Article

Experimental Investigation of the Heat Transfer Characteristics and Operation Limits of a Fork-Type Heat Pipe for Passive Cooling of a Spent Fuel Pool

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Abstract: A fork-type heat pipe (FHP) is a passive heat-transport and air-cooling device used to remove the decay heat of spent nuclear fuels stored in a liquid pool during a station blackout. FHPs have a unique geometrical design to resolve the significant mismatch between the convective heat transfer coefficients of the evaporator and condenser parts. The evaporator at the bottom is a single heat-exchanger tube, whereas the condenser at the top consists of multiple finned tubes to maximize the heat transfer area. In this study, the heat transfer characteristics and operating limits of an FHP device were investigated experimentally. A laboratory-scale model of an FHP was manufactured, and a series of tests were conducted while the temperature was varied to simulate a spent fuel pool. As an index of the average heat transfer performance, the loop conductance was computed from the measurement data. The results show that the loop conductance of the FHP increased with the heat transfer rate but deteriorated significantly at the operating limit. The maximum attainable heat transfer rate of the unit FHP model was accurately predicted by the existing correlations of the counter-current flow limit for a single-rod-type heat pipe. In addition, the instant heat transfer behaviors of the FHP model under different temperature conditions were examined to interpret the measured loop conductance variation and operating limit.

Keywords: fork-type heat pipe; loop conductance; operation limit; passive air cooling; spent fuel pool

1. Introduction

A heat pipe is a thermal device which allows the efficient transport of thermal energy by use of an intermediate heat transfer fluid. Heat pipes working under gravity with the condenser above the evaporator do not need any external power or capillary structure for the return of the heat transfer fluid to the evaporator. This type of heat pipe is known as a gravity-aided heat pipe. Gravity-aided heat pipes have received considerable attention in terrestrial applications due to their efficient heat transfer capability, simple construction and durability [1]. Therefore, it has numerous applications in heat recovery systems [2,3], air conditioning [4], solar energy [5,6] and, recently, in light-water nuclear reactors [7].

When a station blackout (SBO) event occurs in a nuclear power plant, active cooling systems lose their ability to remove heat from spent fuel storage pools. However, the decay heat continuously generated in the stored spent fuel rods heats the stagnant coolant, eventually causing it to evaporate. As a result, spent fuel exposed to air becomes overheated and severely damaged, resulting in release of radioactive materials. Hence, the necessity of passive cooling systems for spent fuel pools has been emphasized in the nuclear industry since the Fukushima nuclear accident in Japan.

Large gravity-aided heat pipes, so-called thermosyphons, are a feasible solution for passive cooling of spent fuel pools during a nuclear power plant SBO. The basic operating principle of a thermosyphon is described in Figure 1. The syphon itself consists of three sections: an evaporator in the heating part, a condenser in the cooling part, and an adiabatic
zone between the two parts. The thermosyphon is gravity operated; thus, the evaporator is placed at a level below the condenser. The working fluid condenses in the condenser and descends to the evaporator by gravity, where it is vaporized. The resulting vapor then rises to the condenser. Thermosyphons are more appropriate for large-scale applications such as nuclear power plants compared to capillary-assisted heat pipes [8].

![Operating principle of a gravity-aided heat pipe (thermosyphon).](image)

To maintain the cooling performance of passive cooling systems for spent fuel pools over time, ambient air is an ideal option. However, due to the significant mismatch between the convection heat transfer coefficients of a liquid-filled spent fuel pool and ambient air, a gravity-assisted heat pipe must be custom-designed to satisfy the operating criteria. Mochizuki et al. [9] proposed a design concept of inserting an air-cooled, single-rod-type thermosyphon heat pipe into a spent fuel pool. The unit heat pipe was designed with a diameter of 34 mm, an evaporator of 6 m, and a condenser of 4 m with extended fins. The change in the temperature of the spent fuel pool according to the number of heat pipes was analyzed. The resulting design included 1662 heat pipes for removing the target decay heat of 3.6 MW. Ye et al. [10] proposed a loop-type thermosyphon design for a passive cooling system with a heat pipe installation in the CAP1400’s spent fuel pool. Their system configuration included 1594 heat pipes with a diameter of 100 mm and a length of 7.7 m, with an average heat transfer capacity of more than 10 kW. The research group later experimentally demonstrated the heat transfer capacity of a unit heat pipe up to ~18 kW [11,12].

There are also a few of numerical simulation studies for large-scale heat pipe designs for applications in spent fuel pools. Kusuma et al. [13] conducted heat transfer experiments on straight vertical heat pipes without wicks of 6 m in height and compared them with the results of analysis using RELAP5/MOD3.2. The condenser part was cooled by forced water convection on the jacket. The amount of heat removed, the temperature of the outer-wall of the heat-pipe, and the internal pressure in the condenser were measured. Parametric studies were conducted using parameters that have included the initial pressure of the heat pipe, the working fluid filling ratio, the evaporator heat load, and the coolant flow rate. Kuang et al. [14] established and verified a flow pattern, analysis-based numerical model to predict the performance of large-scale heat pipes for use in a spent fuel pool. The characteristics of the heat pipes used in the validation experiments were annular flow path evaporator and finned-tube condenser. Validation experiments were conducted with three working fluids (Water, R134a, Ammonia) and the developed numerical model showed a prediction error of less than 30% for water.

