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DEVELOPMENT OF A COMPUTATIONAL CODE FOR THERMODYNAMIC ANALYSIS OF ANGRA 2 AND 3 NUCLEAR POWER PLANTS

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Abstract. *With the 4th Industrial Revolution, many industry sectors are incorporating digital features, such as Internet of Things, Artificial Intelligence, Digital Twin, among others. In this context, the nuclear energy industry also demands for computational tools able to deal with design and operating data. This paper presents a thermodynamic analysis model for Angra 2 and 3 Nuclear Power Plants. It describes the development of a computational code in python that provides process data – temperature, pressure and mass flow rate – under steady state regimes. The basic mathematical models for the power plants processes are presented. The model covers the whole power generation process, including the primary circuit, which is composed by the reactor and cold and hot legs, the secondary cycle, which is the most complex one and consists on a regenerative Rankine power cycle, and the tertiary cycle, which is the sea water inlet and outlet. Results are shown for validation cases, by comparisons with heat balance diagrams of Angra 2 NPP for three different cases: (i) normal operation at 100% power load, (ii) normal operation at partial power load and (iii) preheating train bypass.*

Keywords: *computational model, thermodynamic analysis, Nuclear Power Plants.*

1. INTRODUCTION

This paper presents a computational model developed with the objective of calculating the heat balance of Angra 2 and 3 power cycles. So given certain specified conditions, it calculates the steady state values of all process variables involved in the steam-water cycle.

Angra 2 and Angra 3 are twin plants, but the former has been operating since 2001 and the latter is under construction. For an existing plant, there are frequent design modifications, for improvements or adaptations, which present the necessity of understanding the system's behavior prior to such modifications, in the phase in which its impact on the system is evaluated. For a new project the demand for a simulation tool is evident. Besides assisting in the design process of the systems, it is used in the demonstration of accident scenarios to the licensing authority.

As point out by Alobaid *et al.* (2016), there are many commercial codes which perform the same task, but the use of such programs has two disadvantages: high costs and the fact that they are general codes, which have to be customized to represent the plant's system. This can be quite a complex task. So the motivation for this work is the idea of having a perfectly adapted simulation tool developed specifically to Angra 2 and Angra 3 designs.

This work is the second phase of development of the work presented in 2019 in the 25th COBEM (Stilck *et al.*, 2019), which consisted in a prototype implemented in VBA language. In this second phase, the code is implemented in MATLAB® and consists of a zero-dimensional model that employs specific models for each component (heat exchangers, pumps, tanks and turbine) and integrates them with the pressure losses of the piping systems. The non-linear interactions between different sets of components (e.g. turbine and pre-heaters) are iteratively solved.

The details of the model are presented in sections 3 and 4. But first, the power cycle of Angra 2 and 3 NPPs is described in section 2. The results consist on three validation cases, presented in section 5. Then section 6 brings the conclusions.

2. DESCRIPTION OF THE POWER CYCLE

Angra 2 and 3 are PWR nuclear power plants (NPPs), with PWR standing for Pressurized Water Reactor, which is the type of 301 out of a total of 410 nuclear reactors in the world, according to the Power Reactor Information System (PRIS) of the International Atomic Energy Agency. The reader can refer to the work of Goldberg and Rosner (2011) to learn more about the types of nuclear reactors and their evolution over the decades. In PWRs, the nuclear reactor is cooled by a primary cycle, with subcooled water, which transfers heat to a secondary cycle through steam generators. The secondary cycle is the responsible for conversion of thermal power into mechanical and electrical power. The share of thermal power not converted into work is transferred to a tertiary (open) cycle, which, for Angra NPPs, uses sea water.

The secondary cycle operates according to a Rankine power cycle with reheat and regeneration (fig. 1): steam is produced in the steam generators (SG), enters the high pressure turbine (HPT) and flows to the moisture separator-reheater (MSR). Then the superheated steam flows to the low pressure turbine (LPT), going through several expansion stages until reaching the main condensers (MC), where it condenses, rejecting heat to the refrigerating circuit (tertiary system, which is sea water). The condensate generated in the main condensers is pumped by the condensate pumps (CP) to the low pressure preheating trains (LPPH), in which the condensate absorbs heat supplied by the low pressure steam extractions. The feedwater tank (FWT) receives the main condensate as well as the condensate of the high pressure extractions. This tank supplies feedwater through the main feedwater pumps (FWP) to the high pressure preheaters (HPPH). The preheated feedwater is supplied to the steam generator, closing the cycle.

