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DEFINITION OF THERMAL HYDRAULIC PROFILE OF A STEAM GENERATOR FOR NAVAL NUCLEAR PROPULSION VIA THERMODYNAMIC ANALYSIS OF ANGRA 1 STEAM GENERATOR

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Abstract. *Temperature profiles of U-tubes Steam Generators, applied to nuclear power plants, were not easy to find in the available literature. In this paper, we studied the heat transferred from the primary to secondary circuit of a nuclear power plant located in Rio de Janeiro, Angra 1, and a simulation for naval nuclear propulsion was obtained from calculated data. A thermodynamic analysis was made to reach out the operation parameters of the Steam Generator such as pressure, temperature and the heat transferred. The temperature profiles in Angra 1 and submarine Steam Generators were determined using numerical solutions of finite differences method, applied to parallel and countercurrent heat exchangers. During the phase change, a constant variation of the enthalpy was considered, making the primary fluid temperature decreases following a linear behavior. The heat rejected from the primary circuit and the tubes length needed for making the secondary fluid reaches the saturation point, in each Steam Generator, was defined, respectively, as 943.14* MW_{th} *and 1.4 m for Angra 1 and 42.85* MW_{th} *and 0.435 m to submarine Steam Generator. The number of tubes to reject the heat in SG was defined in, approximately, 950 U-tubes. Using the developed methodology called "Enthalpy Ruler", the encountered results were considered adequate, since the defined lengths are compatible with the constant variation of the enthalpy from the compressed liquid to saturated steam.*

Keywords: *Nuclear Engineering, Energy, Thermodynamics, Mechanical Engineering, Steam Generator.*

1. INTRODUCTION

The Scorpène-class submarine, designed by DCNS, is designed to be able to add an extra AIP (Air-Independent Propulsion) module to increase the autonomy of the submarine without the need to return to the surface to feed the diesel engines with air. Thus, using operational parameters of the Angra 1 nuclear power plant, a simulation of the Steam Generator thermal hydraulic parameters of a naval nuclear propulsion was made for a hypothetical replacement of the AIP module by a nuclear module.

In this work, we carried out an analysis of the mass and energy balance through thermodynamics and heat transfer in steady state of the interface of primary/secondary circuit of a PWR nuclear power plant – Angra 1. The study is based on literature involved in 2-loop Westinghouse plants, which is technically similar to Angra 1, and the use of data on properties such as pressure, temperature, flow, among other data available in Eletronuclear (2014) and Westinghouse (2005), aiming for reliable results.

From the prior analysis of Angra1, an initial simulation is applied to a hypothetical nuclear submarine Steam Generator, based on Scorpène Class (Deagel, 2012) using predefined operating parameters of Angra 1, such as the dimensions of the plant equipment which are reduced. Thus, all thermodynamic parameters were recalculated in order to obtain the heat transferred from the primary to secondary circuit to reach the parameters in accordance with the operating limitations and complexities of a submarine.

2. METHODOLOGY

A thermodynamic analysis of a Steam Generator needs to consider the thermal powers involved on operation of this equipment into a nuclear power plant.

The simplified PWR power plant, according to Todreas and Kazimi (2011), is showing in Fig. 1:

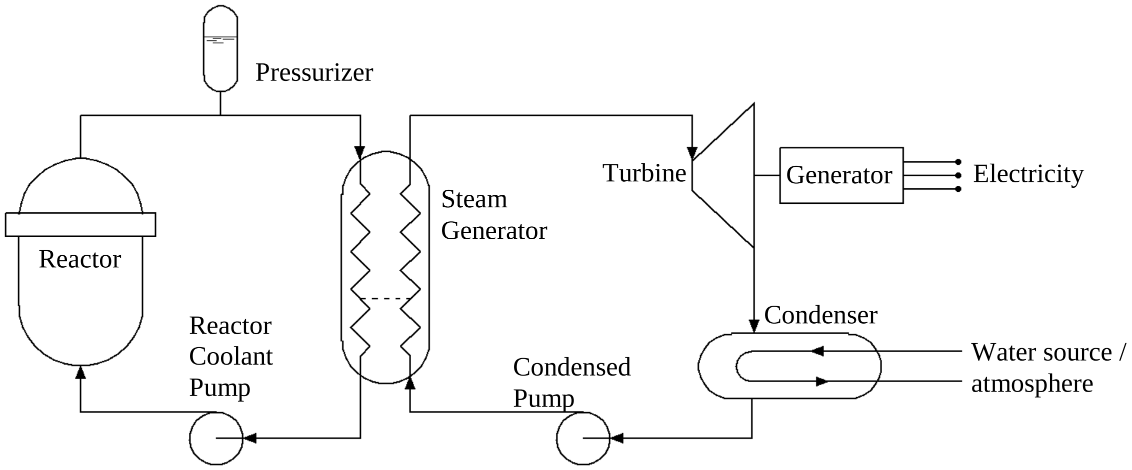

Figure 1: Simplified flow chart of typical PWR plant. (Todreas and Kazimi, 2011)

To make this analysis, a control volume was applied in the interface primary/secondary circuit, as showing in Fig. 2.

