OFF DESIGN MODELING AND SIMULATION OF THE HRSG COMPONENTS IN COMBINED CYCLE POWER PLANTS

Marcelo Assato, assato@puc-rio.br
Rafael S. Barbosa, rafaelbarbosa@puc-rio.br
Luís Fernando Figueira da Silva, luisfer@esp.puc-rio.br
Sandro Barros Ferreira, sandro@puc-rio.br
Pontifícia Universidade Católica do Rio de Janeiro – PUC-Rio
Rua Marquês de São Vicente, 225, Gávea, Rio de Janeiro, RJ, Brasil, 22453-900

Abstract. *This paper presents the modeling and the results of the performance simulation of a heat recovery steam generator (HRSG) with two pressure levels (high pressure, HP and low pressure, LP) under both design-point and off-design or part-load operating conditions. In this last operation mode, the key components which must be accurately modeled are the heat exchangers in HRSG. Besides these, other components such as pumps, valves, deaerator and desuperheater are modeled and their behavior is presented. For the off-design calculations the pressure drops and heat transfer in superheaters, evaporators and economizers are modeled using the design-point pressure on the gas and water/steam sides of the heat exchanger surfaces and the overall heat transfer coefficients. In the equations, some exponents are fitted to the power plant operation data in the part-load regime. The off-design performance prediction for the dual-pressure HRSG is compared with data supplied by boiler manufacturer.*

Keywords*: off-design, HRSG, power plants, heat exchangers, performance simulation*

1. INTRODUCTION

Nowadays combined cycle (CC) power plants represent a good choice to produce energy, because of their good economic performance: high efficiency in utilizing different energy sources, low capital cost, operation and maintenance, short period from beginning of construction and operation, good availability and flexibility in the choice of fuels, low emissions of pollutant to the atmosphere, among other benefits. Ordinary combined cycle power plants couple a Brayton based cycle, gas turbine engines, with a Rankine based cycle, the steam cycle. The hot exhaust gases, available at the exit of the gas turbine are used to produce high-pressure steam for the steam cycle. The device where the steam production takes place is the heat recovery steam generator (HRSG).

High efficiency in CC plants (up to 58%) has been reached mainly for two reasons: (i) improvements in the gas turbine technology, i.e. higher turbine inlet temperature, and, (ii) improvement in the HRSG design with multiple pressure levels.

It is a known fact that power plants will hardly operate at their design point conditions. Off-design operations occur during a considerable part of the lifetime, if not all, due to the natural changes in ambient conditions and dispatch schedule. Also, the CC power plants work at off-design mode due to abnormal changes, which may be related to component/device degradation and failure in the plant operation. Thus, such plants usually undergo frequent load changes. Therefore, it is very important to account for the performance not only at design, but also at part load conditions. Efforts have been made to adopt optimal part load operating modes. An example of one such effort has been to improve heat recovery performance, and thus to enhance performance of the entire combined cycle. Modern HRSGs are designed to have multiple-pressure levels in order to increase plant performance. However, this introduces operational complexity involving a careful monitoring of thermodynamic parameters of the HRSG components. In this context a performance prediction tool is very useful to foresee the off-design behavior of existing plants, thus preventing inefficient operation modes. This tool may also be used to evaluate retrofit opportunities.

Several research efforts have been conducted to construct mathematical model to simulate and analyze the off-design operation of the HRSG. Erbes and Eustis (1986) and Erbes and Phillips (1987) developed a design and off-design model to analyze the performance of power plants. Several equipments, such as gas turbine, HRSG, steam turbine are described and simulated and the corresponding results are compared to measured data in thermal plants. Dechamps *et al.* (1994) describe a methodology used to evaluate part load performance of combined cycle plants. The method was used to simulate one or multiple pressure arrangements, incorporating supplementary firing and possibly reheat. More recently Sanaye *et al.*, (2006) presented a mathematical model, which involves theory and equations of mass/energy balance and heat transfer coefficients, for predicting the off-design and transient operation of HRSG. This model includes an arbitrary number of pressure levels, usually up to three, and any number of components, i.e., superheater, evaporator, economizer, desuperheater and duct burner.

In the present study a mathematical model that describes the off-design operational behavior of a dual-pressure HRSG is introduced. A computational program based on this model was developed and the main components of a typical HRSG were simulated. The performance results during the part-load regime and steady state conditions were compared with the original equipment manufacturer (OEM) data.

