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Original Article

Investigation of the Thermal Performance of a Vertical Two-Phase Closed Thermosyphon as a Passive Cooling System for a Nuclear Reactor Spent Fuel Storage Pool

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ABSTRACT

The decay heat that is produced by nuclear reactor spent fuel must be cooled in a spent fuel storage pool. A wickless heat pipe or a vertical two-phase closed thermosyphon (TPCT) is used to remove this decay heat. The objective of this research is to investigate the thermal performance of a prototype model for a large-scale vertical TPCT as a passive cooling system for a nuclear research reactor spent fuel storage pool. An experimental investigation and numerical simulation using RELAP5/MOD 3.2 were used to investigate the TPCT thermal performance. The effects of the initial pressure, filling ratio, and heat load were analyzed. Demineralized water was used as the TPCT working fluid. The cooled water was circulated in the water jacket as a cooling system. The experimental results show that the best thermal performance was obtained at a thermal resistance of 0.22°C/W, the lowest initial pressure, a filling ratio of 60%, and a high evaporator heat load. The simulation model that was experimentally validated showed a pattern and trend line similar to those of the experiment and can be used to predict the heat transfer phenomena of TPCT with varying inputs.

1. Introduction

One of the most important initiating events in a nuclear power plant that could cause a severe accident is station blackout (SBO). During SBO, the core risk is that the plant is not properly cooled. The excess residual heat generated can cause a fuel meltdown and, consequently, radioactive release to the environment [1].

The critical accident at the Tokyo Electric Power Company's Fukushima Daiichi nuclear power plant on March 11,

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2011, occurred as a result of an SBO following an unprecedented earthquake and tsunami [2]. This event has impacted the safety design development of nuclear power stations and research reactors all over the world, including spent fuel storage pools (SFSPs). With regard to the SFSP, this accident showed the possible benefits of using a passive cooling system to remove residual heat produced in the SFSP when an SBO occurs. The main goals are to ensure the inherent safety of the nuclear reactors and to maintain no radioactive release to the environment when an accident occurs.

Although the amount of decay heat generated from the SFSP is less than the heat released in the nuclear power plant core, it must be removed to prevent heat accumulation from damaging the integrity of the SFSP spent fuel cladding. SFSPs are generally used to store spent fuel that has been used for 10–30 years of reactor operation. The cooling water for the SFSP must maintain its operation level. If the water level lowers because of water leakage or water evaporation due to decay heat, feed water will be added to maintain the water level required for operation [3–5]. If the SFSP is dry due to loss of coolant, the spent fuel could melt due to the decay heat and could cause an accumulation of hydrogen gas in the SFSP [6]. The probability of hydrogen explosion is given special attention in this case.

One method to remove the decay heat is by using a heat pipe. This is an effective technology that transports large amounts of heat with a small temperature difference, working in two phases; its operation does not require any additional external driving source (i.e., it is a passive system). A heat pipe naturally circulates condensate from the condenser to the evaporator utilizing a capillary pumping system of porous media [or without porous media, which system is called a two-phase closed thermosyphon (TPCT)]. The amount of heat per unit time transferred by the heat pipe will be greater because of the latent heat of the working fluid [7–13]. Although it has a simple structure, a TPCT has 200–500 times the thermal conductivity of copper. A TPCT has a number of working fluids in the evaporator with a certain volume. The heat source is placed in the evaporator, and then heat is conducted through the heat pipe wall. The working fluid in the liquid pool boils and then evaporates. The vapor rises to the condenser due to the higher pressure through the adiabatic section. In the condenser, the vapor releases its latent heat to the condenser wall and turns into a liquid [14–17]. Furthermore, the liquid on the inner wall of the condenser will flow down to the evaporator by gravity. With its reliance on gravity, a TPCT cannot be used at an inclination approaching horizontal or in a horizontal position [16].

