Developing an operating map to define the best thermal scheme, in off-design conditions, for a utilities plant.

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Abstract:
With the purpose of optimizing operation, in off-design conditions, this paper presents the development of a methodology for creating an “operating map” of an existing utilities plant. The objective of this map is to define the best thermal scheme and equipment mass flow for any given operating demands (*e.g.* energy and steam supply) at a specific time. The influence of off-design conditions in the performance of equipment, the dynamic nature of utilities plants, especially in cogeneration, as well as the changes that occur (and have occurred) to local and national energy scenario highlight the importance of having an operating map based on a mathematical model of a plant. Focused on reducing fuel consumption of a plant with a high idle capacity, this study simulated operation for specially selected operating demand scenarios and compared the exergy efficiency of the configurations generated by the map to those of typical operating configurations. This comparison, between the optimized and typical configurations, shows that highly unconventional changes to configuration as well as the understanding of the cycle as a whole can lead to an increase in exergy efficiency that varies from 1-5% (for a production that ranges from 65-110 MW of electricity and 535-735 tons per hour of steam at 117.7 bar and 811K). In order to evaluate the components in off-design conditions, developed analytical equations, equipment historical data as well as manufacturer provided operation curves were used. The GRG (generalized reduced gradient) algorithm was used as the resolution method to provide the minimum fuel consumption condition.

Keywords:
Thermoelectric plant, thermal scheme, operating map, steam generation and distribution, operational gain.

1. Introduction

The importance of thermoelectric plants for the petrochemical industry is widely known, due, primarily, to the need for steam in the industry’s processes. Thus, electricity production in cogeneration is a common choice. The search for more efficient thermal cycles has led to the implementation of complex configurations, combining gas turbines (GT) with heat recovery steam generators (HRSG), steam cycles with several pre-heating stages, more than one pressure level and numerous components. These systems offer the opportunity for substantial savings; however, the minimization of expenditures represents a very challenging task due to the large number of design choices and complex interactions among its units. This complexity is even greater when the components have their performances altered at the same time, due to off-design conditions, as pointed out by Aguilar et al. [1], Toffolo and Lazzaretto [2], Zaleta et al. [3] and Zaleta et al. [4]. In Aguilar et al. [1], the occurrence of equipment redundancy is highlighted and this fact allied to the variations in demand gives the plant idle capacity and great operating flexibility, allowing for numerous component configurations that are able to supply the steam and electricity demands. The incorrect choice of configurations can lead critical equipment to operate far from their parameter values, impacting significantly their performance and that of the cycle. In Luo et al. [5] it is emphasized that many simulation and optimization studies of utility systems have been performed taking into account only the macroscopic mass and energy balances of these types of plants; however, this approach does not allow one to account for non-linear variations in the process due to complex interrelations of components. In El-Sayed [6] the fuel consumption of power plants with known energy demands was optimised using correlations to calculate efficiency, whereas in Lazzaretto and Carraretto [7] a plant model was simulated using operation data.

The system developed for this paper proposes to simulate the best thermal scheme, for a certain demand of steam and electricity, taking into consideration both supplier parameter correlation information and actual operational data of the studied plant. In order to achieve the proposed minimal fuel consumption in these conditions, the Generalized Reduced Gradient (GRG) algorithm method was used.

2. Thermal cycle description

The thermal cycle comprises different parts of the petrochemical plant. The main focus of this study is the utilities plant, but due to the fact that it is a cogeneration cycle, a part of the petrochemical process is taken into consideration, more specifically the olefins plant and its furnaces. The petrochemical plant will be called CEMAP and will be considered as a single control volume. The utilities plant features, as its main products, steam at four pressure levels, electricity and compressed air; this last product was not considered a relevant topic due to its low energy content (density). The main objective of the generated steam is to meet the demands of CEMAP's processes, though it is also used to generate electricity and is sold to neighbouring plants (external clients). The pressure levels of the steam are 11.78 MPa (~120 bar), 4.12 MPa (~41 bar), 1.47 MPa (~15 bar) and 0.34 MPa (~3 bar). The general flowchart, based on a complete flowchart shown in Torres and Gallo [8], for the cycle is represented in Fig. 1.

2.1. Steam generation

The steam is generated by five CSGs (Conventional Steam Generators) as well as one HRSG. All the boilers generate steam at 11.78 MPa and a mean temperature of 538°C. The conventional boilers have a design capacity of 400 tons per hour while the heat recovery boiler is designed to generate only 100 tons per hour. Lower pressure levels are generated by the expansion of the steam through two back pressure turbines, through CEMAP's process or pressure reducing valves.

