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Thermal-hydraulic design and transient analysis of passive cooling system for CPR1000 spent fuel storage pool

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Abstract This paper proposes a design of passive cooling system for CPR1000 spent fuel pool (SFP). Our design can effectively manage the SFP temperature not to exceed 80 °C. Then the transient analysis of the CPR1000 SFP with designed passive cooling system is carried out in station blackout (SBO) accident by the best-estimate thermal-hydraulic system code RELAP5. The simulation results show that to maintain the temperature of CPR1000 SFP under 80 °C, the numbers of the SFP and air cooling heat exchangers tubes are 6627 and 19 086, respectively. The height difference between the bottom of the air cooling heat exchanger and the top of the SFP heat exchanger is 3.8 m. The number of SFP heat exchanger tubes decreases as the height difference increases, while the number of the air cooling heat exchanger tubes increases. The transient analysis results show that after the SBO accident, a stable natural cooling circulation is established. The surface temperature of CPR1000 SFP increases continually until 80 °C, which indicates that the design of the passive air cooling system for CPR1000 SFP is capable of removing the decay heat to maintain the temperature of the SFP around 80 °C after losing the heat sink.

Keywords Spent fuel pool · CPR1000 · Passive cooling system · RELAP5

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List of symbols

k	Total heat transfer coefficient W/(m^2 K)
d_i	Tube inner diameter (m)
d_o	Tube outside diameter (m)
h_i	Tube inner heat transfer coefficient W/(m^2 K)
h_o	Tube outside heat transfer coefficient W/(m^2 K)
η_f	Heat rejection efficiency of fins
λ	Heat conduction coefficient W/(m^2 K)
A_i	Total internal surface area of the heat exchanger (m^2)
A_h	Total external surface area of the heat exchanger (m^2)
A_f	Total surface area of the fins (m^2)
A_r	External surface area except the fin surface area (m^2)
Φ	Heat transfer power (W)
ΔT_m	Logarithmic mean temperature difference
Nu	Nusselt number
Gr	Grashof number
Pr	Prandtl number
g	Gravitational acceleration (m/s^2)
l	Tube length (m)
α_v	Expansion coefficient (1/K)
v	Kinematic viscosity (m^2/s)
T_w	Wall temperature (K)
T_∞	Water average temperature (K)
H	Height of the interlayer (m)
δ	Thickness of the interlayer (m)
f	Friction factor
De	Equivalent diameter (m)
u	Fluid velocity (m/s)
u_1	Velocity in the upstream (m/s)
u_2	Velocity in the downstream (m/s)
A_1	Upstream area (m^2)

A_2 Downstream area (m^2)
F_t Structure correction factor
N_s The number of serial heat exchangers
N_p The number of tube passes
u_3 Velocity in the tube (m/s)
Δh The height difference between the top of the SFP heat exchanger and the bottom of the air cooling heat exchanger (m)
T_{in} Inlet temperature of the heat exchanger (°C)
T_{out} Outlet temperature of the heat exchanger (°C)
T_i Inner tube wall temperature of the heat exchanger (°C)
T_o Outer tube wall temperature of the heat exchanger (°C)
G Mass flow of the passive cooling loop (kg/s)
Q Decay heat (MW)

1 Introduction

The spent fuel pool (SFP) is one of the most important equipments in the nuclear power plant. It plays a major role in removing the decay heat and shielding the radiation produced by the spent fuel during the process of spent fuel storage. After the Fukushima nuclear accident in 2011, investigations on the thermal-hydraulic characteristics and the safety performance of the spent fuel pool have become the research focus in the world.

In 2012, China promulgated an 863 project named "nuclear safety research." One of the subtopics is aimed to analyze and improve the safety of the spent fuel pool for generation II+ pressurized water reactor, such as CPR1000 which is the major reactor type in service or in construction in China. However, the cooling system of the SFP for generation II+ PWR is depending on the driving forces. Once without the driving forces, the water of the SFP will be boiling and the fuel will be melt. The passive safety systems were put forward in the 1980s, which depend on the gravity and natural circulation instead of operator actions or electronic feedback in the emergency event. The passive system has been widely applied in generation III reactors, such as the Westinghouse AP1000 [1], the Japanese N.G.P, the Chinese ACPR1000 [2], the Russian VVER, the PIUS with inherent safety features designed by ABB Power Generation Ltd in Switzerland [3], the PWR with SIP-1 presented by SIET laboratory in Italy [4].

After Fukushima accident, a number of researchers have studied the safety of the spent fuel pool. Carlos et al. used TRACE thermal-hydraulic code to simulate the steady-state and transient conditions with a loss of cooling and coolant of spent fuel pools. Their calculations of steady state have a good agreement with the measurements of Maine Yankee [5]. Chen et al. have developed a 3-D CFD model to analyze the localized distributions of the flow and heat transfer of the spent fuel pool in a loss of cooling event. The results show that the temperature rising rate calculated by the 3-D CFD is identical to that calculated by Procedure 597.1 for the Maanshan NPP [6]. Wang et al. [7] applied RELAP5 to investigate the behavior of the spent fuel pool for CPR1000 with the loss of heat sink.