Numerical simulation studies using heat pipes and thermosyphons in the field of nuclear reactors have been conducted by Alizadehdakhel et al [15], Kafeel and Turan [18], and Tung et al [19]. Their results indicate that the performance of a heat pipe heat sink is reliable when there is excess heat that could damage the nuclear safety system. Ye et al [1] conducted a simulation using computational fluid dynamics to study the residual heat removal system in an SFSP using loop heat pipes. Their results showed that the performance of the loop heat pipes is reliable for the removing of the residual heat generated from the SFSP. Loop heat pipes can maintain the temperature of the SFSP below the boiling temperature. Using RELAP5 code and ANSYS FLUENT (ANSYS, Inc. Pennsylvania, United States), Fu et al [20] investigated the heat transfer characteristics, and temperature and velocity distributions of a two-phase thermosyphon loop as a long-term passive cooling system for a nuclear reactor SFSP. Their results showed that a two-phase thermosyphon loop can be used in the SFSP of the generation III advanced nuclear reactor AP1000 design. Using RELAP5/MOD 3.2 simulation, Kusuma et al [21] investigated the heat flux effect in a straight heat pipe as a passive residual heat removal system in a light water reactor. Their results showed that the steady state condition and the heat transfer phenomena of a straight heat pipe can be achieved rapidly with a higher flux in the evaporator section. A higher heat flux will increase the pressure of the straight heat pipe, and the working fluid evaporation will affect the higher saturation temperature of the working fluid.

Many experimental studies of TPCT phenomena have been conducted. A TPCT has been used for many applications such as the de-icers for roadways [15], heat pipe heat exchangers in power generators [16], and water heaters [17]. Xiong et al [22] experimentally investigated a loop-type heat pipe system for a passive cooling system of AP1000 SFSP. Their results show that a large-scale heat pipe loop can remove significant heat from the SFSP.

From previous studies, it is known that a vertical TPCT for passive cooling of a nuclear SFSP has not yet been studied or used. The objective of this study was to investigate the thermal performance of a prototype model of a large-scale vertical TPCT designed as a passive cooling system for a nuclear research reactor SFSP. This research used experimental investigations and RELAP5/MOD 3.2 simulations. The initial pressure, filling ratio, and heat load of the TPCT were used as the variables in this research. The experimental results were then used to validate the simulation model. In this investigation, only steady-state phenomena were studied to determine when the natural circulation in this TPCT will establish sustainable cooling.

2. Methodology

2.1. *Experimental setup*

Fig. 1 shows the experimental setup of the TPCT, which was made from a copper tube that was 1,500 mm long with inner and outer diameters of 25.4 mm and 26.4 mm, respectively. The TPCT was divided into three sections: a 500-mm long evaporator section at the bottom, a 500-mm long adiabatic section at the middle, and a 500-mm long condenser section at the top. The initial TPCT experiment did not consider the lengths of the evaporator, adiabatic, or condenser sections. At the evaporator section, heat was generated by Ni–chrome wire resistance veiled with ceramic. The heat input was controlled using an analog voltage regulator, and the electrical current was measured using a clamp meter. For the adiabatic section, the copper tube was isolated to minimize heat loss to the environment. In the condenser section, a water jacket was installed to absorb heat in the condenser and was connected to a circulating thermostatic bath to provide a constant

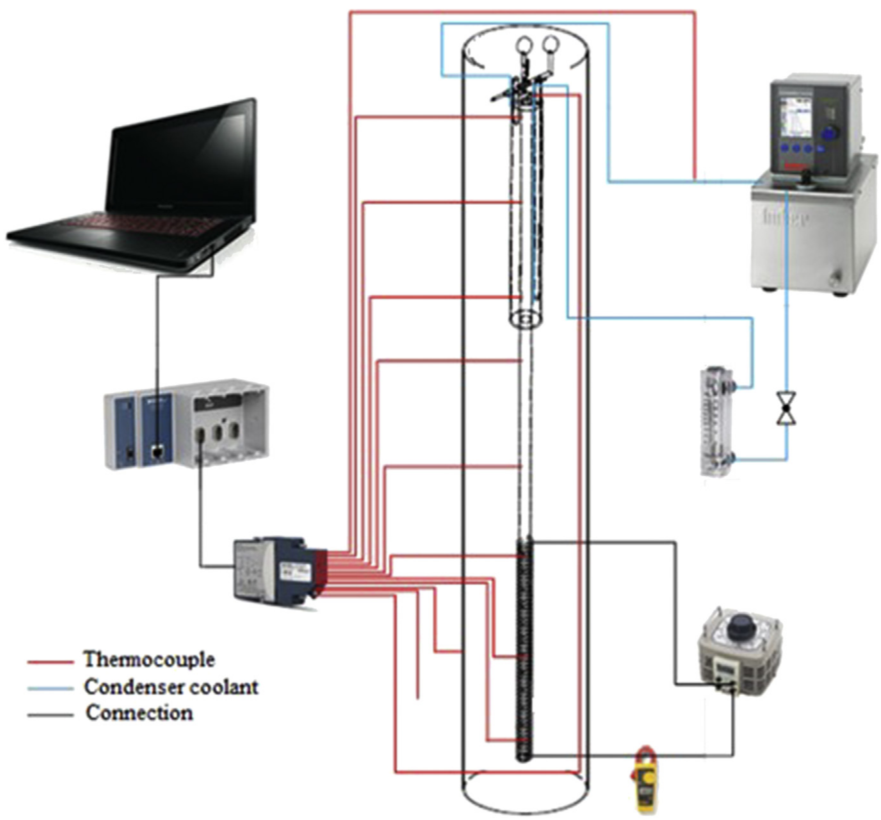

Fig. 1 – Experimental setup.