2.2. Electricity generation

Electricity is currently produced by the two BPTs (back pressure turbines), which admit steam at 11.78 MPa, have an extraction at 4.12 MPa and exhaust at 1.47 MPa, as well as one CT (condensing turbine), which can operate admitting steam at either 1.47 MPa or 0.34 MPa, and two gas turbines. The generation capacities for the three types of turbines are respectively 45 MW, 45 MW and 38 MW. The exhaust of one of the gas turbines (GT G) is used to supply energy demands of CEMAP and the exhaust of the other gas turbine (GT F) is used alongside supplementary firing to generate steam in the HRSG. The electricity generated by these machines, in addition to the electricity that is acquired from the power company (electric grid), is used in the plant and is distributed to other companies in the industrial hub. The amount of electricity that must be generated depends on two important factors; contract limitations with the power company and, due to the instability of the power company's supply, the will to be more self-sufficient, despite the greater cost of producing the energy as opposed to buying it from the power company.

2.3. Water and steam cycle

The water used in the steam generators is the sum of re-circulated water from the petrochemical cycle and demineralized water that is pumped from a treatment facility nearby. While still in the treatment facility, the water is pre-heated using residual heat from the CT and is then sent to five deaerators. The water enters the deaerators (which protect the plant against corrosion) at 80°C.

