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Development of a Thermal-Hydraulic Design Model for Once-Through Steam Generators

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1. Introduction

A steam generator (SG) is a key component whose design concept and methodology must be established at an early stage in the development of a new reactor concept. In SMRs, which are currently under active development, once-through SGs employing a superheated steam cycle are widely adopted to improve plant thermal efficiency and to produce high-temperature steam for applications such as industrial process heat supply and hydrogen production. [1]

In a once-through SG, the tube-side fluid (feedwater) enters in a subcooled state and exits as superheated steam. The shell-side fluid flows outside the tubes in a counter-current direction and transfers heat to the tube-side fluid. Depending on the tube arrangement, once-through SGs are generally classified as straight-tube or helical-coil types. The straight-tube SG features a relatively simple geometry, facilitates fabrication, and benefits from extensive design experience. However, compared to the helical-coil SG, it typically requires a larger heat transfer surface area to achieve comparable heat transfer performance. The helical-coil SG enables a high heat transfer surface area to be accommodated within a relatively small volume. Compared to the straight-tube SG, it provides enhanced heat transfer efficiency, improved thermal–hydraulic stability, and a more compact configuration. [2]

This study presents the development of an analytical model intended to provide fundamental data for the thermal-hydraulic design of once-through SGs. The model evaluates both helical-coil and straight-tube configurations and predicts key thermal–hydraulic parameters, including temperature and pressure distributions, the required heat transfer area and other design variables, based on the heat transfer tube geometry and inlet conditions on the tube and shell sides. Water and helium (He) are considered as the shell-side working fluids. The applicability of the developed model is assessed through comparisons with available reference data.

2. Analysis Models

For the thermal-hydraulic analysis of the once-through SG, the heat transfer tubes are modeled as a single representative tube. The governing equations consist of the continuity equation, momentum equation, energy equation, and equation of state. The following assumptions are made in the present analysis:

(1) One-dimensional flow
(2) No axial heat conduction along the tube
(3) Homogeneous flow and thermodynamic equilibrium in the two-phase flow region
(4) No thermodynamic non-equilibrium effects, such as subcooled boiling and mist evaporation
(5) Forward flow on both the tube and shell sides
(6) Constant shell-side pressure

The governing equations are given as follows:

$$\frac{\partial(\rho u)}{\partial x} = 0 \tag{1}$$

$$\frac{\partial(\rho u^2)}{\partial x} + \frac{\partial P}{\partial x} + \frac{f}{2D_e}\rho u^2 + \rho g \cos\theta = 0 \tag{2}$$

$$\frac{\partial}{\partial x}[\rho u\,(h + \frac{u^2}{2})] + \rho u g = Q \tag{3}$$

$$\rho = \rho\,(P, h) \tag{4}$$

In Eqs. (1)-(4), ρ, u, P, D_e, g, θ, h, Q denote the density, velocity, pressure, equivalent diameter, gravitational acceleration, inclination angle, enthalpy, and volumetric heat input to the fluid, respectively. By integrating Eqs. (1)-(4) along the flow direction from node i to node i+1, the following finite difference equations are obtained:

$$(\rho u)_{i+1} = (\rho u)_i \tag{5}$$

$$\begin{aligned} &P_{i+1} + \rho_{i+1}[g\frac{\Delta x}{2} + (1 + \frac{f}{2D_e}\frac{\Delta x}{2})u_{i+1}^2] = \\ &P_i + \rho_i[g\frac{\Delta x}{2} + (1 + \frac{f}{2D_e}\frac{\Delta x}{2})u_i^2] \end{aligned} \tag{6}$$

$$h_{i+1} + \frac{u_{i+1}^2}{2} = h_i + \frac{u_i^2}{2} - g\,\Delta x + \frac{Q_i\,\Delta x}{(\rho u)_i} \tag{7}$$

$$\rho_i = \rho\,(P_i, h_i) \tag{8}$$

In Eq. (6), f represents the single-phase friction factor. In the two-phase flow region, it is multiplied by a two-phase multiplier. In Eq. (7), Q_i is defined for control volume i, located between nodes i and i+1, based on the outer heat transfer area ($A_{o,i}$) of the tube as follows:

$$Q_i\ V_i = U_{o,i}\ A_{o,i}\ \Delta T_{LMTD,i}\ Z \tag{9}$$

Here, V_i, $U_{o,i}$, $\Delta T_{LMTD,i}$, Z are the volume, the overall heat transfer coefficient, the log mean temperature difference and the surface utilization factor, respectively. The overall heat transfer coefficient is expressed as

$$\frac{1}{U_{o,i}} = \frac{1}{h_{i,i}}\frac{d_o}{d_i} + \frac{d_o}{2k_{w,i}}\ln\frac{d_o}{d_i} + \frac{1}{h_{o,i}} \tag{10}$$

where $h_{i,i}$, $h_{o,i}$, and $k_{w,i}$ represent the tube-side heat transfer coefficient, the shell-side heat transfer coefficient, and the thermal conductivity of the tube wall, respectively.

