

LOAD CONTROL AND PART-LOAD PERFORMANCE OF A COMBINED CYCLE OPERATING IN COGENERATION CONDITIONS

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Abstract. *Combined cycle technology is not widespread in Brazil. Therefore, there is a need for simulation tools that can take into account the effects of site conditions and part load operation, to obtain dependable technical and economic evaluations in the early stages of a project.*
In this paper a gas-turbine based combined-cycle producing electricity and industrial steam (cogeneration operation) is modeled. The cycle reproduces a proposed power plant to be constructed near REPLAN - the largest Brazilian refinery. The combined cycle consists of three gas turbines, each one with its individual heat recovery steam generator (HRSG), a single steam turbine supplying process steam at constant pressure, temperature and flow rate.
This study presents the hypothesis adopted and the simulation strategies, at full load and part-load conditions. Two gas turbine load control methods were evaluated: the conventional temperature decrease control (with constant air flow and decreasing fuel flow) and the combined effect of air flow control with constant temperature (through variable inlet guide vane geometry) followed by the temperature control when the former can no longer operate. The steam turbine load is controlled by sliding pressure method until the industrial steam pressure reaches its nominal value; when this condition is attained, constant pressure method prevails in the high pressure module.
The full-load performance results were analyzed and compared with traditional combined cycle behavior. The part-load results show the effects of the load control methods on the overall performance of the combined cycle, on the exhaust gas temperature, on the high, medium and low pressure steam flows, on the desuperheater water flow, and on the sliding pressure for the three pressure levels.
All the technical results were employed to perform a second-law analysis of the cycle and an exergoeconomic analysis [11], *which are not included in this paper.*

Keywords: *Combined cycle, Cogeneration, Part-load, Simulation, Efficiency*

1. Introduction

The huge Brazilian dependence on hydropower to generate electric energy is noteworthy. Although a renewable source, hydropower is subject to climatic changes and this may impact the electricity offer. The need of other sources to produce electricity became apparent in 2001, when the Brazilian society was compelled to reduce its consumption of electricity.

The use of natural gas with gas turbines to generate electricity, both in simple cycle or combined cycle, is a possibility which have some advantages, such as short implementation period, low investment cost and a positive impact on natural gas pipeline investments.

Modern combined cycle plants have high efficiency values (greater than 55%), using equipment of proven technologies and always considered an alternative for power generation when there is an associated high natural gas availability. According to Heppenstall [13], combined cycle plants offer superior performances to any other existing thermal cycle that may become available in the next years for applications with large power generation. According to Horlock [14] and Korobitsyn [17], a gas turbine based combined cycle plant is the most efficient way to approach Carnot efficiency because heat is added to the cycle at high temperatures and heat rejection occurs at ambient temperature.

Combined cycle plants can be classified according to different criteria, the most important of which is the number of pressure levels of the Heat Recovery Steam Generator (HRSG). Power plants with one or two pressure levels present lower efficiency values compared to the most recent ones with three pressure and reheat.

2. Description of the combined cycle plant

The plant analyzed is proposed to operate by the side of the largest Brazilian oil refinery (REPLAN). The plant will consist of three GE PG7241 FA gas turbines, each one with its own HRSG, and a single steam turbine. The steam produced by the three HRSGs supplies a constant demand of 250.000 kg/h of process steam at 92 bar and 485°C, and the additional steam generated will be expanded in the steam turbine. Of the 250.000 kg/h process steam, 200.000 kg/h will be sold to the oil refinery and the balance to other consumers expected around the refinery in the near future. A condensate return of 45% is expected from process steam.