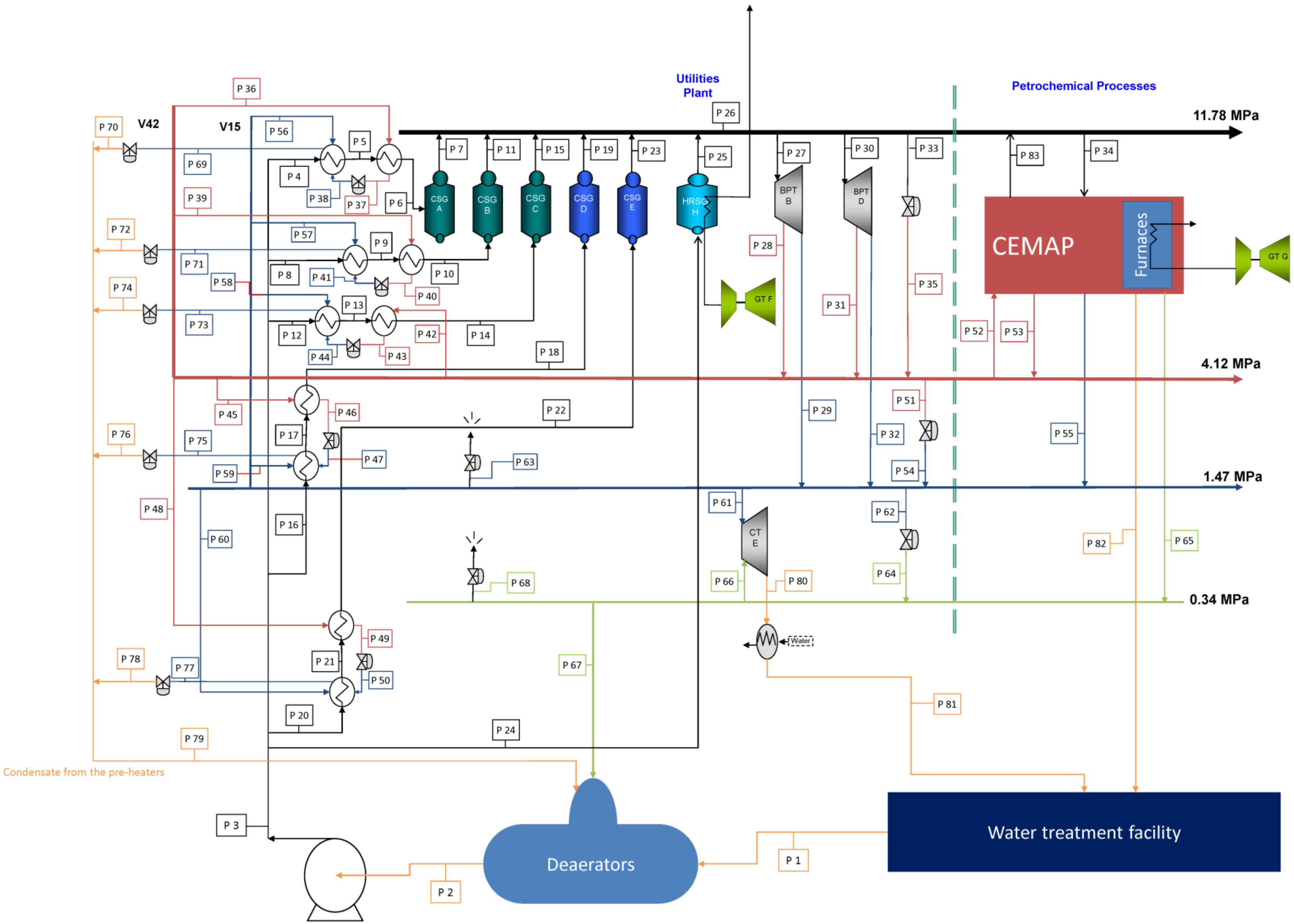

Fig. 1. General flowchart of the petrochemical utilities plant (equipment and structure).

The deaerated water is then sent to the boilers through six centrifugal pumps which elevate the water pressure to 11.78 MPa. However, before arriving at the boilers, the water flows through two pre-heaters where, in the first one, it exchanges heat with steam and saturated water at 1.47 MPa and in the second one, with steam at 4.12 MPa. The saturated water at 1.47 MPa used in the first pre-heater consists of the steam of the second pre-heater, that, after leaving this equipment, is expanded by a pressure reducing valve and goes to the first exchanger. The heat exchanger system will be more thoroughly explained in subtopic 3.2.6. In the boilers, the water turns into steam and it then joins the high pressure steam (11.76 MPa) produced by the furnaces and other components in CEMAP's processes. The high pressure steam (11.76 MPa) is then distributed to the process and to the back pressure turbines. As a result of the operation of these systems, steam at lower pressure levels is generated and this generated steam will subsequently be used in CEMAP, in the pre-heaters, in the CT and, lastly, will be sold to clients in the industrial hub. If there is excess of steam, it's exhausted by the valves.

3. Method

The method created and implemented to achieve the aforementioned operating map is based on developing a mathematical model (simulator) of the plant, capable of analysing the performance of the equipment and the cycle to indicate the thermal scheme that has the smallest energy consumption. The operating conditions of a specific equipment are determined by the flow rate, the pressure and temperature conditions, as well as possible adaptations due to design flexibility. The operating conditions of the plant are determined by the ensemble of its components.

3.1. Method description

The first step to develop the mathematical model is to create a graphic representation of the petrochemical plant, containing the utilities plant and CEMAP. The next step is to collect historically typical values for steam flow rate and energy flow between the utilities plant and CEMAP and external clients. By knowing the steam demands of external processes (CEMAP and clients) it is possible to evaluate the flow, internal demands and their inter-relations. Considering the main scope of the paper, *i.e.* creating an analysis that considers the impact of changes in operating conditions to equipment performance, equipment supplier performance curves (in graph and table form) were used when available. Due to the relatively small amount of previous studies in this field, a series of considerations and constraints are defined. These considerations may be divided in three types: general considerations, specific equipment considerations and mathematical model considerations.

General considerations:

1. The equipment that interact with the thermodynamic cycle, but are not physically present in the utilities plant, are grouped in one control-volume (CEMAP);
2. The fluctuation of pressure and temperature of the steam supplied by CEMAP are not considered. Typical values are adopted;
3. Heat and pressure loss in pipes are not considered a significant factor for this study.
4. The thermoelectric plant operates in a steady state.

The specific approach and considerations made about the equipment will be presented below. The software used to generate equations and to run the simulator is Microsoft Excel® using its data analysis tool and its "Solver" add-in tool. In addition to these tools, an auxiliary spread sheet with steam thermodynamic properties was also used.

3.2. Method applied to equipment

The following topic will present the approach adopted for the cycle and each of its components. The approach consists of considerations made, data sources and equation modelling.