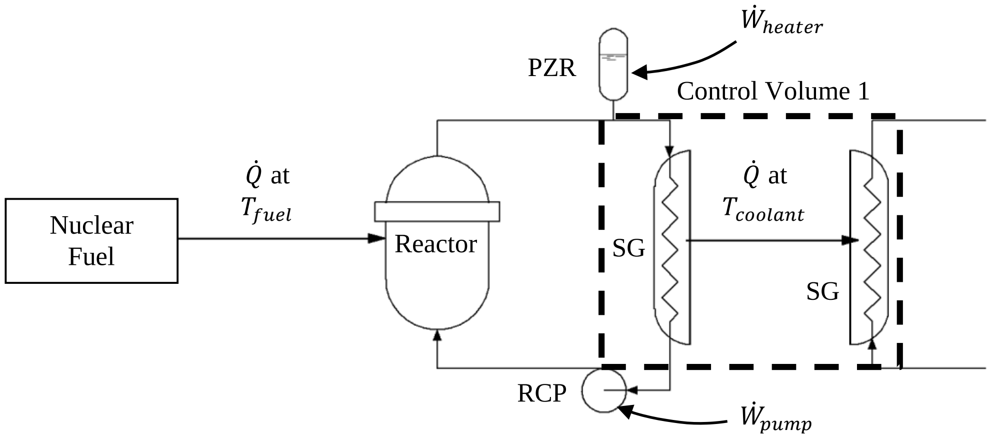

Figure 2: Simplified primary circuit with analyzed control volumes. (Todreas and Kazimi, 2011)

The Control Volume 1 contains only the Steam Generator because in this paper, only the Steam Generator was studied considering the primary and the secondary fluid. It is important to note that the secondary fluid flows always from the bottom to the top, passing through the tubes of the Steam Generator. The secondary flow is represented by the blue arrows and the primary fluid by the red and green arrows in Fig. 3.

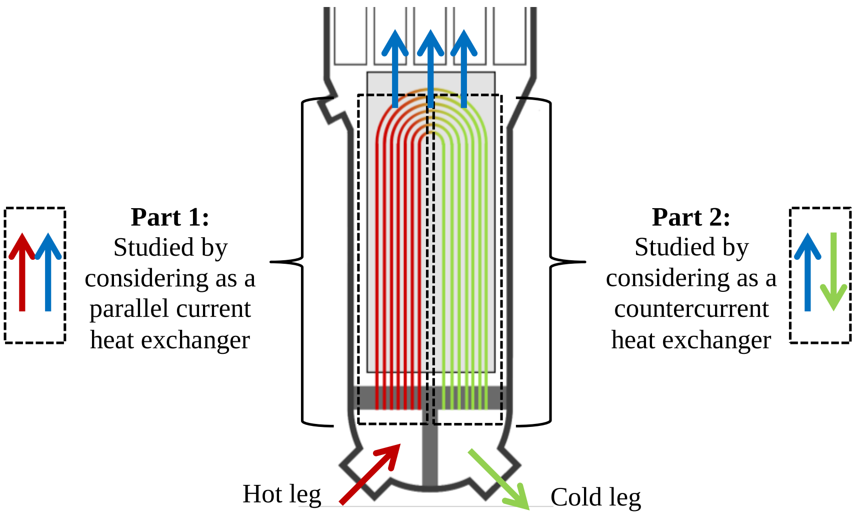

Figure 3: Selection parts to Steam Generator analysis.

In order to optimize the analysis of the Steam Generator, the heat transferred from primary to secondary fluid was studied assuming that the inlet tubes (Part 1) was considered as a parallel current heat exchanger and the outlet tubes (Part 2) was considered as a countercurrent heat exchanger, as shown in Figure 3.

For the Part 1, a typical profile of the parallel current heat exchanger was analyzed to obtain a general equation to model it in Microsoft Excel software, as following:

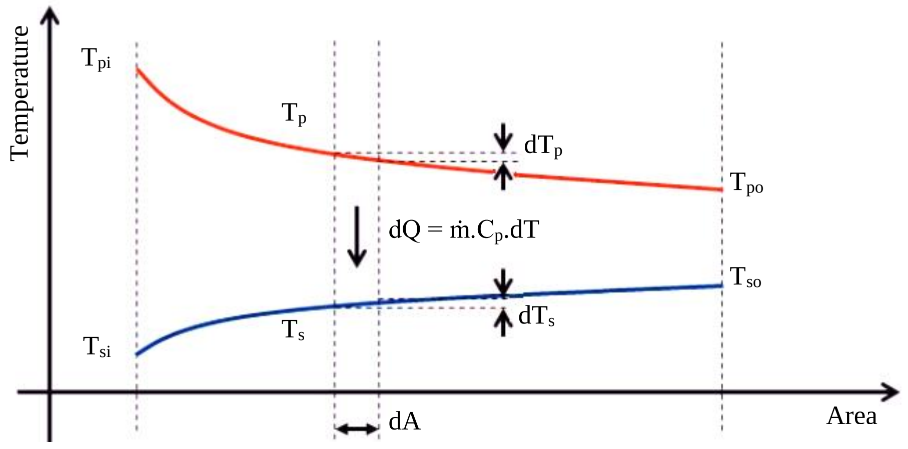

Figure 4: Analysis of a typical profile of the parallel current heat exchanger.