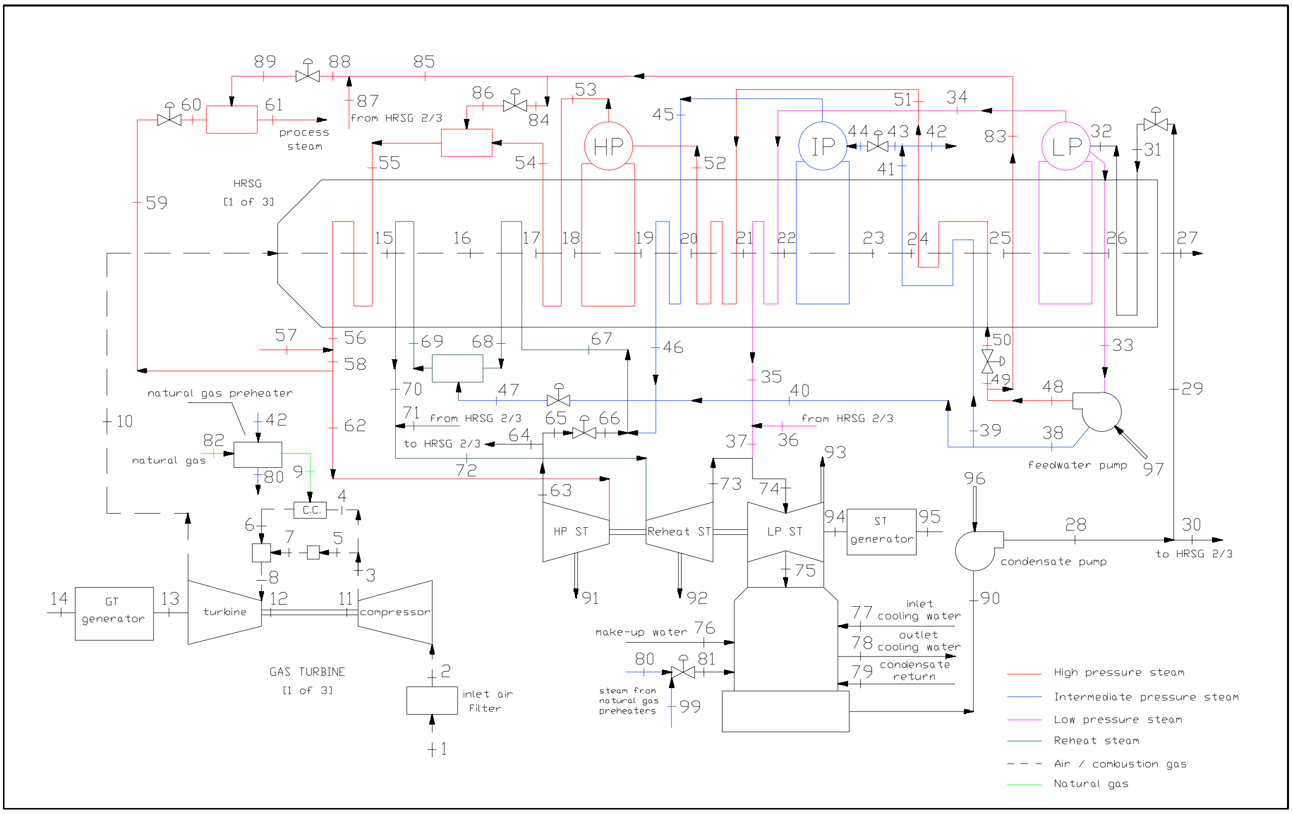

Figure 1: Combined cycle scheme.
Reference: TPP [18] - modified.

The unfired three pressure HRSGs with reheat and the high gas turbines exhaust temperature (600°C) contribute to the high efficiency of the combined cycle plant analyzed. The proposed combined cycle plant is shown schematically in Fig. (1), with only one gas turbine and HRSG depicted, as the others two blocks (GT and HRSG) have similar parameters in the simulation performed.

The high pressure superheaters and reheaters are split in two sections each with attemperators located between each respective pair of heaters in order to control the temperature values, imposed by the maximum temperature (566, 8°C) supported by steam turbine materials rotor. The natural gas is preheated to 185°C by intermediate pressure water from the HRSG economizer (stream 42), before entering the gas turbines combustion chamber.

3. Performance assessment parameters

The exergetic efficiency (η_{ex}) and economic efficiency (η_{eco}), given respectively by Eqs. (1) and (2), were used to evaluate the cogeneration plant performance:

$$\eta_{\text{ex}} = \frac{\overbrace{\dot{\text{E}}\text{x}_{98}}^{\text{power output}} + \overbrace{\left(\dot{\text{E}}\text{x}_{61} - \dot{\text{E}}\text{x}_{76} - \dot{\text{E}}\text{x}_{79}\right)}^{\text{exergy increase of process steam}}}{\underbrace{3 \cdot \dot{\text{E}}\text{x}_{82}}_{\text{fuel exergy}}} \tag{1}$$

$$\eta_{\text{eco}} = \frac{\overbrace{\dot{\text{E}}_{98}}^{\text{power output}} + \left(\frac{1}{3}\right) \cdot \overbrace{\left[\dot{\text{m}}_{61} \cdot (\text{e}_{61} - \text{e}_{79})\right]}^{\text{process heat}}}{3 \cdot \dot{\text{m}}_9 \cdot \text{LHV}} \tag{2}$$

where $\dot{\text{E}}\text{x}$, $\dot{\text{E}}$, e, LHV refer to exergy rate, energy rate, specific energy and lower heating value of the fuel, respectively.

For the exergetic efficiency, the products of the cogeneration plant are power output and the exergy increase of process steam (process steam exergy less make-up water and condensate return exergy). Stream 98 corresponds to cogeneration plant net power, which is not represented in Fig. (1).

As regards the economic efficiency, Huang [16] suggested a typical value of $\frac{1}{3}$ for the ratio of process heat unit price to electrical energy unit price.

4. Modeling and simulation of the combined cycle plant at design condition

Most of the data for the modeling of the combined cycle cogeneration plant were taken from TPP [18], mainly for the HRSG.

The parameters that affect the combined cycle performance have been taken into account in the simulation model, like temperature, ambient pressure, relative humidity, HRSG pressure drop, pinch-point and approach temperatures, steam turbine isentropic efficiency, compressor pressure ratio, condenser pressure and fuel gas composition. Table (1) presents the parameters used in the simulation of the combined cycle plant.

Table 1: Combined cycle parameters.

GT pressure ratio[a]	15,5
GT flow (kg/s)[b]	432
GT combustion temperature (K)[c]	1588
HRSG pressure loss (mbar)[d]	31,1
Maximum temperature for high pressure and reheat steam (°C)[e]	566,8
HRSG heat loss (%)[f]	2
Pinch-point for high pressure level (°C)[g]	10
Approach (°C)[h]	10
Terminal temperature difference (K)	30[1]
Steam turbine isentropic efficiency (%)[i]	85
Steam turbine generator efficiency (%)	96

Reference (a) : GTW Handbook [9]
Reference (b,c) : GE Power Systems [10]
Reference (d,e): TPP [18]
Reference (f): Ganapathy [8]
Reference (g,i): Dechamps [2]
Reference (h): Desideri & Fibbi [5]

It was considered that the three HRSGs operate identically, thus the streams "from HRSG 2/3" in Fig. (1) have the same temperature and pressure to the upstream flow where the mixing process occurs. Similarly, the streams "to HRSG 2/3" present the same pressure and temperature to the upstream flow where it gives the separation process. Therefore each one of the three blocks (gas turbine and HRSG) operate identically. This was the procedure adopted in TPP [18] and also in the present simulation.