3.2.1. Conventional steam generators

The mathematical evaluation of the CSGs adopted an indirect approach to model the losses due to operating conditions. This approach was based on real time measurements of the operating parameters, such as flow rate and conditions of the feedwater and exhaust gases. The results of this indirect method of calculating boiler efficiency, considering ASME PTC 4.1, were considered reliable. The steam generator's efficiency equation (1) has the following structure:

$$\eta_{CSG} = C_1 \dot{m}_s^{\,2} + C_2 \dot{m}_s + C_3 \tag{1}$$

This behaviour is considered for all conventional steam generators in the utilities plant with a reliability for the calculated equation of R^2=88%. Specific considerations made for conventional steam generators:

1. Changes to the type of fuel used in the generators were not considered;
2. In the scope of this study, it was considered that the deactivation of one or more steam generators is a viable option and can be implemented;
3. The output conditions of the steam do not vary.

3.2.2. Heat recovery steam generator

In the case of the heat recovery steam generator, the supplier, Confab, provided the values of natural gas consumption for the supplementary firing based on a series of values of electricity production by the gas turbine that operates associated to the generator, as well as steam flow rate values. The supplier information also included some scenarios of flow rate for specific operating conditions of the gas turbine with no supplementary firing in the generator. Based on the supplier's data books, applying the linear regression method, the flow rate of natural gas for supplementary firing, $\dot{m}_{NG}$ (2), was determined:

$$\dot{m}_{NG} = C_1 \cdot E_{GT}^{\,2} + C_2 \cdot E_{GT} + C_3 \cdot \dot{m}_s + C_4 \tag{2}$$

The reliability of the calculated equation is R^2=98%. Specific considerations made for heat recovery steam generator:

- The minimal electricity production in the gas turbine that allows the HRSG to operate is 60% of its capacity, according to the supplier.

3.2.3. Back pressure turbines

The supplier of the back pressure turbines provided performance curves for its turbines in its operation manual (ALSTOM, 2000). Considering that they are 2 stage turbines, two performance curves were identified. For the first stage, the supplier curve shows the relation between extraction steam temperature and input flow rate, while for the second stage the curve correlates exhaust steam temperature and exhaust flow rate for three different input flow rate values. To determine the equations of the temperatures of the extraction (3) and exhaust steam (4) a trend line was used in the first case and a linear regression in the second case. The equations are as follows:

$$T_{ext} = C_1 \cdot \dot{m}_s^{\,2} + C_2 \cdot \dot{m}_s + C_3 \tag{3}$$

$$T_{exh} = C_1 \cdot \dot{m}_s^{\,2} + C_2 \cdot \dot{m}_{exh} + C_3 \cdot \dot{m}_{exh} + C_4 \tag{4}$$

The reliability of the first calculated equation is R^2=100% and R^2=94% for the second equation. Considerations for back pressure turbines:

- The electric generator of these turbines are considered ideal, thus the production of electricity is equal to the axle's work

3.2.4. Condensing turbine

The information provided for the condensing turbine in the supplier's operation manual (AKZ, 1991) is in the form a table of control values, that presents six operating conditions (output pressure and flow rate) and their respective electricity production. These control values were analysed as the guaranteed performance by the supplier, thus allowing the inference of the turbines behaviour in any conditions. For future calculations the operating ratio, r, result of the normalization of the flow rate by the design maximum flow rate is used. The equation for this equipment, a result of the trend line method applied to the supplier's table determines the behaviour of the isentropic efficiency of the turbine (5):

$$\eta_{iso} = C_1 \cdot r^2 + C_2 \cdot r + C_3 \tag{5}$$

The table provided by the supplier utilizes an input pressure of approximately 1.47 MPa, which is one of the operating conditions of this turbine. The same behaviour was considered for operation at 0.34 MPa. Specific considerations made for the condensing turbine:

1. To determine the output steam's temperature and pressure, typical values for temperature of the cooling tower as well as ambient temperature were used:
    - Wet-bulb temperature (WBT) = 25°C;
    - Approach temperature for the cooling tower = 7°C;
2. The first law of thermodynamics was applied for the condenser. The overall heat transfer coefficient was calculated from the suppliers data and its variation was considered negligible (constant cooling water flow).

3.2.5. Gas turbine

The analysis of the gas turbines was based on energy consumption versus electricity production curves when compared to design capacity provided in the "Operation and Maintenance Guide" (ALSTOM, 1991). Applying the trend line method to the available data, the natural gas consumption versus electricity production (6) was defined:

$$\dot{m}_{NG} = C_1 \cdot E_{GT} + C_2 \tag{6}$$

This equation was used for both gas turbines and its reliability is R^2=100%. Specific considerations made for the gas turbines:

1. For the gas turbine installed in CEMAP, the turbine's energy consumption was analysed as the sum of the energy consumed in the turbine and the energy consumed by furnaces operating with its exhaust gases;
2. The ambient temperature was considered constant; therefore, there are no variations of performance or power due to environment conditions (it is quite true for the site in which the plant is installed).