The low pressure and high pressure preheating trains consist in set of heat exchangers in which some are those where the extractions are condensed, called preheaters, and some are the coolers of the condensate formed in the preheaters. Although they are all shell and tube heat exchangers, they are differently modeled in this work, as will be described in sec. 3. In particular, the preheaters play a very important role because they have a strong influence in the flow rate of the turbine extractions.

As shown by the diagram of fig. 1, the extractions from the high pressure turbine (HPT) going to the high pressure preheaters (HPPH) determine the condition of the feedwater that enters the steam generators (SG) and then, consequently, the HPT. This coupling is solved by means of a iterative procedure. The same occurs with the LP pressure sector (LPT and LPPH).

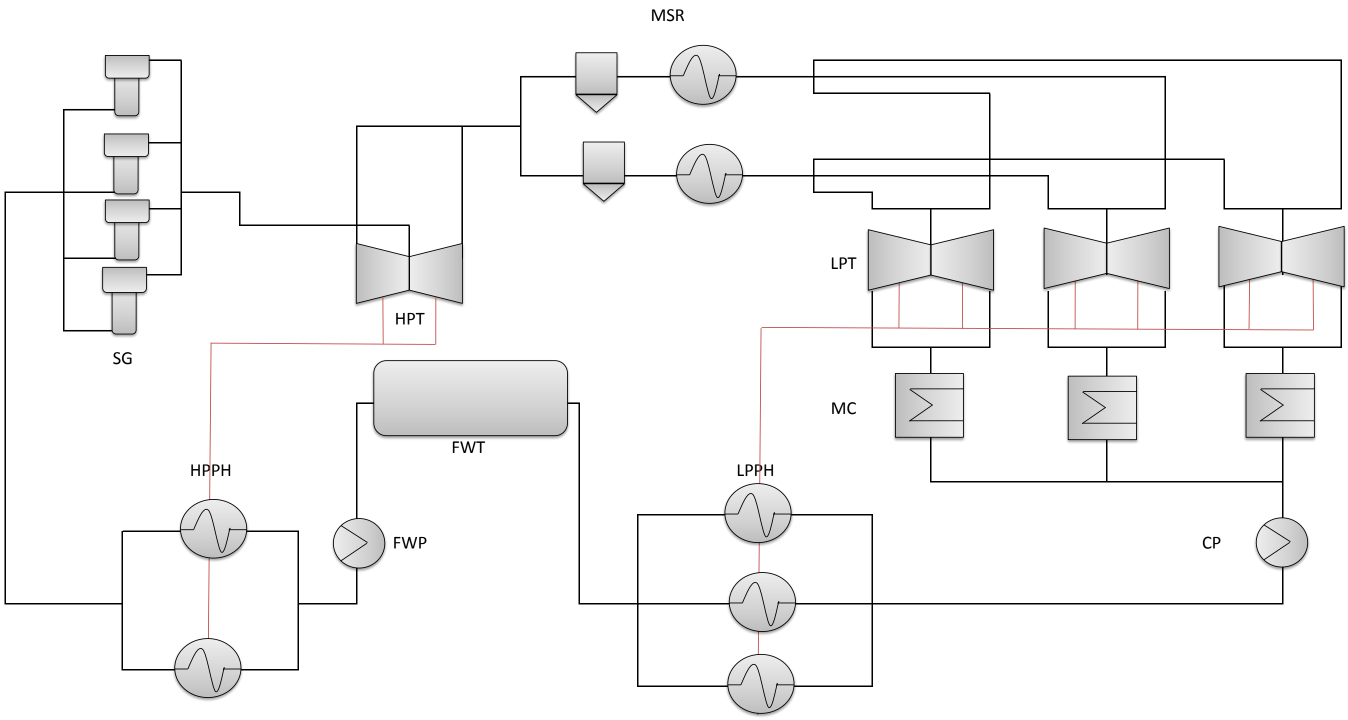

Figure 1. Power cycle (or secondary cycle) of Angra 2 and 3.

