Scaling analysis for PWR steam generator

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Abstract: To test the performance of a nuclear power plant safety system and to verify the relevant safety analysis code, a widely used approach is to design and construct a scaled model based on a scaling methodology. For a pressurized water reactor (PWR), the SG scaling analysis is important before designing a scaled model, which is expected to simulate well the system response of the prototype system in an accident. This work first presents a review of the transient process in SG during a loss of coolant accident (LOCA), and then describes a brief scaling analysis for a natural circulation to get the basic scaling criteria for the SG. The U-tube scaling design showed that if the diameter ratio was different from the length ratio for a model, the thermal height center would be enlarged because the length of the U-tube should be scaled by the length ratio. Therefore the improperly scaled buoyant force would cause a distortion in natural circulation simulation. By single phase heat transfer scaling analysis, a relation between the U-tube diameter ratio and the height ratio was obtained. It showed that the diameter ratio decreased with the decrease of the height ratio. Finally, the transition of the role played by the SG, from heat sink to heat source, was analyzed. The results showed that the inventory of the secondary side of the SG and the total metal heat capacity should be properly scaled in order to represent the transition correctly.

Keyword: scaling analysis; U-tube; integral test facility; decay heat removal; steam generator

1 Introduction

The integral thermal hydraulic (TH) test is of great importance to the safety assessment of nuclear power plant designs. On the one hand, through the integral TH simulation, the performance of the plant safety system is tested to prove that the design can meet the demand of the safety criteria set forth by nuclear regulators. On the other hand, it should be proved that the code prediction is realistically conservative, using the respective safety analysis code. Given the large cost and the engineering scale involved, building a full size test plant is not likely to be under consideration. In practice, a scaled-down facility is the preferred option in order to realize the integral TH test under reasonable costs. The scaled-down approach has been widely used around the world, with the support of the scaling analysis approach or technique, in order to achieve appropriate test results. The full height scaled-down and reduced height scaled-down are two types of similarity design approaches. The full height scaled-down facility is commonly a tall-thin feature of one-dimensional nature, while the reduced height one may be designed in order to better represent the multi-dimensional feature.

An integral TH test facility for a PWR is designed to represent both the reactor coolant system (RCS) and the emergency core cooling system (ECCS) for the safety test focusing on the simulation of the loss of coolant accident (LOCA). Steam generators (SGs) are the key components in RCS to remove the reactor power by generating the steam to drive the turbine during normal operation. Under an accident scenario, SGs become the heat sink that removes the decay power by the natural circulation occurring in the RCS during the early period of a small break LOCA (SBLOCA) transient. It is effective in maintaining core cooling and in preventing the fuel from overheating. For example the results in the phenomena identification and ranking table (PIRT) for both the AP600 and the AP1000 show that the SG has a medium impact on the peak cladding temperature (PCT)[1]. In order to use SG models to accurately represent, in prototype, the real SGs, the scaling analysis needs to be performed and the scaling distortion also needs to be evaluated. During the early period of a SBLOCA, SGs operate in a natural circulation (NC) mode, driven by the buoyant force generated from the heated fluid through the

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core and the cooled fluid through the SG. Extensive research has been undertaken on the NC scaling analysis for both single-phase and two-phase flows [2], and the NC similarity criteria are derived and used to determine the system scale ratio [3]. In this paper, the SG scaling analysis is discussed, as referring to a detailed scaling analysis of the SG thermal hydraulic behavior during a LOCA transient, including the NC and heat transfer scaling, resulting in an adequate simulation of the decay remove function.

2 Transient in SG during a LOCA

When a LOCA caused by a pipe break triggers the "S" (safety shutdown) signal, the reactor and the main pumps will trip, and then the forced circulation in the RCS loop will transfer to natural circulation, first with a single-phase flow and then changing to a two-phase flow once the pressure drops to the saturate pressure caused by the break blowdown and the vapor produced. When the RCS inventory decreases and the pressurized water level drops, the U-tubes in the SG drain out, which cuts off the flow circulation in the primary loop. The duration of the natural circulation phase depends on the break size, which will be long with a small break and short with a large break.