Lim [15] recently proposed a fork-type heat pipe (FHP) thermosyphon for the passive cooling of spent fuel water pools. A conceptual drawing of the FHP is shown in Figure 2.
The FHP has an evaporator, similar to a conventional thermosyphon; however, the FHP is also equipped with multiple condensers, which make it unique. Given that this design effectively enhances the surface area for condensing the working fluid and the heat transfer area for air cooling, it should be applicable to an air-cooled heat sink. Figure 3 shows a conceptual drawing of a passive cooling system made up of FHPs. The FHPs are installed near the sidewall of the spent fuel pool. The decay heat energy in the coolant is transferred to the heat sink via the FHPs. The air heated by the FHPs flows upward by natural convection. Therefore, an additional structure, i.e., a stack building or cooling tower, is installed on top of the fuel building for effective natural convection. The condensers of the FHP are positioned nearly parallel to the ground and perpendicular to the air flow to allow for maximum face velocity for effective air-cooling heat transfer. Later, Choi et al. [16] conducted a series of preliminary experimental tests to assess the heat transfer performance of the FHP using a laboratory-scale model, based on Lim’s proposed concept. Through heat transfer experiments, operational characteristics such as the periodic power fluctuations and operating limits of heat pipes were observed. The average heat transfer performance of the heat pipe measured in the quasi-steady state and the predicted performance using the concept of thermal resistance were compared, and a reasonable agreement was confirmed in the normal operating range.

Figure 2. Conceptual drawing of a fork-type heat pipe (FHP).

Figure 3. Conceptual drawing of a passive spent fuel pool cooling system using FHPs.

A review of previous studies of gravity-assisted, heat pipe system designs for the passive air cooling of spent fuel pools in nuclear power plants revealed that the most previous experimental and numerical studies attempted to estimate the average heat
removal performance of a unit heat pipe module, and the number of heat pipes required to reliably remove the decay heat generated in the spent fuel pool. The prime consideration, of course, is the size of the system and its ability to stably remove the decay heat. However, a practical thermosyphon has a maximum heat transfer limit that is affected by its specific design, as well as diverse unstable operating characteristics related to two-phase flow instability [8]. Therefore, a detailed experimental investigation and understanding of the fundamental thermal-hydraulic phenomena that occur in a new design of heat pipe are necessary to predict the thermal performance of the designed system accurately.

In this study, we experimentally investigated the diverse thermal characteristics of the FHP proposed by Lim [15], for application to the passive cooling of spent fuel pools in light-water nuclear power plants. A laboratory-scale model of the FHP was fabricated, and a series of thermal performance tests was carried out with respect to variations in the coolant temperature of the simulated spent fuel pool. The heat transfer characteristics and maximum heat transfer rate of the FHP model were examined experimentally. The observed thermal-hydraulic operating characteristics of the FHP model as a function of the operating temperature were interpreted for plausible two-phase flow regimes. The experimentally determined maximum heat transfer rate of a unit FHP model was compared and predicted using the existing operation limit correlation.

2. Materials and Methods

2.1. Laboratory-Scale Model of a Fork-Type Heat Pipe

To investigate the fundamental heat transfer characteristics of the FHP design experimentally, a laboratory model was custom-fabricated by Daehong Enterprise Co. (Siheung, Korea). The FHP model was made of copper and consisted of three main parts: a condenser with twelve air-cooling pipes, an adiabatic header, and an evaporator with a single water-heating pipe, as illustrated in Figure 4. The outer diameters of the evaporator and condenser pipes were 19.05 and 12.7 mm, respectively, and the pipe wall was 0.8 mm thick. The evaporator and condenser pipes were 1.0 m long. In addition, there were two adiabatic regions between the evaporator/condenser pipes and the header, 0.2 m and 0.15 m in length, respectively. While the single-pipe evaporator was inserted vertically into the space between the pool wall and the fuel racks, the air-cooling condenser pipes were inclined slightly by 7°, as suggested by Griffith and Silva [17], to prevent condensation-induced water hammer. In addition, 300 annular fins were manufactured on the outer surface of each air-cooling condenser tube to enlarge the effective heat transfer area. Each annular fin had an outer diameter of 32 mm and a thickness of 0.5 mm. The annular fins were attached to the outer surfaces of the pipes by pressurizing the inside and expanding at a high temperature.

The FHP model used water as a working fluid. Here, water filling occurred under high vacuum conditions of $7.0 \times 10^{-4}$ Pa to remove non-condensable gases, which degrade condensation and evaporation heat transfer. The practical operating temperature range for a copper/water heat pipe is approximately 25–150 °C, which is appropriate for passive cooling applications of spent fuel pools using ambient air. The working fluid in the evaporator tube was filled to a height of 1.2 m, and its filling ratio relative to the evaporator volume was 1.38.

An absolute pressure gauge and four thermocouples were installed in the heat pipe to determine the thermal properties of the working fluid during operation and to investigate the heat transfer characteristics. The pressure gauge was installed on top of the header as representative of the system pressure of the FHP. To measure static pressure, the pressure gauge was aligned perpendicular with respect to the flow direction of the FHP via a tube of 1/4 inch inner diameter. A PX172 pressure sensor by OMEGA (Norwalk, CT, USA), with a measurement accuracy of 0.25% FS, was used. The working fluid temperature was measured in each part of the FHP in each part of the FHP. The measurement points at the bottom of the evaporator and the top of the header corresponded to the liquid and vapor temperatures, respectively. Additional measurement points were installed at the top ends of the edge and central
condenser tubes. All temperature sensors were T-type thermocouples manufactured by OMEGA, with a measurement accuracy ±1 °C.

![Figure 4. Schematic diagram of a laboratory-scale FHP model.](image)

### 2.2. Test Apparatus

Figure 5 shows the test apparatus used to measure the heat transfer characteristics of the model FHP, consisting of two main parts: a water-heating loop and an air-cooling duct. The major components and detailed specifications of the test apparatus are summarized in Table 1.

