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Steam Generator Overall Heat Transfer Area Estimation for PWR NPP

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Abstract:

There are over 1,300 steam generators (SGs) installed in the 450 nuclear power plants worldwide. The technology in SG design been continuously improved through experience and the continuous research activities in the industry. The heat transfer phenomena have continuously been studied to improve on the effectiveness and efficiency of SG around the world. In most PWR plant recirculating steam generators (RSG) are in use basically shell and tube heat exchangers and mostly of the vertical orientation besides the VVER models which are horizontal in orientation. The most advanced PWR plants use RSG with and without economizer respectively. This forms the basis for this conceptual research problem to establish the effectiveness of the economizer on the reduction of the overall heat transfer area (OHTA) of the PWR SG. The OHTA was estimated in two steps where subcooled boiling region was evaluated by the general heat transfer equation while the saturated boiling region used the modified Rohsenow equation by Westinghouse. From the results this conceptual study the estimated OHTA of the PWR SG design is 17.15% higher since it is just a preliminary study. Optimization of the models is a topic for further studies through experiments. For the objectives of the study the economizer reduced the OHTA by 3.11% from 18,459.77m2 to 17,906.41m2 which on optimization might be further increased

Keywords: Steam Generators, SG Economizer, Overall Heat Transfer Area of SG, Heat Balance of SG

I. INTRODUCTION

Steam generators in pressurized water reactors (PWR) plant are shell and tube heat exchangers typically two to four per reactor [1]. The steam generator is used to (1) produce dry saturated steam for the turbine generator and auxiliary systems, (2) act as a heat sink for the reactor coolant system (RCS) during normal, abnormal, and emergency conditions, and (3) provide a reactor coolant pressure boundary avoiding release of the radioactive primary coolant into the secondary system[2], [3]. To produce steam the steam generator transfers the heat generated in the nuclear reactor core from the primary reactor coolant to the secondary feedwater through the u-tube walls to generate steam ensuring continuous removal of the steam from the core [2].

The rate of the heat transfer between the primary and the secondary side is dependent on the temperature gradient between the average coolant temperature and the saturation temperature of the steam generator, the overall heat transfer area of the u-tubes, and the overall heat transfer coefficient which is dependent on tube material, geometry, and the convective heat transfers of the tube inner and outer surfaces [2]. To effectively remove the heat generated special optimization of the overall heat transfer area of the u-tubes need to be effectively deter-mined[2], the material selection is done carefully to protect the tube from damage [1][4] while ensuring high rate of heat transfer[3], and standard geometries are being employed by many manufacturers of steam generators in the industry.

Steam generators in operation in PWR's are recirculating u-tube type and once through types [1]. The recirculating types only partial evaporation of feedwater hence generating saturated steam while the once through type allow for 100% evaporation of feedwater generating superheated steam[5]. They can also be either vertical like the APR1400 [3] and AP1000[2] or horizontal like the Russian VVER models.

In case of the Advanced Pressurized Reactor 1400 (APR1400) has two steam generators (SG) designed to transfer 4000MWt from the primary reactor coolant system (RCS) to the secondary main steam system (MSS) and power conversion system. The heat is transferred from the primary to secondary system through SG tubes are Ni-Cr-Fe alloy 690 (Inconel 690); 3/4-inch OD and with a 0.042-inch nominal wall thickness provided with a plugging margin of 10%. The reactor coolant enters the SG through a 42inch inlet nozzle, flows through the u-tubes, and leaves through two 30inch outlet nozzles both at the bottom spherical head. The SG has an integral economizer (Fig 1 and Fig 2) which is a separation of the cold side of the tube bundle from the hot side through vertical divider plates. This allows feedwater to be preheated before its introduction to the evaporator region of the SG [3] hence increasing efficiency of heat transfer. The steam quality is improved to a limit of maximum moisture content of 0.25wt % by moisture separators and steam driers in the shell side during normal operation at full power. The pressure drop on the secondary side from the feedwater nozzles to the steam nozzles is 42psid (2.981kg/cm2). The steam generator design parameters are listed in Table 1 [3].

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Table 1. A typical steam generator main parameters.

Parameter	Value
Primary Side	
Design pressure/temperature (psia/ °F) (kg/cm^2 / °C)	2,500/650 (175.76/343.33)
Coolant inlet temperature, °F (°C)	615 (323.88)
Coolant outlet temperature, °F (°C)	555 (290.55)
Coolant flow rate, each, lb/hr (kg/hr)	83.3 x 10^6 (37.78 x 10^6)