The combined cycle was simulated subject to a constant process steam demand of 250.000 kg/h at 92 bar and 485°C. Steam mass flow rates for low and intermediate pressure levels were kept unchanged in the simulation (unless for the case of combined cycle simulation at part-load) and the high-pressure steam flow was calculated until one the following constraints is reached:

1) pinch-point of 10°C for the high pressure level
2) minimum allowable temperature of 75°C for the HRSG exhaust gas, as recommended by Dechamps [2].
3) minimum pinch-point of 8°C for any pressure level [1].

Figure (2) shows exergetic and economic efficiency for the combined cycle plant.

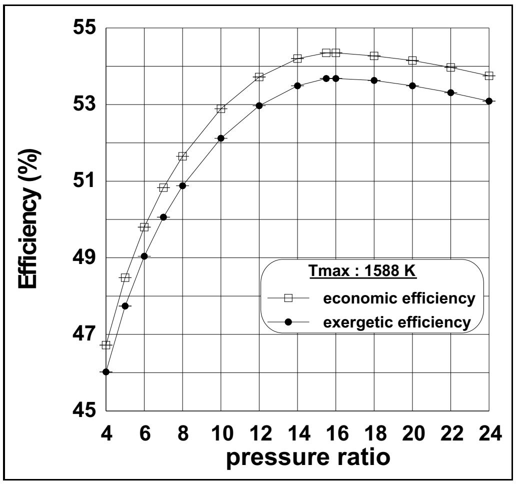

Figure 2: Combined cycle efficiency.

[1]Dechamps [3] recommends a terminal temperature difference (gas turbine exhaust temperature less high pressure steam temperature) between 20 and 30°C.

It can be noted that the optimum pressure ratio for maximum combined cycle efficiency (16) is close to that for maximum gas turbine specific work (15) as was presented by Gomes [11], which turns out to be an advantage for the combined cycle power output.

As efficiency does not vary considerably for pressure ratios between 12 and 24, the gas turbine (design pressure ratio of 15,5) can be designed with a pressure ratio close to that for maximum gas turbine specific work (15), without a great decrease in the combined cycle performance. Such characteristic of the combined cycle is pointed by Horlock [14] as its main advantage. In accordance with Dechamps [3] the optimum pressure ratio for maximum combined cycle efficiency lies in the range 10-18.

Figure (3) presents the combined cycle specific work, where it is noticed that the optimum pressure ratio for maximum specific work is around 6. Similarly as for the Brayton cycle, combined cycle plants have optimum pressure ratios for maximum specific work lower than that for maximum efficiency.

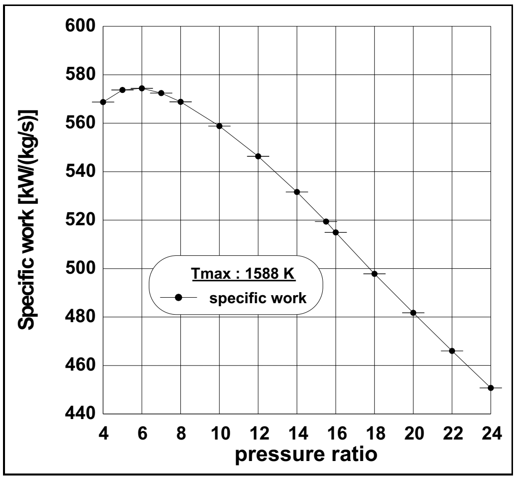

Figure 3: Combined cycle specific work.

Figure (4) shows efficiency of Brayton cycle, Rankine cycle and Combined cycle. It can be concluded that combined cycle efficiency changes little around the optimum pressure ratio for maximum efficiency. While Brayton cycle efficiency increases with pressure ratio, Rankine cycle efficiency decreases owing to the smaller gas turbine exhaust temperatures.

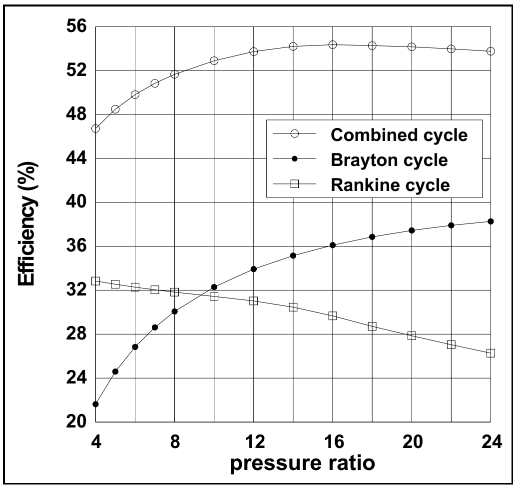

Figure 4: Efficiency of Brayton, Rankine and Combined cycles.

Figure (5) presents the derivative of Brayton and Rankine cycle efficiency to compressor pressure ratio. As already remarked by Horlock [15], the optimum pressure ratio for maximum combined cycle efficiency (around 16) occurs when "Brayton cycle efficiency is increasing with pressure ratio at about the same rate as the Rankine cycle efficiency is decreasing", which can be represented by Eq. (3):

$$\left(\frac{\partial \eta_{\mathrm{Brayton}}}{\partial \mathrm{pr}}\right) \approx -\left(\frac{\partial \eta_{\mathrm{Rankine}}}{\partial \mathrm{pr}}\right) \tag{3}$$

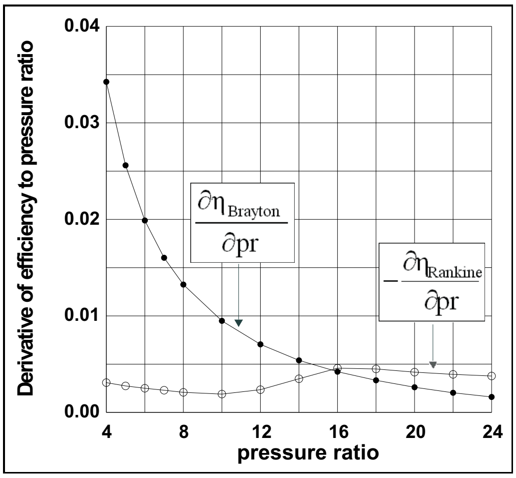

Figure 5: Derivative of Brayton and Rankine cycle efficiency to compressor pressure ratio.