![Figure 5. Schematic diagram of the FHP heat transfer test apparatus.](image)

| Component       | Specification                              |
|-----------------|--------------------------------------------|
| FHP            |                                            |
| Heat exchanger  |                                            |
| Heating chamber |                                            |
| Heater         |                                            |
| Pump           |                                            |
| Flow meter     |                                            |
| Rota-meter     |                                            |
| Air duct       |                                            |
| Fan            |                                            |
| Honeycomb flow straightener |                          |
Table 1. Specifications of the major devices in the test apparatus.

| Component          | Model        | Type                 | Specification                                      |
|--------------------|--------------|----------------------|---------------------------------------------------|
| Heater             |              | Immersion heater     | 2 kW / 3 kW                                       |
| Pump               | NPY-2251-MK  | Regenerative turbine pump | Total head: 490 kPa (50 m)  
Flow rate: 1.8 m³/h |
| Flow meter         | DGT-010AI    | Oval gear            | Range: 0.02–0.7 m³/h  
Accuracy: ±0.5% RS |
| Temperature sensor | OMEGA P-M-1/10-1/4-6-0-T-3 | 1/10 Din RTD | Accuracy: 0.04 °C (@ 20 °C)  
0.12 °C (@ 100 °C) |
|                    | OMEGA SCPSS-040 | T-type             | Range: −60 °C to 100 °C  
Accuracy: ±1 °C |
| Fan                | Innotech TIP-350S | Total head: 784.5 Pa (0.8 m)  
Flow rate: 6900 m³/h |
| Air velocity sensor | E + E Elektronix EE75 | Hot wire anemometer | Range: 0–2 m/s  
Accuracy: ±0.03 m/s |
| Temperature sensor | OMEGA SCPSS-040 | T-type             | Range: −60 °C to 100 °C  
Accuracy: ±1 °C |

The water-heating loop was designed to simulate heating of the single-pipe evaporator of the FHP by single-phase convection of water coolant heated in the spent fuel pool. It comprised a fluid flow system, a single-tube heat exchanger, a heating chamber, and heaters. The fluid flow system was divided into two flow paths, for main and bypass flows. The heating chamber was located at the top of the fluid flow system and was filled with distilled water under ambient pressure. Two immersion heaters (2 kW and 3 kW) to heat the water were inserted from the bottom of the heating chamber. As described in Figure 6, an evaporator pipe with an outer diameter of 19.05 mm was inserted into a single-tube concentric heat exchanger with an inner diameter of 38.7 mm. The shell side of the heat exchanger was connected to the water-heating loop. To supply heat to the single-pipe evaporator convectively, the heated water in the loop was circulated by a turbine pump through the shell side of the single-tube heat exchanger. The flow rate into the heat exchanger was controlled by adjusting a needle valve and the level of bypass flow, and was measured using an oval gear flow meter with an accuracy of ±0.5% RS. The heat source temperature, which indicates the water temperature through the shell side of the heat exchanger, was measured at the four points using T-type thermocouples (OMEGA). Two four-wire 1/10 D in-class RTD from OMEGA were installed at the inlet and outlet of the heat exchanger. The inlet temperature was used as representative of the heat source as an independent experimental variable this study.

The air-cooling duct consisted of an air-duct body and frame, a cooling fan, and a flow straightener. In the air duct, the air flowed vertically from the bottom toward the top. Several honeycomb meshes were installed at the entrance to ensure uniform air flow. A suction fan was mounted at the exit of the air duct. The air velocity was measured by a hot-wire anemometer at the center of the air duct. The anemometer calculated the air velocity using the temperature of the hot wire. The relationship between the air velocity and the hot-wire temperature was pre-calibrated under the air-cooling condition; hence, it was installed at the inlet of the air duct (in front of the FHP) to avoid error induced by heated air.
Figure 6. Detailed illustration of a single-tube concentric heat exchanger in a water-heating loop.

2.3. Data Reduction and Uncertainty Analysis

The heat transfer rate of the FHP model during operation was calculated using calorimetry based on the measured temperature and the flow-rate data of the water-heating loop:

\[ q = \rho \dot{V} c_p \left( T_{\text{in}} - T_{\text{out}} \right) \]  

(1)

where \( \rho \) and \( c_p \) are the water density and specific-heat capacity in the annular channel, respectively. These thermal properties were referred to in an IAPWS R7-97 report corresponding to the average temperature along the annulus. \( T_{\text{in}} \) and \( T_{\text{out}} \) are the inlet and outlet temperatures of water in the shell side of the heat exchanger, respectively, and \( Q \) is the volumetric flow rate measured by the oval gear flow meter.

In the experiment, pressure and temperature were allowed to fluctuate in a regular manner, to simulate the typical operational characteristics of a heat pipe, as discussed in the literature review. Thus, a quasi-steady state was employed to oscillate the parameters with the same amplitude and period. All measurements in this study were conducted while the thermal quantities of interest showed steady-state or quasi-steady-state characteristics. Measurement data were gathered every 4 s for 30 min using an Agilent 34980A system (Agilent Technologies, Santa Clara, CA, USA).

The uncertainties in the measurement data can be estimated using Equations (2)–(4) [18,19]:

\[ u_n = \sqrt{\left( u_{A,n} \right)^2 + \left( u_{B,n} \right)^2} \]  

(2)

\[ u_{A,n} = \frac{s_n}{\sqrt{N_n}} \]  

(3)

\[ u_{B,n} = \frac{a_n}{\sqrt{3}} \]  

(4)

Here, \( u_A \) indicates the uncertainty in the random error caused by the average, which is defined as the sample standard deviation divided by the square root of the sample number. Another uncertainty, \( u_B \), refers to the uncertainty in the systematic error, which is related to the estimate (note ‘a’ in Equation (4)) provided as part of the device specifications. The estimate is usually obtained from statistical analyses conducted by the vendor. If only the upper/lower boundary of the estimate is provided, it can be assumed to be a continuous uniform distribution. In this case, the systematic uncertainty would be described by Equation (4), based on its conservative assumption. The subscript ‘n’ in the equations corresponds to the specific measurement values, such as the temperature and flow rate.
Finally, the total uncertainty in measuring the heat transfer rate in this study can be calculated as:

\[
\frac{U_q}{q} = k \left[ \left( \frac{u_{q,\text{lin}}}{q} \frac{\partial q}{\partial T_{\text{lin}}} \right)^2 + \left( \frac{u_Q}{q} \frac{\partial q}{\partial Q} \right)^2 + \left( \frac{u_{\text{m},\text{lin}}}{q} \frac{\partial q}{\partial T_{\text{m},\text{lin}}} \right)^2 + \left( \frac{u_{\text{out},\text{lin}}}{q} \frac{\partial q}{\partial T_{\text{out}}} \right)^2 \right]^{1/2}
\]