Some researchers have developed the advanced passive safety system for SFP. Arndt et al. [8] introduced a design based on the concept of cooling the spent fuel storage pool only depending on air cooling. Westinghouse Corporation [9] designed a floating passive cooling system using gravitate heat pipe principle to manage the spent fuel pool cooling. Their system is composed of many modular gravitate heat pipes. Each heat pipe may at most remove 300 kW of heat, and the cooling system can control the temperature of the pool water under 60 °C. Ye et al. [10] designed a passive cooling system for spent fuel pool of CAP1400. Their cooling system adopts separate heat pipe principle and removes the decay heat of spent fuel to atmospheric environment depending on the air cooling tower. This passive cooling system can keep the pool water from boil, even under station blackout (SBO).

In order to take away the decay heat of the SFP effectively, the heat exchanger used in nuclear industry needs to be studied to improve heat exchange efficiency. Heat exchange efficiency is determined mainly by the structures of heat exchangers. Partial modification of the tubular heat exchanger can raise its heat exchange efficiency. Ye et al. [11] and You et al. [12] developed a powerful numerical model tool to predict the thermal hydraulic performances of the shell-and-tube heat exchanger, and they found the shell-and-tube heat exchanger (STHXs) with flower baffles has a better overall thermal hydraulic performance than the STHXs with segmental baffles. Then they conducted the experimental and numerical investigations of shell-side thermo-hydraulic performances for the STHXs with trefoil-hole baffles. Both results showed the heat transfer rate on the shell side of the STHXs with trefoil-hole baffles was significantly enhanced, and the flow resistance increased substantially as well.

In the present paper, a passive system with air cooling for the SFP of CPR1000 is designed to control its water temperature under 80 °C. The safety performance of the SFP with passive cooling system is analyzed under station blackout using RELAP5 code. The research of this paper is useful for the safety improvement of CPR1000 spent fuel pool system.

2 Thermal-hydraulic design of passive cooling system

The CPR1000 (improved Chinese PWR) which is based on the French 900 MW PWR with upgraded net power output of 1000 MW is the main reactor type in service and under construction in China. So its safety behavior is a critical issue for Chinese nuclear industry.

When SBO accident takes place, the water temperature of the SFP will continue to rise until boiling. If no active cooling measures are available, the water will evaporate away. Then the radioactive materials will release into the environment to cause danger. The passive cooling system is applied to remove the decay heat effectively and control the water temperature below an acceptable level when the active devices failure or loss of heat sink occurs.

The sketch of the designed passive cooling system is shown in Fig. 1. The heat exchangers in the SFP and air are all shell-and-tube type. The SFP heat exchanger is immersed in the pool around the pool wall to load or unload the spent fuels assemblies conveniently. Water is used as the working fluid in the closed loop. The atmosphere is used as the final heat sink. The air cooling heat exchanger is equipped with uniform section circular fins to increase the heat transfer area.

Under normal operating conditions, the tube side of the heat exchangers and the connecting pipes are filled with water and isolated by a check valve and an isolation valve. After the SBO occurs, the check and isolation valves will open automatically and the passive cooling system will start to work automatically without any active operation. The water heated in the SFP heat exchanger flows upward and enters the tube side of the air cooling heat exchanger, where it is cooled by the shell side air by natural convection. Finally, the cooled water flows back to the tube side of the SFP heat exchanger by gravity. Consequently, the passive SFP cooling system removes the decay heat by natural circulation under station blackout accident.

2.1 The heat transfer model in the passive cooling system

Ignoring the fouling resistance, the total heat transfer coefficient correlations for SFP and air heat exchangers based on tube outer surface are, respectively, as follows

$$\frac{1}{k} = \frac{1}{h_\mathrm{i}}\frac{d_\mathrm{o}}{d_\mathrm{i}} + \frac{d_\mathrm{o}}{2\lambda}\ln\frac{d_\mathrm{o}}{d_\mathrm{i}} + \frac{1}{h_\mathrm{o}}, \tag{1}$$

$$\frac{1}{k} = \frac{A_\mathrm{h}}{h_\mathrm{i}A_\mathrm{i}} + \frac{A_\mathrm{h}}{2\pi\lambda l}\ln\frac{d_\mathrm{o}}{d_\mathrm{i}} + \frac{A_\mathrm{h}}{h_\mathrm{o}(A_\mathrm{r} + A_\mathrm{f}\eta_\mathrm{f})}. \tag{2}$$

In Eq. (2), η_f is the heat rejection efficiency of fins and can be found according to the fin efficiency curve [13]; A_i and A_h is the total internal and external surface area; A_f is the fin surface area; A_r is the external surface area except the fin surface area; $A_\mathrm{h} = A_\mathrm{r} + A_\mathrm{f}$. Figure 2 shows the heat transfer process through the rib wall.