5. Modeling and simulation of the combined cycle plant at part-load

Equations proposed by El-Sayed [6] were used to simulate HRSG heat exchangers and steam turbine at part-load. For the heat recovery steam generator the equations express the ratio of heat exchanger effectiveness at part-load (ϵ) to that at design condition (ϵ_{d}) as function of water/steam and gas mass flow rates at part-load and design condtion (indicated by a subscript 'd'). The effectiveness for economizers (ϵ_{eco}), evaporators (ϵ_{evap}) and reheat/superheaters (ϵ_{sup}) are related respectively to Eqs. (4), (5) and (6).

$$\frac{\epsilon_{\mathrm{eco}}}{\epsilon_{\mathrm{eco,d}}} = \left(\frac{\dot{\mathrm{m}}_{\mathrm{gas}}}{\dot{\mathrm{m}}_{\mathrm{gas,d}}}\right)^{0,15} \cdot \left(\frac{\dot{\mathrm{m}}_{\mathrm{water}}}{\dot{\mathrm{m}}_{\mathrm{water,d}}}\right)^{-0,05} \tag{4}$$

$$\frac{\epsilon_{\mathrm{evap}}}{\epsilon_{\mathrm{evap,d}}} = \left(\frac{\dot{\mathrm{m}}_{\mathrm{gas}}}{\dot{\mathrm{m}}_{\mathrm{gas,d}}}\right)^{-0,05} \cdot \left(\frac{\dot{\mathrm{m}}_{\mathrm{steam}}}{\dot{\mathrm{m}}_{\mathrm{steam,d}}}\right)^{0,01} \tag{5}$$

$$\frac{\epsilon_{\mathrm{sup}}}{\epsilon_{\mathrm{sup,d}}} = \left(\frac{\dot{\mathrm{m}}_{\mathrm{gas}}}{\dot{\mathrm{m}}_{\mathrm{gas,d}}}\right)^{0,20} \cdot \left(\frac{\dot{\mathrm{m}}_{\mathrm{steam}}}{\dot{\mathrm{m}}_{\mathrm{steam,d}}}\right)^{-0,15} \tag{6}$$

Pressure loss for HRSG was predicted in accordance with Eq. (7):

$$\frac{\Delta \mathrm{P}_{\mathrm{HRSG}}}{\Delta \mathrm{P}_{\mathrm{HRSG,d}}} = \left(\frac{\dot{\mathrm{m}}_{\mathrm{gas}}}{\dot{\mathrm{m}}_{\mathrm{gas,d}}}\right)^{1,75} \tag{7}$$

where $\Delta \mathrm{P}_{\mathrm{HRSG}}$ and $\Delta \mathrm{P}_{\mathrm{HRSG,d}}$ are respectively HRSG pressure losses at part-load and design condition. From this HRSG pressure loss it is possible to calculate gas turbine back-pressure which influences its power output.

According to El-Sayed [6] the steam turbine efficiency can be predicted as indicated in Eq. (8):

$$\eta_{\mathrm{ST}} = \eta_{\mathrm{ST,d}} \cdot \left(\mathrm{A}_1 + \mathrm{A}_2 \cdot \mathrm{M}_{\mathrm{r}} + \mathrm{A}_3 \cdot {\mathrm{M}_{\mathrm{r}}}^2\right) \tag{8}$$

where η_{ST} and $\eta_{\mathrm{ST,d}}$ are respectively steam turbine isentropic efficiency at part-load and at design condtion, M_{r} is the ratio of steam mass flow rate at part-load to that at design point. The coefficients A_1, A_2 and A_3 are shown respectively in the following equations:

$$\mathrm{A}_1 = 0,247917 + 0,128125 \cdot \mathrm{P}_{\mathrm{r}} - 0,0101042 \cdot {\mathrm{P}_{\mathrm{r}}}^2 \tag{9}$$

$$\mathrm{A}_2 = 1,23125 - 0,221875 \cdot \mathrm{P}_{\mathrm{r}} + 0,0215625 \cdot {\mathrm{P}_{\mathrm{r}}}^2 \tag{10}$$

$$\mathrm{A}_3 = -0,479167 + 0,09375 \cdot \mathrm{P}_{\mathrm{r}} - 0,0114583 \cdot {\mathrm{P}_{\mathrm{r}}}^2 \tag{11}$$

where P_{r} is the ratio of steam turbine pressure ratio at part-load to that at design point.

In accordance with TPP [18], the steam turbine is of the sliding pressure type, with a constraint of a fixed pressure for process steam (92 bar). Dechamps et al. [4] recommend the use of Stodola relation at steam turbine inlet, as indicated in Eq. (12):

$$\frac{\dot{\mathrm{m}}_{\mathrm{steam}} \cdot \sqrt{\mathrm{T}_{\mathrm{steam}}}}{\mathrm{P}_{\mathrm{steam}}} = \frac{\dot{\mathrm{m}}_{\mathrm{steam,d}} \cdot \sqrt{\mathrm{T}_{\mathrm{steam,d}}}}{\mathrm{P}_{\mathrm{steam,d}}} \tag{12}$$

where $\dot{\mathrm{m}}_{\mathrm{steam}}$, $\mathrm{T}_{\mathrm{steam}}$ and $\mathrm{P}_{\mathrm{steam}}$ denote respectively steam mass flow rate, temperature and pressure at turbine inlet.

