Steady State Modelling and Validation of Once Through Steam Generator

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Abstract. One dimensional steady state model is developed for straight vertical once through counter current shell and tube steam generator for PFBR with water flowing through tubes and sodium through shell. All the heat transfer processes i.e. preheating, evaporation of water and superheating of the steam happen along the length of the tube. The length of the steam generator is divided into a number of control volumes. For modelling water side, continuity, momentum and energy equations are solved for single phase water, two phase steam and super heated vapour over the length of the tube using homogeneous model. The density variation of sodium is not considered and only energy equation is solved for sodium. The differential equations are discretised to linear algebraic equations using finite difference scheme. All three terms advection, frictional and gravitational are considered in momentum equation of water, while only advection and source term are considered in energy equation of water and sodium. The linear algebraic equations are then solved simultaneously using iterative method to get the temperature and pressure profiles in the tube and shell side. Water side heat transfer coefficient is calculated using Steiner and Taborek asymptotic model and Subbotin correlation is used for sodium side heat transfer coefficient. The code developed can be applied to simulate similar steam generators of any length with any number of tubes. Towards validation of the mathematical model and the solution method, 19 tube steam generator tested in in-house steam generator test facility (SGTF) has been simulated and the predicted results are compared with the measured data. Results from the code match well with experimentally observed data from the facility. In addition, grid sensitivity studies have been carried out to establish consistency in the solution.

Key Words: Once through steam generator, Steady-state model , Asymptotic Steiner-Taborek correlation.

1. Introduction

Steam generator is a critical component in a nuclear power plant forming the interface between nuclear and steam water systems. Proper functioning of the steam generator is of utmost importance for removal of nuclear heat and power generation. A code for steady state model of straight vertical once through steam generator is developed. The developed code can be used as a tool to support design and analysis of steam generator of any size, with water on tube and sodium on shell side. The 19 tube steam generator of in-house steam generator test facility was simulated using the developed code and the predicted results were compared with the measured data towards model validation.

2. Nomenclature

A is the cross sectional area,
A_{ht} is the heat transfer area,
C_f is friction factor,
d_i is inside diameter of tube,
ΔE is change in internal energy of water across control volume,
$F(M)$ is residual molecular weight correction factor,
F_{nb} is the nucleate boiling correction factor,

F_{pf} is pressure correction factor,
F_{tp} is the two phase multiplier accounting for the enhancement of convection due to higher velocity of a two-phase flow compared to single-phase flow of the liquid,
f_L is Fanning friction factor for liquid,
f_G is Fanning friction factor for gas,
G is the mass flux,
g is acceleration due to gravity,
H is enthalpy,
h is specific enthalpy,
k_L is thermal conductivity of liquid,
k_G is thermal conductivity of gas,
M is molecular weight of water molecule,
P is pressure in tube,
Pr_L is Prandtl number for liquid,
Pr_G is Prandtl number for gas,
P_r is reduced pressure,
p is inside perimeter of tube,
Q is heat transferred from sodium to water control volume,
q_{onb} is minimum heat flux for onset of nucleate boiling,
q is heat flux,
q_o is reference heat flux,
Re_{Lt} is Reynold's number for full flow as liquid only,
Re_{Gt} is Reynold's number for full flow as gas only,
R_p is mean surface roughness,
T is temperature,
ΔT is the difference of mean of inlet and outlet temperature of sodium and water of the control volume (Fig. 2),
U is overall heat transfer coefficient,
v is the average velocity of mixture,
W is total mass flow rate,
x is quality,
x_{crit} is critical heat flux,
z is length in the direction of flow,
$\alpha_{nb,o}$ is the local nucleate pool boiling coefficient at reference heat flux q_o,
α_{Lt} is the local liquid only forced convection heat transfer coefficient,
α_{Gt} is the local gas only forced convection heat transfer coefficient,
α_{tp} is the sum of the nucleate boiling contribution and convection contribution,
μ_L is the dynamic viscosity of liquid,
μ_G is the dynamic viscosity of gas,
ρ_m is in-situ mean density,
ρ_l is liquid density,
ρ_g is gas density,
τ_w is wall shear stress,
θ is angle from the vertical (for vertical pipe θ equals 0),
ϑ_1 is specific volume of liquid,
ϑ_2 is specific volume of gas,