3.2.6. Pre-heaters

Considering the variety and auxiliary nature of the pre-heaters, no performance curves were informed by the suppliers, thus, the analysis of this equipment was adopted from existing studies. Assuming that the thermal resistance of the water and steam sides are greater than the conductive resistance through the pre-heaters' tubes, the overall heat transfer coefficient (U), in off-design conditions (7), may be represented as according to Valdés and Rapún [9]:

$$U = U_{design} \cdot \left(\frac{\dot{m}_{fw}}{\dot{m}_{fw_{design}}} \right)^A \cdot \left(\frac{T}{T_{design}} \right)^B \tag{7}$$

Since the operating temperature has not change over the years, it is negligible parameter for this analysis. The coefficient, U, for off-design conditions, becomes a function of the feed water mass flow, m_{fw}, and an empirical constant A. The constant A was considered equal to 0.8 as suggested by Silva et al. [10] in a similar model. Figure 2 is a rough representation of the pre-heaters:

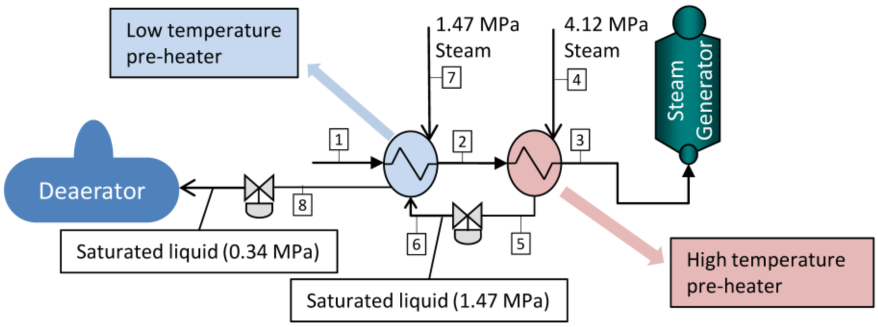

Fig. 2. Schematic representation of the pre-heaters operating scheme.

By calculating the overall heat transfer coefficient, based on the design conditions, it's possible to determine the mass flow of heating steam for both pre-heaters, which corresponds to the mass flow of points for 4 and 7. For both pre-heaters a system of three equations was solved iteratively. The equations correspond to the heat absorbed by the steam generators feed water (8), the heat supplied by the heating steam (9), and the heat transfer between the two currents inside the pre-heater (10).

$$Q = \dot{m}_{fw} \cdot c_p \cdot \Delta T \tag{8}$$

$$Q = \dot{m}_{hs} \cdot c_p \cdot \Delta T \tag{9}$$

$$Q = U \cdot A \cdot \Delta T_{LM} \tag{10}$$

In order to solve this system of equations, the values of temperature and pressure of the fluids in points 5, 6 and 8 of Fig. 3 are known and constant and are presented in Table 1. The values for points 4 and 7 vary independently and are calculated during the simulation.

Table 1. Temperature, pressure and enthalpy for specific points of the pre-heaters' configuration

Points	**Temperature (◦C)**	**Pressure(MPa)**	**Enthalpy (kJ/kg)**
5	207.4	4.12	886.72
6	197.4	1.47	886.72
8	154.5	0.34	652.27

Specific considerations for the pre-heaters:

1. The first law of thermodynamics was applied for both pre-heaters;
2. The variation of the output temperature of the high temperature pre-heater (T_5) is a function of both heating steams' mass flows (mass flows of points 4 and 7).

3.2.7. Deaerator

Specific considerations for the deaerators:

1. The five deaerators were considered as a single equipment;
2. The approach to this equipment is the direct application of the first law of thermodynamics and conservation of mass;
3. The output conditions of the deaerator are assured by the adjustment of the flow rate of low pressure steam (0.34 MPa) to the deaerator.

3.2.8. Pumping system

Specific considerations for the pumping system:

1. The six pumps that operate in parallel were considered as a single pump;
2. The calculation of the energy consumed by the pumps was based on a constant isentropic efficiency of 85%;

3. The impact of the variation of the pump efficiency was considered negligible. According to Moran and Shapiro [11], the work necessary to operate the pumps is usually smaller than 1% of the work provided by the turbine.
4. The state of operation of the pump allows it to vary its rotation to assure a constant output pressure, independent of the water flow rate.