To analyze combined cycle plants at part-load is of fundamental importance to know the part-load performance of gas turbines, because, according to Facchini [7] the steam cycle adjusts itself to variations of gas flow and temperature at gas turbine exhaust. Gomes and Gallo [12] studied gas turbines at part-load for two load control methods that are also used to simulate the combined cycle, which are described below:

1) TIT (*turbine inlet temperature*): gas turbine load is decreased by reducing the turbine inlet temperature, without decreasing the air admission in the compressor. The gas turbine inlet temperature is controlled by regulating the fuel flow in the combustion chamber.

2) IGV (*inlet guide vane*) + TIT: gas turbine load is reduced by first diminishing the air inlet flow without changing TIT, until a limit of 80% of air flow at base load, and when this limit is set the turbine inlet temperature control as described previously is turned on.

It is worth noting that in the computational model the load is reduced by decreasing each gas turbine load in equal percentages, which also reflects in the same temperature profile in the operating HRSGs.

At almost 60% plant load, two gas turbines with its respective HRSGs were also simulated, but only one block could not meet the required process steam flow. As the HRSGs do not operate with supplementary firing, the simulation was stopped when gas turbine exhaust temperature could not produce steam for process at the required temperature of 485°C.

Figure (6) presents the exergetic efficiency as load is reduced for the two control modes described. Regardless of the control mode used, the efficiency decreases with load but it is greater for the combined IGV/TIT method than the solely TIT control mode. This is opposite to the results concerning gas turbine performance operating in simple cycle at part-load, as shown by Gomes and Gallo [12].

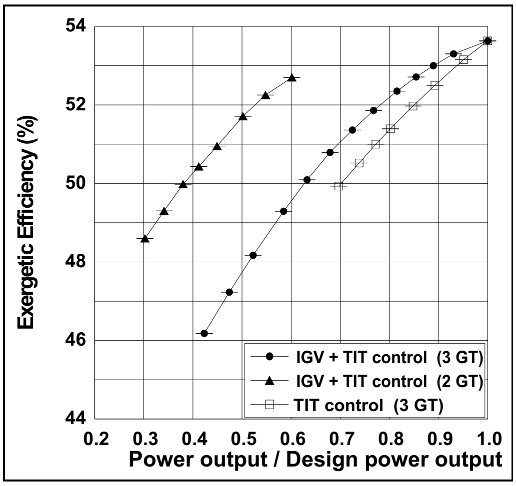

Figure 6: Exergetic efficiency for combined cycle at part-load.

For cogeneration plants the TIT mode has a narrow operating range than the IGV/TIT mode owing to its smaller gas turbine exhaust temperature as had been shown by Gomes and Gallo [12]. It is important to note from Fig. (6) that for loads below 60% it is preferable to turn off one gas turbine and operate the plant with only two gas turbines at full load. For loads below 60% the two GTs load have been decreased in a similar way as done for the three GT sets.

Figure (7) shows HRSG exhaust gas temperature where it can be observed that TIT method causes an increase in exhaust temperature as load is reduced while IGV method has an opposite effect. Desideri and Fibbi [5] obtained similar results when analyzing a combined cycle plant with one pressure level for the HRSG.

Figure (8) presents mass flow rates for high-pressure steam, reheat steam, intermediate-pressure steam and low pressure steam, from where it may be concluded that:

1) whichever the control mode, mass flow rates for high-pressure and reheat steam decreases with load, although IGV/TIT method can produce greater steam flow rates due to its higher gas turbine exhaust temperatures.

2) for the intermediate and low pressure levels steam mass flow rates increase as load is reduced for the TIT method, while it decreases in IGV's operating range and then turns to increase in TIT's operating range for the combined method. As opposed to high-pressure and reheat steam, mass flow rates for intermediate and low pressure levels are greater for the solely operation with TIT method.

3) due to the lower gas turbine exhaust temperatures for TIT method, heat recovery is most efficient for the pressure levels of lower thermodynamic value (intermediate and low pressure). As shown in Fig. (8) for TIT method it can be noted that smaller the gas turbine exhaust temperature, greater will be the mass flow rates for intermediate and low pressure levels.

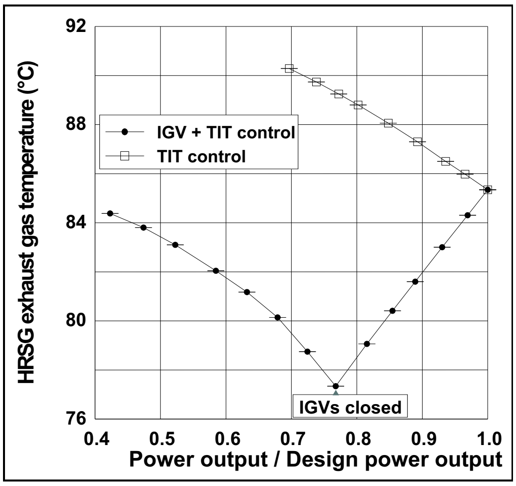

Figure 7: HRSG exhaust gas temperature.

4) steam mass flow rate decreases for all pressure levels with IGV control, although this method attains steam flow rates greater than TIT method for the pressure levels of highest thermodynamic value, which contributes for a higher combined cycle efficiency.