temperature of cooling water at the inlet section. The temperature and mass flow rate of the coolant were controlled using a circulating thermostatic bath for a constant cooling water temperature of 30°C and a cooling water mass flow rate of 1 L/min. A flow meter with an accuracy of ±4% was used to measure the coolant mass flow rate in the water jacket. Demineralized water was selected as the TPCT working fluid. The TPCT system was insulated with a ceramic blanket and glass wool to reduce the heat loss. Preheating and vacuum processes were performed to remove dissolved gas inside the TPCT and in the working fluid.

The experimental data were recorded using a National Instrument data acquisition system. K type thermocouples with an accuracy of ±0.05°C were used: 12 were placed on the TPCT outside wall, three on the outside condenser wall, two on the outside adiabatic wall, three on the outside evaporator wall, one on the coolant inlet, one on the coolant outlet, and one on the evaporator isolation wall; one other was used to measure the ambient temperature. The placement of the thermocouples on the TPCT wall is shown in Fig. 2. At the top of the vertical TPCT, a pressure gauge with an uncertainty of 2.07 Pa was installed to measure the pressure inside the vertical TPCT when the vacuum process was performed.

2.2. *Experiment*

The experiment was divided into two steps. The first step was to determine the best filling ratio in the TPCT and the heat load given to the evaporator section. The results obtained were used as fixed parameters in the subsequent experiments. In the first experiment, the heat load varied among 48.75 W, 103.35 W, and 187 W. The filling ratio (defined as the ratio of the filling liquid volume to the evaporator section volume) varied among 45%, 50%, 55%, 60%, 65%, and 70%. The constant parameters were TPCT initial pressure of −62 cmHg, cooling water mass flow rate of 1 L/min, and temperature of 30°C. The results obtained show that the best thermal performance, which produced a thermal resistance of 0.26 °C/W, was obtained for a filling ratio of 60% and a head load of 187 W, as shown in Fig. 3.

The second experiments were conducted by varying the TPCT initial pressure among −62 cmHg, −64 cmHg, and −66 cmHg. The constant parameters were a heat load of 187 W, a filling ratio of 60%, a cooling water mass flow rate of 1 L/min, and a temperature of 30°C.

2.3. *Simulation using RELAP5/MOD 3.2*

The computer code RELAP5/MOD 3.2 is commonly used for thermal-hydraulics simulation of light water reactor coolant systems during postulated accidents. In this research, this code was used to simulate the TPCT as a prototype model for a nuclear reactor SFSP passive cooling system. The model has the same geometry as the test equipment. The nodalization used in the simulation is shown in Fig. 4. As can be seen in the figure, several PIPE models were used to denote the TPCT. The PIPEs of P01, P02, and P03 represent the vapor lines of the evaporator, adiabatic, and condenser sections, respectively.

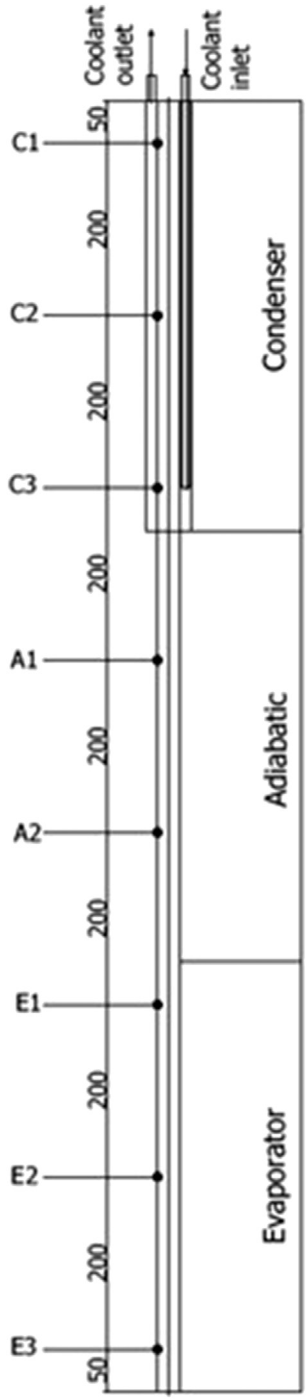

Fig. 2 – Placement of thermocouple on two-phase closed thermosyphon (TPCT) wall.