2. HRSG DESCRIPTION

The HRSG analyzed in this study supplies two pressure levels, i.e., high pressure (HP) and low pressure (LP), to a steam turbine. A scheme of the HRSG is shown in Fig. 1. Gas turbine exhaust gas (ENTG 1) flows through the HRSG, thus providing thermal energy which is used for steam production. The HRSG contains four superheaters (three HP SUPQ and one LP SUPQ), five economizers (three HP ECON, one LP ECON and one as a preheater of the feedwater), two evaporators which are linked with the drums, one desuperheater (DSPQ), one evaporator-deaerator (EVDE). Some auxiliary equipments such as: pumps, valves, splitters are also accounted for. In the present model the pegging steam line is disabled, for that is the original project of this HRSG. The feedwater supplied from the condenser by the boiler feed pump (not shown in this scheme) is heated at the preheater (ECON 21) before it enters at the deaerator. The deaerated water flow is then split among high and low pressure circuits.

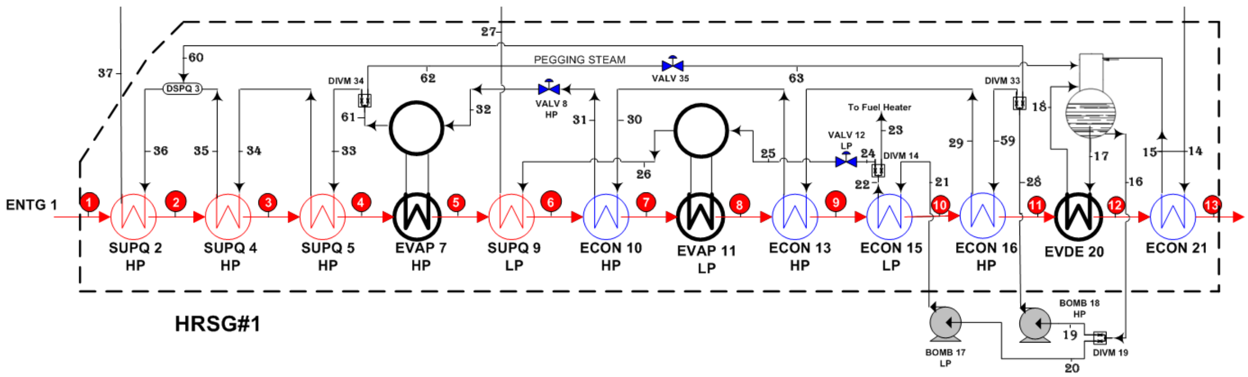


Figure 1. Scheme of the HRSG with dual-pressure configuration.

3. OFF DESIGN MODELING OF HRSG COMPONENTS

In order to analyze the performance of power plant equipments over typical ranges of operation, it is necessary to use mathematical models which can predict performance under both design-point and off-design, or part-load, operating conditions.

Individual models have been proposed and discussed in the literature for typical individual HRSG heat exchanger components. However, these are generally too complex and thus inadequate to be used in plant modeling. Therefore, more useful for thermal equipment simulation are sets of relatively simple correlations which capture the key off-design performance parametric variations in heat exchanger pressure drops and heat transfer coefficients under varying plant operating conditions. These correlations are expressed as ratios between the design-point values and the off-design values. Therefore, the present modeling involves, firstly, the design-point determination, and, then the off-design performance.

The thermodynamic properties of water at liquid and steam phases are determined following the formulation proposed by International Association for the Properties of Water and Steam, IAPWS – IF97 (Wagner et al. 2000), whereas, for the hot gas path, the thermodynamic properties are calculated using the formulations described by McBride et al. (1993).

3.1. Heat Exchanger Pressure Drops

All the heat exchangers are assumed to be composed of tube bundles with pressurized water/steam flowing on the inside and surrounded by the hot combustion products.

The off-design modeling determines the pressure drops on the gas and water/steam sides of heat exchanger surfaces.

(a) Water/steam side: it is assumed that the flow is fully turbulent. The friction factor, *f*, is defined as, (Minoner, 1979):

$$f = -\frac{\Delta P_v / \rho g}{(l/D)\left(\overline{V}^2 / 2g\right)}, \tag{1}$$

where,

ΔP_v is the pressure drop;

l and D represent the tube length and diameter, respectively;

ρ is the specific mass of fluid;
g is the gravity acceleration;
$\overline{V}$ is the average velocity.