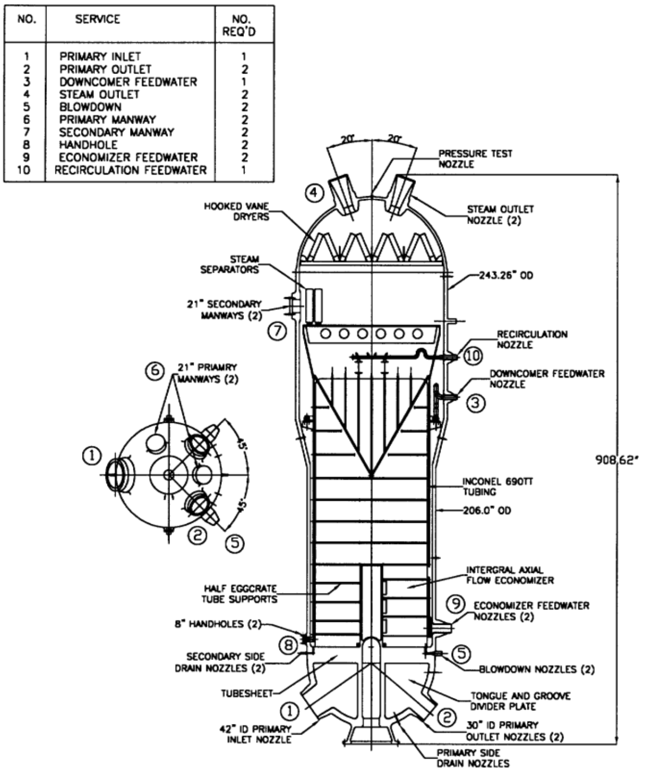

Figure 1. A typical PWR SG major components

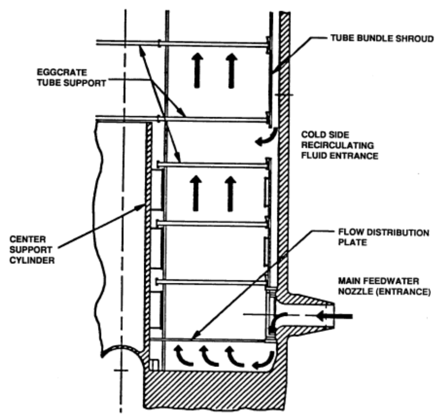

Figure 2. A cross sectional view of SG economizer region

The APR1400 SG designed in compliance with all legal, regulatory requirements, codes, and standards [3] is one of the most recent advanced power reactors in the industry. One of the most distinguishing features is that it has an integral economizer at the cold exit side of the u-tubes as described above. The economizer is a preheater that helps in the improvement of the thermal efficiency in the steam generator by preheating the fluid before it enters the evaporator section [5]. Other more recent high output models like the AP1000 do not possess such a feature but rather consist of just the evaporator region on the lower side of the steam generator [2]. The AP1000 SG receives feedwater from the main feedwater nozzles through a feedwater ring which combines with the recirculating water from the moisture separators and flows down the downcomer into the evaporator region [2]. The main objective of this project is to understand the concept in the design of SGs which would be used as an input in conceptual design of smaller capacity SGs while establishing the effect of the economizer provision on the APR1400 SG on OHTA

II. REVIEW OF HEAT TRANSFER IN PWR STEAM GENERATORS

The main purpose of the steam generator is to produce steam for the turbine, using the heat generated by the core and carried by the primary coolant. This function is vital to safety of the reactor core as it ensure continuous removal of heat that is generated thus preventing core damage due to elevated temperatures. The basic modes of heat transfer are conduction, radiation, and convection. The temperature gradient, medium (solid, liquid, gas, or vacuum) determines the mode of heat transfer, where in some cases; all these forms of heat transfer may take place at the same time.

II-I. Conduction heat transfer

Conduction is the transfer of energy from more energetic particles to less energetic particles due to interaction between the particles [7]. At molecular level, interactions of gas and liquid molecules or the lattice vibrations in solids enhance the heat transfer in the media [7]. In the steam generators heat is generated on the primary side and transferred to the secondary side using the heat conduction mechanism in the wall of tubes in the bundle. While designing tubes, engineers select materials that have high thermal conductivity in order to improve heat transfer[8]. The conduction heat transfer process is quantified in terms of appropriate rate equations per unit time. Conduction follows the Fourier law where for one dimensional plane wall having a temperature distribution T(x) [7].

$$q'' = -K\frac{dT}{dx} \quad (1)$$

Where

K is the thermal conductivity of the medium ($\frac{W}{mK}$)

II-II. Convection heat transfer

Convection occurs by random molecular motion (diffusion)

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and the bulk macroscopic motion of the fluid. In this study the interest is in the convection heat transfer between a fluid in motion and a bounding surface when the two are at different temperatures. The fluid surface interaction leads to the development of a hydrodynamic, velocity, or boundary layer where the fluid velocity varies from zero (no slip condition) at the surface to a finite value associated with the flow. The convection heat transfer is sustained by random molecular motion and the bulk fluid motion within the boundary layer. Therefore convection heat transfer can be classified by the nature of flow as (1) ***Forced convection*** when the flow is caused by external means like fan, pumps or atmospheric winds (2) ***Free or natural convection*** when the flow is induced by buoyancy forces due to density differences caused by temperature variations in the fluid [7]. Regardless of the nature of the heat transfer process, the appropriate rate equation for convection is of the form [7], [8].

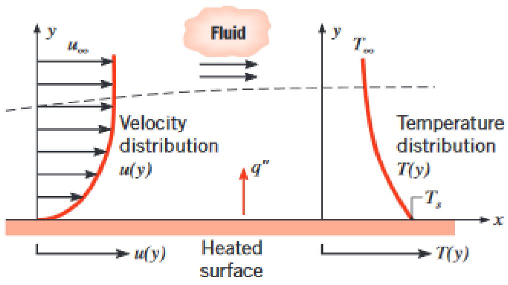

Figure 3. Boundary layer development in convection heat transfer

$$\ddot{q} = h(T_s - T_\infty) \quad (2)$$

Where h is the convection heat transfer coefficient $\left(\frac{W}{m^2K}\right)$,

T_s and T_∞ are the surface and bulk temperatures (K).