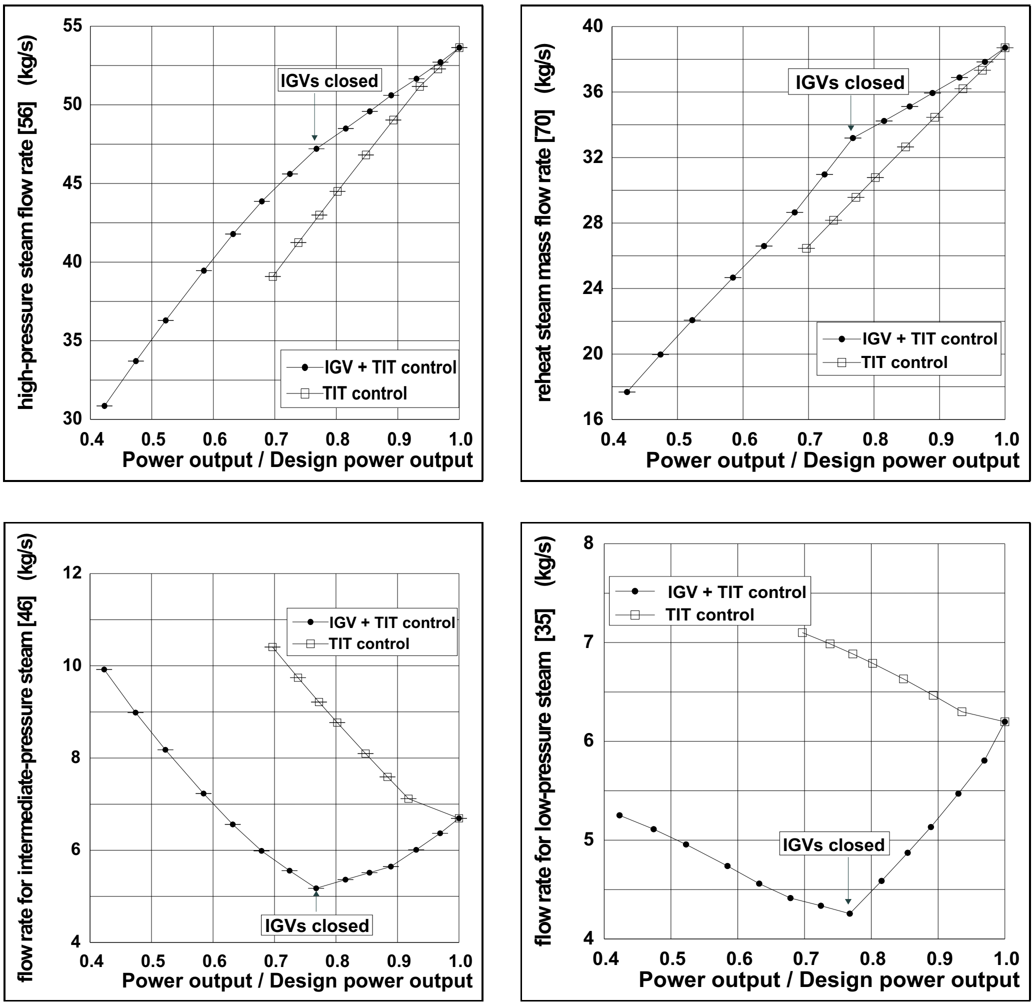

Figure 8: Mass flow rates for high-pressure, reheat, intermediate and low pressure steam.

Figure (9) ilustrates the superheated steam pressure for all pressure levels, from which the following remarks may be arrived at:

1) whichever the part-load control method considered, superheated steam pressure decreases for all pressure levels as load is reduced.

2) steam pressure subject to IGV control is higher compared to TIT control for all pressure levels.

3) as regards the high-pressure steam, there is a constraint related to the required process steam pressure (92 bar). Therefore the steam pressure could vary only from its design value (102,5 bar) to a minimum pressure of 93 bar, thus meeting process steam pressure with a suitable margin. It is worth noting that this minimum allowable pressure is attained by IGV/TIT control at a smaller part-load than that for TIT control.