Through this analysis, it was possible to develop the general equation of the parallel current heat exchanger, in accordance to Seleghim (2014) methodology. The following equation can be modeled in *Microsoft Excel* software:

$$\frac{dT_p}{dA} + U.\boldsymbol{X}'.T_p = -\frac{\dot{m}_p.c_{p_p}}{\dot{m}_s.c_{p_s}}\frac{dT_p}{dA} + U.\boldsymbol{X}'.\left(T_{si} + \frac{\dot{m}_p.c_{p_p}}{\dot{m}_s.c_{p_s}}.(T_{pi} - T_p)\right) \quad (1)$$

With $\boldsymbol{X}' = \left(\frac{1}{\dot{m}_p.c_{p_p}} + \frac{1}{\dot{m}_s.c_{p_s}}\right)$

For the Part 2, a typical profile of the countercurrent heat exchanger was analyzed to obtain a general equation to model it in *Microsoft Excel* software, as following:

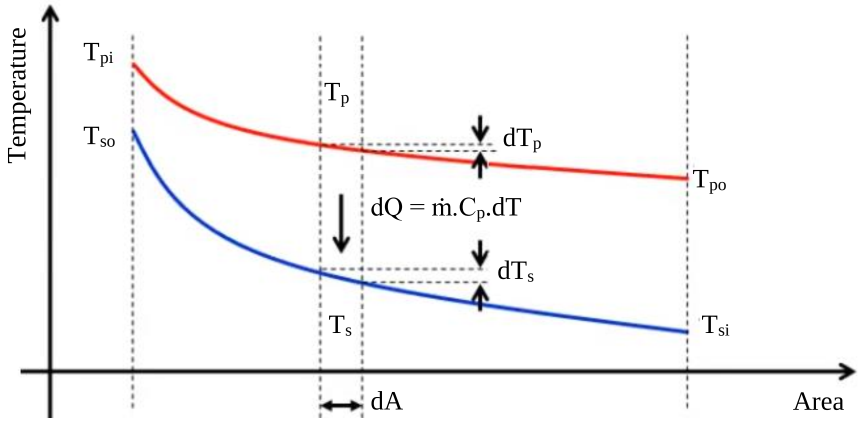

Figure 5: Analysis of a typical profile of the parallel current heat exchanger.

Through this analysis, it was possible to develop the general equation of the countercurrent heat exchanger, in accordance to Seleghim (2014) methodology. The following equation can be modeled in *Microsoft Excel* software:

$$\frac{dT_p}{dA} + U.\boldsymbol{X}''.T_p = \frac{\dot{m}_p.c_{p_p}}{\dot{m}_s.c_{p_s}}\frac{dT_p}{dA} + U.\boldsymbol{X}''.\left(T_{si} + \frac{\dot{m}_p.c_{p_p}}{\dot{m}_s.c_{p_s}}.(T_p - T_{po})\right) \quad (2)$$

With $$\boldsymbol{X}'' = \left(\frac{1}{\dot{m}_p . c_{p_p}} - \frac{1}{\dot{m}_s . c_{p_s}}\right)$$

Where:
T = coolant temperature [°C]
A = heat transfer area [m^2]
$\dot{m}$ = coolant mass flow in single channel [kg/s]
c_p = coolant specific heat [kJ/kg.°C]
Subscripts $_p$ = referring to primary circuit
Subscripts $_s$ = referring to secondary circuit
Subscripts $_{pi},\ _{po}$ = referring to primary inlet and outlet, respectively
Subscripts $_{si},\ _{so}$ = referring to secondary inlet and outlet, respectively

The overall heat transfer coefficient (U) depends on the convective coefficient of the primary circuit and the secondary circuit, which are determined by the number of Reynolds (Re), Prandtl (Pr), Nusselt (Nu) and friction factor of the flow tubes, internally and externally. In this study, the Nusselt number was obtained by Gnielinski correlation (Incropera et al., 2008) instead of the Dittus-Boelter correlation because the Gnielinski correlation considers the tubes friction factor, aiming for more reliable results, as following:

$$Re = \frac{\rho V D}{\mu} \tag{3}$$

$$Pr = \frac{\mu C_p}{k} \tag{4}$$

$$Nu = \frac{hD}{k} \tag{5}$$

$$Nu = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12{,}7\left(\frac{f}{8}\right)^{0{,}5}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{6}$$

Where:
f = tube friction factor (obtained by Moody chart or correlations of Colebrook, Churchill, etc.)
ρ = coolant density [kg/m³]
V = coolant average velocity [m/s]
D = pipe diameter [m]
μ = coolant dynamic viscosity [N.s/m²]
h = coolant convective coefficient [W/m².°C]
k = coolant thermal conductivity [W/m.°C]

It is important to note that the Eq. (1) and Eq. (2) must be applied only in the subcooled region of the secondary fluid. In other words, it should be used only where the secondary fluid presents liquid phase.