Ratio between the off-design and the design-point pressure drops is determined using Eq. (1) and $\overline{V} = \dot{m}/\rho A$. After some algebraic manipulation the following expression is obtained:

$$\frac{\Delta P_v}{\Delta P_{DP}} = \left(\frac{f}{f_{DP}}\right)\left(\frac{\dot{m}^2/\rho}{(\dot{m}^2/\rho)_{DP}}\right), \tag{2}$$

where $\dot{m}$ represents the mass flow in the heat exchanger tube.
For simplicity, an empirical form of the Karman-Nikuradse equation is used for the friction factor, which is explicit in f (Kays and Crawford, 1980):

$$f = 0.046\,\mathrm{Re}^{-0.2}. \tag{3}$$

Substituting Eq. (3) into Eq. (2) and canceling out common terms, one may obtain:

$$\Delta P_v = \Delta P_{DP}\left(\frac{\dot{m}}{\dot{m}_{DP}}\right)^x\left(\frac{T}{T_{DP}}\right)^y\left(\frac{P}{P_{DP}}\right)^z\left(\frac{v}{v_{DP}}\right)^a, \tag{4}$$

where, T, P and v represent the temperature, pressure and specific volume of the fluid, respectively. The design point values, designated by the index DP, are known values from the design point operation modeling.
The exponents x, y, z and a in Eq. (4) are adjusted by comparisons with the manufacturer data. The HRSG manufacturer supplies the temperature, pressure, and mass flow values at the inlet and outlet of the heat exchangers for three part-load operation conditions, 75%, 50% and 40%.

(b) Hot gas side: for the prediction of the pressure drop on the gas side of the heat exchanger, the following formulation (Holman, 1976) is employed:

$$\Delta P_g = \frac{2 f V_{\max}^2 N}{\rho}\left(\frac{\mu_w}{\mu_b}\right)^{0.14}, \tag{5}$$

where, $V_{\max}$ represents the velocity at minimum flow area;
N is the number of rows;
μ_w and μ_b represent the viscosity at the wall and the bulk flow viscosity, respectively.
Again, the ratio between the design and off-design pressure drops may be expressed as:

$$\frac{\Delta P_g}{\Delta P_{DP}} = \left(\frac{f}{f_{DP}}\right)\left(\frac{\dot{m}^2/\rho}{(\dot{m}^2/\rho)_{DP}}\right). \tag{6}$$

Note that it has been assumed that the viscosity ratio variation is negligible when compared to those of the mass flow and friction factor.
The friction factor across tube banks can be expressed as (Holman, 1976):

$$f = \lambda\,\mathrm{Re}_{\max}^{-0.16}, \tag{7}$$

where λ is a function of geometry.
From Eqs (6) and (7), and definition of Reynolds number, using the perfect gas equation of state the state equation and, upon canceling the constant terms, is obtained the expression:

$$\Delta P_g = \Delta P_{DP} \left(\frac{\dot{m}}{\dot{m}_{DP}} \right)^{\alpha} \left(\frac{T}{T_{DP}} \right)^{\beta} \left(\frac{P}{P_{DP}} \right)^{\gamma}, \tag{8}$$

The exponents, α, β and γ are to be adjusted using the manufacturer data for the part-load operation (75%, 50% and 40%).

3.2. Overall Heat Transfer Coefficient

The modeling of heat exchangers considers that the heat transfer is expressed by:

$$Q = U\,A\,\Delta T_{\log}, \tag{9}$$

Where, U is the overall heat transfer coefficient, A represents heat exchange surface area, and $\Delta T_{\log}$ is the Logarithmic Mean Temperature Difference (LMTD), given by:

$$\Delta T_{\log} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln(\Delta T_{in} / \Delta T_{out})}, \tag{10}$$

where $\Delta T_{in} = (T_1 - T_3)$ and $\Delta T_{out} = (T_2 - T_4)$ represent the temperatures difference at inlet and output of heat exchanger tubes. Thus, T_1 and T_2 represent the inlet and output hot gas temperatures, respectively, and T_3 and T_4 are the inlet and output water/steam temperatures, respectively.

It is assumed that the flow over the tubes is turbulent; therefore the heat transfer coefficient can be expressed by:

$$Nu = C\,\mathrm{Re}^n\,\mathrm{Pr}^m, \tag{11}$$

where, *Re* and *Pr* represent the Reynolds number and Prandtl number, respectively, C, n and m are constants.