The analysis is performed using Eqs. (5)-(10) with heat transfer and friction factor correlations which are summarized in Table I. The thermal conductivities of the considered tube materials are taken from Ref. [7], the water–steam thermophysical properties from Ref. [8], and the helium properties from Ref. [9].

Table I: Correlations for heat transfer coefficients and friction factors

Tube	Tube-side	Shell-side
Helical	1-phase heat transfer	1-phase heat transfer
	SKBK[3]	SKBK[3]
	2-phase heat transfer	2-phase heat transfer
	SKBK[3]	N/A
	Friction factor	Friction factor
	SKBK[3]	N/A
Straight	1-phase heat transfer	1-phase heat transfer
	Dittus-Boelter[4]	Weisman(water) [4], Presser(He) [4]
	2-phase heat transfer	2-phase heat transfer
	Chen[5]	N/A
	Friction factor	Friction factor
	Colebrook[6]	N/A

Given the tube geometric parameters (e.g., diameter, pitch, number of tubes) and the inlet boundary conditions on both the tube and shell sides (temperature, pressure, and mass flow rate), the developed model can be used to determine the tube length (heat transfer area) required to meet a specified heat load, or to evaluate the removable heat load for a given tube length. Fig. 1 presents the flowchart for determining the tube length needed to satisfy the specified heat load.

3. Results and Discussions

To evaluate the applicability of the developed model, comparative calculations are performed using the ONCESG code [3], which was developed for the thermal-hydraulic design of the helical-coil SG of SMART. In addition, benchmark analyses are conducted for once-through SGs used in light-water reactors and high-temperature gas-cooled reactor.

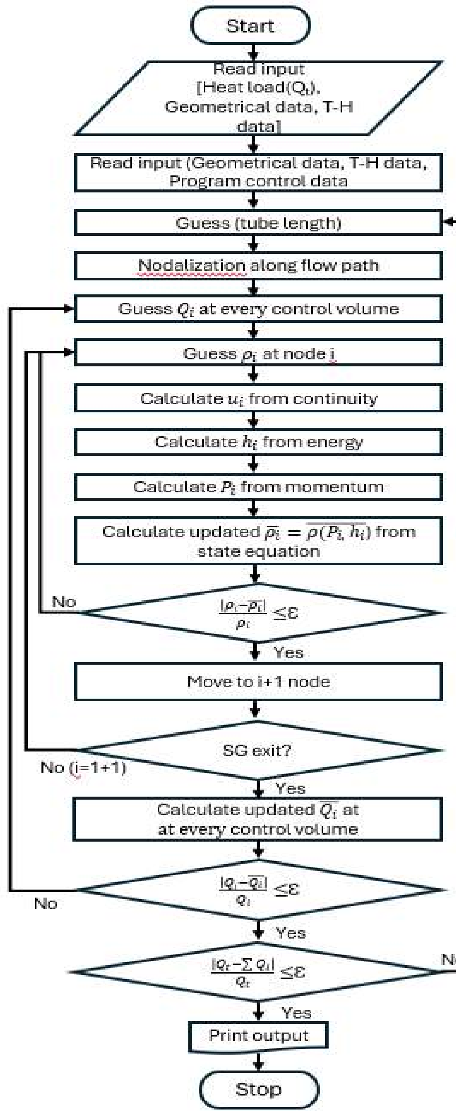

Fig. 1. Flow chart

For the comparative evaluation with the ONCESG code, an option to calculate the heat load for a specified heat transfer tube length is applied. The input data and calculation results are summarized in Table II, and Fig. 2 compares the calculated temperature distributions. In Table II, L, D_i, D_o, D_{coil1}, P_r, P_l, N_{tube}, N_r, $m_{p,i}$, $T_{p,i}$, $P_{p,i}$, $m_{s,i}$, $T_{s,i}$, $P_{s,i}$, Q_t, $T_{p,o}$, $T_{s,o}$, $P_{s,o}$, D_{coilH}, D_{coilN}, A_o are the average tube length, tube inner diameter, tube outer diameter, innermost coil diameter, radial pitch, axial pitch, number of tubes, number of coil layers, shell-side mass flow rate, shell-side inlet temperature, shell-side pressure, tube-side mass flow rate, tube-side inlet temperature, tube-side inlet pressure, heat load, shell-side outlet temperature, tube-side outlet temperature, tube-side outlet pressure, tube bundle height, outermost coil diameter, and heat transfer area based on D_o, respectively. The heat load is calculated to be 5.4527 MW_t, showing good agreement with the ONCESG result of 5.4519 MW_t. In addition, the temperature distribution is also well predicted.

