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Proposal and Comparison of Passive Cooling Systems for Long-term Cooling of i-SMR

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1. Introduction

Small Modular Reactors (SMRs) are gaining attention as a next-generation electric power source due to their enhanced safety, efficiency, and flexibility compared to large-scale reactors. Among various types of SMRs, light water reactor (LWR)–based SMRs are considered the most promising, as they leverage the accumulated technological advancements of large LWRs and can be deployed within the next decade.

Notable examples of LWR-type SMRs include VOYGR (developed by NuScale Power Corporation) and i-SMR (developed by KHNP and KAERI) [1]. These designs incorporate passive safety systems to improve reactor safety and utilize a water pool as the final heat sink for passive cooling. However, if the water in the final heat sink is depleted, the cooling capability is significantly reduced, necessitating an auxiliary cooling system to ensure long-term cooling.

Several efforts have been made to develop auxiliary systems to enhance the long-term passive cooling capability of large LWRs. KAERI proposed an Air-Cooled Heat Exchanger (ACHX), which passively condenses evaporated steam from the Passive Condensation Cooling Tank (PCCT)—the final heat sink of APR+—to delay water depletion in the PCCT [2,3]. Additionally, Kim designed the Air-Cooled Passive Decay Heat Removal System (APDHR), which replaces the existing Passive Condensate Heat Exchanger (PCHX) to improve air-cooling effectiveness during the air-cooling phase [4].

In this study, an auxiliary passive cooling system is developed to enhance the long-term cooling capability of an i-SMR. Two design concepts, both of which passively condense evaporated steam from the final heat sink of i-SMR (Emergency Cooling Tank: ECT), are proposed and evaluated using a nuclear thermal-hydraulic system code. The performance of the two designs is then compared, and their respective characteristics are analyzed and discussed.

2. Design Proposal

The first design proposed in this study is the Large Loop Heat Pipe (LLHP) cooling tower (Figs. 1, 2). A wickless loop heat pipe, also known as a loop thermosyphon, is applied in this design due to its superior manufacturability compared to a wick-type heat pipe. This design absorbs heat by condensing steam at the evaporator and releases it into the air at the condenser. As the heated air rises, a natural air circulation path is established, as shown in Fig. 2.

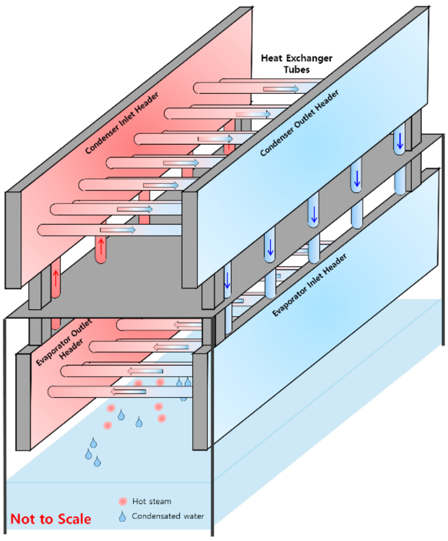

Fig. 1. Large loop heat pipe 3D conceptual diagram

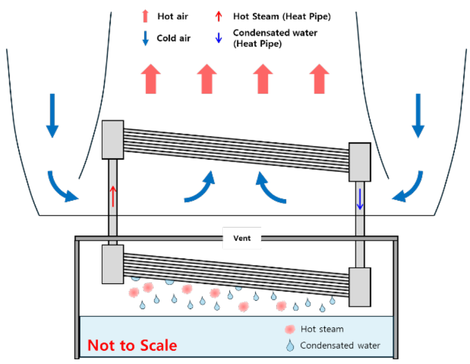

Fig. 2. Large loop heat pipe cooling tower side view

The second design proposed in this study is the Condensation Heat Exchanger (CHX) cooling tower (Fig. 3, 4). This design is structurally similar to the LLHP cooling tower; however, unlike the LLHP, which serves as a heat transfer medium, the CHX directly absorbs steam, condenses it within the heat exchanger tubes, and returns the condensate to the water pool. As the air is heated by the released heat from the heat exchanger tubes, a natural air circulation path is formed, similar to that in the LLHP cooling tower.