The ratio between off-design point and design values is expressed as:

$$\frac{U}{U_{DP}} = \frac{Nu}{Nu_{DP}} = \left(\frac{\mathrm{Re}}{\mathrm{Re}_{DP}} \right)^n \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_{DP}} \right)^m. \tag{12}$$

Applying the definition of Reynolds number, Prandtl number and canceling the constant terms, the correlation used for the HRSG heat exchanger becomes:

$$U = U_{DP} \left(\frac{\dot{m}}{\dot{m}_{DP}} \right)^n \left(\frac{T}{T_{DP}} \right)^k \left(\frac{P}{P_{DP}} \right)^p. \tag{13}$$

The usual value for n, 0.8 is employed, which is valid over a Reynolds number range of 4×10^4 to 4×10^5 (Holman, 1976). The other exponent, k and p, are adjusted using the manufacturer data for the part-load operation (75%, 50% and 40%).

3.3. Computational methodology

The computational program was developed in FORTRAN language. It has several calculation routines defined as modules which represent the HRSG components. The modules are solved sequentially with a small number of initial guesses. The exponents of pressure drops and overall heat transfer coefficient are adjusted to the manufacturer data, and the HRSG components parameters are iteratively calculated until a convergence criterion is reached.

Next, it is presented the HRSG simulation results under design and partial load regimes of a typical combined cycle power plant. The calculation program is validated by comparison with manufacturer data for the stations shown in Figure 1.

4. RESULTS OF HRSG OFF-DESIGN OPERATIONS

Figure 2 shows the GT exhaust mass flow, that represents the HRSG inlet as a function of the gas turbine load percentage, for a complete operational range at base load condition (T=32 ^{o}C). It is observed two different operation modes of the gas turbine (GT) exit. For high power, the GT is controlled by the IGV (Inlet Guide Vane) angle, whereas for lower loads, it employs the control by fuel mass flow to the combustion chamber. Inlet guide vane control is interesting because the exhaust temperature is kept constant during load variation for operating regimes at the vicinity of the base load.

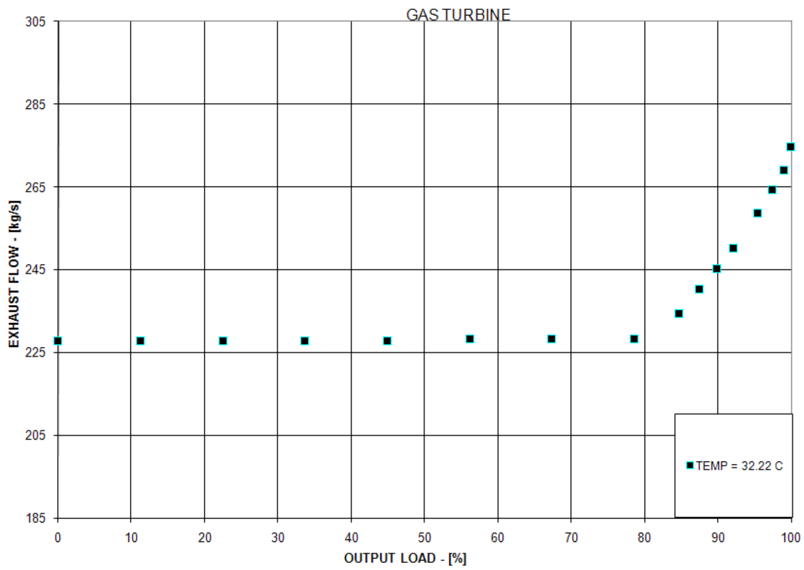


Figure 2. GT exhaust mass flow variations in function of GT load at base load condition.

4.1. Overall HRSG Off Design Performance

Heat recovery steam generator operation simulations involve the comparisons of thermodynamic properties: temperature, pressure and mass flow, supplied by OEM for loads of 100%, 75%, 50% and 40%, i.e., corresponding to the gas turbine known performance for those loads.

Table 1 and Table 2 show the obtained deviations between the design data supplied by manufacturer, defined as "plant" and those predicted by computational program for four investigated load conditions.

$$deviation\ [\%] = \frac{\left(\phi_{plant} - \phi_{prog}\right)}{\phi_{plant}} 100\ . \tag{14}$$

It is possible to observe that the deviations of pressure drops in heat exchangers predicted by the present model are lower than 0.5%. Both the gas and water/steam sides exhibit small deviations that, for practical effects may be considered insignificant. Concerning now the predicted mass flow values of the HP and LP evaporators, deviations smaller than ± 2.44% are observed with respect to OEM data. These deviations can have several origins, for example, the calculation of overall heat transfer coefficient, the model based on logarithmic mean temperature difference, heat losses to the environment, among others. Temperature deviations also are small, because they are related to the energy balance, which involves the water/steam mass flow and the steam pressure.