During the NC, the decay heat will be removed by SG through heat transfer from the coolant in the primary side to the secondary side. The main steam line is shut off when the "S" signal is actuated. The power-operated relief valve (PORV) on the SG will open when the secondary side pressure increases over the set point for "valve open" due to the heat transfer from the primary side to the secondary side by the U-tubes, and it will close once the pressure drops to the set point for "valve closed" due to relief of the high pressure vapor in the secondary side. So the PORV operates circularly between the open and the closed statuses, in order to help remove the decay heat taken by the naturally circulated coolant and the coolant discharged when the PORV is open.

As Fig.1 shows, after a break, the system pressure will drop rapidly and the NC will be established in the RCS loop, which is transitioned from the forced circulation driven by the coast down of the main pumps. The SG secondary side pressure will increase due to the heat removal by the SG in forced circulation, and then by the NC. The NC loop will be cut off when the U-tubes are empty. By that time the NC phase ends. The SG secondary side is then not heated anymore and the pressure remains relatively stable with the valve closed, while in the primary side (in the U-tubes), the pressure drops with the decrease in RCS pressure, and finally reaches the pressure level near the containment pressure. After the end of the NC phase, the SG has no significant effect on the afterward transient, and the SG effect on the core cooling is no longer a concern. Therefore, the SG scaling analysis focuses on the early transient in a LOCA where the SG primary side NC plays the main role in the decay heat removal.

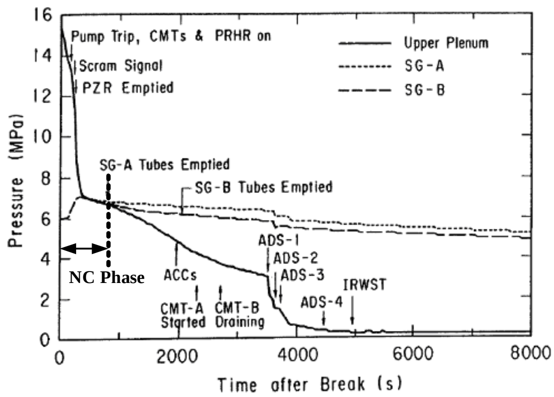

Fig.1 The pressures in primary and secondary side of SG in AP600 during a 1-in. cold leg break test on ROSA [4].

3 Scaling analysis of SG NC in the primary loop

In a PWR, the SG elevation is higher than the reactor core and thus a natural circulation can be induced after the pump trip. As shown in Fig. 2, the PWR RCS can be treated as a natural circulation loop with the core as a heat source and the SG at higher location as a heat sink.

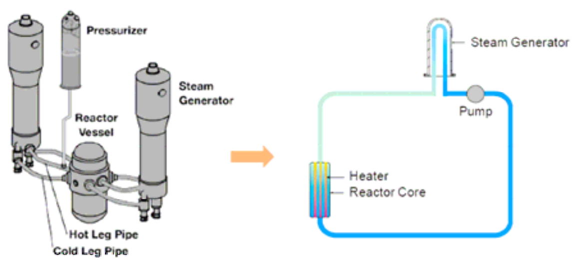

Fig.2 RCS loop can be treated as a natural circulation loop.

As Fig.2 shows, the fluid density decreases from ρ_0 to ρ_1 through heating from the core, and then increases from ρ_1 back to ρ_0 by the heat removal through the SG. For simplicity, the heat loss is not taken into consideration. In order to simplify the problem and to show the basic understanding of the NC scaling, a single phase liquid NC model is used. The fluid mass continuity in the entire loop can be expressed by:

$$\rho_i u_i a_i = \rho_c u_c a_c \tag{1}$$

where u is the fluid velocity, a is the flow cross area, and subscript i means the ith component in the loop. The momentum balance of the loop fluid is written as:

$$\frac{d\rho_c u_c}{dt}\sum_i\left(\frac{a_c}{a_i}l_i\right) = \beta g\rho_c \Delta T_c l_{th} - \frac{\rho_c u_c^2}{2}\sum_i\left[\left(\frac{fl}{d_h}+K\right)_i\left(\frac{a_c}{a_i}\right)^2\right] \tag{2}$$

where l is the length of the flow component, β is the thermal expansion coefficient of the coolant, ΔT is the temperature difference of the reactor core, f is the friction loss factor, d_h is the hydraulic diameter, and K is the form loss coefficient. The subscript c denotes core entrance, and th means the thermal height.