The heat transfer equation is as follows:

$$\Phi = kA_\mathrm{h}\Delta T_\mathrm{m}. \tag{3}$$

In Eq. (3), A_h is the total external surface area and ΔT_m is the logarithmic mean temperature difference, and it can be expressed by

$$\Delta T_\mathrm{m} = \frac{\Delta T_\mathrm{max} - \Delta T_\mathrm{min}}{\ln\frac{\Delta T_\mathrm{max}}{\Delta T_\mathrm{min}}}. \tag{4}$$

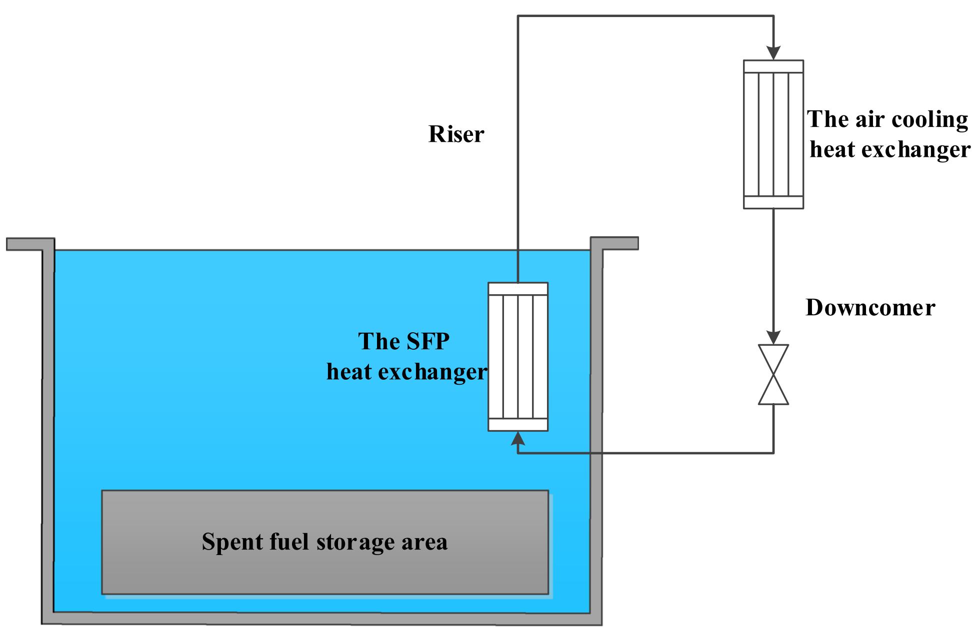

Fig. 1 Design diagram of passive cooling system

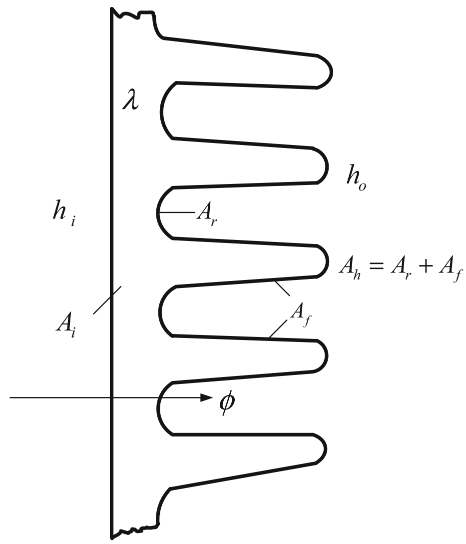

Fig. 2 Heat transfer process through the rib wall

The heat transfer coefficient between the wall and water in the tubes of the SFP or air cooling heat exchanger is calculated by Churchill–Chu correlation [14]:

$$\mathrm{Nu} = \left[0.825 + \frac{0.387(\mathrm{Gr}\times\mathrm{Pr})^{1/6}}{\left[1+\left(\frac{0.492}{\mathrm{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2, \tag{5}$$

where

$$\mathrm{Nu} = \frac{hl}{\lambda}, \tag{6}$$

$$\mathrm{Gr} = \frac{g\alpha_V |T_\mathrm{w} - T_\infty| l^3}{v^2}. \tag{7}$$

The Churchill–Chu correlation is reported to be valid over the full laminar and turbulent Rayleigh number range. It is usually used for natural convection, especially for a vertical flat plate.