Table 1. Results and comparison for the stations shown in Figure 1: operation conditions of 100% load and 75% load.

	100 % LOAD					
	Plant			Deviation* (%)		
Point	P [kPa]	T [°C]	Flow [kg/s]	P [kPa]	T [°C]	Flow [kg/s]
1	3,09	548,30	278,10	0,00	0,00	0,00
2	2,97	534,90	278,10	0,00	0,05	0,00
3	2,84	511,40	278,10	0,00	0,03	0,00
4	2,73	475,70	278,10	0,00	0,02	0,00
5	2,08	321,60	278,10	0,00	-0,58	0,00
6	2,05	317,50	278,10	0,00	-0,60	0,00
7	1,71	278,50	278,10	0,00	-0,68	0,00
8	1,39	238,10	278,10	0,00	-0,76	0,00
9	1,09	220,60	278,10	0,00	-0,84	0,00
10	1,02	211,80	278,10	0,00	-0,88	0,00
11	0,75	175,00	278,10	0,00	-1,74	0,00
12	0,61	148,50	278,10	0,00	-2,10	0,00
13	0,55	129,70	278,10	0,00	-2,42	0,00
14	230,00	47,10	41,79	0,00	0,02	-0,41
15	220,00	79,10	41,79	0,00	0,00	-0,41
16	220,00	123,90	41,79	0,00	0,00	-0,41
21	3510,00	125,20	6,63	0,00	-0,14	0,00
24	3510,00	217,40	6,63	0,26	0,00	0,00
25	2660,00	217,50	6,63	0,26	0,02	0,00
26	2660,00	227,20	6,63	0,26	0,03	0,00
27	2570,00	294,50	6,63	0,26	0,00	0,00
28	11540,00	123,90	35,16	0,00	-2,03	0,00
29	11450,00	197,40	35,16	0,00	0,00	0,00
30	11360,00	230,90	35,16	0,00	0,00	0,00
31	11280,00	299,60	35,16	0,00	0,00	0,00
32	9880,00	310,10	35,16	0,00	0,00	0,00
33	9850,00	309,80	35,16	0,00	-0,03	0,00
34	9790,00	385,50	35,16	0,00	0,00	0,00
35	9730,00	460,10	35,16	0,00	0,00	0,00
36	9630,00	454,60	35,16	0,00	-0,01	0,00
37	9390,00	502,00	35,16	0,00	0,00	0,00

	75 % LOAD					
	Plant			Deviation* (%)		
Point	P [kPa]	T [°C]	Flow [kg/s]	P [kPa]	T [°C]	Flow [kg/s]
1	2,10	547,20	230,10	0,00	0,00	0,00
2	2,01	532,80	230,10	0,00	0,00	0,00
3	1,93	509,70	230,10	0,00	-0,02	0,00
4	1,85	471,50	230,10	0,00	-0,10	0,00
5	1,42	307,40	230,10	0,00	-0,01	0,00
6	1,39	304,10	230,10	0,00	0,00	0,00
7	1,17	269,30	230,10	0,00	-0,18	0,00
8	0,95	234,70	230,10	0,11	-0,03	0,00
9	0,74	218,90	230,10	0,00	-0,19	0,00
10	0,70	211,40	230,10	0,00	-0,22	0,00
11	0,51	174,00	230,10	0,20	-0,68	0,00
12	0,42	147,30	230,10	0,24	-0,78	0,00
13	0,38	126,70	230,10	0,26	-1,20	0,00
14	240,00	44,80	33,93	0,00	-0,87	-0,57
15	240,00	79,90	33,93	0,00	-0,43	-0,57
16	240,00	125,40	33,93	0,00	-0,44	-0,57
21	3860,00	128,80	4,68	0,00	1,11	-2,44
24	3860,00	217,80	4,68	0,47	0,57	-2,44
25	2630,00	217,90	4,68	0,00	0,58	-2,44
26	2630,00	226,60	4,68	0,00	-0,03	-2,44
27	2550,00	289,10	4,68	-0,02	0,11	-2,44
28	12350,00	125,40	29,25	0,00	-2,35	-0,27
29	12280,00	199,20	29,25	0,00	-0,80	-0,27
30	12210,00	229,20	29,25	0,01	-0,42	-0,27
31	12150,00	290,90	29,25	0,01	-0,04	-0,27
32	8400,00	298,40	29,25	0,00	-0,01	-0,27
33	8360,00	298,10	29,25	0,00	0,00	-0,27
34	8310,00	388,40	29,25	0,00	0,09	-0,27
35	8260,00	466,50	29,25	0,00	0,12	-0,27
36	8160,00	450,20	29,25	0,00	0,07	-0,27
37	7930,00	502,00	29,25	0,00	0,10	-0,27