In the steam generator design, both forced and free convection occur. Forced convection occurs since the liquid on the primary side of the SG is pumped by RCPs, through the reactor core, before it enters the SG tube bundles. On the secondary side of SG, both free and forced convection are driving heat transfer. In the economizer region, since the fluid is pumped by the feedwater pumps, forced convection heat transfer occurs. On the other hand, in the evaporator region, the heat transfer mechanism can be described is free convection.

II-III. Boiling heat transfer in Steam Generators

Boiling occurs when a surface is exposed to a liquid and maintained at a temperature above the saturation temperature of the liquid with the heat flux dependent on the temperature difference between the surface and the saturation temperature [8]. Pool boiling occurs when the heated surface is submerged below the free surface of the liquid. Subcooled (local) boiling is when temperature is below saturated temperature while bulk (saturated) boiling occurs if the temperature is maintained at saturated temperature [8].

Boiling occurs in distinct regions as indicated in Fig 4 below where in region I free convection is responsible for fluid motion near the surface since the fluid near the surface is superheated slightly leading to its rise hence the region can be evaluated using free convection correlations[8]. Bubbles form on the surface and are dissipated in the fluid after breaking away from the surface in region II marking the onset of nucleate boiling. There is a rapid formation of bubbles which rise to the fluid surface on region III. Eventually the bubbles are too many that they blanket the heating surface which leads to the increase of thermal resistance and hence reduction of the heat flux marked by region IV. This is a very unstable region as it marks the transition between the nucleate boiling and the stable film boiling in region V. The surface temperature required to maintain a stable film boiling are high, and once the condition is reached most of the heat transfer is by thermal radiation as indicated in region VI [8].

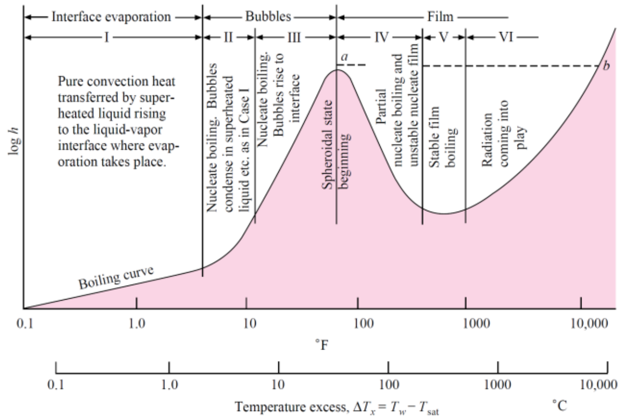

Figure 4. Boiling regimes from Farber and Scorah experiment [7]

Rohsenow had correlated experimental data for nucleate pool boiling with the following relation [7][8],

$$\frac{C_l \Delta T_x}{h_{fg} {Pr_l}^s} = C_{sf}\left[\frac{q/A}{\mu_l h_{fg}}\sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}}\right]^{1/3} \quad (3)$$

where;

C_l = Specific heat of saturated liquid, $\left[\frac{BTU}{lbm°F}\right]$ or $\left[\frac{J}{kg°C}\right]$

ΔT_x = Excess temperature = $T_w - T_{sat}$ [°F] or [°C]

h_{fg} = enthalpy of vaporization, $\left[\frac{BTU}{lbm}\right]$ or $\left[\frac{J}{kg}\right]$

Pr_l^s = Prandtl number of saturated liquid

q/A = heat flux per unit area, $\left[\frac{BTU}{ft^2h}\right]$ or $\left[\frac{W}{m^2}\right]$

μ_l = liquid viscosity, $\left[\frac{lbm}{ft\,h}\right]$ or $\left[\frac{kg}{m\,s}\right]$

σ = surface tension of liquid-vapor interface, $\left[\frac{lbf}{ft}\right]$ or $\left[\frac{N}{m}\right]$

g = gravitational acceleration, $\left[\frac{ft}{s^2}\right]$ or $[\frac{m}{s^2}]$

ρ_l = density of saturated liquid, $\left[\frac{lbm}{ft^3}\right]$ or $\left[\frac{kg}{m^3}\right]$

ρ_v = density of saturated vapor, $\left[\frac{lbm}{ft^3}\right]$ or $\left[\frac{kg}{m^3}\right]$

C_{sf} = constant, determined from experimental data

s = 1.0 for water and 1.7 for other liquids

This equation can be rearranged to

$$T_w - T_{sat} = \frac{h_{fg}}{c_{p,l}} Pr_l^s C_{sf} \left[\frac{q''}{\mu_l h_{fg}} \left(\frac{\sigma}{g(\rho_l - \rho_v)}\right)^{1/2}\right]^{1/3} \quad (4)$$

Also the heat transfer in the evaporator region can be expressed in terms of heat transfer coefficient as

$$q'' = h(T_W - T_{sat}) \quad (5)$$

Where the heat transfer coefficient is obtained after substituting eq (4) into (5) as

$$h = K_R(q'')^{2/3} \quad (6)$$

$$K_R = f(p_{sat}) = a_1 + a_2 p + a_3 p^2 = b_1 \exp\left(\frac{p}{b_2}\right) \quad (7)$$

The constants in the above equation can be determined through experimental work. An example is the Westinghouse calculation where they determined the constant K_R is determined by the equation [5].

$$K_R = 4.1\{1 + 1.4 \times 10^{-4}(p_{sat} - 800)\} \quad (8)$$

Where the unit of p_{sat} is in psig.