3. RESULTS AND DISCUSSION

3.1 Angra 1:

Table 1 contains the main data to Steam Generator analysis and for determining the temperatures profiles:

Table 1: Angra 1 power plant Steam Generator data
Adapted from Eletronuclear (2014); Westinghouse (2005)

Steam Generators operational data	
Primary side pressure [kgf/cm²]	158
Primary side coolant mass flow [kg/s]	4479

Secondary side pressure [kgf/cm²]	64.7
Secondary side coolant mass flow [kg/s]	515
Primary side inlet temperature [°C]	324.3
Primary side outlet temperature [°C]	287.5
Feedwater inlet temperature [°C]	221.1
Steam outlet temperature [°C]	279.2
Steam outlet quality	0.999
Heat rejected in Steam Generator [MW_{th}]	943.14
Tube wall thickness [mm]	1.05
Number of tubes	5428
Inconel tubes thermal conductivity [W/m.°C]	12.1

According to Bueno and Garcia (2012) the type of Steam Generator used in Angra 1, similar to Almaraz, has its tubes following a triangular pitch, as shown in Fig. 6.

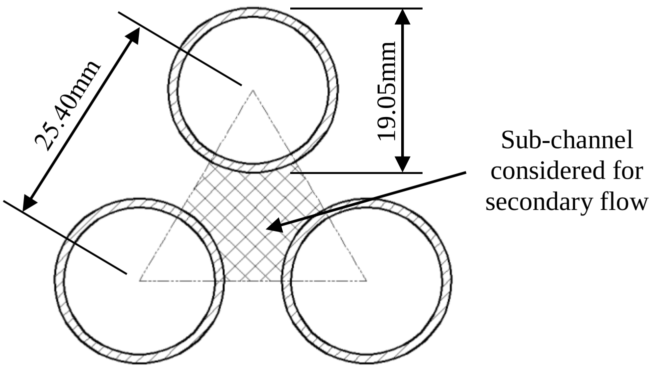

Figure 6: Triangular pitch, similar to Angra 1, with 19.05mm of tubes diameter.

Using the Eqs. (1) to (6), it was possible to model the profiles in *Microsoft Excel* software and find the values of Nusselt numbers (Nu) and the convective coefficients (h). According to NORSOK (2006), the roughness of titanium and nickel alloy tubes are defined as 0.05 mm. This value was used to visual interpolation on Moody chart to determine the tubes friction factor and applying on the Gnielinski correlation (Eq. 6).

The sub-channel area for flowing the secondary's fluid was calculated, according to Fig. 6, as 136.85 mm² and, applying the visual interpolation on Moody chart, the friction factors was defined as f=0,025 and f=0,030 for the primary and secondary sides, respectively.

The Nusselt numbers calculated to fluid conditions at inlet and outlet, respectively, are:

1718.05 $\leq$ Nu $\leq$ **1887.85** – Primary fluid

96.67 $\leq$ Nu $\leq$ **108.04** – Secondary fluid

Using the Eq. (5), it was possible to determine the convective coefficients to fluid conditions at inlet and outlet, respectively, as following:

52.56 kW/m² .°C $\geq$ h_p $\geq$ **51.11 kW/m².°C** – Primary fluid

3.25 kW/m².°C $\leq$ h_s $\leq$ **3.37 kW/m².°C** – Secondary fluid

The temperature profiles of the tubes in subcooled regions are shown in the Fig. 7 and Fig. 8. The length of the tubes required for the secondary fluid for beginning phase change was determined at approximately 1.4 m, as shown in Fig. 7 and Fig. 8.

The overall temperature profile in the Steam Generator can be modeled using the previous calculated data. This temperature profile can be seen in Fig. 9, as follows:

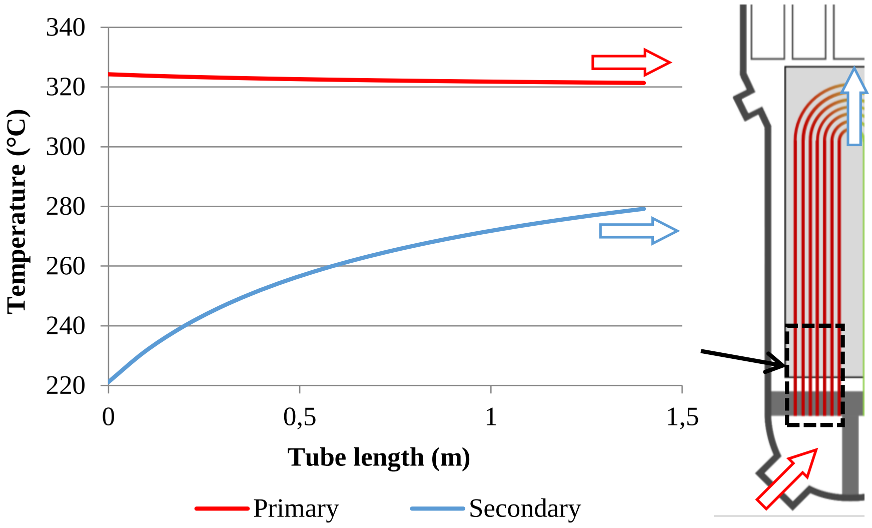

Figure 7: Temperature profile on the subcooled region of the Part 1.