Table 2. Results and comparison for the stations shown in Figure 1: operation conditions of 50% load and 40% load.

	50 % LOAD					
	Plant			Deviation* (%)		
Point	P [kPa]	T [°C]	Flow [kg/s]	P [kPa]	T [°C]	Flow [kg/s]
1	1,97	447,20	227,70	0,00	0,00	0,00
2	1,90	443,10	227,70	0,00	0,04	0,00
3	1,82	432,30	227,70	0,00	0,00	0,00
4	1,75	408,20	227,70	0,00	-0,12	0,00
5	1,36	279,90	227,70	0,00	0,01	0,00
6	1,33	278,20	227,70	0,00	0,00	0,00
7	1,12	263,10	227,70	0,00	0,02	0,00
8	0,91	237,40	227,70	0,00	0,00	0,00
9	0,71	233,00	227,70	0,00	-0,03	0,00
10	0,66	228,60	227,70	0,00	-0,07	0,00
11	0,46	206,00	227,70	0,00	-0,32	0,00
12	0,36	182,20	227,70	0,00	-0,40	0,00
13	0,33	156,20	227,70	0,00	-0,57	0,00
14	690,00	44,80	23,95	0,00	1,79	-0,21
15	680,00	107,40	23,95	-0,12	2,23	-0,21
16	680,00	163,90	23,95	-0,12	2,21	-0,21
21	4460,00	165,10	3,52	0,00	2,07	-0,40
24	4450,00	232,80	3,52	0,00	1,46	-0,40
25	2870,00	231,40	3,52	0,00	0,85	-0,40
26	2870,00	231,40	3,52	0,00	0,00	-0,40
27	2790,00	270,80	3,52	-0,01	0,44	-0,40
28	13120,00	163,90	20,44	0,00	0,60	-0,12
29	13070,00	225,10	20,44	0,00	0,57	-0,12
30	13020,00	236,70	20,44	0,00	0,56	-0,12
31	12970,00	275,50	20,44	0,00	0,53	-0,12
32	5700,00	272,30	20,44	0,00	0,01	-0,12
33	5670,00	271,90	20,44	0,00	-0,01	-0,12
34	5630,00	362,80	20,44	0,00	-0,02	-0,12
35	5590,00	415,40	20,44	0,00	-0,06	-0,12
36	5490,00	414,70	20,44	0,00	-0,05	-0,12
37	5280,00	435,70	20,44	0,00	-0,08	-0,12

	40 % LOAD					
	Plant			Deviation* (%)		
Point	P [kPa]	T [°C]	Flow [kg/s]	P [kPa]	T [°C]	Flow [kg/s]
1	1,93	410,00	227,10	0,00	0,00	0,00
2	1,86	407,70	227,10	0,00	-0,01	0,00
3	1,79	400,80	227,10	0,00	-0,14	0,00
4	1,72	382,80	227,10	0,00	-0,46	0,00
5	1,34	276,40	227,10	0,15	-0,02	0,00
6	1,32	274,90	227,10	0,15	-0,07	0,00
7	1,11	264,10	227,10	0,09	-0,16	0,00
8	0,89	239,80	227,10	0,11	-0,01	0,00
9	0,69	237,70	227,10	0,14	-0,11	0,00
10	0,64	234,30	227,10	0,16	-0,26	0,00
11	0,45	218,40	227,10	0,22	-0,76	0,00
12	0,35	197,20	227,10	0,29	-1,36	0,00
13	0,31	168,80	227,10	0,33	-2,01	0,00
14	1040,00	40,10	20,01	0,00	-0,37	1,84
15	1030,00	121,70	20,01	-0,04	-1,04	1,84
16	1030,00	181,20	20,01	-0,04	-0,75	1,84
21	4870,00	182,10	3,35	0,00	-1,00	1,81
24	4870,00	237,60	3,35	0,01	1,16	1,81
25	3020,00	234,20	3,35	0,00	-0,29	1,81
26	3020,00	234,20	3,35	0,00	-0,01	1,81
27	2950,00	268,60	3,35	-0,02	0,77	1,81
28	13360,00	181,20	16,66	0,00	-2,31	1,85
29	13320,00	232,90	16,66	-0,01	-1,16	1,85
30	13280,00	239,60	16,66	-0,01	-0,90	1,85
31	13240,00	273,90	16,66	-0,02	-0,46	1,85
32	5490,00	269,80	16,66	0,00	-0,02	1,85
33	5460,00	269,40	16,66	0,00	-0,04	1,85
34	5430,00	351,10	16,66	-0,01	0,60	1,85
35	5390,00	390,90	16,66	-0,03	0,99	1,85
36	5290,00	390,10	16,66	-0,04	0,99	1,85
37	5100,00	403,90	16,66	-0,09	1,20	1,85