3. MATHEMATICAL MODELS

3.1 Low pressure coolers

The low pressure preheating trains are composed of two coolers and three preheaters, in a sequence cooler A1→cooler A2→pre-heater A1→pre-heater A2→preheater A3. The two low pressure coolers are modeled by the $\epsilon - NTU$ method, and are solved coupled through a linear system.

According to Shah and Sekulic (2003), we have that $NTU = UA/C_{\min}$.

$$\epsilon = \frac{2}{1 + C + (1 + C^2)^{0.5}\coth(\Gamma/2)} \tag{1}$$

where $\Gamma = NTU(1 + C^2)^{0.5}$ and $C = C_{\min}/C_{\max}$, recalling that $C_h = (\dot{m}c_p)_h$, where the subscript stands for hot, indicating that these are the parameters of the hot side. Likewise, we have that $C_c = (\dot{m}c_p)_c$ for the cold side. And finally we have that $C_{\min} = \min(C_h, C_c)$ and $C_{\max} = \max(C_h, C_c)$.

After these definitions, the ϵ of each heat exchanger can be defined by

$$\epsilon = \frac{(T_{\text{out}})_{\text{tube}} - (T_{\text{in}})_{\text{shell}}}{(T_{\text{in}})_{\text{tube}} - (T_{\text{out}})_{\text{shell}}} \tag{2}$$

The two equations using ϵ are combined with the energy balance equations $(mc_p\Delta T)_h = (mc_p\Delta T)_c$, and the system can be solved, providing the outlet temperatures.

3.2 Low pressure preheaters

The model developed for the preheaters are different than that of the coolers. For the pre-heaters, an iterative approach is performed to converge the heat transfer capacity of the equipment and the energy balance between the two sides. In other words, the extraction that enters the pre-heater is assumed to fully condense, and leaves the equipment as saturated liquid, in accordance to what actually occurs in the plant. With this assumption, the total amount of heat exchanged is determined, but it has to be coherent with the heat change capacity, i.e., $UA(LMTD)$. This is achieved by correcting the mass flow of the turbine extraction. This approach is employed to the three low pressure pre-heaters and also to the two high pressure pre-heaters.

Of the three low pressure preheaters, two are horizontal heat exchangers, which means that the fluid inside the tubes flows horizontally, and one is vertical.

The overall heat transfer U coefficient is calculated by

$$U = \frac{1}{1/\alpha_{\text{tube}} + 1/\alpha_{\text{shell}} + e/\kappa} \tag{3}$$

where α_{tube} and α_{shell} are the convection heat transfer coefficients of tube and shell sides, and e/κ is the inverse of the conduction heat transfer of the tube wall divided by the tube thickness, in which κ is the conduction heat transfer and e is the tube thickness. The fouling factor is considered zero.

For the horizontal heat-exchangers, α_{tube} is calculated by

$$\alpha_{\text{tube}} = \frac{\mathrm{Nu}\,\kappa_c}{d} \tag{4}$$

where κ_c is the thermal conductivity of the cold fluid (which in this case is the main condensate), d is the internal diameter of the tubes and Nu is the Nusselt number, given by

$$\mathrm{Nu} = 0.037(\mathrm{Re}^{0.75} - 180)\,\mathrm{Pr}^{0.42} \tag{5}$$

with Re and Pr being the Reynolds and Prandtl numbers related to the main condensate.

And the coefficient α_{shell} is given by

$$\alpha_{\text{shell}} = 0.726\left(\frac{\kappa_h^3\rho_h^2 g\Delta h}{\mu_h\Delta T d}\right)^{0.25}(n/2)^{-0.1} \tag{6}$$

The properties κ_h, ρ_h and μ_h are the thermal conductivity, specific mass, dynamic viscosity of the hot fluid; Δh is the specific enthalpy variation of the steam extraction and ΔT is the temperature difference between the fluid temperature at the tube wall and the bulk flow; g is the gravity acceleration, d is the tube external diameter and n is the number of tubes.

For the vertical preheater, α_{tube} is obtained in the same way as for the horizontal pre-heaters, but α_{shell} is calculated by a different equation, expressed as

$$\alpha_{\text{shell}} = 0.003 \left(\frac{\kappa_h^3 \rho_h^2 g L \Delta T}{\mu_h^3 \Delta h} \right)^{0.5} \tag{7}$$

where the height difference L appears in the numerator.