3.2.9. Pressure reducing valves

Specific consideration for the valves:

- All the valves, reducing and relief, are considered isenthalpic.

3.3. Optimization approach

To optimize a specific parameter using Excel's "Solver" application, the three concepts on which it is based (objective, constraints and variables) must be understood:

- The **objective** is the parameter being optimized, be it by minimizing, maximizing or converging it to a specific value. In the case of this study the objective is to minimize the energy consumption of this utilities plant;
- The system's **constraints** are defined by the user and vary for every case. Some examples of constraints in this study are the equipment capacities, fixed production values and temperature limits. In this study 83 constraints were defined;
- The **variables** represent the path to achieve the objective without breaking any constraints. These variables are values that the application modifies and iterates until it finds the optimal solution. The variables represent the final product of the mathematical model. In this case 29 variables were pointed out and their final values will define the thermal scheme.

3.3.1. Nonlinear GRG algorithm

The nonlinear GRG (Generalized Reduced Gradient) method is used in mathematical models where the objective and constraints are smooth functions of the variables, *i.e.* there are no edges in their graphs. This method was chosen since it is considered highly effective in solving non-linear programming models as stated by Sacoman [12] and Graueer et al. [13]. This algorithm returns local optimal solutions. To lower the chance of finding local optimal solutions (as opposed to global optimal) one may use his/her knowledge of the problem to define initial values to the variables that are close to those expected as a solution. The software used (MS Excel) also allows the user to change the optimization settings, choosing tools like the automatic scale and the multistart method. The automatic scale analyses the initial values and performs iterations with alterations to these values according to the order of magnitude of the initial values for each variable. The multistart method performs a series of simulations with different initial values in order to broaden the spectrum of solution and maximize the chance of finding the optimal global solution.

4. Results

The results of the mathematical simulations will be presented and discussed in this topic in the form of tables containing values of flow rate, electricity production and energy consumption for the main variables and equipment analysed. The simulation was based on a few probable energy and steam demand scenarios that will be presented below.

4.1. Scenario 1: current situation

Scenario 1 was based on the company's current situation. The typical electricity demand for the petrochemical plant is 190 MW, of which it buys 110 MW from the power company and its utilities plant is responsible for 80 MW. Table 2 provides the typical values of supply and consumption of steam by CEMAP and the clients.

Table 2. Steam consumption and supply by external clients and CEMAP at different pressure levels

Pressure Levels (MPa)	**Consumption (t/h)**		**Supply (t/h)**
	External Clients	CEMAP	CEMAP
11.78	-	1036.53	630.13
4.12	107.87	654.93	593.88
1.47	73.93	415.3	673.46
0.34	-	-	120.0

4.2. Scenario 2: reliability

The second scenario was drawn up considering the unstable reality of the local electric grid. For this scenario the electricity and steam demands are the same as those of scenario 1, but the roles of the utilities plant and the power company are inverted, meaning that the plant is now responsible for the production of 110 MW. The greater production assures greater stability to the process and helps protect CEMAP from possible blackouts. The values in Table 2 are applicable to this scenario.

4.3. Scenario 3: maintenance shutdown

The third scenario considers the temporary shutdown of one of the plants of CEMAP. This shutdown has an effect on the steam supplied by CEMAP and on the general consumption of electricity. The shutdown considered was regarding one (the newest) of the two olefins plants that form CEMAP. The electricity demand of this olefins plant is 14.4 MW, thus, for this scenario, the production of electricity in the thermoelectric plant must be 65.6 MW. Table 3 presents the values for steam consumption and supply.

Table 3. Steam consumption and supply by external clients and CEMAP at different pressure levels

Pressure Levels (MPa)	**Consumption (t/h)**		**Supply (t/h)**
	External Clients	CEMAP	CEMAP
11.78	-	598.0	341.07
4.12	107.87	555.74	593.88
1.47	73.93	264.18	228.80
0.34	-	-	120.0

4.4. Results

Tables 4 to 6 present values of mass flow rate for the main studied equipment, values of electricity production and the energy consumption for the different scenarios.