Hence reducing the Rohsenow equation to [5]:

$$h = 3.97[1 + 1.41 \times 10^{-4}(P_{sat} - 800)](q'')^{2/3} \quad (9)$$

Other correlations that have been developed for the nucleate pool boiling convective heat transfer coefficient include:

Jens and Lottes: $h = \frac{10^{3/2}}{60} \exp\left(\frac{p_{sat}}{900}\right)(q'')^{3/4}$ (10)

Weatherhead: $h = \frac{10^{3/2}}{0.18(500 - 0.707\, T_{sat})}(q'')^{3/4}$ (11)

Thom: $h = \frac{1}{0.072} \exp\left(\frac{p_{sat}}{1260}\right)(q'')^{1/2}$ (12)

Levy: $h = \left(\frac{p_{sat}^{4/3}}{495}\right)(q'')^{2/3}$ (13)

III. THEORETICAL DEVELOPMENT OF TOTAL HEAT TRANSFER AREA ESTIMATION

III-I. Energy Balance and Thermal Sizing

Heat generated by the nuclear fission in the reactor is equal to the mass flow rate through the reactor multiplied by the change in the enthalpy of the coolant for normal operating condition of reactor. Heat generated in the reactor is expressed by the equation.

$$Q = \dot{m}(h_{in} - h_{out}) \quad (1)$$

Thus

$$Q_p = \dot{m}_p(h_{hot} - h_{cold}) \quad (2)$$

The heat generated in the reactor is transferred to the secondary side through the steam generator U-tubes. The heat transferred in these U-tubes is calculated as a function of the mass flow rate, the isobaric heat transfer rate at the average temperature, and the change in temperature at the entry and exit of the SG. Heat transferred in SG from primary is expressed by the equation.

$$Q = \dot{m}_p c_p \, \Delta\mathrm{T} \quad (3)$$

Thus

$$Q = \dot{m}_p \, c_{p_c}^{\;h} \, (T_{hot} - T_{cold}) \quad (4)$$

where $c_{p_c}^{\;h}$ = isobaric heat capacity at the average of hot and cold temperature

The heat removed from the primary RCS is calculated as a function of the mass flow rate in the secondary side ($\dot{m}_s$) and the change in enthalpy between the feed-water and the steam.

$$Q_s = \dot{m}_s(h_s - h_{fw}) \quad (5)$$

Figure 5 represents the fluids flow parameters with the left side indicating the inlet and outlet temperature of coolant as it passes through the core as it enters through cold leg and leaves through hot leg. The central part represents the steam generator showing the primary coolant temperature reducing as it passes through the steam generator tubes. The secondary side shows different behavior since in the first section. There is a rise in temperature of the feed-water as it passes through the economizer region and then it remains constant at the saturation temperature of steam.

The right side represents the turbine and the condenser. In the turbine, the steam expands and losses heat as it drives the turbine leading to a temperature reduction. The steam from the turbine is discharged to the condenser where is converted back to feed-water by removing heat and use of water from the ultimate heat sink.

Approximation of the total heat transfer area can be calculated by calculating the overall heat transfer in the steam generator which is evaluated from the equation **Error! Reference source not found.**. Heat flows in the steam generator from the hot coolant by convection, through the wall by conduction, and then by convection from the surface to the cold fluid. In the three regions the heat flow is calculated as:

Region I: hot liquid to solid convection

$$q_x = h_{hot}(T_h - T_{iw})x\, A_i \quad (6)$$

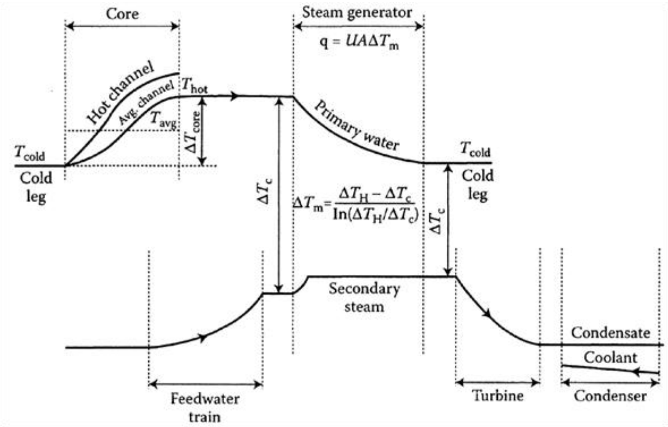

Figure **5.** A depiction of the fluid flow

Region II: conduction across the wall,

$$q_x = \frac{2\pi Lk(T_{ow} - T_{iw})}{Ln\frac{D_o}{D_i}} \quad (7)$$

Region III: solid to cold liquid convection

$$q_x = h_{cold}(T_{ow} - T_c)x\, A_o \quad (8)$$

From the three equations above the overall heat transfer is determined as:

$$q_x = UA(T_h - T_c) \quad (9)$$

Generally expressed as:

$$q_x = UA_o\text{LMTD} \quad (10)$$

Where LMTD is calculated from the equation **Error! Reference source not found.**:

$$LMTD = \frac{\Delta T_h - \Delta T_c}{\ln\frac{\Delta T_h}{\Delta T_c}} \quad (11)$$