3. Physical Model

The steam generator module is shown in Fig. 1. Sodium and water flow on the shell and tube side respectively in counter current directions. The inlet flow, temperature and pressure of the water side form the tube side boundary condition and sodium inlet temperature and flow rate form the shell side boundary condition and are kept constant for simulation. As water passes through the tubes, it is heated to the boiling point, vaporised through the mechanism of nucleate and film boiling and finally superheated. The length of the steam generator is divided into a number of control volumes. The continuity momentum and energy equations were solved over each control volume on the tube side while on shell side only energy equation is solved as the density change with pressure is very less in sodium in the operating temperature range. The following simplifying assumptions are made.

1. Homogeneous model is used for water side modelling.
2. Uniform flow is assumed and radial variation of flow in tube and shell is not considered.
3. The steam generator is modelled as two concentric tubes with water flowing in inner tube and sodium flowing in the outer tube in counter current direction.
4. Heat loss to the surroundings from the shell is neglected.

Steiner and Taborek asymptotic model is used to calculate water side heat transfer coefficient and Subbotin correlation for Sodium side heat transfer coefficient. A large number of flow boiling models are available in literature but Steiner and Taborek correlation is based on a large database for vertical channels. The overall heat transfer coefficient based on the tube outer area is calculated taking into account the tube wall resistance. The pressure drop in the two phase zone is obtained by multiplying the single phase pressure drop by Martinelli-Nelson factor. The thermodynamic properties of water/steam are taken from IAPWS IF 97[1].

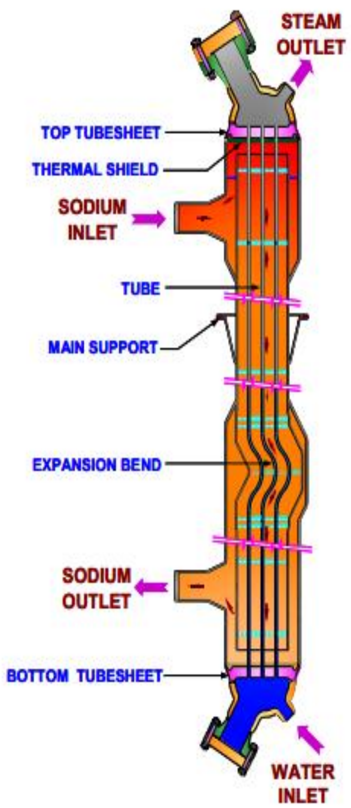

FIG. 1. Steam generator module.

4. Mathematical Model

The mathematical model is formed using the steady state continuity, momentum and energy equations in one dimensional space in the direction of flow. These equations are solved simultaneously to get temperature and pressure profiles.

4.1 Water Side Conservation Equations

4.1.1 Continuity Equation

$$W = \rho_m Av \tag{1}$$

$$G = \rho_m v \tag{2}$$

$$\frac{1}{\rho_m} = x\vartheta_2 + (1-x)\vartheta_1 \tag{3}$$

The above equations represent single phase continuity equation, when single phase density is used and two-phase continuity equation when homogeneous mean density given by equation 3 is used.