3.3 Feedwater tank

The feedwater tank is a very large component, with approximately 570 m^3 of internal volume, and is responsible for the deaeration of the secondary cycle and works as a buffer for all the pressure, mass and energy variations. In the low pressure turbine, there is a steam extraction which flows directly to the feedwater tank.

For this component, the model seeks for the equilibrium condition by adjusting the pressure and the mass flow of the turbine extraction.

3.4 High pressure preheaters

The high pressure preheating trains are composed of three coolers and two preheaters, in a sequence cooler A5→preheater A5→cooler A6→pre-heater A6→reheater cooler.

In the high pressure preheating trains, a similar approach to that used in the low pressure is employed, with the difference that in the high pressure the equations for the five heat exchangers are coupled to form a single linear algebraic system, composed of ten equations, two for each component. For the coolers, the same approach employed to the low pressure coolers is used, whereas for the preheaters, the strategy used for low pressure pre-heaters is applied. In the high pressure case, the two preheaters are vertical oriented heat exchangers.

3.5 Steam Generators

The steam generators are the frontier between the primary (nuclear) side and the secondary (non-nuclear) side. It transfers the heat produced from the fission of uranium, in the reactor core, to the water-steam cycle, which converts it to electricity. The geometry of the steam generators turns its modeling into is a complex process. The primary system flows inside a U-shaped tube bundle, and the secondary side enters at a subcooled condition. So the secondary side goes through two stages of thermal processes – sensible heat transfer and latent heat transfer – in which the first phase depends on the second.

The method implemented to solve this is based on a trial value for the primary outlet temperature. First, two amounts of exchanged heat are calculated:

- the heat exchanged in the sensible heat stage in the upward part of SG tubes
- the heat exchanged in the downward part of the SG tubes.

The calculation is performed in small sections of tube, i.e., the height of the SG is divided into a number of small segments (fig. 2), creating a mesh in which small parcels of heat are calculated. The calculation proceeds until phase change starts. Then it is interrupted. At this point, the specific enthalpy of the secondary fluid is calculated. From this value, the remaining energy required for total evaporation of the feedwater is calculated. This energy amount is compared to another energy amount which corresponds to the enthalpy variation in the primary side, which is tied to the outlet temperature value.

The ratio of the two energy values are then used to correct the trial value of the outlet primary temperature. The iterative process continues until convergence is reached.

3.6 Steam turbine

The main objective of the turbine model is to determine the steam pressure after each expansion stage. The steam turbine of Angra 2 and 3 is a 1800 RPM turbine with 4 cylinders, one high pressure cylinder and three identical low pressure cylinders. In total, there are six steam extractions, two high pressure and four low pressure extractions (one of the high pressure extractions is actually located in the moisture separator reheater).

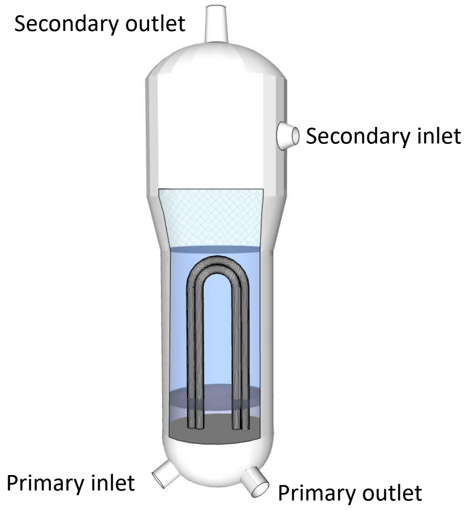

Figure 2. Detail of subcooled region inside the steam generators.

To calculate the pressure at each extraction stage, the Stodola model is employed (Cooke, 1983; Dettori *et al.*, 2017), with the Stodola coefficients being calculated from curves generated from Angra 2 operating data. See example of the high pressure turbine in fig. 3).