(5)

where the coverage factor, \( k \), is related to the confidence level. When the 95% confidence level is applied, the value of the coverage factor is 2. In this analysis, the tolerance of the density and specific heat capacity of water were 0.003% and 0.2%, respectively [20]. The water flow rate on the shell side of the heat exchanger and the inlet temperature of the air duct remained constant for all experimental cases: 0.053 \((\pm 7 \times 10^{-4})\) kg/s and 26 \((\pm 0.7)\) °C, respectively. Figure 7 shows the uncertainty calculated using Equation (5), together with the measured heat transfer rate. The resulting uncertainty is strongly dependent on the value of the heat transfer rate: 120% at the low heat transfer rate of 0.03 kW and 3% at 2.2 kW. This is because in Equation (1), the measurement error of each RTD temperature sensor (of \( \pm 0.05 \) °C) is comparable to the fluid temperature rise at low values of \( q \) \((\sim 0.1 \) °C at 0.03 kW), whereas with high values of \( q \) it is negligibly small.

![Figure 7. Maximum measurement uncertainty as a function of the heat transfer rate.](image)

3. Results and Discussions

3.1. Average Heat Transfer Rate and Working-Fluid Temperature

The heat transfer rate of the FHP model was measured by varying the temperature of the circulating water in the heating loop and the velocity of the air flow in the cooling duct. The temperature of the circulating water in the heating loop was controlled in the range of 60–100 °C, which corresponded to the allowable maximum temperature of spent fuel pool coolant (water) during normal operation and its boiling temperature during an SBO accident, respectively [21]. The air-cooling velocity was set to 0.5 or 0.7 m/s.

Figure 8 shows the measured average heat transfer rates with respect to the inlet temperature of the heating water inside the heat exchanger. The associated characteristic tendency can be understood by using the data for the air-cooling velocity of 0.5 m/s. No measurable heat transfer was observed until the heat source temperature reached 67 °C. At 67 °C, the minimum measurable heat transfer rate of \( \sim 30 \) W, appeared. From this point up to 92 °C, the average heat transfer rate of the FHP model increased almost linearly in proportion to the heat source temperature. When the heat source temperature increased beyond 92 °C, the rate of the increase in heat transfer weakened. Finally, \( \sim 2.2 \) kW was measured at 97 °C. Thereafter, a further increase in the heat source temperature did not lead to any additional increase in heat transfer. Similar tendencies for the changes in the heat transfer rate were observed for an air velocity of 0.7 m/s. The average heat transfer rate at 0.7 m/s increased more rapidly than the temperature of the heat source. Additionally, the limit point of the heat transfer rate appeared at the lower heat source temperature of \( \sim 91 \) °C, and the peak value was slightly smaller: \( \sim 2.1 \) kW. However, an increase in the air-cooling velocity from 0.5 to 0.7 m/s resulted in a corresponding increase in the average heat transfer rate of the FHP model at the same heat source temperature.
This observed behavior can be interpreted using the concept of a thermal resistance circuit for the FHP model. Equation (6) represents the thermal resistance circuit for the FHP, where the heat transfer rate is calculated by dividing the overall temperature difference by the total thermal resistance in the circuit. The first thermal resistance ($R_{source}$) is associated with the convective heat transfer in the concentric heat exchanger, and the second resistance ($R_{hp}$) corresponds to the effective thermal resistance of the heat pipe. The final resistance term ($R_{sink}$) is related to the air-side convection heat transfer. An increase in the air-cooling velocity directly improves the heat transfer efficiency in the heat sink ($h_{sink}$); hence, it reduces the total thermal resistance and improves heat transfer through the FHP:

$$q = \frac{T_{source} - T_{sink}}{R_{source} + R_{hp} + R_{sink}} = \frac{T_{source} - T_{sink}}{(h_{source}A_{e,o})^{-1} + R_{hp} + (h_{sink}A_{c,o})^{-1}}$$  \hspace{1cm} (6)$$

Figure 9 shows how the temperature of the operating fluid in the tested FHP model changes as a function of the temperature of the heated part. With the increasing temperature of the heating loop as the heat source, the parameter showed a proportional increase. An increase in the velocity of the cooling air (as the heat sink) reduced the temperature of the working fluid. This behavior can also be interpreted using the concept of a thermal-resistance circuit. The thermal circuit from the heat source through the FHP model to the heat sink consists of seven thermal-resistance terms associated with convective heat transfer in the heat exchanger ($R_{source}$), wall conduction via the evaporator ($R_{we}$), boiling heat transfer ($R_{c}$), adiabatic pressure head loss through the heat pipe ($R_{loss}$), condensation heat transfer ($R_{c}$), wall conduction via the condenser ($R_{wc}$), and air-cooling convective heat transfer ($R_{sink}$). Equation (6) can be decomposed for each thermal resistance term, as shown in Equation (7):

$$q = \frac{T_{source} - T_{wc,o}}{R_{source}} = \frac{T_{wc,i} - T_{e}}{R_{wc}} = \frac{T_{e} - T_{c}}{R_{loss}} = \frac{T_{c} - T_{wc,i}}{R_{c}} = \frac{T_{wc,i} - T_{wc,o}}{R_{wc}} = \frac{T_{wc,o} - T_{sink}}{R_{sink}}$$  \hspace{1cm} (7)$$

