

The degree of thermodynamic perfection (ε') relates all exergy outputs to all inputs. By this way it can be calculated of the cogenerative system as a whole and not for the power cycle taken isolate.

CONCLUDING REMARKS

The exergy analysis of the cogeneration system studied here allowed to identify the sub-systems that present the worst thermodynamic performance. This kind of information is relevant to optimize the system performance from a economical point of view. Since the present analysis was applied for an existing system, new operation strategies can arise.

Even if not always possible, exergetic analysis recommends that purely dissipative devices (such as expansion valves) be avoided, although from the viewpoint of energy conservation (first law) such elements do not induce energy losses to the environment (isenthalpic processes). On the other hand, exergetic analysis reduces the relative importance of the enthalpic flows lost as

Table 5. Overall efficiencies for the plant

	First law efficiency (η)	Rational efficiency (ε)	Degree of thermodynamic perfection (ε')
Power cycle alone	9.62	8.96	—
Cogeneration	88.40	38.75	49.13

exhaust gas or boiler blow-down, since the exergy lost with these flows are lower than the enthalpic losses.

Partial load operation of the cogeneration system, analysed here, may seem very efficient, if analysed exclusively by means of the first law. The electrical generation can be modulated at will, apparently without much impact on the first-law efficiency for the system, since there is no difference if the useful energy is in electrical or thermal form.

However, this is not the figure shown by the second-law analysis. Special care must be taken when operating the system at partial load. Exergetic analysis enhances the value of electrical power production, whereas it reduces the value of steam (to the available part of the enthalpic flow). From this point of view, care has to be taken with regard to the modulation of the electrical load of the system, since expansion of great amounts of steam through the valves introduces enormous irreversibility into the process. In the same way, the operation of gas turbine or boilers under partial loads must be avoided.

The electric power generation in Brazil is highly based on hydro-electricity. The national distribution grid offers low prices to customers. Then, thermal generation must be regarded as complementary; this fact affects the cogeneration systems negatively, since electric power can be taken from the grid. In such a situation, the correct operation of the system has a strong influence on the economic performance of the cogeneration system.

The analysis of this cogeneration system will be continued, through the investigation of other load conditions. Steam demand of the industrial consumers will also be investigated, since it seems there is some kind of unbalance between the pressure of the steam supplied and its end use inside each industry.

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