4.1.2 Momentum Equation[2]

$$\frac{dP}{dz} = -G\frac{dv}{dz} - \frac{p}{A}\tau_w - \rho_m gCos\theta \tag{4}$$

The three terms on right are

$$\frac{dP}{dz}_A = -G\frac{dv}{dz} \quad \text{Acceleration pressure drop.} \tag{5}$$

$$\frac{dP}{dz}_F = -\frac{p}{A}\tau_w \quad \text{Friction pressure drop.} \tag{6}$$

$$\frac{dP}{dz}_G = -\rho_m g \quad \text{Gravitation term.} \tag{7}$$

$$v = \frac{G}{\rho_m} \quad \text{The average velocity of phases.} \tag{8}$$

After substitution of the average velocity from equation 8 and mean density from equation 3 in the acceleration pressure drop term, equation 5 takes the form given in equation 9.

$$-\left(\frac{dP}{dz}\right)_A = G^2 \ \frac{dx}{dz}(\vartheta_2 - \vartheta_1) + \frac{dP}{dz} \ (1-x)\frac{d\vartheta_1}{dP} + x\frac{d\vartheta_2}{dP} \tag{9}$$

Substituting τ_w with $C_f \rho_m \frac{v^2}{2}$ and replacing the expression for v and ρ_m from equation 8 and equation 3 respectively, the frictional term yields equation 10.

$$-\left(\frac{dP}{dz}\right)_F = \frac{p}{A}\tau_w = \frac{C_f 2(x\vartheta_2+(1-x)\vartheta_1)G^2}{d} \tag{10}$$

The gravitation term after substitution of ρ_m gives $\rho_m g = \frac{g}{(x\vartheta_2+(1-x)\vartheta_1)}$,

Combining the three term the momentum equation becomes

$$-\frac{dP}{dz} = \frac{G^2\left(\frac{dx}{dz}*(\vartheta_2-\vartheta_1)\right)+\frac{C_f*2*(x\vartheta_2+(1-x)\vartheta_1)*G^2}{d}+\frac{g}{(x\vartheta_2+(1-x)\vartheta_1)}}{1+G^2*\left((1-x)\frac{d\vartheta_1}{dP}+x\frac{d\vartheta_2}{dP}\right.} \tag{11}$$

In the equation 11 the only coefficient which is unknown is C_f which can be obtained using two phase multiplier from Nelson and Martinelli parameter [3]. All other quantities are constant or can be obtained from steam water properties. Since there will be no significant flashing, for computation of term $\frac{dx}{dz}$, quality can be considered as a function of enthalpy only i.e. $x = x(h)$ [2]. The quality gradient can be computed from energy equation. The term $\frac{dx}{dz}$ forms the coupling between momentum and energy equation.

4.1.3 Energy Equation

The energy equation is written from the first law of thermodynamics

$$Q = \Delta E + Work \tag{12}$$

In absence of external shaft work and neglecting kinetic and potential energy changes the water side energy equation for a control volume becomes

$$Q = H_{out\ water} - H_{in\ water} \tag{13}$$

$$Q = W_{water}\,(h_{out\ water} - h_{in\ water} \tag{14}$$

4.2 Sodium Side Conservation Equations

Sodium side only energy conservation equation is solved. Momentum and continuity equations are neglected as the density variations are very small.

$$Q = H_{in\ Sodium} - H_{out\ Sodium} \tag{15}$$

$$Q = W_{Sodium}\, C_p\,(T_{in\ Sodium} - T_{out\ Sodium} \tag{16}$$

The heat transferred Q from sodium to water is the coupling term between the water side and sodium side energy equations. Q is determined using equation 17.

$$Q = UA_{ht}\Delta T \tag{17}$$

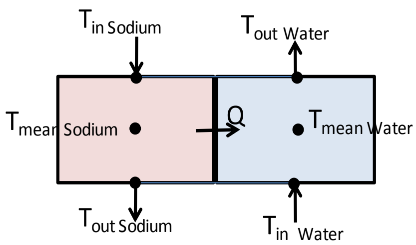

FIG. 2. Representative control volume.

The control volumes are initialized with an assumed initial value and solved iteratively till the values of temperature and pressure converge. The water side control volume outlet enthalpy is calculated first from energy equation, outlet quality is calculated using the calculated enthalpy then momentum equation is solved to get the pressure profile and using the computed enthalpy and pressure, temperature profile is computed.