It is thought that the air-cooling convective heat transfer thermal resistance ($R_{sink}$) is directly affected by the air-flow velocity, whereas the other factors are indirectly affected by changes in the thermal properties. Hence, the increased air velocity predominantly decreases the air-side thermal resistance ($R_{sink}$), whereas the other factors remain almost constant. According to Equation (7), this increases the temperature difference between the heat source and the heat pipe wall. As a result, the working fluid temperature also decreases in proportion to the increase in the air-cooling velocity at the same heat source temperature.
Equation (6) can be decomposed for each thermal resistance term, as

\[
R \approx 22 (T_R - T) 
\]

The loop conductance can be used to evaluate the characteristic thermal performance of a heat pipe, as it excludes the influences of external thermal resistances, such as \(R_{\text{source}}\) and \(R_{\text{sink}}\). The loop conductance of the FHP can be expressed as Equations (8) and (9), as follows:

\[
q = (T_{we,o} - T_{wc,o}) / R_{hp} = (T_{we,o} - T_{wc,o}) / \left( \frac{1}{h_{hp} A_e} \right) \approx (T_e - T_c) / \left( \frac{1}{h_{hp} A_e} \right) 
\]

\[
h_{hp} = q / [A_e(T_e - T_c)] 
\]

In this study, the wall of the FHP model used in the present experiment was very thin (0.8 mm) and was made from highly conductive copper. The values of the phase-change heat transfer coefficients in the evaporator and condenser parts were very large. Hence, the temperature difference between the outer wall and the working fluid of the FHP was sufficiently small to assume that the outer wall temperature was approximately the same as the working fluid temperature in both the evaporator and condenser parts.

Figure 10 shows the loop conductance of the tested FHP model as a function of the heat source temperature. It also illustrates the loop conductance transition as a result of increasing heat source temperature. All data points on the trend line in Figure 10 correspond to those in Figure 9, in order, from the origin. The loop conductance of the FHP shows the same trend and a similar order of magnitude for loop conductance at heat transfer rates of less than 1.9 kW, regardless of the air velocity. However, the loop conductance increases rapidly at a certain heat transfer rate in both cases. The maximum loop conductance for each air velocity mostly corresponds to a point just before the heat transfer rates converge, as discussed in Section 3.1. With an air velocity of 0.7 m/s, the loop conductance decreases suddenly after reaching its maximum. The adverse decrease in loop conductance is directly affected by the increase in the temperature difference inside the FHP after the heat transfer rates converge. Physically, this means that the thermal resistance of the FHP increases.
In this study, the wall of the FHP model used in the present experiment was very thin (0.8 mm) and was manufactured from highly conductive copper. The values of the phase change heat flux reach the critical heat flux (CHF). Two correlations predict the CHF at the evaporator with a stagnant liquid pool: Zuber’s correlation [23], for a horizontal heated surface immersed in a pool; and Engineering Sciences Data Unit’s correlation, ESDU 81038 [24], developed for a two-phase closed thermosyphon. They have the same formulations, but with slightly different coefficients to account for the effects of different geometries, as summarized in Table 2.

### 3.3. Operation Limits

The maximum heat transfer rate observed was at a heat source temperature of above 90 °C, as shown in Figure 8. Two relevant operation limits ultimately resulted in the failure of the thermosiphon due to the drying out of the working fluid in the evaporator: the boiling limit and the flooding limit.

The boiling limit occurs under pool-boiling conditions when there is sufficient working fluid in the evaporator part. Dry-out happens on the heated surface when the wall heat flux reaches the critical heat flux (CHF). Two correlations predict the CHF at the evaporator with a stagnant liquid pool: Zuber’s correlation [23], for a horizontal heated surface immersed in a pool; and Engineering Sciences Data Unit’s correlation, ESDU 81038 [24], developed for a two-phase closed thermosyphon. They have the same formulations, but with slightly different coefficients to account for the effects of different geometries, as summarized in Table 2. Figure 11 shows comparisons between the calculated boiling limits based on the correlations and the measured surface heat flux of the FHP. The measured heat-flux capacity of the FHP was lower than the predicted CHF by one order of magnitude. Therefore, it can be concluded that the maximum heat transfer rate in the FHP was not limited by the boiling limit mechanism.

### Table 2. Boiling limit correlations for a gravity-aided thermosyphon.

| Author       | Correlation                                                                 | Note                                               |
|--------------|-----------------------------------------------------------------------------|----------------------------------------------------|
| Zuber [23]   | \[ q_{b, \text{max}} = 0.13 h_f \rho_v \sqrt{\frac{1}{2} \sigma g (\rho_l - \rho_v)} \]^{1/4} | Typical CHF correlation for pool boiling           |
| ESDU 81038 [24] | \[ q_{b, \text{max}} = 0.12 h_f \rho_v \sqrt{\frac{1}{2} \sigma g (\rho_l - \rho_v)} \]^{1/4} |                                                    |

![Figure 10. Estimated loop conductance of an FHP.](image)
The flooding limit occurs due to the counter-current flow limit. The existing correlations used to predict the flooding limit in a heat pipe are summarized in Table 3. The correlations of Wallis [25–27] and Kutateladze [28,29] were developed based on adiabatic experimental data of water–gas counter-current flow without phase change. Wallis considered the balance between the inertial force and the hydrostatic force, whereas Kutateladze considered the balance among the dynamic head, surface tension, and gravitational force. Tien and Chung [30] developed improved correlations that accounted for the effects related to phase change. Specifically, Tien and Chung found a characteristic length in the Wallis correlation that they identified as the critical wavelength of the Taylor instability by comparing the two correlations of Wallis and Kutateladze. The experiments showed that the diameter of the thermosyphon is applicable to the characteristic length; hence, the Kutateladze constant can be redefined as a function of the Bond number within the range of the critical wavelength of the Taylor instability. This correlation generally shows a good agreement for several refrigerants. In contrast, for water, and especially considering a small Bond number, the correlation shows significant error. Imura et al. [31] developed a flooding-limit correlation using a non-dimensional analysis to improve the agreement of various conditions such as geometry, working fluids, and working fluid temperature. This correlation shows ± 30% accuracy, regardless of the various conditions. Faghri et al. [32] proposed a generalized correlation applicable to various working fluids, including refrigerants and water. They proposed a modified Kutateladze number as a function of the liquid–vapor density ratio by combining the correlations of Tien and Chung [30] and Imura et al. [31].
Table 3. Flooding limit correlations for a gravity-aided thermosyphon.