And U is the Universal Heat Transfer Coefficient calculated from:

$$U = \frac{1}{R_i + R_w + R_o} \quad (12)$$

Where:

$$R_i = \frac{d_o}{h_i d_i} \quad (13)$$

$$R_w = \frac{d_o}{2k_w} ln\,\frac{d}{d_i} \quad (14)$$

$$R_o = \frac{1}{h_o} \quad (15)$$

Where h_i is calculated from the Dittus-Boelter equation for heat transfer presented as follows:

$$h = 0.023\,\frac{k}{D_h} Re^{0.8} Pr^n \quad (16)$$

Where n is 0.3 for primary side (cooling) and 0.4 for secondary side (heating)

From the calculations above then the overall heat transfer area can be estimated as:

$$A_o = \frac{q_x}{U\ LMTD} \quad (17)$$

A more precise calculation of the heat transfer are can be calculated by subdividing heat transfer area into evaporator region and economizer region.

a. Heat transfer area calculation for evaporator region

Evaporator region of heat transfer is characterized by the secondary side saturated condition, hence, temperature maintains constant boiling temperature. A small segment of heat transfer tube is shown in Fig 6 below which shows heat flow from hot water of primary side to the secondary side cold water. This region is dictated by Rohsenow's correlations.

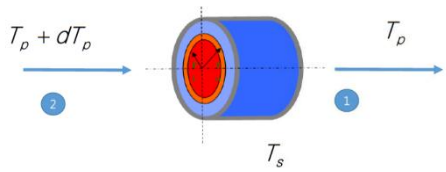

Figure 6. A differential tube element

A modified Rohsenow equation developed through experiment applicable to this case is

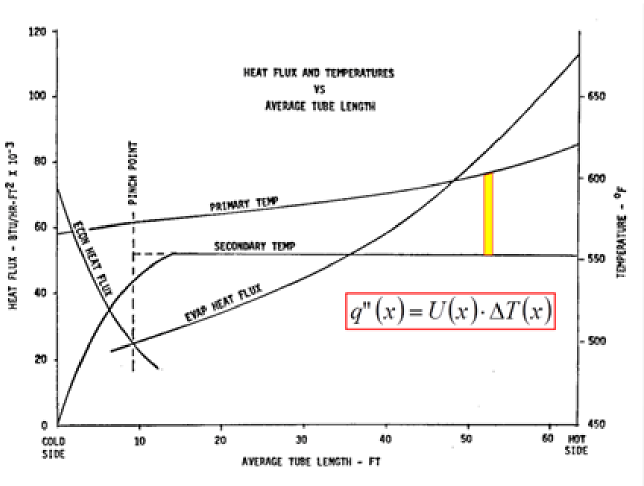

Figure 1 Heat flux and Temperature against Average Tube length

$$h_s = K_s \cdot \left(\frac{q}{A}\right)^{2/3} \quad (18)$$

Where K_s is a constant that is a function of the saturation pressure and is calculated as,

$$K_s = 4.1\{1 + 1.4 \times 10^{-4} \cdot (p_{sat} - 800)\} \quad (32)$$

In the differential element above, the heat transfer can be calculated as follows,

$$dq = \dot{m}_p c_p dT_p \quad (19)$$

But the total temperature difference through primary to secondary is

$$\Delta T_t = T_p - T_s \tag{20}$$

And since secondary side is saturated, the steam temperature, T_s, is constant.

$$d(\Delta T_t) = dT_p \tag{21}$$

Hence the equation of heat transfer from primary side above becomes:

$$dq = \dot{m}_p c_p d(\Delta T_t) \tag{22}$$

If R_r is the summation of all the resistances, R_r, except the boiling film resistance, R_o, from (22), and it is rewritten in different form.

$$q = UA(T_h - T_c) \text{ where } U = \frac{1}{R_r + R_o} \tag{22}$$

$$R_r q + R_o q = A(T_p - T_s) = A\Delta T_t$$

$$\Rightarrow R_r dq + R_o dq = dA \Delta T_t$$

$$\Rightarrow \Delta T_t = R_r \frac{dq}{dA} + R_o \frac{dq}{dA}$$

For boiling heat transfer, using Rohsenow correlation,

$$R_o = \frac{1}{h_o} \text{ and } h_o = K_s \left(\frac{q}{A}\right)^{2/3}, \text{ and } R_o = \frac{1}{K_s}\left(\frac{q}{A}\right)^{-2/3}$$

Where for infinitesimal area $\frac{q}{A} = \frac{dq}{dA}$

$$\Rightarrow R_o \frac{dq}{dA} = \left(\frac{1}{K_s}\right)\left(\frac{dq}{dA}\right)\left(\frac{dq}{dA}\right)^{-2/3} = \left(\frac{1}{K_s}\right)\left(\frac{dq}{dA}\right)^{1/3}$$

Using the modified Rohsenow equation for the film boiling in above:

$$\Delta T_t = R_r \left(\frac{dq}{dA}\right) + \left(\frac{1}{K_s}\right)\left(\frac{dq}{dA}\right)^{1/3} \tag{37}$$

Differentiation of equation **Error! Reference source not found.**) by dq:

$$\frac{d\Delta T_t}{dq} = R_r \frac{d}{dq}\left(\frac{dq}{dA}\right) + \left(\frac{1}{K_s}\right)\frac{d}{dq}\left(\frac{dq}{dA}\right)^{1/3} \tag{23}$$