4.2. LP and HP Evaporator Off-Design Performance

Figure 3 and Figure 4 show the (a) saturation pressure and (b) pinch point values behavior as a function of several off-design regimes, which are imposed by the gas turbine exhaust parameters for the EVAP 7 (HP) and EVAP 11 (LP), respectively. Note that these values are also dependent of the different operation strategies that are adopted for the steam turbine. For partial loads between 100 and 75%, the sliding pressure control technique is used. In this operation mode the steam mass flow and pressure are controlled by HRSG pump which is interesting because steam temperature remains approximately constant in off-design conditions. For low partial loads the mode of control known as “arc throttling” is employed, where steam turbine valves act in the steam mass flow and pressure control. As a consequence, it can be observed an abrupt variation of the saturation pressure and pinch point for the values smaller than 75% of load.

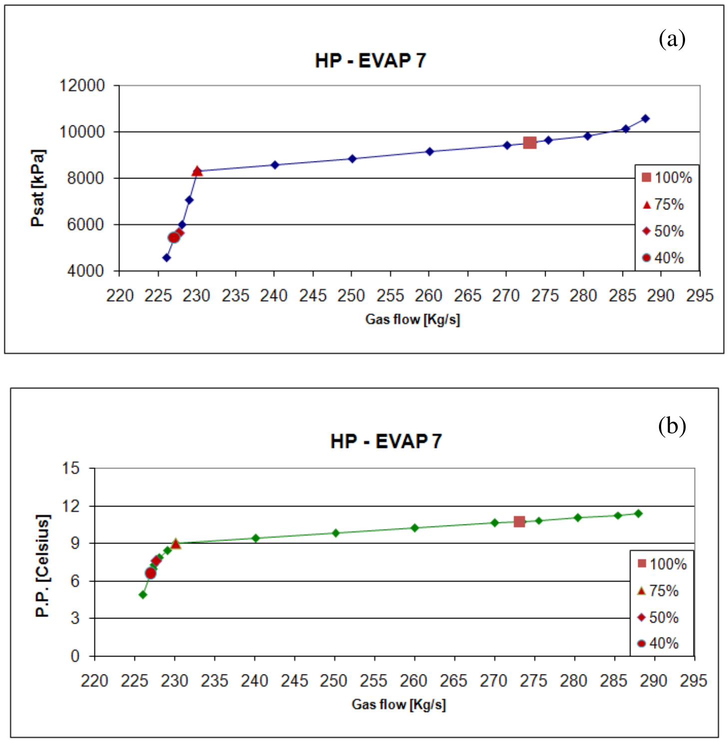
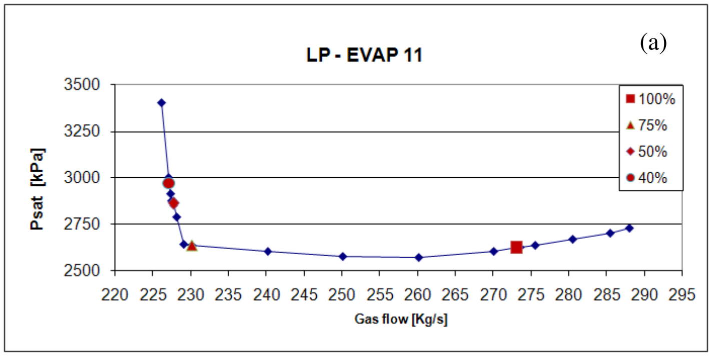


Figure 3. Values of saturation pressure (a) and pinch point (b) for the HP evaporator in several part-load conditions.