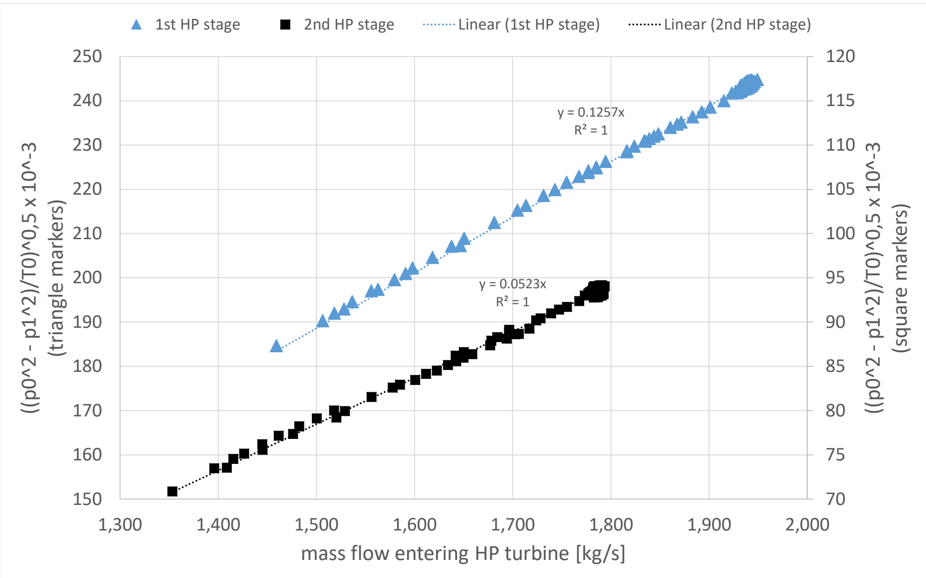

Figure 3. Stodola curves of the high pressure turbine generated from Angra 2 operating data.

The Stodola model consists on the assumption that, for each expansion stage,

$$\frac{W^2 T}{p} = \text{constant} \tag{8}$$

where W is the inlet mass flow, T is the absolute temperature and p is the total pressure. Dettori *et al.* (2017) extend this relation to an expression relating the inlet and outlet pressures of a particular expansion stage, represented by

$$W = K\sqrt{\frac{p_{\text{in}}^2 - p_{\text{out}}^2}{T}} \tag{9}$$

in which the Stodola coefficient K determines the proportionality between both sides. There is one coefficient K_i associated to each expansion stage i, and they are obtained by the operating data of the turbine.

3.7 Main condensers

After the final expansion in the turbine, the exhaust steam flows to the main condensers, closing the cycle. In the main condensers, the generated main condensate is combined with the condensate that comes from the low pressure preheater coolers. There is also a small extraction taken from the steam generators, the blowdown system, that flows to the main condensers. The blowdown system has the objective of making a chemical regeneration of the work fluid. After leaving the main condensers, the main condensate flows to the low pressure preheating trains, restarting the cycle.

4. HEAT BALANCE CALCULATION

For the heat balance calculation, all the models presented in section 3are integrated in an iterative procedure, as shown in figure 4.