For the heat exchanger without fins, the heat transfer mode between the tube outer wall and the coolant belongs to natural convection in infinite space. For the heat exchanger with fins, the heat transfer mode may belong to natural convection in infinite space or natural convection in finite space depending on the situation. Seigel et al. [15] found if the ratio of the fin pitch to fin height is larger than 0.28, the heat transfer mode between the fins and the coolant is natural convection in infinite space. Otherwise, it will be natural convection in finite space. The natural convection heat transfer correlation in infinite space is as follows [13]:

$$\mathrm{Nu} = C(\mathrm{Gr}\times\mathrm{Pr})^n. \tag{8}$$

The natural convection heat transfer correlations in vertical and horizontal finite space can be presented, respectively, as follows [13]:

$$\mathrm{Nu} = C(\mathrm{Gr}\times\mathrm{Pr})^n\left(\frac{H}{\delta}\right)^m, \tag{9}$$

$$\mathrm{Nu} = C(\mathrm{Gr}\times\mathrm{Pr})^n. \tag{10}$$

Equation (10) is fully applicable to gas. But for liquid, considering the relation between the properties and the temperature, the right side of this equation should be multiplied by a correction factor depending on the property change. So the heat transfer correlation for the tube outside is modified as

$$\mathrm{Nu} = C(\mathrm{Gr}\times\mathrm{Pr})^n(\mathrm{Pr_f}/\mathrm{Pr_w})^{0.11}. \tag{11}$$

In Eqs. (8)–(11), C, n, and m are determined by experiments, depending on the heat transfer surface shape and position, turbulent or laminar flow, and thermal boundary conditions. Their values can be found in the literatures [13].

2.2 The pressure drop model in the passive cooling system

The establishment of the stable natural circulation is based on the situation that the pressure drop produced by the density difference between the riser and downcomer must overcome the resistance pressure drop in the loop:

$$\Delta P_\mathrm{el} = \sum\Delta P_\mathrm{f} + \sum\Delta P_\mathrm{c,s} + \sum\Delta P_\mathrm{c,c} + \sum\Delta P_\mathrm{c,e} + \sum\Delta P_t, \tag{12}$$

where

The elevation pressure drop:

$$\Delta P_\mathrm{el} = P_\mathrm{down} - P_\mathrm{up}. \tag{13}$$

The frictional pressure drop:

$$\Delta P_\mathrm{f} = f\frac{l}{D_\mathrm{e}}\frac{\rho u^2}{2}. \tag{14}$$

The elbow, junction, or valve pressure drop:

$$\Delta P_\mathrm{c,s} = K\frac{\rho u^2}{2}. \tag{15}$$

The area abrupt contraction pressure drop:

$$\Delta P_\mathrm{c,c} = 0.4\left[1-\left(\frac{A_2}{A_1}\right)^2\right]\frac{\rho u_2^2}{2}. \tag{16}$$

The area abrupt expansion pressure drop:

$$\Delta P_\mathrm{c,e} = \left(1-\frac{A_1}{A_2}\right)^2\frac{\rho u_1^2}{2}. \tag{17}$$

The heat exchanger tube pressure drop:

$$\Delta P_t = \left(f\frac{l}{D_\mathrm{e}} + 3\right)F_t N_\mathrm{s} N_\mathrm{p}\frac{\rho u_3^2}{2}, \tag{18}$$

where K is form drag coefficient; F_t is structure correction coefficient with the value of 1.4 for the $\phi 25 \times 2.5$ pipe; f is friction factor which can be checked from the Moody figure according to the pipe roughness and fluid Reynolds number Re.

2.3 Thermal-hydraulic design

According to the design basis of the spent fuel pool for CPR1000, the max permissible temperature of the pool water is 80 °C. In this paper, 80 °C is designed as the final temperature of the spent fuel pool in a loss of heat sink event. The environment temperature is designed to 30 °C. According to the target, the numbers of the SFP heat exchanges tubes N_{SFP} and the air cooling heat exchanges tubes N_{air} and the height difference Δh between the top of the SFP heat exchanger and the bottom of the air cooling heat exchanger are calculated by other designed parameters.

The first step is to calculate the outlet temperature of the SFP heat exchanger T_{out} by the inlet temperature of the SFP heat exchanger T_{in} and the mass flow G of the passive cooling loop. The second step is to calculate the tube number of the SFP heat exchanger N_{SFP} by the decay heat Q of the SFP and the other designed geometrical parameters. The number of the air cooling heat exchanger tubes N_{air} is calculated using the same method. The designed geometrical parameters of the passive cooling system are shown in Table 1. From the table, the value of the fin pitch to the fin height of the air cooling exchanger is 0.33. So the heat transfer mode between the fins and the coolant of the air cooling exchanger is natural convection in infinite space. The last step is to calculate the height difference Δh between the top of the SFP heat exchanger and the bottom of the air cooling heat exchanger by the mass flow G and the pressure drop of the passive cooling loop. The form drag coefficient K here is 0.56.

The detailed calculation procedure is shown in Fig. 3.

Other parameters of the passive cooling system are given in Table 2. The power of the spent fuel assemblies is set to 11.98 MW, which is the maximum power in abnormal conditions.