Through normalization with the initial or boundary conditions, the normalized equations Eq.(1) and Eq.(2) can be obtained. They are presented in many papers and hence are herewith omitted. As a similarity requirement, the buoyant force should match the resistance force and therefore the Richardson number and the Friction number should be kept identical for the prototype and the model, yielding:

$$\Pi_{Ri,R} = \left(\frac{\beta_0 g \Delta T_0 L_{th}}{u_0^2}\right)_R = 1 \tag{3}$$

and

$$\Pi_{F,R} = \left[\sum_i\left(\frac{F_i}{A_i^2}\right)_0\right]_R = 1 \tag{4}$$

The subscript 0 refers to the initial value. F_i is the friction factor of the ith flow section. Scaling analysis generally employs the length ratio l_R instead of the thermal height ratio $L_{th,R}$ because they are approximately identical and when each component length ratio is identical, $l_{iR}=l_R$. Additionally, to support this condition, the thermal power distribution along the reactor core and the SG are kept identical in both the prototype and the test model, which is normally considered reasonable when using the proper core axial power profile.

If the property similitude condition is met, the most important relation for single-phase flow NC scaling can be obtained from Eq. (3), namely $(u_0)_R^2= l_R$ and $(\tau_0)_R=l_R^{1/2}$. Therefore in a reduced height system the event occurs faster than in the prototype.

When the facility works under the same pressure as the prototype, the fluid property similitude can be realized, and the temperature rise across the core should be identical. The relation as shown below is satisfied under a steady state condition,

$$\rho_0 u_0 a_c C_{p0} \Delta T_0 = \dot{q}_{core,0} \tag{5}$$

where C_p is the heat capacity of the coolant and $\dot{q}_{core}$ is the core power. Hence, the core power ratio needs to be set as:

$$\dot{q}_{core,R} = \left(a_c l^{\frac{1}{2}}\right)_R \tag{6}$$

Therefore, when the reduced height model works under the same system pressure as the prototype, by adjusting the loop resistance and core power to satisfy Eq.(4) and Eq.(6), the model can simulate the single-phase natural circulation properly.

The above analysis works under the single-phase condition. In the two-phase condition, the same scaling result can also be obtained from the detailed two-phase NC scaling analysis [3].

4 U-tube bundle design

The U-tube bundle design for a SG is a key issue because it is the essential part within a NC loop for removing the decay heat from the primary side to the secondary side through tube wall heat transfer.

Therefore the U-tube design needs both to meet the NC scaling criteria and to have proper heat transfer surface area to remove the decay heat.

4.1 U-tube dimension design

The above NC scaling shows that in order to properly simulate a NC loop, the component length along the flow path should be kept in the same length ratio l_R, making the fluid residence time ratio the same as the system time ratio τ_R. Moreover, for the SG as a heat sink, the thermal center height difference between the core and the SG l_{th} should be reduced to scale as specified by the length ratio l_R. Since the length of core is scaled to l_R, the SG thermal center height (from the U-tubes bottom) l_{SGth} should also be scaled to l_R. Thus, the following equation should be satisfied:

$$l_{SG,R} = l_{SGth,R} = l_R \tag{7}$$

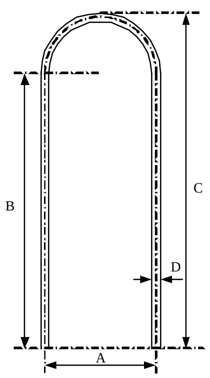

Fig.3 The dimensions of U-tube.

The U-tube has a U shape as shown in Fig.3, and A, B, C and D represent the U-tube span length, the vertical part length, the total height and the inner diameter respectively. There are over ten thousand U-tubes in each SG for a one thousand megawatts electric power class PWR, and therefore to model each U-tube in the scaled-down test facility is unrealistic. Normally, the number of U-tubes is reduced to a reasonable scale designed to properly represent the prototype. When performing U-tube scaling design, the U-tubes can be treated by geometric average with the average valve <A>, <B>, and <C> , which can be calculated by:

$$< A >= \frac{A_1 N_1 + A_2 N_2 + \cdots A_n N_n}{N_1 + N_2 + \cdots + N_n} \tag{8}$$

$$< B >= \frac{B_1 N_1 + B_2 N_2 + \cdots B_n N_n}{N_1 + N_2 + \cdots + N_n} \tag{9}$$

$$< C >=< B > + \frac{< A >}{2} \tag{10}$$

where the subscript number denotes identification of group with the same geometry, and N_i is the total number of tubes of the i group. The average tube length can be expressed by:

$$< l_{SG} >= 2 < B > + \frac{\pi}{2} < A > \tag{11}$$

It is very difficult to accurately predict the thermal center height due to the complex heat transfer process in SG, and therefore in the scaling analysis, the assumption of a linear temperature distribution along the U-tube is normally used to simplify the problem to a reasonable extent [5]. Under a linear temperature distribution condition, the SG thermal center height can be obtained by integration calculation, yielding:

$$< l_{SGth} >= \frac{< B >}{2} + \frac{< A >}{4\frac{< B >}{< A >} + \pi} \tag{12}$$