| Author                  | Correlation                                                                 | Note                                                      |
|-------------------------|------------------------------------------------------------------------------|-----------------------------------------------------------|
| Wallis [25–27]          | $q_{f, \text{max}} = C_w^2 h_f g A\left[\frac{g L (\rho_L - \rho_v)}{\rho_v}\right]^{1/2} \left[\rho_v^{-2} + m \rho^{-1}\right]^{-2}$ | Whalley [27] recommended use of a tube diameter < 10 mm    |
|                         | $C_w = 1.0$ for total flooding                                               |                                                           |
|                         | $L$ is characteristic length ($L = D$ in thermosiphon)                      |                                                           |
| Kutateladze [28,29]    | $q_{f, \text{max}} = C_k^2 h_f g A\left[\frac{g \sigma (\rho_L - \rho_v)}{\sigma}\right]^{1/4} \left[\rho_v^{-2} + m \rho^{-1}\right]^{-2}$ | Whalley [27] recommended use of a tube diameter > 10 mm    |
|                         | $C_k = 3.2$ for total flooding                                               |                                                           |
| Tien and Chung [30]     | $q_{f, \text{max}} = C_k^2 h_f g A\left[\frac{g \sigma (\rho_L - \rho_v)}{\sigma}\right]^{1/4} \left[\rho_v^{-2} + m \rho^{-1}\right]^{-2}$ |                                                           |
|                         | $C_k = \sqrt{3.2 \tanh \left(0.5 Bo^{1/4}\right)}$                        |                                                           |
|                         | $Bo = D \left[\frac{g (\rho_l - \rho_v)}{\rho_v}\right]^{1/2}$             |                                                           |
| Imura et al. [31]       | $q_{f, \text{max}} = 0.64 A h_f g \left(\frac{\rho_v}{\rho_v}\right)^{0.13} \left[\frac{g \sigma (\rho_L - \rho_v)}{\sigma}\right]^{1/4}$ |                                                           |
| Faghri et al. [32]      | $q_{f, \text{max}} = K h_f g A\left[\frac{g \sigma (\rho_L - \rho_v)}{\sigma}\right]^{1/4} \left[\rho_v^{-2} + m \rho^{-1}\right]^{-2}$ |                                                           |
|                         | $K = \left(\frac{\rho_v}{\rho_l}\right)^{0.14} \tanh^2 (Bo^{1/4})$       |                                                           |

The predicted correlation of the operation limits of a heat pipe is a strong function of the thermal properties of the working fluid (Table 3). Therefore, it is directly affected by the operating temperature of the heat pipe, which varies according to the heat source temperature in the experiment. Qualitatively, an increased operating temperature and corresponding saturation pressure raise the vapor density and lower the volumetric velocity of the vapor flow. Thus, a higher vapor production rate by evaporation is required to reach the counter-current flooding limit, which is strongly affected by the velocity difference between the vapor and liquid flows. Based on the experimental results shown in Figure 9, the operating temperature of the heat pipe in this study increased in proportion to the heat source temperature; however, beyond a heat source temperature of 90 °C, such behavior stopped and an almost constant value of ~55 °C was observed. This observation is the characteristic behavior expected in the region where the heat transfer in the heat pipe is restricted by the operation limits. The resulting maximum operating temperature of the FHP model obtained in the present experiments was about 57 °C at 0.5 m/s and 55 °C at 0.7 m/s.

Figure 12 shows a comparison of the measured heat transfer rates and the flooding limit correlations as a function of the operating temperature. The trend of the average heat transfer rate measured in the experiment decreased significantly at and above the operating temperature at which the loop conductance peaks in Figure 10 (e.g., 50.1 °C at 0.7 m/s). In particular, the measured heat transfer rate as a function of the operating temperature asymptotically approaches the flooding limit predicted by the correlation of Faghri et al. [32], showing the best agreement among the existing correlations introduced in Table 3.
3.4. Instantaneous Operation Characteristics

To interpret the measured thermal performance limits and the loop conductance variation of the FHP model, instant operation characteristics are systematically analyzed in this section. Note that, for clarity, only the dataset obtained at an air-cooling velocity of 0.7 m/s is analyzed.

For each heat source temperature, the instant variations in three working fluid temperatures were compared. One was the working fluid temperature measured at the adiabatic header. The other two were the working fluid temperatures measured at the bottom of the evaporator and the thermodynamic saturation temperature corresponding to the vapor pressure calculated using the measured pressure at the header and the hydrostatic head of the working fluid in the evaporator:

$$T_{sat,e} = T_{sat}(P_{sys} + \rho g H)$$  \hspace{1cm} (10)

where ‘H’ indicates the level of the liquid-phase working fluid in the evaporator, which was assumed to be the same as the initial water level during operation. In addition, the instant variations in the overall heat transfer rate and the axial distribution of the local heat transfer rate from the heat source to the single-pipe evaporator were analyzed. The local heat transfer rates ($q_{12}$, $q_{23}$, and $q_{34}$) were calculated in a calorimetric manner using the water-heating temperatures measured at the four serial points along the annular channel, as illustrated in Figure 6.