The number of each heat exchanger tubes and height difference between the two heat exchangers are calculated by the design parameters, as given in Table 3. The height difference is 3.8 m, which is easy to achieve from the view of engineering. These results indicate the designed passive cooling system for the spent fuel pool of CPR1000 is practical significance.

2.4 Height difference sensitivity analysis

As shown in Sect. 2.3, the establishment of the stable natural circulation is based on the situation that the pressure drop produced by the density difference between riser and downcomer must overcome the resistance pressure drop of the loop. So the height difference between the bottom of the air heat exchanger and the top of the SFP heat exchanger which has a significant influence on pressure drop will affect the stable natural circulation.

Figure 4 shows the curve of the mass flow of the passive cooling loop with the height difference.

Keeping other parameters constant, the numbers of the air and SPF heat exchangers tubes are changed with the height difference between the bottom of the air heat exchanger and the top of the SFP heat exchanger. Figures 5 and 6 show the curves of numbers of the air and SFP heat exchangers tubes with the height difference, respectively. As they show, the number of SFP heat exchanger tubes decreased as height difference increased, while the number of the air cooling heat exchanger tubes increased.

This is because the mass flow of the natural circulation increases with the height difference. According to Eq. (19),

$$Q = C_\mathrm{p} G (T_{\mathrm{out}} - T_{\mathrm{in}}), \tag{19}$$

where the decay heat Q and inlet temperature T_{in} are designed to be a constant, so T_{out} decreases with the increase in the height difference. This leads the logarithmic mean temperature difference of the SFP heat exchanger increase and the logarithmic mean temperature difference of the air cooling heat exchanger decrease. Finally, the heat transfer area of the SFP heat exchanger decreased and the heat transfer area of the air cooling heat exchanger increased as height difference increased shown in Eq. (3).

The ratio of the material area is defined as the ratio of the material area of the passive cooling system to the minimum of it. Figure 7 shows the curve of the ratio of the material area to height difference. In terms of economy, the height difference is better between 3 and 10 m.

Table 1 Geometrical parameters of the passive cooling system

SFP heat exchanger				Air cooling heat exchanger		
Tube length (m)	Tube type	Tube length (m)	Tube type	Fin height (m)	Fin thickness (m)	Fin distance (m)
6.5	$\phi 25 \times 2.5$	12.0	$\phi 25 \times 2.5$	0.03	0.002	0.01