It can be easily seen from Eq. (11) and Eq. (12) that if <B> and <A> are both scaled to l_R, the scaling criteria can be met. It can be deduced that:

$$< l_{SGth} >_R = 1 + \frac{\pi}{4\lambda}\left(l_R - D_R\right) \tag{13}$$

Where $\lambda =< B > / < A >$. So under a linear scaling, the thermal center height ratio and length ratio match each other.

If the span length can be adjusted in horizontal direction to the length scale ratio, one can still make the thermal height ratio in the model the same as in the prototype. However, in a reduced scale system, the length ratio is normally different from the diameter ratio, which is particularly the case for a full height and reduced diameter system. This makes the average span length <A> scaled to a relatively smaller value than the vertical length <B>, and if one wants to keep the length ratio, the vertical length of the model will be increased and thus the thermal center height will be enhanced, making the NC flow rate faster than expected. To solve this problem, one can lower the elevation of the SG tube sheet by a proper value to get the same thermal height elevation while keeping the U-tube length scale ratio and system height scale ratio identical.

4.2 Heat transfer and U-tube diameter

The system energy balance equation during the NC phase after the break can be expressed as:

$$\frac{dU}{dt}=(\dot{m}h)_{in}-(\dot{m}h)_{out}+\dot{q}_{core}-\dot{q}_{SG}-\dot{q}_{loss}-P\frac{dV}{dt} \quad (14)$$

Where U is the internal energy of the fluid mixture within the control volume, $\dot{m}$ is the mass flow rate leaving or entering the system (subscripts *in* and *out* represent entering and leaving), *h* is the flow enthalpy, $\dot{q}_{core}$ is the decay heat power, $\dot{q}_{SG}$ is the heat removal power by SG, $\dot{q}_{loss}$ is the heat loss, P is the system pressure and V is the RCS control volume, which remains constant. The heat loss can be neglected relative to the core decay power and SG power, and therefore the energy balance equation can be simplified as:

$$\frac{dU}{dt}=(\dot{m}h)_{in}-(\dot{m}h)_{out}+\dot{q}_{core}-\dot{q}_{SG} \quad (15)$$

Using the initial break power relief value $(\dot{m}h)_{out,0}$ to non-dimensionalize the equation, it becomes:

$$\begin{aligned}\frac{U_0}{(\dot{m}h)_{out,0}}\frac{dU^+}{dt}&=\frac{(\dot{m}h)_{in,0}}{(\dot{m}h)_{out,0}}(\dot{m}h)^+_{in}-(\dot{m}h)^+_{out}\\&+\frac{\dot{q}_{core,0}}{(\dot{m}h)_{out,0}}\dot{q}^+_{core}-\frac{\dot{q}_{SG,0}}{(\dot{m}h)_{out,0}}\dot{q}^+_{SG}\end{aligned} \quad (16)$$

The system pressure P and the temperature T are related to its inner energy U. In order to represent the prototype behavior, the coefficient ratio in Eq.(16) should be unit, leading to:

$$\left[\frac{(\dot{m}h)_{in,0}}{(\dot{m}h)_{out,0}}\right]_R=\left[\frac{\dot{q}_{core,0}}{(\dot{m}h)_{out,0}}\right]_R=\left[\frac{\dot{q}_{SG,0}}{(\dot{m}h)_{out,0}}\right]_R=1 \quad (17)$$