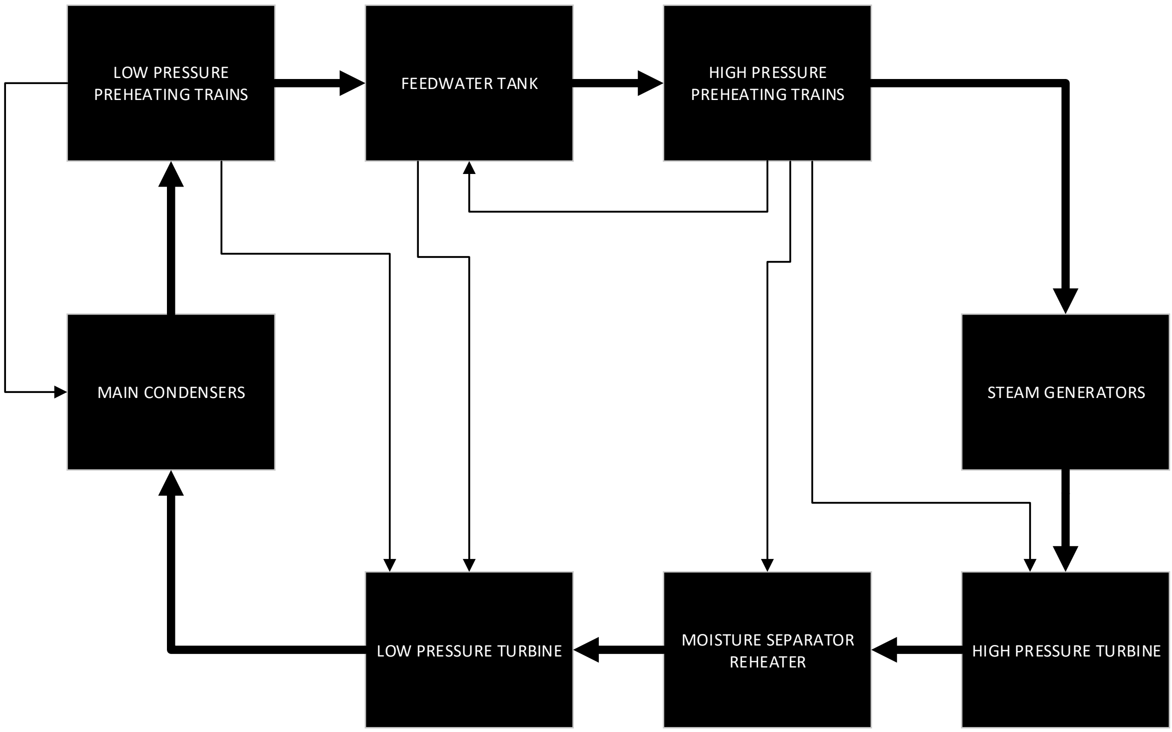

Figure 4. Structure of the code.

To initiate the iterative process, the code reads a data set from Angra 2 operating data. After processing the operating data, each function of the model updates this data set with calculated values.

The code performs the calculations until convergence is achieved, following the flux presented in the diagram of figure 4.

The code is configured to calculate the heat balance for a defined electrical power output. In other words, an electrical power output is set, and then the code starts the iterative process, adjusting the total feedwater mass flow to achieve the wanted power output.

5. RESULTS

Results are presented for three cases, corresponding to the following defined electrical power outputs P_{el}:

- P_{el} = 1093 MW
- P_{el} = 1221 MW
- P_{el} = 1387 MW

The reference for comparison are the heat balances obtained from the Data Reconciliation Program of Angra 2 NPP. Details about the reconciliation program for process data can be found in the works of TranQuang *et al.* (2011) and

Valdetaro (2012). In practice, it represents the actual operating data from the plant. Each one of the three power output values were taken from three selected heat balances from the data reconciliation program of Angra 2, and the results obtained from the model are compared to the values fo the corresponding heat balance.