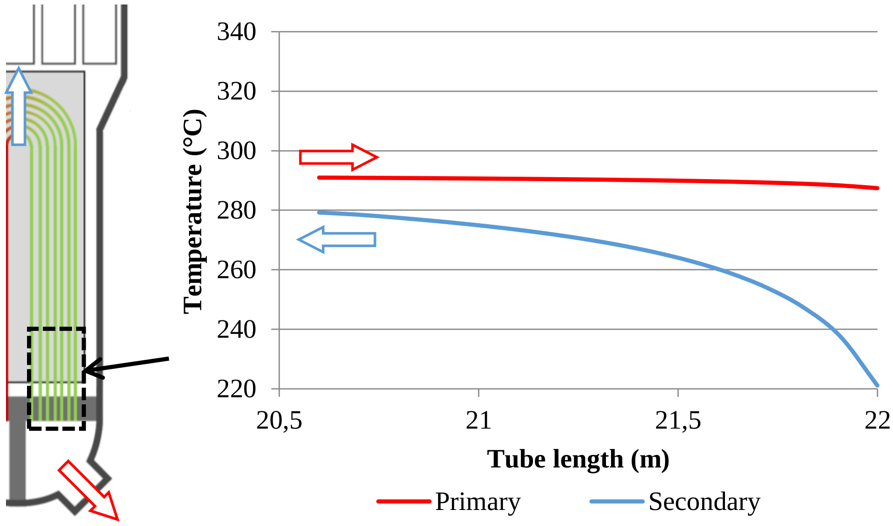

Figure 8: Temperature profile on the subcooled region of the Part 2.

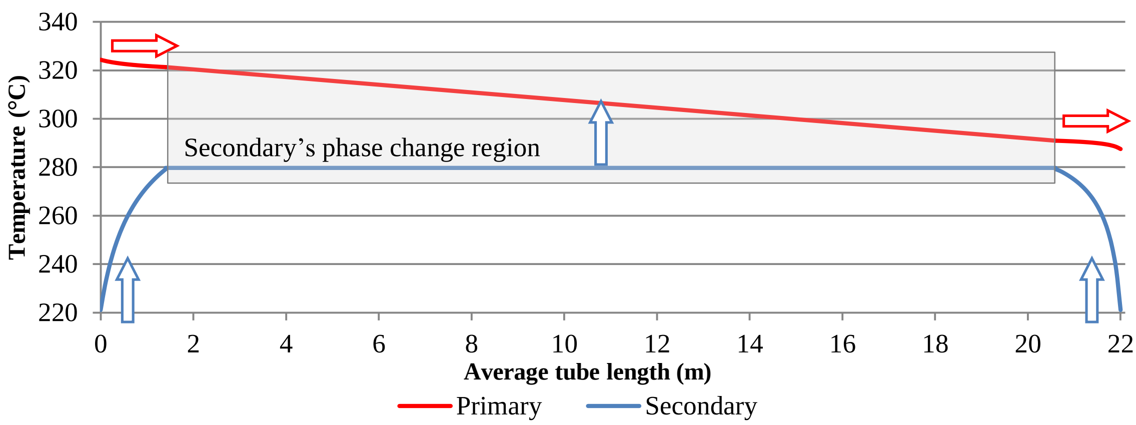

Figure 9: Overall Steam Generator temperature profile

3.2 Submarine:

To fit on a Scorpène Class submarine, with 11.7m diameter [6], the Angra 1 primary circuit was reduced on a 1 : 3.5 scale. Therefore, all parameters were recalculated, obtaining the average SG tube length as 6,3m.