Table 4. Values for mass flow rate of the main equipment

Equipment	**Scenario 1**	**Scenario 2**	**Scenario 3**
Conventional SG A (t/h)	0	400	400
Conventional SG B (t/h)	355	204	36
Conventional SG C (t/h)	381	0	0
Conventional SG D (t/h)	0	0	0
Conventional SG E (t/h)	0	0	0
HRSG H (t/h)	0	100	100
Turbine B extraction(t/h)	44.7	240	147.7
Turbine B exhaust (t/h)	5.1	0	118.3
Turbine D extraction(t/h)	240	0	0
Turbine D exhaust (t/h)	0	0	0
Condensing Turbine (P-t/h)	14.7 (bar)-174.7	14.7 (bar)-171.8	3.4 (bar)-54.5
Reducing Valve 11.78→4.12 (t/h)	40.1	59.27	13.1

Reducing Valve 4.12 →1.47 (t/h)	0	0	0
Reducing Valve 1.47→0.34 (t/h)	0	0	0
Relief Valve 1.47 (t/h)	0	0	0
Relief Valve 0.34 (t/h)	57.62	57.59	17.0

Table 5. Values for electricity production in the turbines

Equipment	Scenario 1	Scenario 2	Scenario 3
Turbine B: 1st Stage (MW)	1.25	13.7	8.9
Turbine B: 2nd Stage (MW)	0.17	0	12.8
Turbine D: 1st Stage (MW)	13.73	0	0
Turbine D: 2nd Stage (MW)	0	0	0
Condensing Turbine (MW)	27.04	26.2	5.9
Gas turbine F (MW)	0	38.0	38.0
Gas turbine G (MW)	37.8	32.1	0
TOTAL	**80**	**110**	**65.6**

Table 6. Values for energy consumption (in MW)

Equipment	Scenario 1	Scenario 2	Scenario 3
Conventional SG A (MW)	0	263	265
Conventional SG B (MW)	231	135	22
Conventional SG C (MW)	248	0	0
Conventional SG D (MW)	0	0	0
Conventional SG E (MW)	0	0	0
HRSG H (MW)	0	22	22
Gas turbine F (MW)	0	121	121
Gas turbine G (MW)	120	106	0
CEMAP Furnaces (MW)	0	8	0
TOTAL (MW)	**599**	**657**	**430**

4.4.1. Analyzing scenario 1

As to be expected, the conventional generators operate close to design conditions, when possible, optimizing the performance of the generator/pre-heater ensemble. For this scenario, however, the operation of the HRSG is not advised, possibly to prioritize the operation of the gas turbine installed in CEMAP, since the extra energy consumption by the furnaces is greater than the savings associated to the use of the HRSG as opposed to CSGs. For the back pressure turbines, it's advised to operate only the 1st stage of one of them and partial flow for 1st and 2nd stage of the other. The second turbine helps to balance the steam demand for the different pressure levels, not contributing much to electricity production. The condensing turbine gives flexibility to the cycle. The choice of operating this turbine with 1.47 MPa steam is due to the greater surplus flow rate and greater electricity production potential. The gas turbines operate to minimize the energy consumption in the furnaces. The pressure reducing valve (11.78→4.12) shows a non-intuitive result; operating at 40.1 t/h. This operating configuration may be explained by the influence this isenthalpic valve has on the feed water temperature (since it has direct influence in the temperature of the steam used in the second pre-heater). The negative impact of not using this steam (transferring greater demand to the gas turbines) is, thus, smaller than the positive impact the higher feed water temperature has on the energy consumption of the steam generator. The 0.34 MPa relief valve is actuate because there are not many options for consuming low pressure steam in the cycle. The only alternative to the valve is the condensing turbine, which is operating with a higher pressure.

4.4.2. Analyzing scenario 2

The greater electricity demand, not altering the steam demand, in relation to scenario 1, allows the cycle to operate both gas turbines close to their design conditions. Due to the combined cycle nature of these turbines, they represent the most efficient option for producing electricity, allowing the back pressure turbines to focus on balancing the flow rates of the different pressure levels. The operation of the HRSG, which does not use pre-heating, decreases the need for 4.12 MPa and 1.47 MPa steam and dismisses the need for operating a second back pressure turbine. The condensing turbines, pressure reducing valves and relief valves operate similarly to those of scenario 1.