Table II: Input and results of comparison with ONCESG

Inputs			
L [m]	10.235	N_r	12
D_i [mm]	7.0	$m_{p,i}$ [kg/s]	28.29
D_o [mm]	10.0	$T_{p,i}$ [°C]	310.0
D_{coilI}[m]	0.114	$P_{p,i}$ [MPa]	14.70
P_r [mm]	14	$m_{s,i}$ [kg/s]	2.00
P_l [mm]	11.5	$T_{s,i}$ [°C]	50.0
N_{tube}	96	$P_{s,i}$ [MPa]	3.77

Results		
	Model	ONCESG
Q_t [MW$_t$]	5.4527	5.4519
$T_{p,o}$ [°C]	273.9	273.9
$T_{s,o}$ [°C]	284.4	284.6
$P_{s,o}$ [MPa]	3.48	3.51
D_{coilH} [m]	1.1117	1.1117
D_{coilN} [m]	0.422	0.422
A_o [m^2]	30.87	30.87

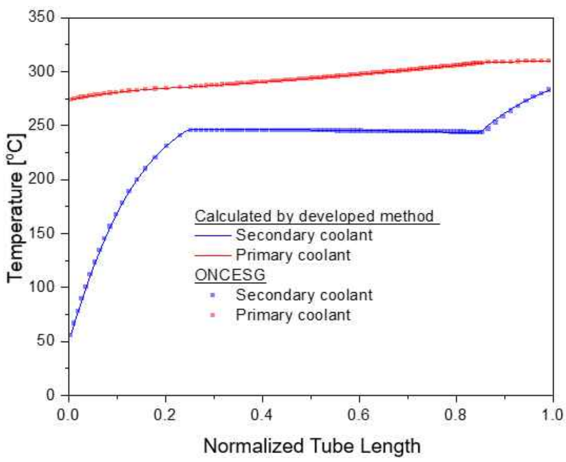

Fig. 2. Comparison of temperature distributions

For the evaluation of the helical-coil SGs, the available design data for IRIS [10] and HTR-PM [11-13] are utilized. For the straight-tube SG, the design data for Babcock & Wilcox presented in Ref. [14] are employed. The parameters used in the analysis for a single SG unit are summarized in Table III. It should be noted that the tube pitches for the helical-coil SGs are adjusted to approximate the tube bundle geometry of IRIS and HTR-PM.

The evaluation results for the main variables are summarized in Table IV, and the calculated temperature distributions are shown in Fig. 3. For the helical-coil SG, the option to calculate the heat load for a given heat transfer area is applied. On the other hand, the option to calculate the required heat transfer area to remove the specified heat load is employed for the straight-tube SG. The number of control volumes used in the calculations is 100.

Table III: Parameters used for benchmark analyses

Parameter	IRIS	HTR-PM	B&W
Tube type	Helical	Helical	Straight
Q_t [MW$_t$]	125	13.1578	1284
D_i [mm]	13.24	13	14.147
D_o [mm]	17.46	19	15.875
L [m]	32	60	15.96
Tube material	Inconel 690	Alloy 800H, 2-1/4Cr1Mo	Inconel 690
D_{coilI} [m]	0.634	0.561	-
D_{coilH} [m]	7.9	8.6	-
P_r [mm]	22.0	27.55	22.225
P_l [mm]	27.3	43.65	-
N_{tube}	655	35	15531
N_r	21	5	-
$m_{p,i}$ [kg/s]	589.0	5.0737	8273.16
$T_{p,i}$ [°C]	328.4	750	317.7
$P_{p,i}$ [MPa]	15.5	7	15.17
$m_{s,i}$ [kg/s]	62.5	4.9947	680.4
$T_{s,i}$ [°C]	223.9	205.0	237.8
$P_{s,i}$ [MPa]	6.096	13.25	6.38

Table IV: Comparison of main variables

	Q_t	$T_{p,o}$	$T_{s,o}$	$P_{s,o}$	D_{coilH}	L
	IRIS (Helical)					
Design	125.0	292.0	317.0	5.80	7.90	-
Cal.	125.6	291.8	323.2	5.78	7.83	-
Err|%|	0.46	0.06	1.97	0.31	0.88	-
	HTR-PM (Helical)					
Design	13.16	250.0	567.0	-	8.600	-
Cal.	13.15	250.4	567.2	12.8	8.604	-
Err|%|	0.08	0.16	0.04	-	0.05	-
	B&W (Straight)					
Design	1284	290.0	312.8	-	-	15.9
Cal.	1284	290.0	311.9	6.33	-	13.2
Err|%|	0.00	0.01	0.27	-		17.1