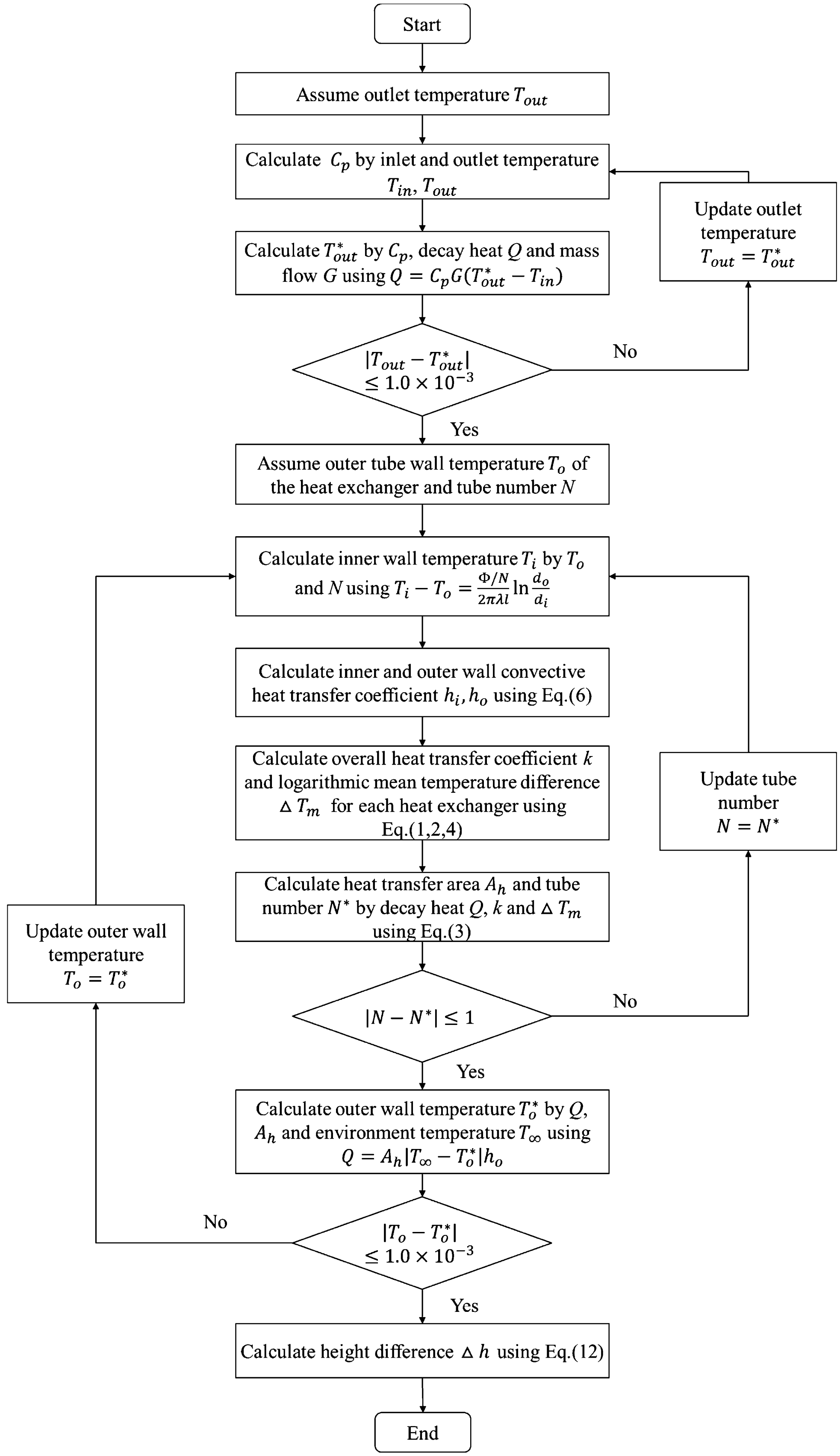

Fig. 3 Calculation flowchart of the passive cooling system

Table 2 Other assumed parameters of the passive cooling system

Parameters	Values
Environment temperature (°C)	30
Pool temperature (°C)	80
Inlet temperature of the SPF heat exchanger (°C)	35
Mass flow of the passive cooling loop (kg/s)	65
Diameter of the passive cooling loop pipe (m)	0.3
Power of the spent fuel assemblies (MW)	11.98
Working fluid	Water

Table 3 The steady-state results of the passive cooling system

Parameters	Values
The number of the SPF heat exchanger tubes	6627
The number of the air heat exchanger tubes	19 086
Height difference (m)	3.8

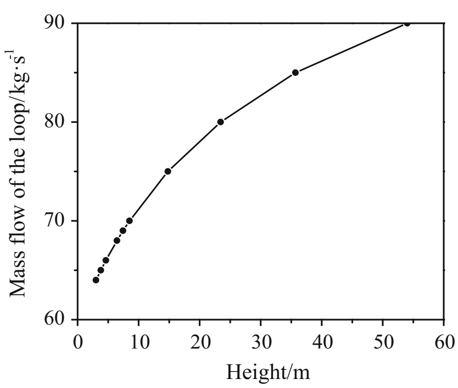

Fig. 4 Curve of the mass flow with height difference

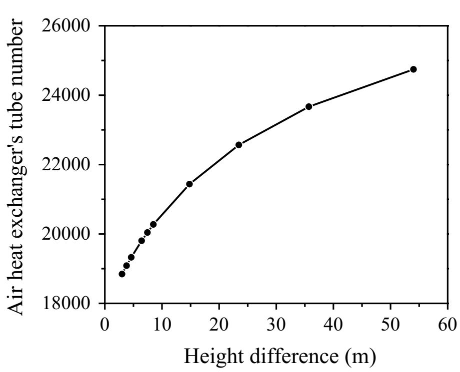

Fig. 5 Curve of the number of the air heat exchanger tubes with height difference

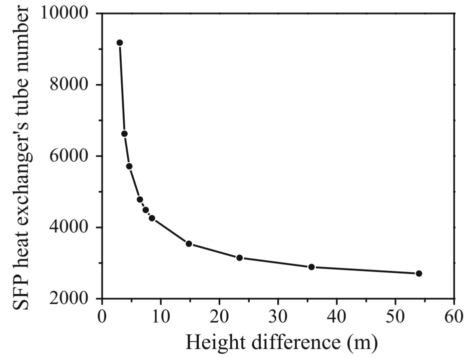

Fig. 6 Curve of the number of the SFP heat exchanger tubes with height difference

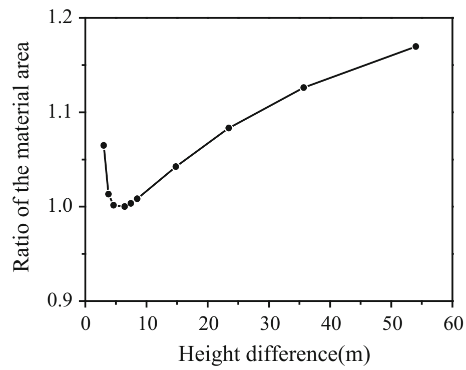

Fig. 7 Curve of the ratio of the material area to height difference

3 The passive cooling system performance analysis based on RELAP5

This section will use RELAP5 to analyze the designed passive cooling system, taking the result of Sect. 2 as the input conditions. We need to verify whether the passive cooling system can control the pool temperature at 80 °C after loss of heat sink occurs.