With the fluid property similitude, to keep the scaling ratio being at unit, the correct value of $(\dot{m}h)_{in}$ and $(\dot{m}h)_{out}$ of the model can be achieved by carefully choosing the orifice size to adjust the break size and the friction of injection line. By NC scaling analysis, the core decay power ratio satisfies the reasonable scaling requirements, so it can be seen from Eq.(17) that the SG power ratio should be:

$$\dot{q}_{SG,R}=a_R\sqrt{l_R} \quad (18)$$

The total U-tubes heat transfer area ratio should be:

$$A_{HT,R}=N_R D_{Utube,R} l_R \quad (19)$$

And the total flow area of the primary side should be:

$$a_R=N_R D^2_{Utube,R} \quad (20)$$

Therefore Eq.(19) can be rewritten as:

$$A_{HT,R}=\frac{a_R l_R}{D_{Utube,R}} \quad (21)$$

And Newton’s law for heat transfer:

$$\dot{q}=A_{TH}h\Delta T \quad (22)$$

For the convection heat transfer, the heat transfer coefficient can be expressed as:

$$h=Nu\frac{k}{D_H} \quad (23)$$

Where, *Nu* is the Nusselt number, *k* is the thermal conductivity of the coolant, and D_H is the heat transfer pipe hydraulic diameter. So the SG decay heat removal power can be expressed as with fluid similitude and the same tube material:

$$\dot{q}_{SG,R}=A_{HT,R}h_R\Delta T_R=l_R Nu_R\frac{a_R}{D^2_{Utube,R}} \quad (24)$$

And the Nusselt number can be expressed by the equation:

$$Nu=C\,\mathrm{Re}^{\alpha}\,\mathrm{Pr}^{\beta}\left(\frac{\mu_w}{\mu}\right)^{\kappa} \quad (25)$$

Where μ_w is fluid viscosity at the wall temperature, C, α, β and κ are the coefficients that depend on the fluid and pipe structure. Pr is the Prandtl number that depends on the fluid property. Re is the Reynolds number: $\mathrm{Re}=\rho u D_H/\mu$. Thus Eq.(24) can be rewritten as:

$$\dot{q}_{SG,R}=\frac{a_R l_R u^{\alpha}_R}{D^{2-\alpha}_{Utube,R}} \quad (26)$$

With the NC scaling relation, $u_R=\sqrt{l_R}$, and Eq.(26), one can see that the diameter of the U-tube should be:

$$D_{Utube,R}=(u_R)^{\frac{1+\alpha}{2-\alpha}}=(l_R)^{\frac{1+\alpha}{2(2-\alpha)}} \quad (27)$$

For the extensively used Nu number, α is 0.8[6]. Therefore the U-tube diameter with a different height ratio can be obtained as shown in Fig.4.

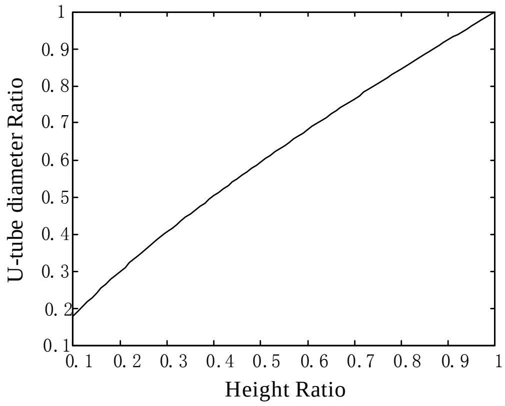

Fig.4 U-tube diameter ratio versus system height ratio.

It can be seen from Fig.4 that the U-tube diameter ratio decreases as the height ratio decreases, and the diameter ratio is slightly larger than the length ratio in the reduced height ratio system. The tube number will greatly increase for a fixed flow area due to the N~1/D^2. So with the small height scale, the thinner U-tube should be selected and also a large number of holes in the tube sheet would have to be drilled. It should be noticed that the heat transfer correlation used above is applicable to the single-phase condition, and if the two-phase fluid condition occurs in the U-tube, the analysis result would be different. Because the two-phase heat transfer analysis becomes increasingly complex, one can take a single-phase design guidance and use another approach, such as numerical simulation, to adjust the design results.

5 The transition from heat sink to heat source

5.1 Decay power removal by the drying-out of SG secondary inventory

When decay power is removed by the SG NC, the proper heat removal rate is given by Eq.(18). And thus the integrated heat removal by the SG also needs to be properly scaled. Given the time ratio $\tau_R = \sqrt{l_R}$, the integrated heat energy removal ratio by the SG can be expressed as:

$$Q_{SG,R} = a_R l_R \tag{28}$$

And Eq.(28) can be used to determine the SG secondary inventory at the time of shut off of the reactor and of the main steam line.