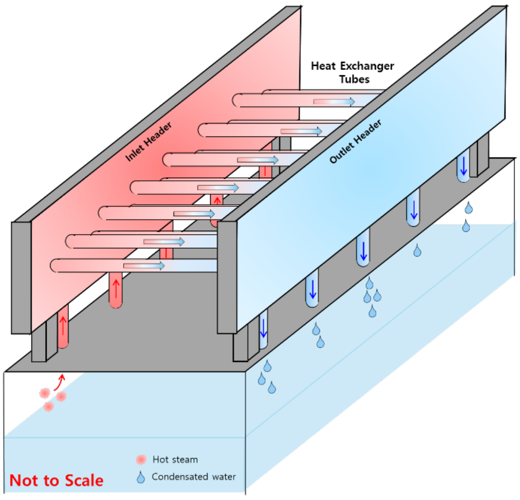

Fig. 3. Condensation heat exchanger 3D conceptual diagram

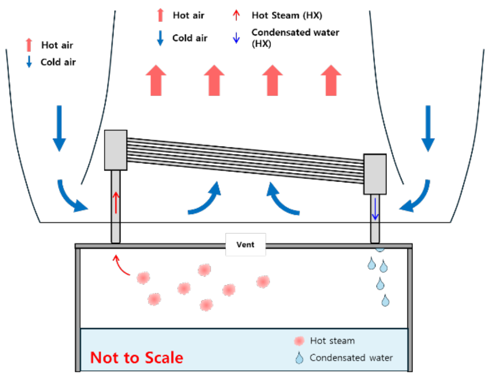

Fig. 4. Condensation heat exchanger cooling tower side view

3. Methods and Results

The MARS-KS code, which is widely used in the Republic of Korea for domestic nuclear regulatory analysis, is employed to assess the heat removal capability of two proposed designs. Additionally, an in-house code is developed to calculate the air velocity inside the cooling tower, as MARS-KS does not account for friction loss in the air crossflow through the tube bundle.

Before performing the code-based evaluation, the design geometry is determined. The fundamental design parameters are presented in Table I and Fig. 5.

Table I. Basic Design Parameters

Design Parameter	Value
Tube diameter	5.08 cm (2 inches)
Tube pitch	7 cm
Tube length	6.5 m
Cooling tower height	4 m
Number of tubes in the single row	244

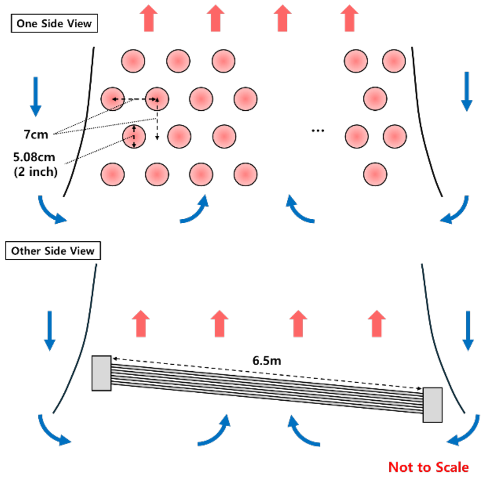

Fig. 5. Heat Exchanger Tube Design

3.1 Development of In-House Code

The in-house code is developed using Python. The total driving pressure head is calculated using Eq. (1). The density change across the tube bundle is neglected, as the air travel distance through the tube bundle is relatively short (26 cm).

$$\Delta p_{driving} = (\rho_{in} - \rho_{out}) g H_{tower} \quad (1)$$

(ρ: Density, H_{tower}: height of the air-cooling tower)

To determine pressure loss as air flows through the tube bundle, the Zukauskas correlation is applied [5]. The total pressure loss is given by Eq. (2).

$$\Delta p_{loss} = \xi n_R \frac{\rho w_e^2}{2} \quad (2)$$

(ξ: Drag coefficient, n_R: number of tubes in the single row, w_e: equivalent velocity)

The Gnielinski correlation is used to evaluate heat transfer from the tubes to the air [6]. The Nusselt number for air passing through a single row of tubes and the entire tube bundle is determined using Eqs. (3) and (4).

$$Nu_{l,0} = 0.3 + \sqrt{Nu_{l,lam} + Nu_{l,turb}} \quad (3)$$

$$Nu_{0,bundle} = f_A Nu_{l,0} \quad (4)$$

($Nu_{l,0}$: Nusselt number for the single row, $Nu_{0,bundle}$: Nusselt number for the whole tube bundle, f_A: adjustment factor)

In the developed in-house code, an iterative process is conducted, adjusting the mass flow rate until the difference between the pressure head and pressure loss falls below a predefined criterion. The air temperature and the condenser surface temperature are set to 50°C and 98°C, respectively.

As a result:

- The air temperature increased by 18°C.
- The air velocity was calculated as 0.46 m/s.