Figure 13 shows the temporal variations in the abovementioned instant operation parameters of the FHP model at a heat source temperature of 67 °C, which was the lowest temperature, resulting in the minimum measurable heat transfer rate of ~30 W. The working fluid temperature in the FHP was quite stable without fluctuations, whereas the measured heat transfer rate was almost zero. The evaporator temperature in Figure 13a was 67 °C; thus, the heat source temperature and the evaporator temperature were nearly in thermal equilibrium. The evaporator temperature was even larger than the estimated thermodynamic saturation temperature. This indicates that, at evaporation, the working fluid is superheated and that this superheating is not sufficient for causing the onset of so-called nucleate boiling. Hence, despite the resulting heat transfer rate being very small, the transient temperature behavior remains relatively stable.
Figure 13. Operating characteristics of the FHP with a heat source temperature of 67.3 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.

When the temperature of the heat source increased by only 2 °C, from 67 °C to 69 °C, the temperature of the working fluid and the heat transfer rate started to fluctuate considerably, as shown in Figure 14. Specifically, when the temperature of the liquid in the evaporator decreased suddenly, the temperature of the vapor in the header correspondingly increased sharply. This behavior in the evaporator part of the vertical, two-phase, closed thermosyphon heat pipe was reported previously as the intermittent boiling regime by Niro and Berretta [33]. The temperature fluctuations in Figure 14a can be interpreted using two characteristic periods of intermittent boiling: the time during which the temperature of the working fluid at the bottom of the evaporator gradually increases, corresponding to the waiting (or transient heating) period; and the remaining time, during which the
temperature of the working fluid decreases rapidly, corresponding to bubble nucleation and growth. In connection with the two characteristic periods of intermittent boiling, the instant heat transfer rate remains almost constant during the waiting period, but jumps instantly during the bubble growth period, as shown in Figure 14b. The intense intermittent boiling characteristics are observed in the bottom part of the FHP model. The local heat transfer rate between T-1 and T-2 (red area) is almost zero during the waiting period but increases dramatically during the bubble growth period. Notably, the fluctuation in the total heat transfer rate shows a strong correspondence to the variation in the local heat transfer rate at the evaporator bottom. Lastly, Figure 14c illustrates the average heat source temperature distribution along the axis direction; the average temperature difference between two adjacent points is relatively small at locations close to the bottom.

Figure 14. Operating characteristics of an FHP with a heat source temperature of 69.0 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.
It was observed that, when intermittent boiling occurs, additional unstable and dynamic behavior occurs, accompanied by a hammering sound. If a heated body of water is deep or if there is a sufficiently large difference in the hydrostatic head between the water surface and the floor, geyser boiling [34–37] can occur. The geyser boiling phenomenon and the expected axial hydrostatic head distribution in the FHP are illustrated in Figure 15. In the first phase, nucleate boiling appears at the upper side of the evaporator, while the liquid at the bottom side is gradually superheated without nucleation. Once intermittent boiling occurs at the lower level, a relatively large bubble may be generated by the accumulated thermal energy in the superheated liquid. Furthermore, the bubble can rapidly expand due to flashing while rising in the vertical evaporator tube. Then, when the bubble becomes large enough to clog the cross-section of the evaporator, the resulting giant slug bubble sweeps the liquid working fluid out of the evaporator. This is the typical sequence of geyser boiling, which leads to the explosive eruption of liquid and vapor from the water body. It is thought that in the FHP model, erupted liquid due to geyser boiling hits the top surface of the adiabatic header, resulting in the observed hammering sound. As evidence of this interpretation, we observed a momentary increase in the header temperature that corresponded to the sudden decrease in evaporator temperature due to geyser boiling, as shown in Figure 14a.

Figure 16 displays the heat transfer characteristics at a heat source temperature of 81.2 °C. The period of geyser boiling was reduced as the heat source temperature increased. The waiting time shortened, and the phase change occurred continuously, which led to a decrease in the fluctuation effects of heat transfer and a reduction in the amplitude. Figure 17 represents the heat transfer characteristics for a heat source temperature of 83.1 °C. The period of geyser boiling was shorter, whereas the waiting and bubble growth times were similar. This effect coincided with fully developed boiling [33]. However, slightly longer waiting periods appeared intermittently, which coincided with a temporary decrease in the local heat transfer rate between T-1 and T-2 (red area). This demonstrated the transition sequence from intermittent boiling to fully developed boiling. The three local heat transfer rates showed similar magnitudes to that of fully developed boiling.
Figure 16. Operating characteristics of an FHP with a heat source temperature of 81.2 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.
Figure 17. Operating characteristics of an FHP with a heat source temperature of 83.1 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.

Figure 18 shows the heat transfer characteristics at a heat source temperature of 96.4 °C; the fluctuation in the temperature of the working fluid differs from that of geyser boiling, with irregular variation. Additionally, the change in the evaporator temperature is large and sudden. The average total heat transfer rate is about 2.1 kW, similar to the maximum heat transfer rate for an air velocity of 0.7 m/s. The difference in the heat source temperature at the bottom of the evaporator decreased compared with under the fully developed boiling condition, although the difference in the temperature at the top of the

Figure 18 shows the heat transfer characteristics at a heat source temperature of 96.4 °C; the fluctuation in the temperature of the working fluid differs from that of geyser boiling, with irregular variation. Additionally, the change in the evaporator temperature is large and sudden. The average total heat transfer rate is about 2.1 kW, similar to the maximum heat transfer rate for an air velocity of 0.7 m/s. The difference in the heat source temperature at the bottom of the evaporator decreased compared with under the fully developed boiling condition, although the difference in the temperature at the top of the
evaporator was more pronounced. This indicates that local cooling through the bottom of the heat pipe decreased over time. Figure 19 shows similar characteristics for a heat source temperature of 98.1 °C to those in Figure 18. However, the evaporator temperature increased more dramatically, whereas the cooling ability at the bottom was more degraded. The evaporator temperature increased to almost that of the heat source. Given that this phenomenon occurred close to the maximum heat transfer capacity, it may correspond to the operation limit condition. This characteristic appeared close to the flooding limit predicted by Faghri et al., highlighting the three major conditions associated with this phenomenon:

- High evaporator temperature at the bottom of the evaporator;
- Low heat transfer rate at the bottom of the evaporator;
- Irregular fluctuations.