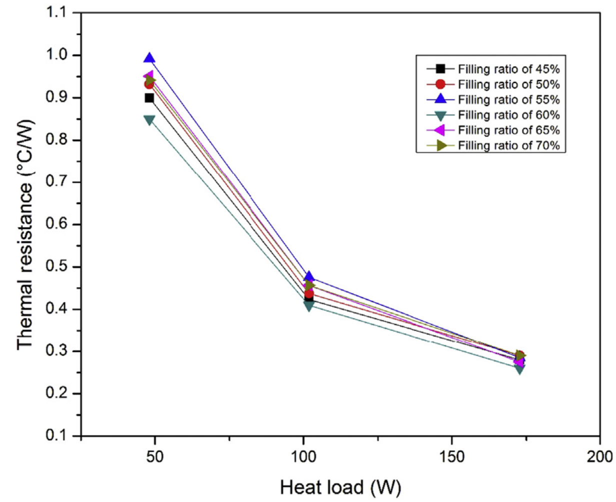

Fig. 3 – Thermal resistance of first step experiment.

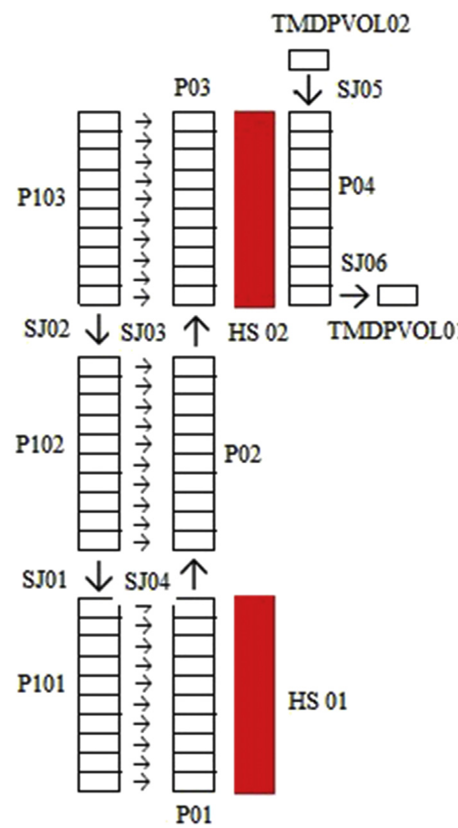

Fig. 4 – Computer code RELAP5/MOD 3.2 (developed at Idaho National Laboratory) nodalization.

Similarly, the PIPEs of P101, P102, and P103 represent the fluid lines of the evaporator, adiabatic, and condenser sections, respectively. Meanwhile, single junction models of SJ01–SJ06 were used to connect the PIPEs. To capture the heat transfer phenomena in the system, heat structure HS 01 was implemented to simulate the heat load from an electrical heater. In addition, heat structure HS 02 was used to model the heat transfer between the cooling water and the condenser section. On the other side of the condenser, the PIPE of P04 was used to flow the coolant into the water jacket. TMDPVOL01 represents a water repository, and TMDPVOL02 represents a source of cooled water for the water jacket.

In the simulation, the effects of the initial pressure were analyzed. The initial pressure was varied among −62 cmHg, −64 cmHg, and −66 cmHg. The working fluid was charged with a filling ratio of 60%, and the heat load for the evaporator section was set to 187 W. Demineralized water was used as the working fluid inside the TPCT. Coolant water with a temperature of 30°C and a mass flow rate of 1 L/min was circulated in the water jacket for the cooling system.

2.4. Data calculation

The heat load inside the evaporator section is defined as follows:

$$Q_{in} = V \cdot I - q_{loss} \tag{1}$$

To ensure that the heat load inside the evaporator was the net heat load, the heat loss that occurred in the TPCT system must be calculated. The analysis used empirical correlations for the external free convection flows for a vertical plate [23]. The long vertical cylinder was divided into many vertical plates, and the heat loss was calculated on each vertical plate. Finally, the heat loss on each vertical plate was summed as the heat loss of the long vertical cylinder.