- The overall heat transfer rate was 998.32 kW.

The computed air velocity is subsequently used as a boundary condition in the MARS-KS code analysis.

3.2 Development of MARS-KS Input

The MARS-KS input geometry for the LLHP cooling tower and CHX cooling tower are shown in Figs. 6 and 7. The two designs are structurally similar, except for their connection to the ECT (Emergency Cooling Tank). In the LLHP cooling tower, the evaporator section is connected to ECT through heat structure 100. In contrast, CHX cooling tower is directly connected to the ECT, allowing it to absorb steam directly.

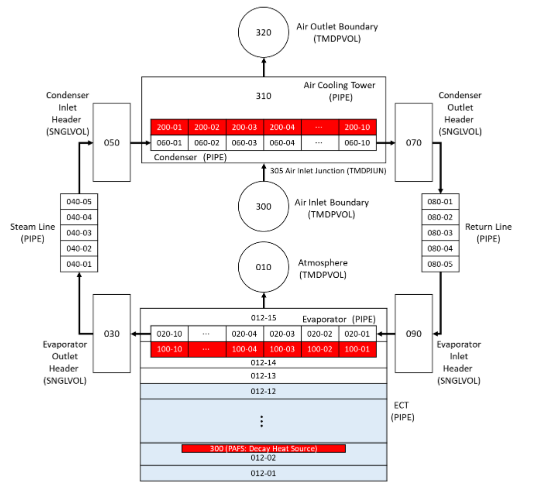

Fig. 6. LLHP Cooling Tower MARS-KS Input Geometry

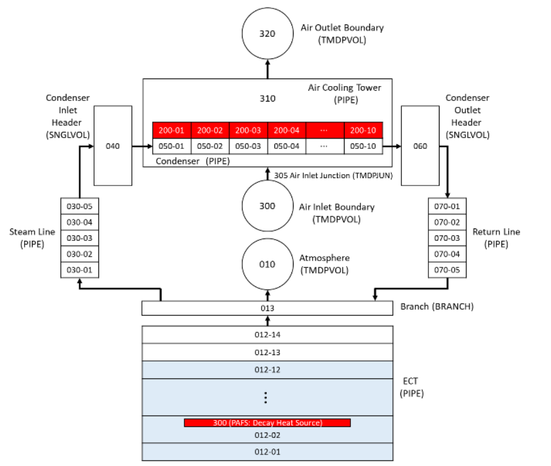

Fig. 7. CHX Cooling Tower MARS-KS Input Geometry

The detailed boundary and initial conditions are presented in Table II. Most volumes are filled with air at 1 bar, 50°C. Air flows through the air-cooling tower at a velocity of 0.46 m/s. The heat flux applied at heat structure 300 is calculated by dividing transient decay heat amount by the total PCHX area. The LLHP is filled with saturated steam and water, whereas the CHX is filled with atmospheric air.

Table II. Boundary and Initial Conditions

Number of the Volume, Junction, or Heat Structure	Condition
Common Volume 10, 300, 310, 320	Air, 1bar, 50°C
Common Volume 12	Water, 1bar, 50°C (Sub-volume 1~12) Steam, 1bar, 50°C (Sub-volume 12~15)
Common Junction 305	Air, 0.46 m/s
Common Heat Structure 300	Heat flux which corresponds to decay heat amount
LLHP Volume 20, 30, 90	Saturated Water, 50°C
LLHP Volume40, 50, 60, 70, 80	Saturated Steam, 50°C
CHX Volume13, 30, 40, 50, 60, 70	Air, 1bar, 50°C

3.3 Results & Discussion

Fig. 8 shows the total gas flow rate in the LLHP. The gas gradually decreased along the steam line (40), the condenser center (60), and the return line (80). This indicates that the saturated steam circulates unidirectionally and undergoes effective condensation within the LLHP.

Fig. 9 illustrates the total gas flow rate in the CHX. Unlike the LLHP, countercurrent steam flow was observed, and the gas flow rate exhibited greater fluctuations. This phenomenon occurred because gas can enter both the steam line and the return line, leading to unstable flow behavior.