3.1 RELAP5 code introduction

RELAP5 developed by the Idaho National Engineering Laboratory for the U.S. Nuclear Regulatory Commission is a general system code for transient behavior analysis, used as the basis for a nuclear plant analyzer. RELAP5 adopts the two-phase flow model and uses the semi-implicit algorithm or nearly implicit algorithm to solve the model. It can simulate transients in LWR systems such as loss of coolant (LOCA), anticipated transients without scram (ATWS), and operational transients such as loss of feed-water, loss of offsite power, station blackout (SBO), and turbine trip. RELAP5 is a highly generic code that it can simulate a wide variety of thermal-hydraulic transients in

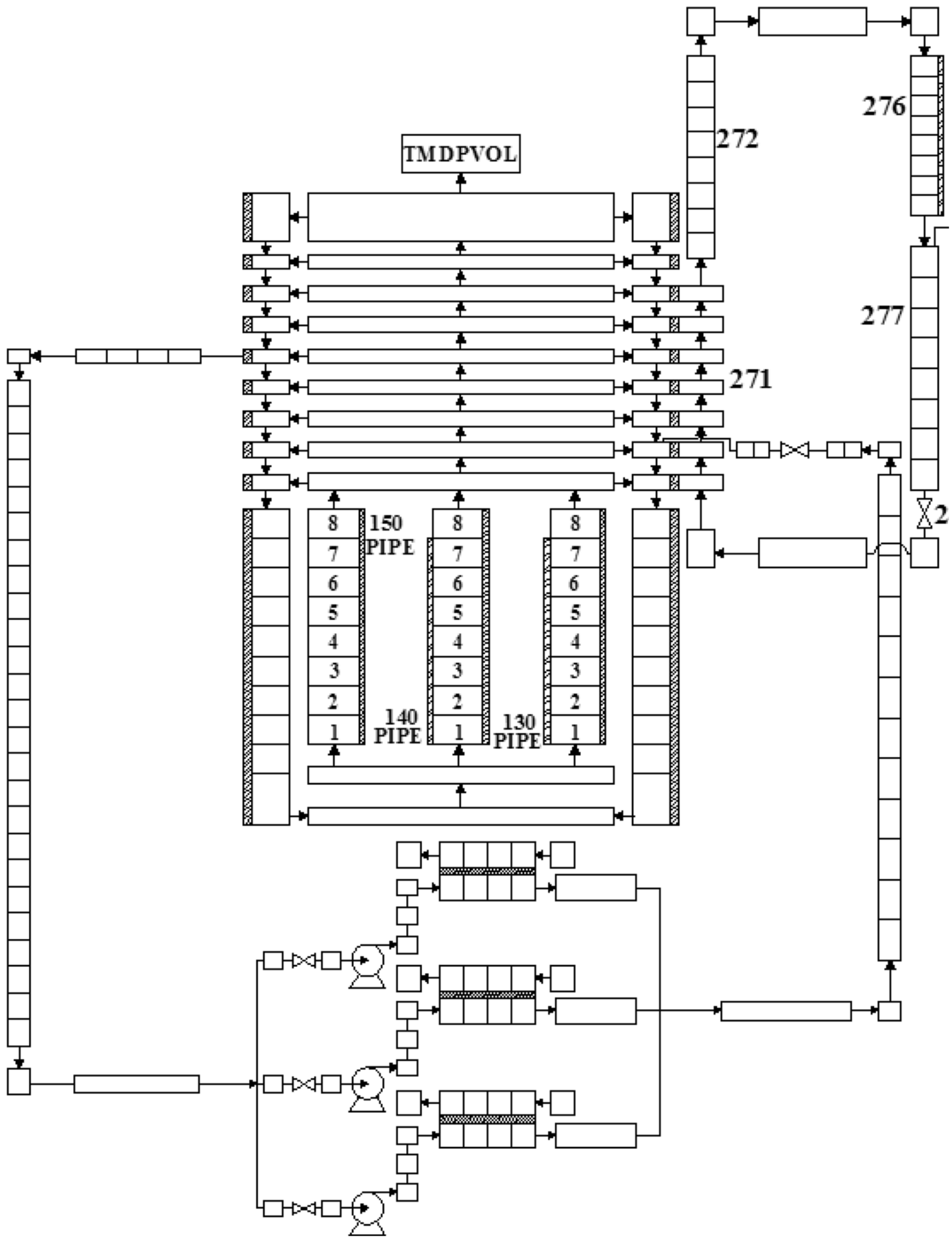

Fig. 8 Nodalization of the passive cooling system

both nuclear and nonnuclear systems involving mixtures of steam, water, noncondensable gas, and solute.

3.2 The passive cooling system for CPR1000 SFP

A nodalization of the passive cooling system for the SFP is shown in Fig. 8. Control volumes 271 and 276 represent the SFP and air cooling heat exchangers, respectively. Control volumes 272 and 277 represent the riser and downcomer of the loop, respectively; 287 is the valve at the downcomer and must be opened immediately to remove the decay heat when the loss of heat sink occurs.