5. Water Side Heat Transfer coefficient

The decision for water side heat transfer coefficient is taken on the basis of the output quality from the control volumes. For calculation of heat flux for i^{th} control volume the output quality from the $(i-1)^{th}$ control volume is checked and the heat transfer coefficient is calculated. A large number of models are available for flow boiling in vertical tube. For the present analysis Steiner-Taborek asymptotic model[4] is used for local heat transfer coefficient in the tube. This model combines the effects of convection and nucleate boiling as given in equation 18.

$$\alpha_{tp} = [(\alpha_{nb,o} * F_{nb})^3 + (\alpha_{Lt} * F_{tp})^3]^{\frac{1}{3}} \quad (18)$$

The model is applicable for quality less than the critical quality. It does not apply for mist flow in the tube. The single phase liquid heat transfer coefficient is given by the Gnielinski equation, given by equation 19.

$$\frac{\alpha_{Lt} * d_i}{k_L} = \frac{\left(\frac{f_L}{8}\right) * (Re_{Lt} - 1000) * Pr_L}{1 + 12.7 * \left(\frac{f_L}{8}\right)^{1/2} * (Pr_L^{\frac{2}{3}} - 1)} \quad (19)$$

The fanning friction factor f_L for the liquid is calculated using

$$f_L = [0.7904 * ln(Re_{Lt}) - 1.64]^{-2} \quad (20)$$

The applicability range for f_L is 4000<Re_{Lt}<5000000 and 0.5< Pr_L<2000 for single phase flows.

Reynolds number is calculated considering the entire fluid as liquid only.

If heat flux is greater than the heat flux for onset of nucleate boiling, heat transfer due to nucleate boiling is also considered in addition to convective heat transfer term. The nucleate boiling coefficient of water for the Steiner-Taborek flow boiling correlation $\alpha_{nb,o}$ is 25580W/m^{2}K at the standard condition of reduced pressure P_r of 0.1, a mean surface roughness of $R_{p,o} = 1\ \mu m$. and the heat flux q_o=150000W. The nucleate boiling correction factor F_{nb} includes correction due to pressure, heat flux, tube diameter, surface roughness and a residual molecular weight correction factor.

$$F_{nb} = F_{pf} * \left(\frac{q}{q_o}\right)^{nf} * \left(\frac{d_i}{d_{i,o}}\right)^{-0.4} * \left(\frac{R_p}{R_{p,o}}\right)^{0.133} * F(M) \quad (21)$$

The pressure correction factor F_{pf} valid for P_r < 0.95, accounts for the increase in the nucleate boiling coefficient with increasing pressure:

$$F_{pf} = 2.81 * P_r^{0.45} + \left\{3.4 + \frac{1.7}{1 - Pr^7}\right. * P_r^{3.7} \quad (22)$$

The nucleate boiling exponent:

$$nf = 0.8 - 0.1 * exp(1.75 * P_r) \quad (23)$$

The standard tube reference diameter $d_{i,o}$ is 0.01m.

The residual molecular weight correction factor, in terms of the liquid molecular weight M (valid for 10 < M < 187) is given by equation 24.

$$F(m) = 0.377 + 0.199 * ln(M) + 0.0002842 * M^2 \quad (24)$$

For points where $x < x_{crit}$ at the tube exit and q > q_{onb}, equation 25 is used. x_{crit} is the limit of applicability of Steiner-Taborek model.

$$F_{tp} = [(1 - x)^{1.5} + 1.9 * x^{0.6} \ \frac{\rho_L}{\rho_G}^{0.35}]^{1.1} \quad (25)$$

F_{tp} is valid for 3.75 < $\left(\frac{\rho_L}{\rho_G}\right.$ < 5000.