As shown in Table IV, the calculated results for the helical-coil SGs of IRIS and HTR-PM agree well with the design values within 2%, provided that the input parameters properly represent the tube bundle geometry. In the case of the straight-tube SG for Babcock & Wilcox, the required heat transfer area satisfying the specified heat load is evaluated to be 17.1% smaller than the design value. Normally, the heat transfer area of an SG is determined by incorporating a design margin at the initial design stage to compensate for design uncertainties and performance degradation due to tube plugging and fouling. From this perspective, the Babcock & Wilcox SG design is considered to include a design margin.

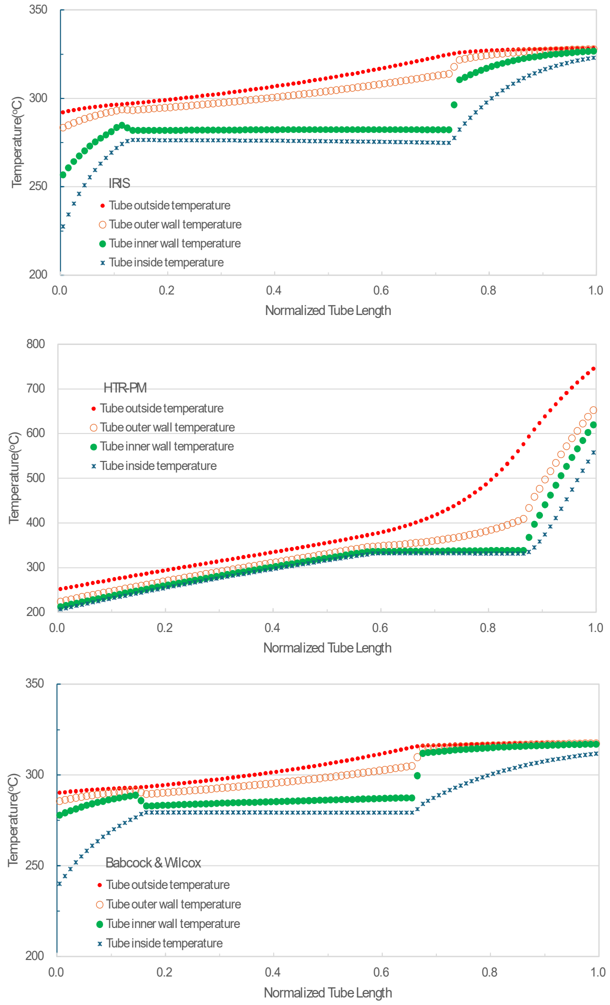

Fig. 3. Calculated temperature distributions

The calculated temperature distributions show that the tube-side fluid temperature remains nearly constant in the phase change region, whereas the tube wall temperature changes relatively rapidly during the transition from two-phase flow to single-phase vapor flow. As heat transfer is enhanced in the two-phase flow region, the temperature difference between the tube-side fluid and the tube wall is reduced. In the case of HTR-PM, the external thermal resistance between the tube outer surface and the helium coolant is larger than the internal thermal resistance, leading to a larger temperature difference between the helium flow and the tube wall. Furthermore, the two-phase flow region inside the tube is relatively short, and the change in the tube wall temperature near the phase transition point is relatively gradual. In the present model, thermodynamic non-equilibrium effects in the transition regions, such as subcooled boiling during the transition from liquid to two-phase (saturated boiling) flow and mist evaporation during the transition from two-phase to vapor flow, are not modeled. Incorporation of these effects is considered to improve the prediction of tube wall temperature variations in the phase change region.

3. Conclusions

A one-dimensional analytical model was developed to evaluate the thermal-hydraulic performance of once-through SGs. To validate the applicability of the model, benchmark analyses were performed using the ONCESG code and the design data for the IRIS, HTR-PM, and Babcock & Wilcox SGs. For the helical-coil SGs, the calculated results showed good agreement with the design values. For the straight-tube SG, the results indicated that the heat load was well predicted when the design margin of the heat transfer area was taken into account. The developed model is expected to be applicable to the thermal-hydraulic design of once-through SGs. Furthermore, by incorporating thermophysical property data and heat transfer models for sodium and molten salt, the model can be extended to sodium-cooled fast reactor and molten salt reactor.

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