Figure 9 shows the coolant mass flow change of the passive cooling system. The mass flux increases rapidly to 46.3 kg/s at 100 s after the accident. Early in the accident the large temperature difference between the SFP and air heat exchangers causes the large density difference and a large driving force in the loop.

As Fig. 10 shows, early in the accident the thermal load of the heat exchanger rises rapidly, but still under the decay heat of the spent fuel (11.98 MW). So the passive cooling system can't remove the decay heat, and the temperature of the pool surface increases continually, as shown in Fig. 11.

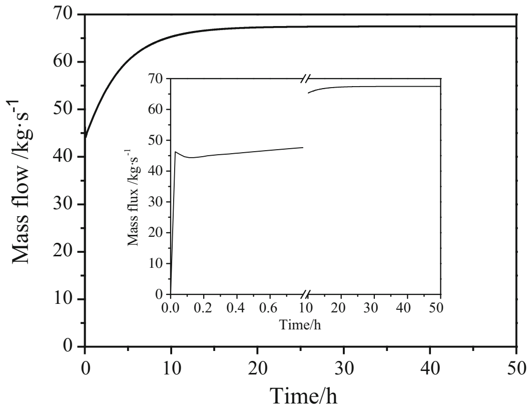

Fig. 9 Coolant mass flux change of the passive cooling system

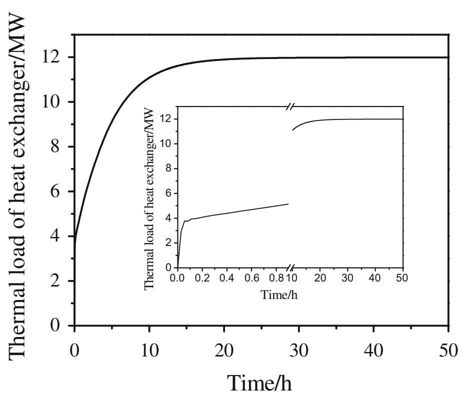

Fig. 10 Thermal load change of the heat exchanger

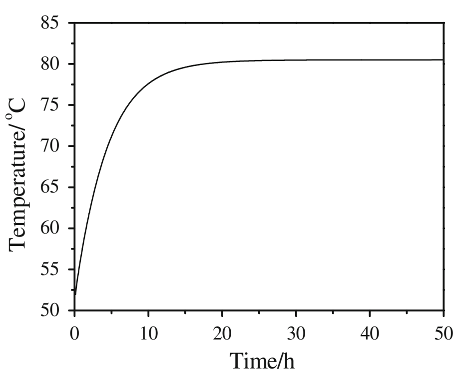

Fig. 11 Temperature profile of the pool surface

The increase of the water temperature results in a larger temperature difference between the cold and heat sources of the loop. Moreover, the driving force of the coolant increases, and the coolant mass flux increases with it. Meanwhile, the heat load of the heat exchanger increases until 11.98 MW to remove the decay heat of the spent fuel

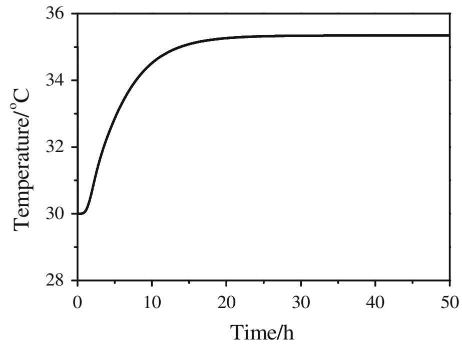

Fig. 12 Inlet temperature of the SFP heat exchanger

exactly. After that every parameter remains constant. It means the passive cooling system has been established successfully.

As shown in Fig. 11, the temperature of the pool surface is 80 °C after the establishment of the passive system. Figure 12 shows the inlet temperature of the SFP heat exchanger is 35 °C. They certify the passive cooling system we designed is effective.

4 Conclusion

In this paper, a passive cooling system for the SFP of CPR1000 is designed. The passive cooling system can maintain the temperature of the SFP around 80 °C after losing the heat sink.

Then the height difference sensitivity was analyzed. The result shows that the number of SFP heat exchanger tubes decreased as height difference increased, while the number of the air cooling heat exchanger tubes increased.

The loss of heat sink accident of the spent fuel storage pool with the passive cooling system was analyzed by RELAP5-MOD3.3, which used the theoretical calculation as the input. After the accident, the temperature of the pool surface increased continually until 80.5 °C. The result suggests that the design of the passive cooling system of the SFP for CPR1000 is successful to remove the decay heat to maintain temperature of the SFP around 80 °C after losing the heat sink. The present results will be useful for the improvement of the CPR1000 spent fuel pool cooling system.

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