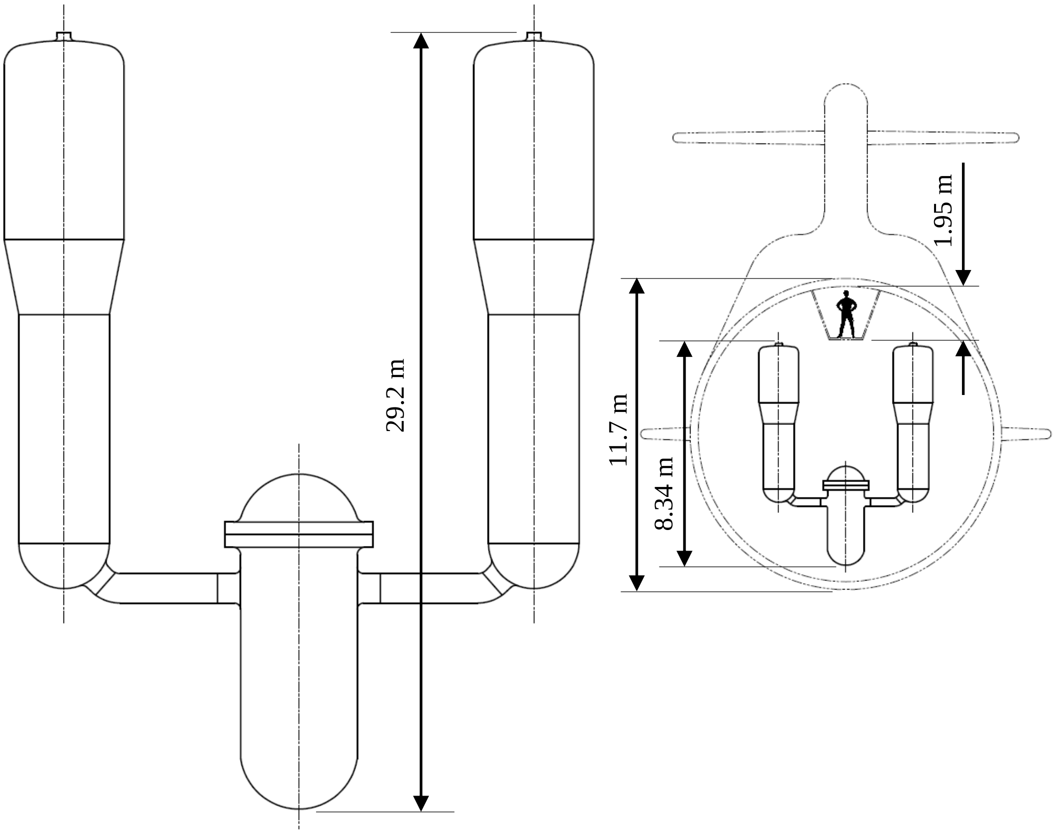

Figure 10: Illustrative scale reduction of Angra 1 primary circuit

The same analysis made to Angra 1, was made for the submarine SG. The heat rejected in each Steam Generator was defined as 42.85 MW_{th} with 23.41 kg/s of flow mass (values reached through thermodynamics first law), according to Chaves (2017).

Making iterations in *Microsoft Excel* software, it is possible to reach the number of tubes necessary to transfer the primary's heat. This value was defined in, approximately, 950 U-tubes.

Also following a triangular pitch, with 25.4 mm and 19,05 mm of tubes diameter and using some parameters from Tab. 1, the Nusselt numbers calculated to fluid conditions at SG inlet and outlet were, respectively:

$\mathbf{860.79} \leq \mathrm{Nu} \leq \mathbf{1050.69}$ – Primary fluid

$\mathbf{31.29} \leq \mathrm{Nu} \leq \mathbf{40.09}$ – Secondary fluid

Using the Eq. (5), it was possible to determine the convective coefficients to fluid conditions at inlet and outlet, respectively, as following:

$\mathbf{26.33\ kW/m^2 .°C} \geq h_p \geq \mathbf{28.45\ kW/m^2.°C}$ – Primary fluid

$\mathbf{1.52\ kW/m^2.°C} \leq h_s \leq \mathbf{1.80\ kW/m^2.°C}$ – Secondary fluid

The temperature profiles of the tubes in subcooled regions are shown in the Fig. 11 and Fig. 12. The length of the tubes required for the secondary fluid for beginning phase change was determined at approximately 0.435 m, as shown in Fig. 11 and Fig. 12.

The overall temperature profile in the Steam Generator can be modeled using the previous calculated data. This temperature profile can be seen in Fig. 13, as follows:

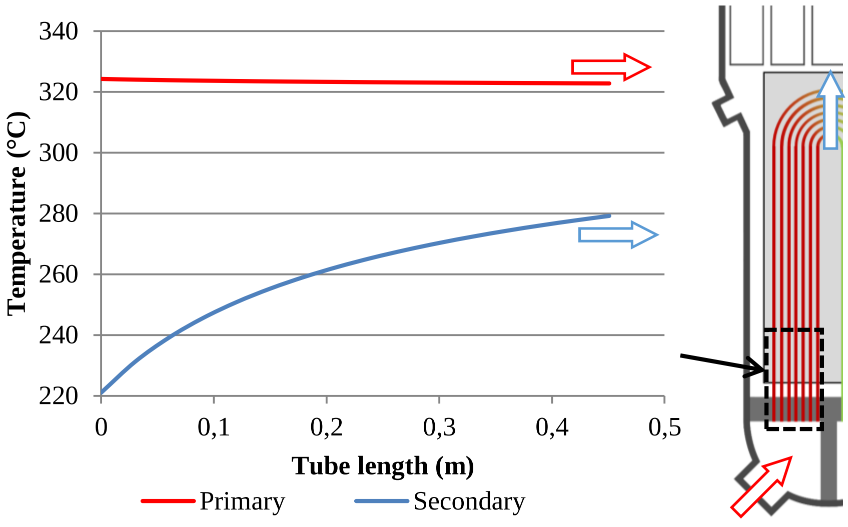

Figure 11: Temperature profile on the subcooled region on Part 1 of submarine SG.