When $q < q_{onb}$, only pure convective evaporation is present. At the limiting case of x = 1.0, the value of α_{tp} corresponds to α_{Gt} as given by equation 26, where the Reynolds no. is calculated considering the entire fluid as gas only, as given in equation 27.

$$\frac{\alpha_{Gt} * d_i}{k_G} = \frac{\left(\frac{f_G}{8}\right) * (Re_{Gt} - 1000) * Pr_G}{1 + 12.7 * \left(\frac{f_G}{8}\right)^{1/2} * \left(Pr_G^{\frac{2}{3}} - 1\right)} \tag{26}$$

$$Re_{Gt} = \frac{\dot{m} * d_i}{\mu_G} \tag{27}$$

In this case equation 28 is used for calculation of F_{tp}.

$$F_{tp} = \left[(1-x)^{1.5} + 1.9 * x^{0.6} * (1-x)^{0.01} * \left(\frac{\rho_L}{\rho_G}\right)^{0.35}\right]^{-2.2} + \left[\frac{\alpha_G}{\alpha_L} * x^{0.01} * \left(1 + 8 * (1-x)^{0.7}\right) * \left(\frac{\rho_L}{\rho_G}\right)^{0.67}\right]^{-2} \tag{28}$$

6. Experimental Validation

5.1.System description of Steam Generator Test Facility (SGTF)

SGTF is test loop of steam generator with sodium flowing on shell side and high pressure water /steam flowing through the tubes. It is a 5.5 MWt, 19 tube model of 157 MWt, 547 tube steam generator of PFBR. Fig. 3 shows the 19 tube steam generator tested in SGTF. The system is discussed in reference [5] in detail, while the experimental results are given in reference [6].

5.2.Results

The code developed was tested using geometrical and process inputs from steam generator of SGTF and the results were compared with experimental values observed.

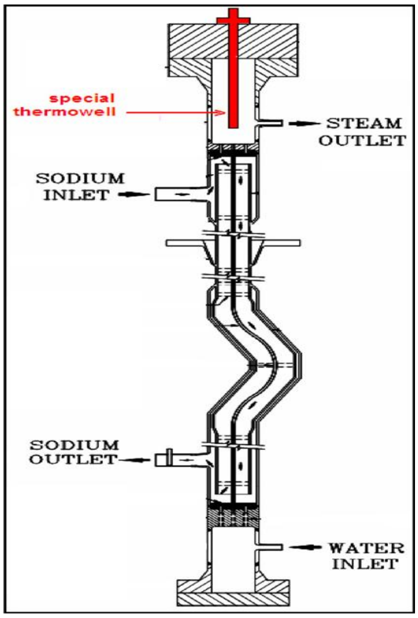

FIG. 3. 19 tube steam generator tested at SGTF.

Experiments were conducted in SGTF with sodium flow rate of 25.2 kg/s, water inlet temperature of 233°C and feed water pressure of 18.01 MPa. Based on these experimental conditions, the following cases have been simulated

- Case-1: Water flow rate 2.452 kg/s and sodium inlet temperature 516.5°C
- Case-2: Water flow rate 2.645 kg/s and sodium inlet temperature 525.1°C
- Case-3: Water flow rate 2.426 kg/s and sodium inlet temperature 518.5°C.

The results from the code match well with the experimentally observed data obtained from in-house SGTF loop for the above cases as shown in Table I. Fig. 4 depicts the typical water and sodium temperature profiles along the length in steam generator for case 2. Fig. 5 shows the pressure variation along the length of the tube for the same case where the initial pressure drop of 10 bar due to orifice is not shown. The percentage error of water temperature change given by equation 29 is ~3%.

$$\frac{(T_{steam\ out} - T_{water\ in})_{experimental} - (T_{steam\ out} - T_{water\ in})_{from\ code}}{(T_{steam\ out} - T_{water\ in})_{experimental}} \quad (29)$$

Grid sensitivity analysis was carried out to establish the consistency of the mathematical model. Fig 6 shows the variation of outlet steam temperature for different number of control volumes. It can be seen that when the length of tube is divided into 1000 or less number of control volumes, the outlet temperature of steam is dependent on the number of control volumes while beyond it, the outlet steam temperature is almost independent of number of grids.