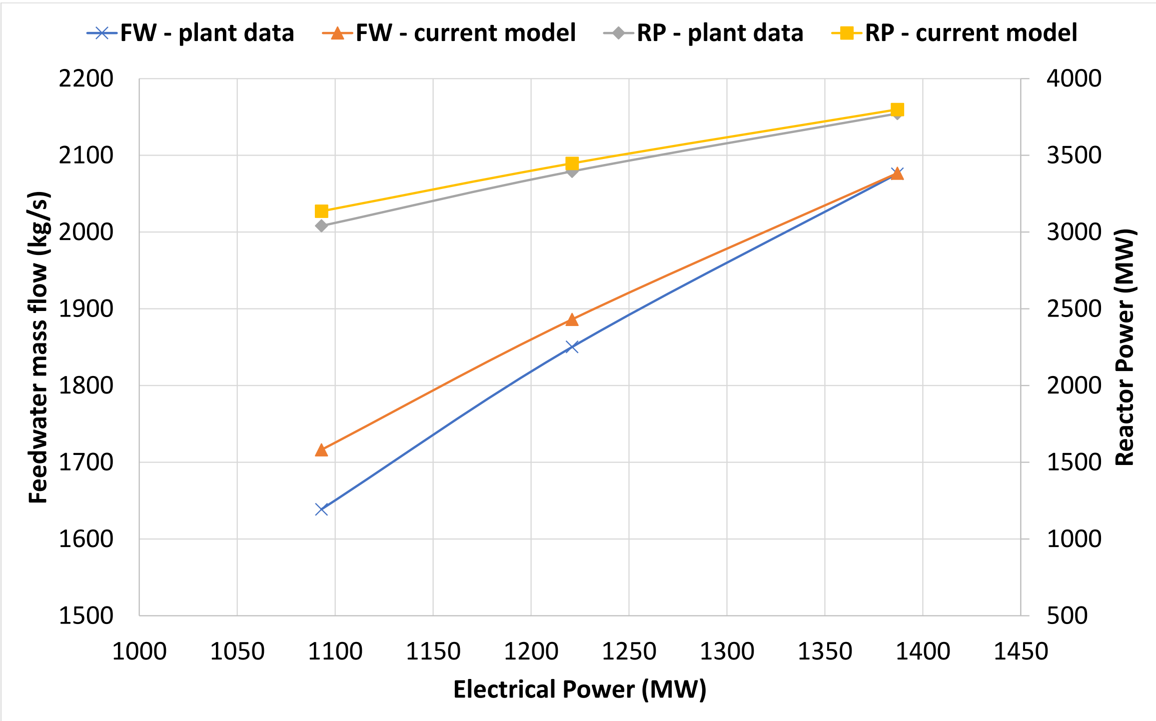

Figure 5. Comparison of the results obtained with the model with the plant operating data for Feedwater mass flow (FW) and Reactor Power (RP).

The results are presented in one graph and three tables. The main calculated variables are the total feedwater (FW) mass flow and the Reactor Power. The graph (fig. 5) shows a comparison of these two process variables for the three operating points – 1093 MW, 1221 MW and 1387 MW.

The three tables show the comparison of the main process variables, together with a measure of error ϵ, calculated by

$$\epsilon = \left| 1 - \frac{X_{\rm model}}{X_{\rm plant}} \right| \tag{10}$$

where $X_{\rm model}$ is a generic process data calculated by the model and $X_{\rm plant}$ is a generic process value from the plant (data reconciliation value). Table 1 shows the results for $P_{\rm el} = 1093$ MW, table 2 shows the results for $P_{\rm el} = 1221$ MW and table 3 shows the results for $P_{\rm el} = 1387$ MW.

It can be noticed that the error increases as the power output is reduced. One reason for this is that some of the heat exchangers have constant overall heat transfer coefficient, corresponding to 100% thermal load, i.e., close to 1350 MW. That is the case of the low pressure coolers. Another reason is that Stodola coefficients for the turbine are obtained by linear regression over the operating data, which are more dense close to 100% thermal load. In general, the great majority of the operating data available correspond to 100% load, since the plant has a very high load factor (constant uninterrupted operation).

6. CONCLUSIONS

The paper presented a computational code developed for calculation of the heat balance of Angra 2 and 3 power plants. For a given electrical power output, it calculates all the process variables of the steam-water cycle.

As commented in section 5, the model achieves its highest accuracy for power outputs close to 100% power output, which correspond to approx. 1350 MW. The reason os that some models were calibrated for this power load condition.

Table 1. Results for main operating data. Comparison with plant data for 1093 MW Electrical Power.

Electrical Power: 1093 MW	**plant data**	**model**	**error**
FW mass flow (kg/s)	**1638.70**	**1716.28**	**4.73%**
Reactor Power (MW)	**3040.41**	**3135.90**	**3.14%**
HP turbine Stage 1 (MW)	225.03	215.35	4.30%
HP turbine Stage 2 (MW)	147.78	158.95	7.56%
LP turbine Stage 1 (MW)	65.85	50.75	22.92%
LP turbine Stage 2 (MW)	42.01	44.62	6.22%
LP turbine Stage 3 (MW)	37.83	45.19	19.46%
LP turbine Stage 4 (MW)	27.26	31.10	14.06%
LP turbine Stage 5 (MW)	38.75	33.43	13.73%
LP turbine Stage 6 (MW)	28.55	34.53	20.93%
FW temp after reheater cond. cooler (°C)	209.80	222.47	6.04%
FW temp after A6 preheaters (°C)	204.75	219.61	7.26%
FW temp after A6 cond. cooler (°C)	179.45	189.71	5.72%
FW temp after A5 preheaters (°C)	175.05	185.12	5.75%
FW temp after A5 cond. cooler(°C)	153.02	164.61	7.58%
Main cond temp after A3 preheater (°C)	117.10	131.60	12.38%
Main cond temp after A2/A1 preheater (°C)	91.43	102.39	11.98%
Main cond temp after A2/A1 cond coolers (°C)	43.43	51.80	19.25%
	maximum error:		22.92%
	mean error:		11.37%

Table 2. Results for main operating data. Comparison with plant data for 1221 MW Electrical Power.