Under the flooding limit condition in the thermosyphon, the high velocity of the upward flow of vapor restricted the return flow of condensed liquid due to the counter-current flow limit. The working fluid eventually dries out at the bottom of the evaporator and the temperature increases sharply. In this state, the heat transfer rate decreases naturally. The temperature of the evaporator is sometimes cooled by the returned liquid due to competition between the drag force and the weight of the falling liquid. This phenomenon is called rewetting. The cycle then repeats. Thus, irregular temperature fluctuations appear, as shown in Figures 18 and 19, as well as degradation of the loop conductance (Figure 10).

Figure 18. Operating characteristics of an FHP with a heat source temperature of 96.4 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.
Figure 19. Operating characteristics of an FHP with a heat source temperature of 98.1 °C: (a) working fluid temperature, (b) cumulative heat transfer rate, and (c) local heat source temperature.

Under the flooding limit condition in the thermosyphon, the high velocity of the upward flow of vapor restricted the return flow of condensed liquid due to the counter-current flow limit. The working fluid eventually dries out at the bottom of the evaporator and the temperature increases sharply. In this state, the heat transfer rate decreases naturally. The temperature of the evaporator is sometimes cooled by the returned liquid due to competition between the drag force and the weight of the falling liquid. This phenomenon is called rewetting. The cycle then repeats. Thus, irregular temperature fluctuations appear, as shown in Figures 18 and 19, as well as degradation of the loop conductance (Figure 10).
4. Conclusions

An FHP is a passive-heat-transport, air-cooled device that is used to remove the decay heat of spent nuclear fuel rods stored in a liquid pool during an SBO accident. An FHP has a unique fork-like geometric design to resolve the significant mismatch between the convective heat transfer coefficients outside the evaporator and condenser parts: the evaporator at the bottom is a single heat exchanger tube, whereas the condenser at the top consists of several finned tubes to create an enlarged heat transfer area.

In this study, various heat transfer characteristics and the operation limit features of a unit FHP with a unique design geometry were investigated experimentally using a manufactured laboratory-scale model. The main findings of this study are:

- When the temperature of the heat source liquid in which the evaporator tube of the sub-atmospheric, water-filled FHP was submerged increased, the working fluid became superheated and regular operation was initiated by characteristic geyser boiling, together with a significant fluctuation in the instant heat transfer rate. Before regular operation began, nominal heat transfer performance was observed.

- Once geyser boiling occurred, cyclic fluctuations in the temperature of the evaporator and the instant heat transfer rate were observed, due to the waiting periods for bubble nucleation and rapid bubble growth. The loop conductance of the FHP as an index of the average heat transfer performance increased with the heat source temperature and corresponding heat transfer rate.

- At high heat source temperatures, the counter-current, two-phase flow pattern in the evaporator was formulated as the condensate liquid descends along the inner wall of the evaporator tube whereas the generated vapor rises in the center. At the upper end of this regime, the maximum loop conductance point appeared.

- A further increase in the temperature of the heat source, however, caused the counter-current flow limit phenomenon to occur inside the evaporator. Irregular, unstable fluctuations in the evaporator temperature appeared due to localized dry-out of the condensate liquid film descending on the inner wall of the evaporator. This restricted further increase in the heat transfer and degraded loop conductance by increasing the temperature difference at the evaporator of the FHP.

- The measured maximum attainable heat transfer rate of the unit FHP model can be accurately predicted by the existing correlations of the counter-current flow limit for a single-rod-type heat pipe. The limit is directly proportional to the cross-sectional area of the evaporator tube, as seen in the existing correlations in Table 3. Therefore, while the current design of the finned condenser for air cooling remains, the maximum attainable heat transfer rate of the unit FHP can be further increased by enlarging the tube diameter of the evaporator.

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### Nomenclature

| Symbol | Description |
|--------|-------------|
| $A$ | Area ($m^2$) |
| $a_n$ | Systematic error for variable $n$ |
| $Bo$ | Bond number |
| $C_k$ | Kutateladze constant |
| $C_w$ | Wallis constant |
| $c_{p,l}$ | Specific heat capacity of liquid (J/kgK) |
| $D$ | Diameter (m) |
| $g$ | Gravitational acceleration (m/s$^2$) |
| $H$ | Elevation/Height (m) |
| $h$ | Heat transfer coefficient (W/m$^2$K) |
| $h_f$ | Latent heat (J/kg) |
| $h_{hp}$ | Loop conductance / Effective heat transfer coefficient for heat pipe (W/m$^2$K) |
| $K$ | Modified Kutateladze number |
| $k$ | Coverage factor |
| $N$ | Sample number/Number of data |
| $P$ | Pressure (Pa) |
| $Q$ | Volumetric flow rate ($m^3$/s) |
| $q$ | Heat transfer rate (W) |
| $R$ | Thermal resistance (K/W) |
| $s_n$ | Standard deviation of variable $n$ |
| $T$ | Temperature ($^\circ$C) |
| $U$ | Combined uncertainty |
| $u_n$ | Uncertainty of variable $n$ (unit of $n$) |
| $u_{A,n}$ | Type A uncertainty of variable $n$ |
| $u_{B,n}$ | Type B uncertainty of variable $n$ |
| $\rho_l$ | Density of liquid (kg/m$^3$) |
| $\rho_v$ | Density of vapor (kg/m$^3$) |
| $\sigma$ | Surface tension (N/m) |

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