Table I: COMPARISON OF CODE RESULTS WITH EXPERIMENTAL DATA FROM SGTF

Parameters	**Case 1**		**Case2**		**Case 3**	
	Experimental value	**Result From present code**	**Experimental value**	**Result From present code**	**Experimental value**	**Result From present code**
Steam pressure (MPa)	17.1	16.87	17.1	16.68	17.1	16.88
Steam outlet temperature (°C)	493	502	493	503	493	502
SG outlet sodium temperature (°C)	342	342	337	336	345	345
Power(MWt)	5.64	5.62	6.05	6.07	5.58	5.6

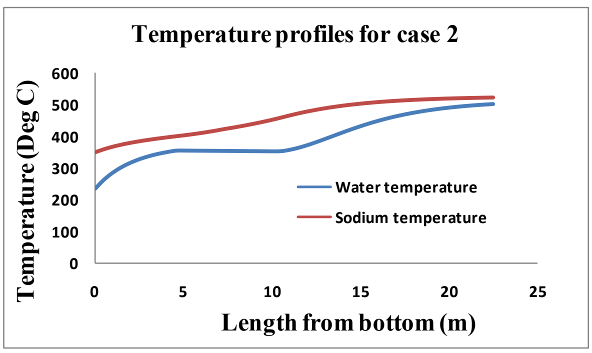

FIG. 4.Temperature profiles along tube length.

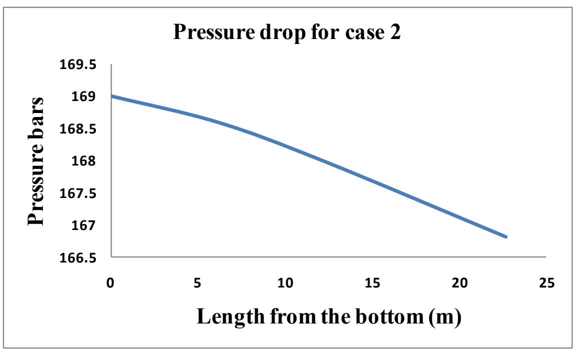

FIG. 5.Pressure drop along the tube length.

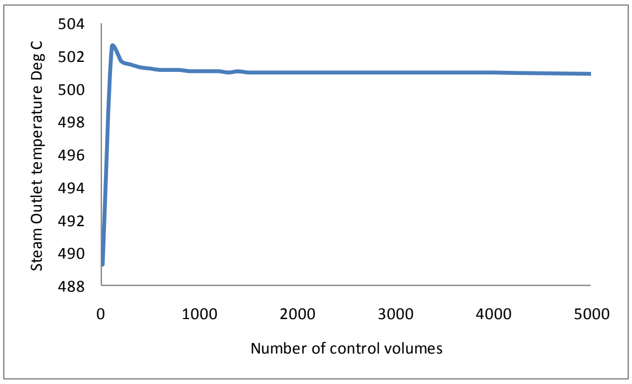

FIG. 6.Variation of steam outlet temperature with number of control volumes.

7. Conclusion

A code for one dimensional steady state model for straight vertical once through steam generator is developed using Steiner Taborek asymptotic model for flow boiling inside vertical tube which is based on a large database for vertical channels. Gnielinski correlation is used for single phase vapour and superheated steam above the critical quality. Consistency of the solution was established through grid sensitivity analysis. Results obtained using the code, simulating the experimental conditions for the steam generator of in-house test facility, are found to be in close agreement with experimentally observed data, thereby validating the model.

The developed code can be used as a design and analysis tool for straight vertical once through steam generators of any length with any number of tubes. The model is intended to be used as a part of integrated process model of the plant. The code will be further developed in future to include transient calculations for plant dynamics studies simulating various transients.

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