5. CONCLUSIONS

In this paper an off-design technique for mathematical modeling was applied to simulate a complete operation range of a dual pressure HRSG. Equations to calculate the pressure drop and heat transfer at heat exchangers were adjusted to the OEM data. Good agreements of temperature, mass flow and pressure for both the sides, gas and water/steam, were obtained for all the investigated components. The deviations of pressure drops and mass flow in HRSG components were smaller than 0.5% and ± 2.44%, respectively, with respect to the manufacturer data. Therefore, from results of HRSG simulations were seen that, despite its simplicity, the partial load modeling of HSRG is effective. The developed computer model can thus be used to simulate actual thermal plants through appropriate adjustments to the operation data, being capable of reproducing the entire operation range of a HRSG.



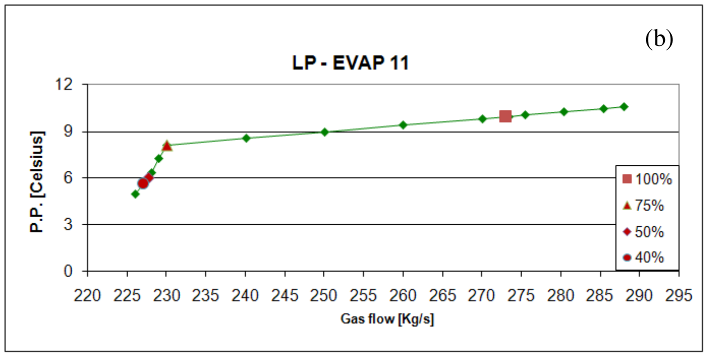


Figure 4. Values of saturation pressure (a) and pinch point (b) for the LP evaporator under several part-load conditions.

6. ACKNOWLEDGEMENTS

The authors would like to thank the financial support of Termorte S/A, through the R&D programme of the ANEEL (Agência Nacional de Energia Elétrica, law 9.991/2000). This work was performed while the third author was a visiting researcher with a scholarship from ANP (Agência Nacional do Petróleo, Gás Natural e Biocombustíveis), on leave from Centre National de la Recherche Scientifique (CNRS, France).

7. REFERENCES

Dechamps, P. J., Pirard, N. and Mathieu, P., 1994, "Part Load Operation of Combined Cycle Plants with and without Supplementary Firing", ASME, Cogen-Turbo 94, Portland (Or), pp. 1-10.

Erbes, M.R. and Eustis, R. H., 1986, "A Computer Methodology for Predicting and Design and Off-Design Performance of Utility Steam Turbine-Generators", in Proceedings of the American Power Conference, Chicago, IL.

Erbes, M.R. and Phillips, J.N., 1987, "Modeling the Off-Design Performance of Power Plants for Systems Studies – Issues and Methodologies", in Analysis and Design of Advanced Energy Systems: Computer-Aided Analysis and Design, ASME, New York, AES- vol.3.3, pp. 15-21.

Holman, J. P., 1976, "Heat Transfer", Fourth Edition, McGraw-Hill Book Company, New York.

Kays, W.M. and Crawford, M.E., 1980, "Convective Heat and Mass Transfer", Second Edition, McGraw-Hill Book Company, New York.

McBride, B. J., Gordon, S., and Reno, M. A., 1993, "Coefficient for Calculating Thermodynamic and Transport Properties of Individual Species", NASA, Technical Memorandum 4513, Washington.

Mironer, A., 1979, "Engineering Fluid Mechanics", McGraw-Hill Book Company, New York.

Sanaye, S., Rezazadeh, M.; Oladi, J., Sadeghpoor, G. H., Bashiri, F. and Sahebi, J., 2006; "Thermal Modeling of HRSG Behavior in Combined Cycle power Plants"; Proceedings of GT2006, ASME Turbo Expo 2006: Power for Land, Sea and Air, May, 8-11, 2006, GT2006-90718, Barcelona, Spain.

Wagner, W., Cooper, J., Dittmann, A., Kijima, J., Kretzchamar, H., Kruse, A., Mares, R., Ogushi, K., Sato, H., Stocker, I., Sifner., O., Takaishi, Y., Trubenbach, J., and Willkommen, T. H., 2000, "IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water", J. Eng. Gas Turbines and Power, ASME, vol. 122: pp. 150-182.

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.