As an integral test system, all the important thermal hydraulic behavior occurring in the model when it simulates a LOCA should have an identical time ratio to keep the global system response similar to that of the prototype. Only in this way, the event sequence can be properly simulated by the model.

During a LOCA, the SG first plays the role of heat sink to remove the decay heat from the primary side, and then after the inventory of the SG secondary side is dried out, the SG become the heat source of the primary side. In order to keep the heat sink transition in the correct event order, the secondary side dry-out time should be preserved with the system time ratio.
The PORV making the pressure of the SG secondary side relatively stable, the water in the secondary side boil-off process can be expressed by:

$$M_{SCE} = \frac{\int_0^{t_d} \dot{q}_{core} dt}{\Delta h_{sub} + h_{fg}} \tag{29}$$

The secondary side water mass ratio can be expressed by:

$$M_{SCE,R} = \frac{\left(\int_0^{t_d} \dot{q}_{core} dt\right)_R}{\left(\Delta h_{sub} + h_{fg}\right)_R} \tag{30}$$

Where the core power is scaled to the following expression according to the scaling analysis and taking in consideration the same SG secondary side fluid property:

$$\frac{\dot{q}_{core,m}(\sqrt{l_R} t)}{\dot{q}_{core,p}(t)} = a_R \sqrt{l_R} \tag{31}$$

Taking the power ratio as $a_R \sqrt{l_R}$ and the time ratio as $\sqrt{l_R}$, from Eq.(30) it results that:

$$M_{SCE,R} = a_R l_R \tag{32}$$

One can see that the result given by Eq.(32) is coherent with Eq.(28). The SG shell inner diameter keeps a_R . The feed water level ratio in the secondary side equals the system height ratio l_R . And it is simple to duplicate the dry-out event sequence.

5.2 Stored energy release by SG

The SG will finally change to a heat source due to its stored energy, and its power should satisfy the scaling requirement, which can be expressed as:

$$\left(\frac{dE_{SG}}{dt}\right)_R = a_R\sqrt{l_R} \quad (33)$$

where E_{SG} is the energy stored in the SG, to be released to the primary side. In general, if one has no heat transfer correlations and the flow discharge equations for the secondary side, one cannot specify the stored energy release process. This will be very challenging to the problem at hand. To properly scale the stored energy release, one common method focuses on the integrated power, and on the total energy released by the metal, regardless of the specified heat transfer process. Thus, the stored energy analysis is achievable, and the total energy stored in the SG metal can be written as:

$$E_{SG} = \eta\Delta T\sum\left(m_{SG,i}C_{vs,i}\right) \quad (34)$$

Where ΔT is the SG temperature difference caused by the temperature dropping from the initial state under normal operation to the saturate temperature under the containment pressure in a long cooling term. And η is the factor representing the energy loss to the ambient. And if the energy loss to the ambient is negligible, the SG component mass should satisfy the equation below:

$$\left[\sum\left(m_{SG,i}C_{vs,i}\right)\right]_R = \frac{a_R l_R}{\Delta T_R} \quad (35)$$

for a full pressure test system, $\Delta T_R = 1$, and for a reduced pressure system $\Delta T_R < 1$. Eq.(34) can be used for SG heat storage evaluation. By the evaluation, one can judge if the SG metal mass used in the model is acceptable or not.

6 Conclusion

During a LOCA the SG first plays the role of a heat sink when it works under a natural circulation mode to remove the decay heat from the primary coolant to the secondary side in the early phase, and then after the drying-out of the secondary inventory, the SG changes its role from heat sink to heat source in relation to the primary side. In order to simulate the transient of the SG properly, the SG scaling analysis is performed. For the U-tube, which is the crucial part in the SG in order to determine the heat transfer performance, its dimension in the model can be figured out using the NC scaling rules. And the heat transfer scaling analysis gives the relation between the U-tube diameter ratio and the height ratio. Moreover, the U-tube length ratio and the thermal center height ratio cannot both meet the scaling requirements when the height ratio and the diameter ratio are different in the scaled model, which can be solved by lowering the tube sheet by a proper value. Finally, the SG transition from the role of a heat sink to a heat source was analyzed, and it showed that the SG secondary inventory and the total SG material heat capacity in the model need to be scaled properly to represent the transition correctly.

A linear heat transfer rate and a widely used convection heat transfer correlation were used in this work. To improve the SG scaling analysis, studying the heat transfer model of a U-tube would be of particular benefit, especially under a two-phase condition.

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