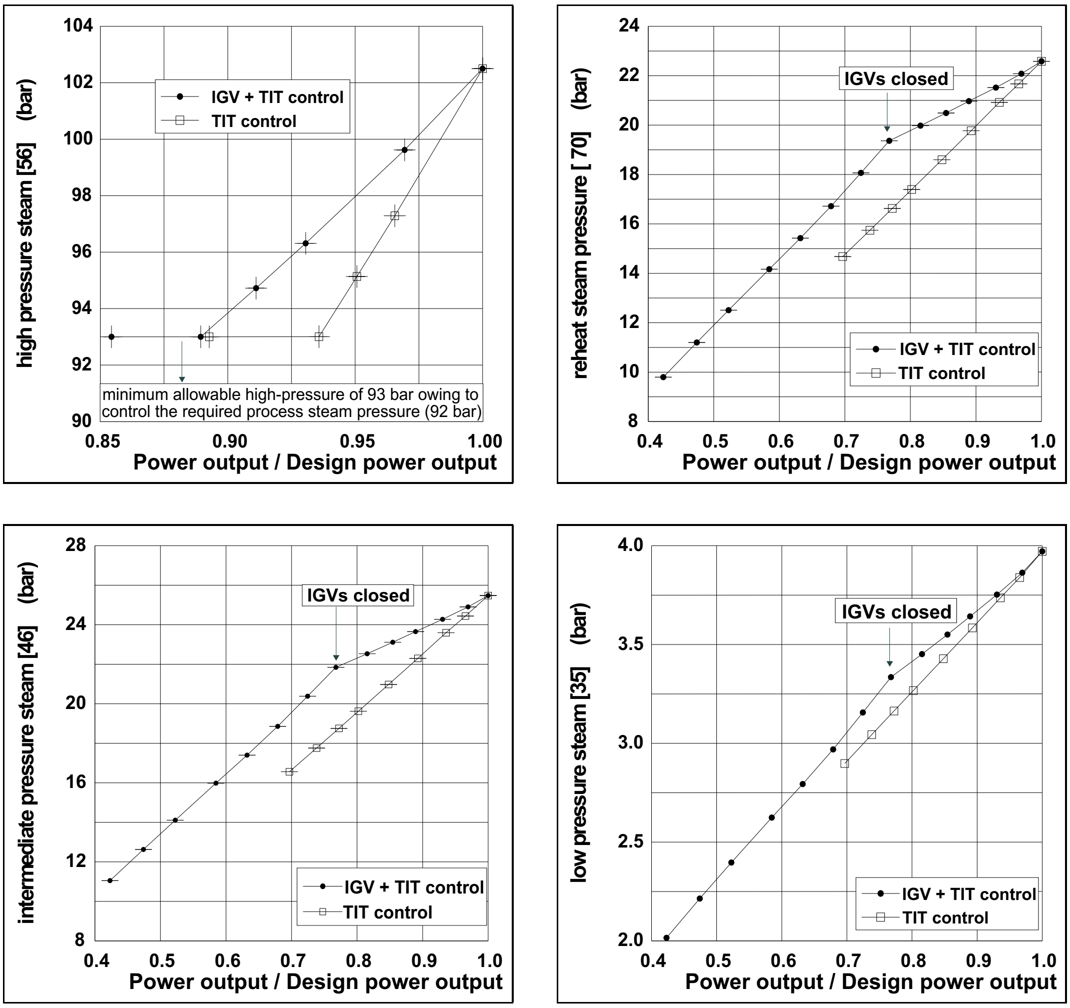

Figure 9: Superheat steam pressure for high-pressure, reheat, intermediate and low pressure levels.

Figure (10) shows the temperature for high-pressure and reheat steam.

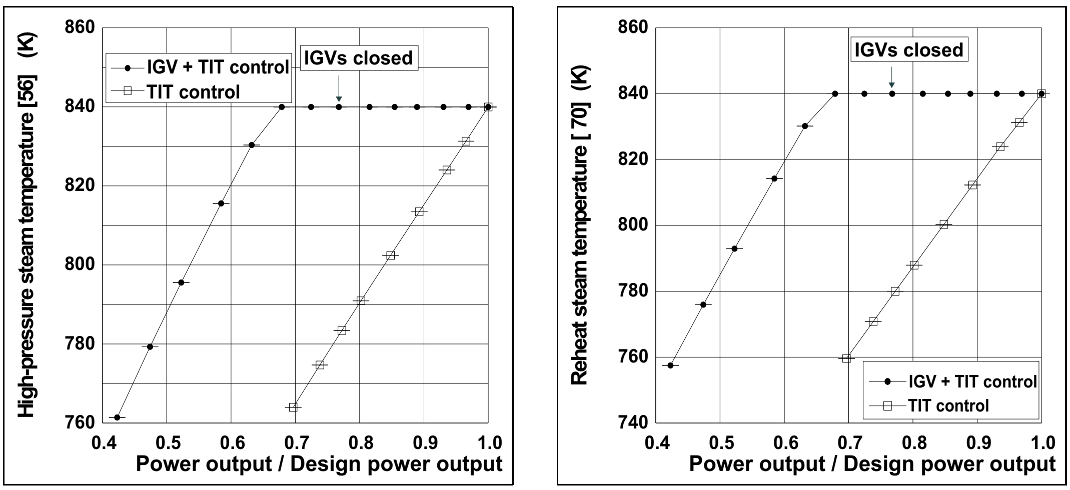

Figure 10: High-pressure and reheat steam temperatures.

It can be noted in Fig. (10) that the steam temperature for the combined method is higher than that for TIT control owing to the higher gas turbine exhaust temperatures presented by the former method. As gas turbine exhaust temperature decreases for TIT control there is an accompanying decrease in high-pressure and reheat steam temperatures, therefore the operating range for this control method is set at the part-load (70%) which could still produce process steam (485°C) subject to gas turbine exhaust temperature decreasing.

As shown by Gomes and Gallo [12] the gas turbine exhaust temperature increases in IGV's operating range compared to the design point, which reflects in higher steam temperatures and thus a much smaller part-load at which the plant could still operate (42%) and also a higher combined cycle efficiency as revealed in Fig. (6).

Due to the increase of gas turbine exhaust temperature in IGV's operating range as load is reduced, it is necessary to inject water into the high-pressure desuperheater (stream 86) and reheat desuperheater (stream 47) because superheat steam temperatures can not be higher than the maximum temperature (566, 8°C) supported by steam turbine materials. Figure (11) shows the water mass flow rates to desuperheaters where it can perceived that water flow increases in IGV's operating range owing to the increase in gas temperature. The water flow is maximum for the load at which the transition from IGV to TIT control occurs and has the maximum gas exhaust temperature. From this point on the gas exhaust temperature decreases and so does the water flow to desuperheaters.

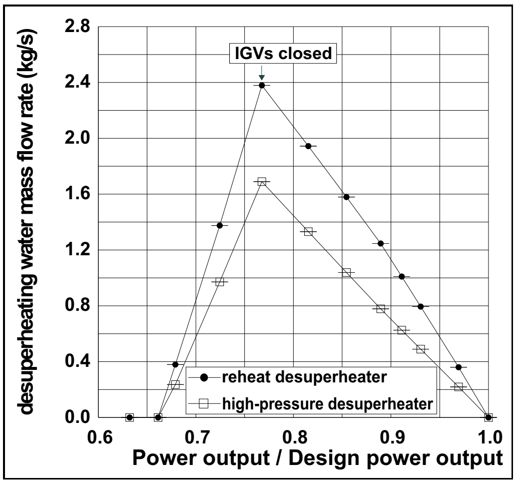

Figure 11: Water mass flow rates for high-pressure and reheat desuperheaters.