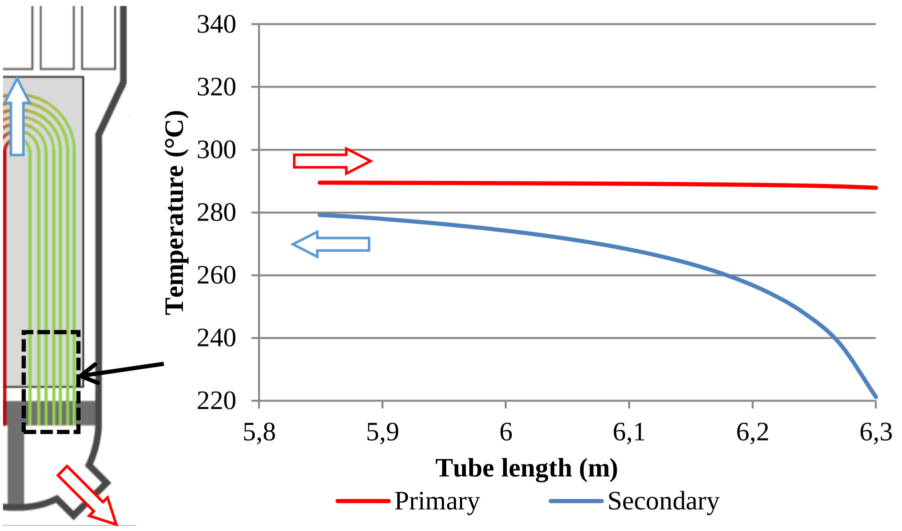

Figure 12: Temperature profile on the subcooled region on Part 2 of submarine SG.

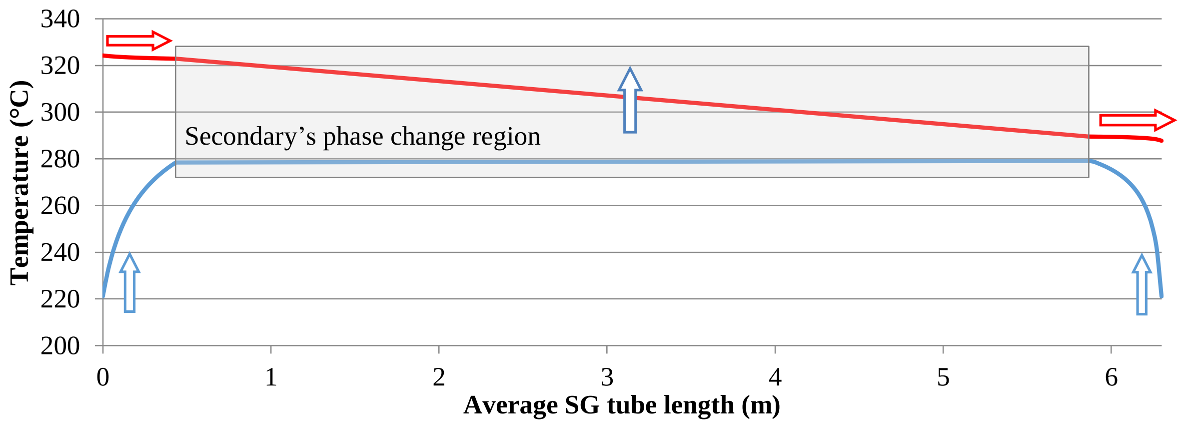

Figure 13: Overall temperature profile on submarine Steam Generator

3.3 Comparison between Angra 1 and the studied Steam Generator:

To evaluate the encountered results, a methodology called "Enthalpy Ruler" was developed to compare the defined lengths necessary to the secondary fluid to reach the saturation point. This methodology consists in a ruler with the values of constant variation of the secondary fluid enthalpy and the average length of the tubes.

A comparison made to evaluate the results is presented in Fig. 14, as follows:

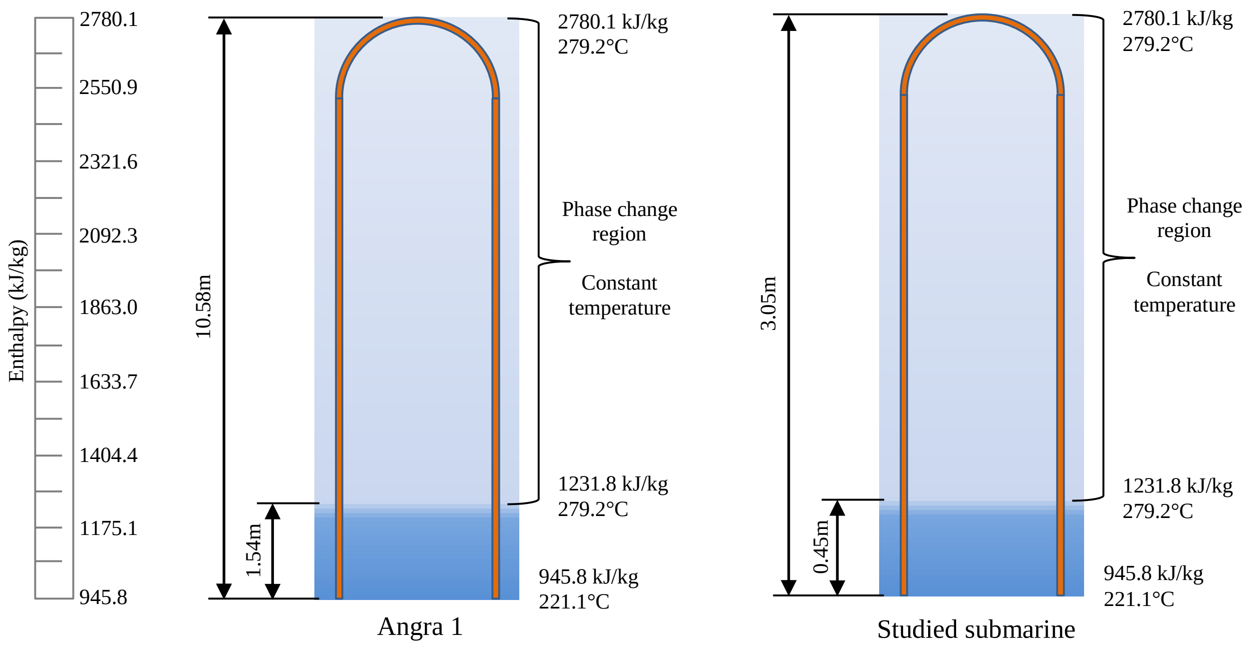

Figure 14: Enthalpy Rulers comparison for Steam Generators of Angra 1 and the studied submarine