Electrical Power: 1221 MW	**plant data**	**model**	**error**
FW mass flow (kg/s)	**1850.30**	**1886.28**	**1.94%**
Reactor Power (MW)	**3395.73**	**3448.08**	**1.54%**
HP turbine Stage 1 (MW)	252.95	243.21	3.85%
HP turbine Stage 2 (MW)	168.31	173.52	3.10%
LP turbine Stage 1 (MW)	67.49	58.86	12.80%
LP turbine Stage 2 (MW)	48.85	49.59	1.50%
LP turbine Stage 3 (MW)	50.15	50.20	0.09%
LP turbine Stage 4 (MW)	29.87	33.83	13.26%
LP turbine Stage 5 (MW)	37.38	37.20	0.49%
LP turbine Stage 6 (MW)	32.97	38.46	16.65%
FW temp after reheater cond. cooler (°C)	213.75	221.98	3.85%
FW temp after A6 preheaters (°C)	209.85	219.11	4.41%
FW temp after A6 cond. cooler (°C)	183.80	189.32	3.00%
FW temp after A5 preheaters (°C)	180.00	184.75	2.64%
FW temp after A5 cond. cooler (°C)	156.79	162.96	3.94%
Main cond temp after A3 preheater (°C)	120.80	129.37	7.10%
Main cond temp after A2/A1 preheater (°C)	93.37	99.27	6.33%
Main cond temp after A2/A1 cond coolers (°C)	46.23	49.47	7.00%
	maximum error:		16.65%
	mean error:		5.80%

However, the accuracy achieved by the model is considered within a range of good agreement. Although some large errors are observed, the FW mass flow and the Reactor Power are calculated with errors below 5%, even for the worst case tested – $P_{\text{el}} = 1093$ MW. And for $P_{\text{el}} = 1387$ MW, the model predicted values with very good accuracy. This means that the model can serve as a tool for thermodynamic analysis of various scenarios of interest for the plant, e.g., heat exchanger tube plugging.

Future work includes the improvement of the thermal model of the low pressure coolers, with calculation of the heat transfer coefficient, implementation of thermal calculation of the main condensers, and integration of the primary mean temperature control.

Table 3. Results for main operating data. Comparison with plant data for 1387 MW Electrical Power.

Electrical Power: 1387 MW	**plant data**	**model**	**error**
FW mass flow (kg/s)	**2076.00**	**2076.76**	**0.04%**
Reactor Power (MW)	**3772.26**	**3799.21**	**0.71%**
HP turbine Stage 1 (MW)	282.71	279.30	1.21%
HP turbine Stage 2 (MW)	192.54	188.90	1.89%
LP turbine Stage 1 (MW)	69.52	69.08	0.63%
LP turbine Stage 2 (MW)	57.63	56.28	2.33%
LP turbine Stage 3 (MW)	58.63	57.45	2.00%
LP turbine Stage 4 (MW)	36.32	36.61	0.80%
LP turbine Stage 5 (MW)	41.03	42.11	2.64%
LP turbine Stage 6 (MW)	40.95	43.64	6.56%
FW temp after reheater cond. cooler (°C)	218.50	221.41	1.33%
FW temp after A6 preheaters (°C)	215.35	218.53	1.48%
FW temp after A6 cond. cooler (°C)	188.45	188.82	0.20%
FW temp after A5 preheaters (°C)	184.40	184.25	0.08%
FW temp after A5 cond. cooler (°C)	160.12	160.52	0.25%
Main cond temp after A3 preheater (°C)	123.50	125.70	1.78%
Main cond temp after A2/A1 preheater (°C)	96.17	93.79	2.47%
Main cond temp after A2/A1 cond coolers (°C)	45.07	44.60	1.04%
	maximum error:		5.56%
	mean error:		1.79%

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