4.4.3. Analyzing scenario 3

This topic will analyse the results that are conceptually different from those of scenario 1. Considering that the furnaces of the olefins plant are not operating in this scenario, it was possible to fully take advantage of the HRSG. The high operating rate of the gas turbine allows the HRSG to produce 100 t/h of steam with one fourth of the equivalent energy consumption of conventional turbines for this flow rate (at maximum efficiency). In this scenario there is a reduction in consumption of 11.78 and 4.12 MPa steam by CEMAP and not much change in its supply of steam. This fact allows the steam generators to reduce their production in 200t/h. Another important consequence of this change in steam demand is the decrease in steam extracted in the back pressure turbines, allowing a more effective electricity production by this equipment. The shutdown of the olefins plant caused a decrease in lower pressure (1.47) steam supply and the condensing turbine executes its role of providing flexibility to the cycle by operating with 0.34 MPa steam, since this pressure level has become the one with most unused surplus. The pressure reducing and relief valves operate similarly to those of scenario 1.

5. Conclusion

A mathematical model was developed for a specific utilities plant in order to obtain the optimal thermal scheme for a given off-design condition. To accomplish this, a series of considerations were made. The model was then tested for three realistic off-design operating scenarios, and the calculated exergy efficiency for scenarios 1, 2 and 3 were, respectively, 43, 46 and 47%. These values represent, in some cases, a substantial gain when compared to the typical operating value of 42%. The main difficulties for the implementation of this model were the complexity of the utilities plant, the large number of variables and of considerations that had to be made due to the relatively unexplored nature of this analysis. The results were useful to identify the components that are leading the plant to operate with performance below optimal and shed light on non-conventional thermal schemes that, when considering the cycle as a whole, can lead to greater energy savings.

6. Nomenclature

Symbols:

A area, (m^2)

c specific heat, J/(kg K)

C calculated constant

E electricity, W

$\dot{m}$ mass flow rate, kg/s

r operating ratio

T temperature, °C

U overall heat transfer coefficient, W/(m^2 K)

Greek Symbols:

η efficiency

Δ variation

Subscripts and superscripts:

A, B calculated constants

CSG conventional steam generator

exh exhaust

ext extraction

fw feed water

GT gas turbine

hs heating steam

LM log mean

NG natural gas

p constant pressure

s steam

7. References

[1] Aguilar O, Smith R, Perry S, Kim J. Optimising the design and operation of industrial utility plants subject to variable demands and prices. Computer Aided Chemical Engineering 2005. 20:907-912.

[2] Toffolo A, Lazzaretto A, Energy system diagnosis by a fuzzy Expert system with genetically evolved rules. Energy; 2007:1-8.

[3] Zaleta A, Munõz G, Rangel V, Valero A. A reconciliation Approach based on a module simulator. an approach to the diagnosis of energy system malfunctions. Int. J. Thermodynamics 2004;7:51-60.

[4] Zaleta A, Royo J, Rangel V, Reyes E. Thermo-characterization of power systems components: a tool to diagnose their malfunctions. Energy 2004;29:361-377.

[5] Luo X, Zhang B, Chen Y, Mo S. Modeling and optimization of a utility system containing multiple extractions steam turbines. Energy 2011; 36:3501-3512.

[6] El-Sayed YM. Revealing the cost-efficiency trends of the design concepts of energy-intensive systems. Energy Convers Manage 1999;40:1599–1615.

[7] Lazzaretto A, Carraretto C. Optimum production plans for thermal power plants in the deregulated electricity market. Energy 2006;31:1567–1585.

[8] Torres. E. A. “Avaliação Exergética e Termoeconômica de um Sistema de Cogeração de um Pólo Petroquímico”. FEM/UNICAMP. Tese (Doutorado). Campinas/SP. 1999

[9] Valdés M, Rapún J. L. Optimization of heat recovery steam generator for combined cycle gas turbine power plants. Applied Thermal Engineering 2001; 21:1149e59.

[10] Silva. J. A. M., Venturini. O. J., Lora. E. E. S., Pinho. A. F., Santos. Thermodynamic information system for diagnosis and prognosis of power plant operation condition. Energy 2011; 36: 4072-4079.

[11] MORAN, M. J., SHAPIRO. H. N. “Princípios de Termodinâmica para Engenharia”. Livros Técnicos e Científicos Editora S.A., 2002.

[12] Sacoman M. Otimização de projetos utilizando GRG, solver e Excel. COBENGE XL Congresso Brasileiro de Educação em Engenharia. Belém/PA. 2012.

[13] Graueer M. Optimization of a complex plant by a GRG algorithm. Computers & Chemical Engineering 1979; 3: 597-602.