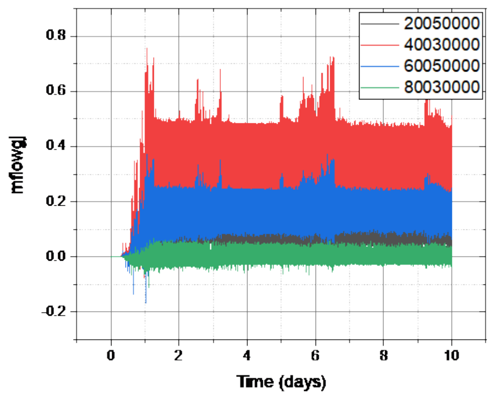

Fig. 8. Gas Flow Rate in LLHP

Fig. 10 shows the heat transfer rate comparison between the LLHP and CHX. A sharp decline in the CHX heat transfer rate was observed at specific time points. This decrease corresponds to the presence of non-condensable gases, as shown in Fig. 11. The non-condensable gas quality exceeding 0.6 coincides with the timing of the heat transfer rate drop, confirming its impact on CHX performance.

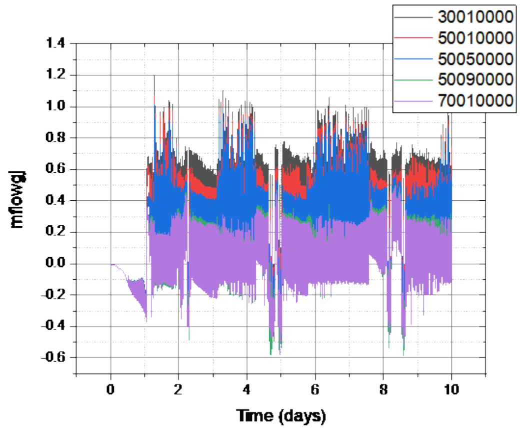

Fig. 9. Gas Flow Rate in CHX

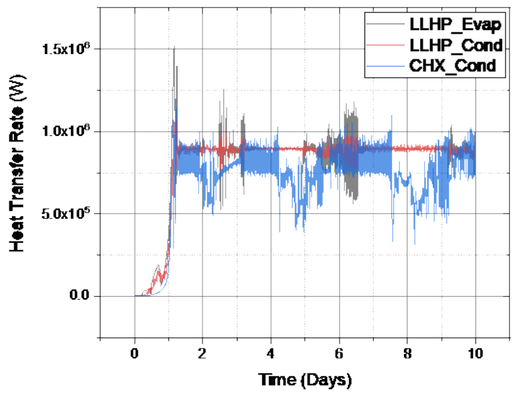

Fig. 10. Heat Transfer Rate Comparison

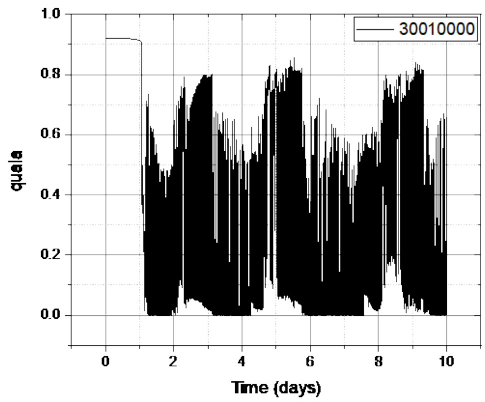

Fig. 11. Non-Condensable Gas Quality in CHX Steam Line

The water depletion rate in the ECT decreased when either LLHP or CHX was applied, confirming the effectiveness of both designs in extending the ECT cooling duration in i-SMRs.

Among the two designs, LLHP exhibited the smallest decrease in water level, indicating its superior cooling performance. This suggests that a separated system like LLHP can provide better and more stable cooling capability by effectively isolating the effects of non-condensable gases within the system.

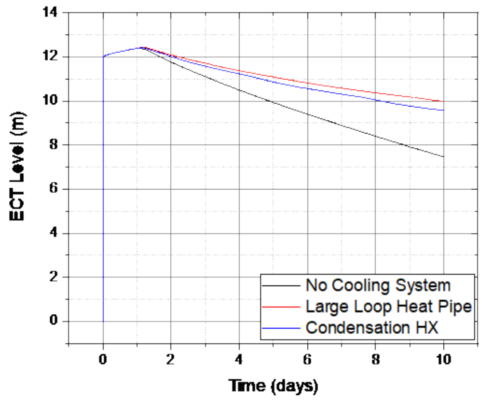

Fig. 12. ECT Water Level Change Comparison

4. Conclusions

An auxiliary passive cooling system for the long-term cooling of i-SMRs was developed. Two design concepts—LLHP and CHX cooling towers—were proposed and evaluated using a custom-developed in-house code and MARS-KS simulations. The results demonstrated that:

- Both designs are feasible for enhancing long-term cooling.
- The LLHP cooling tower exhibited superior cooling performance due to less effect of non-condensable gases.

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