6. Conclusion

This paper presented a model to predict the behavior of combined cycle plants at part-load by means of the simulation for a specific combined cycle plant operating in cogeneration mode. The model includes equations to describe HRSG and steam turbine components at part-load, where the gas turbine operation at part-load was analyzed in the first part of this paper for two control modes: inlet guide vane variable geometry (IGV) and turbine inlet temperature (TIT).

The optimum pressure ratio for maximum combined cycle efficiency is greater than that for maximum specific work, similar to the case of gas turbine. In addition, both optimum pressure ratios for the combined cycle are lower than the respective pressure ratios for the gas turbine.

It was shown that the optimum pressure ratio for the combined cycle occurs when Brayton cycle efficiency is increasing with pressure ratio at about the same rate as the Rankine cycle efficiency is decreasing. For the combined cycle plant studied, this was attained at a pressure ratio around 16.

With regard to the part-load methods analyzed, combined cycle efficiency is higher for IGV/TIT control than the solely TIT method as opposed to the technical results presented for gas turbine in simple cycle. For loads below 60% results showed that it is preferable to operate with two gas turbines at full load rather than three gas turbines at part-load and then decrease power output for the two GTs as load is reduced.

Whichever the part-load control considered, both methods decrease gas turbine efficiency and steam pressure for all HRSG pressure levels, although higher steam pressures are achieved with IGV/TIT control compared to TIT control alone.

Steam mass flow rate decreases for all pressure levels with IGV control as opposed to TIT control, which presented an increase in intermediate and low-pressure steam mass flow rates as load was reduced.

The results showed that high-pressure and reheat steam temperatures are considerably higher for the combined method (IGV/TIT) than for TIT control, which contributed both to a wider part-load operating range and higher efficiency for the former method.

7. Acknowledgments

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8. References

[1] Combustion Engineering Inc,1991, *Combustion Fossil Power*, 4th edition, Joseph G. Singer (Editor), Rand Mcnally, USA, 1100 p.

[2] Dechamps, P.J.,1998, *Advanced combined cycle alternatives with the latest gas turbines*, Journal of Engineering for Gas Turbine and Power, Vol.120, pp. 350-357.

[3] Dechamps, P.J.,1998, *Lecture Notes of Biomass Power Generation*, ITA: São José dos Campos, 297 p.

[4] Dechamps, P.J., Pirard, N. and Mathieu, Ph.,1995, *Part-Load Operation of combined cycle plants with and without supplementary firing*, Journal of Engineering for Gas Turbine and Power, Vol.117, pp. 475-483.

[5] Desideri, U. and Fibbi, A.,1993, *A simplified approach to off-design performance evaluation of combined cycle powerplants with single-pressure steam cycles*, ASME Cogen-Turbo,IGTI, Vol.8, pp. 199-207.

[6] El-Sayed, Y.M.,1997, *Predicting part-load performance of an energy system concept*, Proceedings of TAIES, Beijing, China, pp. 176-187.

[7] Facchini, B.,1993, *A simplified approach to off-design performance evaluation of single shaft heavy duty gas turbines*, ASME Cogen-Turbo, IGTI, Vol.8, pp. 189-197.

[8] Ganapathy, V.,1991, *Waste Heat Boiler Deskbook*, Fairmont Press, 399 p.

[9] *Gas Turbine World 1998-1999 Handbook*,1998, Fairfield: Pequot Publishing Inc, 232 p.

[10] GE F-class Gas Turbines. `http://www.gepower.com/products/gas_turbines/79fa_perf.htm`. 23 Feb. 2000.

[11] Gomes, C.M., 2001, *Thermodynamic, exergetic and thermoeconomic analysis of a cogeneration combined cycle plant*. Campinas: Faculty of Mechanical Engineering, State University of Campinas, Brazil, M.Sc. Thesis, 155 p. (in Portuguese)

[12] Gomes, C.M. and Gallo, W.L.R.,2003, *Modeling part-load performance of a gas turbine*, Proceedings of the 17th International Congress of Mechanical Engineering, São Paulo, Brazil, 9 p. (paper submitted to COBEM 2003)

[13] Heppenstall, T.,1998, *Advanced gas turbine cycles for power generation: a critical review*, Applied Thermal Engineering, Vol.18, pp. 837-846.

[14] Horlock, J.H.,1995, *Combined power plants-past, present, and future*, Journal of Engineering for Gas Turbine and Power, Vol.117, pp. 608-616.

[15] Horlock, J.H.,1995, *The optimum pressure ratio for a combined cycle gas turbine plant*, Proceedings of Institution of Mechanical Engineers-PART A: Journal of Power and Energy, Vol.209, pp. 259-264.

[16] Huang, F.F.,1996, *Performance assessment parameters of a cogeneration system*, Proceedings of ECOS, Stockholm, Sweden, pp. 225-229.

[17] Korobitsyn, M.A.,1998, *New and advanced energy conversion technologies.Analysis of cogeneration, combined and integrated cycles*. Enschede: Department of Mechanical Engineering, University of Twente, Netherlands, PhD. Thesis, 135 p.

[18] TPP(Companhia Termelétrica do Planalto Paulista),1999, *Estudo de impacto ambiental-EIA*, Vol.1, pp. 92-119.