Through the analysis of Fig. 14, it was possible to consider the encountered values as adequate results. Considering that the boiling causes a reduction in heat transfer coefficients, the length necessary to reach the saturation, obtained through the Rulers, tends to decrease, compared with in a real case, approaching to the encountered values. So the results could be considered with good approximation with the real values.

Fig. 15 represents the behavior of the secondary fluid through the water saturation curve, obtained by thermodynamics tables. It is important to note that the following behavior occurs inside the SG at the pressure of 64.7 kgf/cm². This kind of analysis, considering the temperature and enthalpy relation, shows that the constant variation of the enthalpy applied to phase change behaviors, makes sense while we are analyzing steady state conditions.

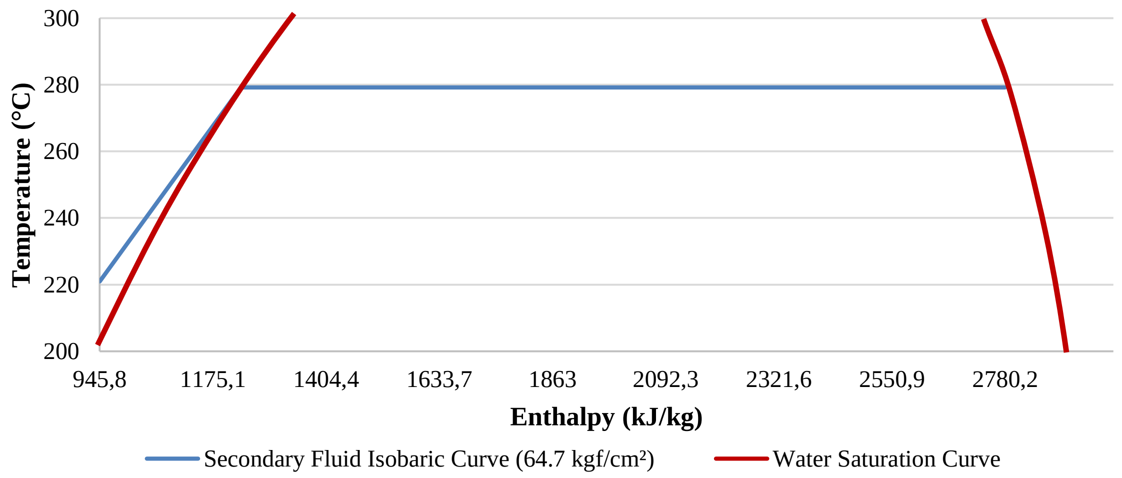

Figure 15: Behavior representation of the secondary fluid in water saturation curve

With the results of the analyzes, it was possible to develop the following table with comparative data between Angra 1 and studied Submarine, to evaluate the encountered results:

Table 2 – Comparative data of the studied nuclear submarine Steam Generator.

Steam Generator	*Submarine*	*Angra1*[1,4]
Feedwater temperature (°C)	221.1	221.1
Outlet vapor temperature (°C)	279.2	279.2
Secondary circuit pressure (kgf/cm²)	64.7	64.7
Secondary coolant average flow (kg/s)	23.41	515
Approximated number of tubes	990	5428
Tubes average length (m)	6.3	22.1
Heat transfer total area (m²)	373	7179
Tubes external diameter (mm)	19.05	19.05

4. CONCLUSION AND CONTRIBUTIONS OF THIS WORK

The methodology used for Angra 1 and extended to thermodynamic parameters of the submarine's Steam Generator proved to be satisfactory since the calculated parameters, such as heat transfer coefficients, mass flow, total area of heat transfer and the dimensionless parameters are enough to reject the heat of primary circuit. The heat rejected in the SG is comparable to others SG used for naval propulsion (Naval Technology, 2017).

Studies related to technology involving nuclear energy contribute to the Brazilian economic and technological development. According to Padilha and Wiltgen (2012), a technology transfer agreement signed between Brazil and France in 2009 provides for the construction of a Scorpène class nuclear submarine. It will be the first Brazilian nuclear submarine and its construction is an entry card for the select group of countries that dominate nuclear technology.

The methodologies applied to the Steam Generators analyzes were considered adequate. The finite differences method is a method largely used to analyze heat exchangers with no phase change. The method considering a constant variation of the enthalpy provide good approximations for the study of phase change in steady state conditions. So the methodology called "Enthalpy Ruler" can be used to evaluate if the encountered values provide results which make sense, avoiding large deviations when the results needs to be interpreted.

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