Where $\frac{d\Delta T_t}{dq} = \frac{1}{\dot{m}_p C_p}$ (24)

And replacing into the equation,

$$\frac{1}{\dot{m}_p c_p} = R_r \frac{d}{dq}\left(\frac{dq}{dA}\right) + \left(\frac{1}{K_s}\right)\frac{d}{dq}\left(\frac{dq}{dA}\right)^{1/3} \tag{25}$$

Replacing $\left(\frac{dq}{dA}\right)$ with S and multiplying the denominator by$\left(\frac{dA}{dA}\right)$

$$\frac{1}{\dot{m}_p c_p} = R_r \frac{dS}{dq\left(\frac{dA}{dA}\right)} + \left(\frac{1}{K_s}\right)\frac{dS^{1/3}}{dq\left(\frac{dA}{dA}\right)} \tag{41}$$

$$\frac{1}{\dot{m}_p c_p} = R_r \frac{dS}{dA\left(\frac{dq}{dA}\right)} + \left(\frac{1}{K_s}\right)\frac{dS^{1/3}}{dA\left(\frac{dq}{dA}\right)} \tag{42}$$

$$\frac{dA}{\dot{m}_p c_p} = R_r \frac{dS}{S} + \left(\frac{1}{K_s}\right)\frac{dS^{1/3}}{S} \tag{43}$$

$$\frac{A}{\dot{m}_p c_p} = R_r \int_1^2 \frac{dS}{S} + \left(\frac{1}{K_s}\right)\int_1^2 \frac{dS^{1/3}}{S} \tag{44}$$

$$\frac{A}{\dot{m}_p c_p} = [R_r \ln S]_1^2 + \left[\left(\frac{1}{K_s}\right)\frac{-S^{-2/3}}{2}\right]_1^2 \tag{45}$$

Re-arranging the formula and replacing S,

$$\frac{A}{\dot{m}_p c_p} = R_r \left[ln\left(\frac{dq}{dA}\right)_2 - ln\left(\frac{dq}{dA}\right)_1\right] + \left(\frac{1}{2K_s}\right) \cdot$$

$$\left[\left(\frac{dq}{dA}\right)_1^{-2/3} - \left(\frac{dq}{dA}\right)_2^{-2/3}\right] \tag{26}$$

And finally the equations becomes,

$$A = \left(\frac{q}{\Delta T_p}\right)\left\{R_r \ln\left[\frac{\left(\frac{dq}{dA}\right)_2}{\left(\frac{dq}{dA}\right)_1}\right] + \left(\frac{1}{2K_s}\right)\left[\left(\frac{dq}{dA}\right)_1^{-2/3} - \left(\frac{dq}{dA}\right)_2^{-2/3}\right]\right\} \tag{27}$$

Where the subscript 1 refers to the primary outlet and 2 to the primary inlet.

This methodology is actualized by establishing the pinch point temperature and dividing the portion of the evaporator region into small differential elements. With the finer the elements, we can get more accurate result. Iterations for next sections are done and finally the summation of all the elemental areas is done. The economizer region is also evaluated using the same methodology, but using the average temperatures for the temperature differences.

b. Heat transfer area calculation for economizer region

Heat transfer in the economizer on the secondary side,

$$Q_{eco} = \dot{m}_s(h_s - h_{fw}) \tag{28}$$

Where $\dot{m}_s$ is the flow rate of steam or feed water (secondary side flow rate).

And the heat transfer in the primary side equals to the secondary side,

$$Q_{eco} = \dot{m}_p \, c_p \, (T_e - T_c) \tag{29}$$

Where: $\dot{m}_s$ is the coolant flow rate of steam generator, and T_e is inlet temperature of primary side economizer and T_c is outlet temperature of primary side (i.e. outlet of economizer).

Hence the pinch point temperature is the temperature of economizer inlet,

$$\frac{Q_{eco}}{\dot{m}\, C_p} + T_c = T_{eco} \tag{30}$$

The T_{eco} is used for the calculation of heat transfer area applying (47).

IV. IMPLEMENTATION OF THEORY AND DETERMINATION OF THE HEAT TRANSFER AREA

This section describes the implementation of the theory developed in the previous section. The procedure is divided in to number of steps as follows.

1. Primary side heat transfer
2. Heat transfer through the U-tube wall
3. Secondary side heat transfer

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   a. Bulk boiling area
   b. Area of the economizer region
4. Estimation of total heat transfer area

IV.I Primary Side Heat Transfer

For the primary side of the Steam Generator, the coolant was assumed to be in subcooled state, thus all parameters were calculated as for the subcooled liquid using Chemica Logic Steam Tab Companion [6]. In order to determine heat transfer in that region, it is important to calculate Nusselt number, which provides a measure of convection heat transfer occurring at the inner surface of the U-tube. The general form of the Nusselt number can be expressed by the equation:

$$Nu = f(x^*, Re_L, Pr) \tag{31}$$

Where:

x^* = spatial variable

Re_L = Reynolds numbers, which specifies flow condition

Pr = Prandtl number, characterizing heat transfer between moving fluid and solid body

In the Steam Generator U-tube, the flow condition is described as fully developed turbulent flow in the smooth tube. Thus, the correlation used for convective heat transfer coefficient can be expressed using Dittus-Boelter correlation [7]:

$$Nu = 0.023\, Re_L^{0.8} Pr^{0.3} \tag{32}$$

Temperature which was used for necessary parameters was assumed to be average temperature of the fluid of the primary side and is expressed by the equation:

$$T_{avg} = \frac{T_H + T_C}{2} \tag{33}$$

Where:

T_H = temperature of the hot leg (input temperature of the primary side)

T_C = temperature of the cold leg (output temperature of the primary side)

IV.II Heat Transfer through the U-Tube Wall

Heat transfer through the U-tube wall is a simple calculation expressing thermal resistance for conduction through a solid wall of the U-tube using Fourier's Law [3],

$$R_k = \frac{d_o}{2k} \ln\left(\frac{d_o}{d_i}\right) \tag{34}$$

Where:

k = thermal conductivity of the Inconel 690

d_o = outer diameter of the U-tube

d_i = inner diameter of the U-tube

IV.III Secondary Side Heat Transfer

In order to determine heat transfer in the secondary side of the Steam Generator, it was important to calculate first bulk boiling area and in the next step to determine pinch temperature

a. Bulk boiling area

This region was divided to 10 segments in order to calculate each segment temperature and heat transfer area, and subsequently total heat transfer area in the bulk boiling region. In this evaluation Rohsenow modified equation by experiment was used.

$$K_s = 4.1\{1 + 1.4 \times 10^{-4}(p_{sat} - 800)\} \tag{35}$$

This equation is taking saturation pressure p_{sat} of steam in $psia$ units. However, because K_s is a constant, it was not required to convert the units.

b. Economizer region heat transfer area

In economizer region, both inside and outside is saturated, hence heat flows From the primary side convective heat transfer on the wall, conductive heat transfer through U-tube wall then convective heat transfer on the secondary side of the wall. In this region, the both side are the force convection. The heat transfer is calculated as following.

$$Q_{eco} = UA\Delta T = \dot{m}(h_e - h_c) \tag{36}$$

And $UA = \frac{\dot{m}(h_e - h_c)}{\Delta T_m} = \frac{Q_{eco}}{\Delta T_m} = 9{,}593{,}017.04 \left[\frac{J}{s°C}\right]$ (37)

Where the rate of Heat transfer through Economizer region as,

$$Q_{eco} = \dot{m}\,(h_e - h_c) = 2.75 \times 10^8 \left[\frac{J}{s}\right] \tag{38}$$

- Temperature difference between inlet and outlet

$$\Delta T_m = \frac{\Delta T_H - \Delta T_C}{ln\frac{\Delta T_H}{\Delta T_C}} = 28.629\ [°C] \tag{39}$$

- ΔT_H depends on the coolant outlet temperature and the feed-water temperature

$$\Delta T_h = T_c - T_{fw} = 290.55 - 232.22 = 58.33\ [°C] \tag{40}$$

- ΔT_C depends on the pinch temperature and the saturated steam temperature,

$$\Delta T_C = T_e - T_{sat} = 295.473 - 284.2 = 11.273[°C] \tag{41}$$

i) Convective heat transfer on the primary side of economizer region:

Since primary coolant flow in tube and in single phase, Dittus-Boelter correlation is used for the primary side of the economizer: $R_i = \frac{d_o}{h_i d_i}$

$$\frac{h_i d_i}{k} = 0.023\left(\frac{G_i d_i}{\mu}\right)^{0.8}\left(\frac{C_p \mu}{k}\right)^{0.3} \tag{79}$$

Mass flow rate per square area:

$$G_i = \frac{\dot{m}_p}{A_x} = \frac{\dot{m}_p}{\frac{n\pi d_i^2}{4}} = 4669.32 \left[\frac{kg}{m^2 s}\right] \quad (80)$$

From equation **Error! Reference source not found.** convective Heat Transfer Coefficient (primary side):

$$h_i = 0.023 \frac{k}{d_i}\left(\frac{G_i d_i}{\mu}\right)^{0.8}\left(\frac{c_p \mu}{k}\right)^{0.3} = 42303.931 \left[\frac{W}{m℃}\right] \quad (81)$$

So the resistance coefficient is determined by (26) as following:

$$R_i = \frac{d_o}{h_i d_i} = 2.662 \times 10^{-5} \quad (82)$$

ii) Conductive heat transfer through the tube wall.

$$R_w = \frac{d_o}{2k_w} ln\ \frac{d_o}{d_i} \quad (83)$$

$$R_w = \frac{d_o}{2k_w}\ln\left(\frac{d_o}{d_i}\right) = 6.54 \times 10^{-5} \quad (84)$$

With thermal conductivity of Inconel 690:

$$k_w = 17.3 \left[\frac{W}{m℃}\right]$$

iii) Convective heat transfer on the secondary side of economizer: $R_o = \frac{1}{h_o}$

Using McAdams correlation for secondary side (force convection region) as following:

$$Nu = \frac{h_o D_e}{k} = 0.36\left(\frac{G_s D_e}{\mu}\right)^{0.55}\left(\frac{c_p \mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (85)$$

Where D_e is hydraulic diameter of wetted area.

For triangular pattern of tubes: $D_e = \frac{4\left(\frac{P_T^2\sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\pi d_o/2}$

For square pattern of tubes: $D_e = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o}$

Shell side mass velocity:

$$G_s = \frac{\dot{m}_s}{A_s} = 500.626 \left[\frac{kg}{m^2 s}\right] \quad (86)$$

Mass flow rate of feedwater: $m = 1142.800852 \left[\frac{kg}{s}\right]$

Cross flow area at the center of the shell:

$$A_s = \frac{\pi D_{SG}^2}{8} - \frac{n\pi d_o^2}{4} = 2.283\ [m^2] \quad (87)$$

Inside Diameter of Shroud: $D_{SG} = 3.615m$

For the squared tube arrangement, thus the hydraulic diameter is:

$$D_e = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o} = 0.02407\ [m] \quad (88)$$

For the triangular tube arrangement, thus the hydraulic diameter is:

$$D_e = \frac{4\left(\frac{P_T^2\sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\pi d_o/2} = 0.018293344\ [m] \quad (89)$$

The tube pitch: $P_T = 0.0254m = 1inch$

μ_w is viscosity at the wall temperature

Substitute the values of parameter into **Error! Reference source not found.**:

$$h_o = 5397.111 \left[\frac{W}{m℃}\right] \quad (90)$$

Thus, the heat transfer resistance coefficient is determined as following:

$$R_o = \frac{D_o}{D_i h_o} = 0.000208654 \quad (91)$$

Now with the known parameter value of R_o, we can calculate economizer region heat transfer area. From (82), (84), (91) into (25)

$$U = \frac{1}{R_i + R_w + R_o} = 3325.87 \left[\frac{W}{m℃}\right] \quad (92)$$

From **Error! Reference source not found.** and **Error! Reference source not found.** the area of the economizer region

$$A = \frac{UA}{U} = 1132.369\ [m^2] \quad (93)$$

IV. DISCUSSIONS AND CONCLUSIONS

In a SG, thermal performance is a key critical metric and is the main concept behind the conceptual design of the APR1400. For the present project, thermal performance used the basic assumptions in the heat balance equations for convection between fluid and wall in the primary side, between wall and fluid in the secondary side, and conduction in the u-tube wall. The process started by establishing the heat balance in both primary and secondary side. Calculated heat capacity on the primary side from the **Error! Reference source not found.** is $1{,}977.31MWt$. This value is less by $23MWt$ (approximately 1.15%) than the APR1400 rated capacity of $2000MWt$ per steam generator. On the other hand, using equation **Error! Reference source not found.**, the secondary side heat rate was $2{,}007.86MWt$, which is 0.4% higher than the rated value. These margins of error could be attributed to the human errors like the use of average temperatures and discrepancies in the steam table values.

From the calculations based on equation **Error! Reference source not found.**, the economizer portion of the APR1400 accounts for $292MWt$ which is about 14.6% of the total heat in the steam generator using an initial estimation of 10,000 tubes. The evaporator region of the steam generator accounts for the $1{,}715MWt$, which is 85.6% of the total heat transfer in the secondary side. To evaluate the heat transfer area, the assumptions made in **Error! Reference source not found.-Error! Reference source not found.** were used. In the evaluation of the differential elements of the evaporator region, it was noted that the heat flux was the highest at the entry region into the SG, hence a lower heat transfer area of 776.5m^2 which increases gradually to $3{,}679.25m^2$ at the economizer exit (pinch point) for 12,000 tubes. The heat transfer area calculations for the economizer used the different way with for the evaporated region. The accuracy of the value of the heat transfer area was affected by the variable temperatures in both the primary and secondary side, hence fluid properties, which complicate the calculations. To simplify this, an assumption of the average temperatures for both the secondary and primary side was used in the estimation of the fluid properties. The economizer surface area was then calculated as $1{,}238.1m^2$ representing a 7.1% of the total surface area for tube bundle.

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REFERENCES

[1] W. Rohsenow, J. Hartnett, and Y. Cho, *Heat transfer*. 1998.

[2] Q. Zhao, D. Deng, S. Nie, W. Chen, J. Wang, and D. Zhang, “The Research of Heat Transfer Area for 55/19 Steam Generator,” *J. Power Energy Eng.*, vol. 3, no. 3, pp. 417–422, 2015.

[3] T. Bergman, A. Lavine, F. Incropera, and D. Dewitt, *Fundamentals of Heat and Mass Transfer*. 2011.

[4] “International Association for the Properties of Water and Steam.” [Online]. Available: http://www.iapws.org/index.html. [Accessed: 10-Mar-2017].

[5] Wolfram Alpha LLC, “Wolfram|Alpha: Computational Knowledge Engine.” [Online]. Available: http://www.wolframalpha.com/. [Accessed: 10-Mar-2017].

[6] SteamTab™ Companion, “Download SteamTab Companion.” [Online]. Available: http://www.chemicalogic.com/Pages/DownloadSteamTabCompanion.aspx.

[7] K. A. A. Abdallah and I. Namgung, “Sensitivity study of APR-1400 steam generator primary head stay cylinder and tube sheet thickness,” *Ann. Nucl. Energy*, vol. 92, pp. 8–15, 2016.

[8] C. Report, *Crane Co., “Flow Of Fluids Through Valve, Fittings and Pipe.” p133*. 1982.

[9] Korea Hydro and Nuclear Power Co., APR1400 Design Control Document Tier 2, Chapters 3, 2014,

[10] Crane Co., “Flow Of Fluids Through Valve, Fittings and Pipe.” p. 133, 1982.

[11] E. Farber and Scorah R, *Heat transfer to water boiling under pressure*. Columbia: Univ. of Missouri, 1948.