The heat loss by free convection from the evaporator insulation wall to the room was given by Newton's law of cooling:

$$q_{loss} = \overline{h} A_s (T_s - T_\infty) \tag{2}$$

which can be obtained from the Rayleigh number for the transition to turbulence that occurs on the vertical plate.

$$Ra_L = \frac{g\beta(T_s - T_\infty)L^3}{\alpha v} \tag{3}$$

can be obtained from the Churchill and Chu equation:

$$\overline{Nu_L} = \left(0,825 + \frac{0,387 Ra_L^{\frac{1}{6}}}{\left[1 + \left\{\frac{0,492}{Pr}\right\}^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right)^2 \tag{4}$$

and can be obtained using:

$$\overline{h} = \frac{\overline{Nu_L}}{L} k \tag{5}$$

The thermal resistance of the TPCT, as a thermal performance indicator, was calculated using the following equation:

$$R_T = \frac{T_e - T_c}{Q_{in}} \tag{6}$$

3. Results and discussion

3.1. Experiment

3.1.1. Transient temperature

The transient temperature values for varying TPCT initial pressures of −62 cmHg, −64 cm Hg, and −66 cmHg are presented in Fig. 5. The evaporator temperature increased due to the continuously supplied heat load, and the fluid inside the vertical TPCT boiled rapidly to the saturation temperature. At an initial pressure of −66 cmHg, the evaporator temperature had the lowest peak temperature. This occurs because decreasing the initial pressure decreases the saturation temperature of the fluid and the evaporator wall temperature.

When the TPCT initial pressure increased, the evaporator temperature increased due to the higher saturation temperature of the fluid. However, natural circulation process did not occur because the vapor starts to rise to the condenser. The high temperature difference increases the thermal resistance. This shows that the condensate was still in the condenser and did not flow down to the evaporator. The absence of condensate falling from the condenser to the evaporator caused the evaporator temperature to increase drastically. The temperature of the adiabatic and condenser sections increased, but not to a level as high as that in the evaporator. The highest value of the temperature in the evaporator is typically called the overshoot. Overshoot occurred because the inner wall of the vertical TPCT was in direct contact with the vapor on the evaporator. The working fluid still did not circulate; therefore, the evaporator wall temperature became high. The condenser temperature approached the value of the coolant temperature during overshoot.

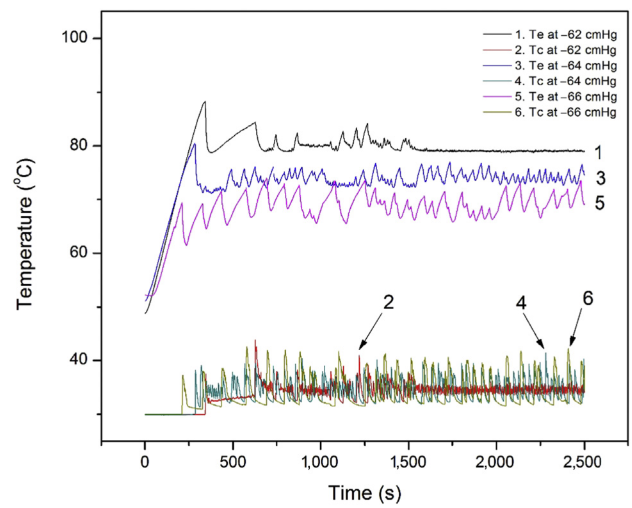

Fig. 5 – Transient temperature at filling ratio of 60% and heat load of 187 Watts.

After overshoot occurred, a zigzag temperature profile was observed, which indicates the start of the natural circulation in the TPCT. This phenomenon is called zigzag and showed the transient condition inside the TPCT. The vapor arrived at the condenser, and the temperature of the condenser section increased. The latent heat was absorbed by the cooling water in the water jacket, passing through the inner wall of the condenser, and the condensate fell to the evaporator through the adiabatic section due to gravity. The temperatures of the adiabatic and evaporator sections started to drastically decrease. This phenomenon repeated until a steady state was reached that was indicated by the temperature profile. These conditions were expected to continue to occur because continuous natural circulation was established.

The pattern obtained in this investigation is in agreement with the results obtained by previous researchers, namely, overshoot, zigzag, and stable periods [24].

3.1.2. Steady-state temperature distribution

Fig. 6 shows the temperature distribution at the TPCT wall with varying initial pressure. It can be seen that the temperature difference between the evaporator and the condenser will be smaller for a lower initial pressure in the TPCT. The initial pressure significantly affects the vertical TPCT

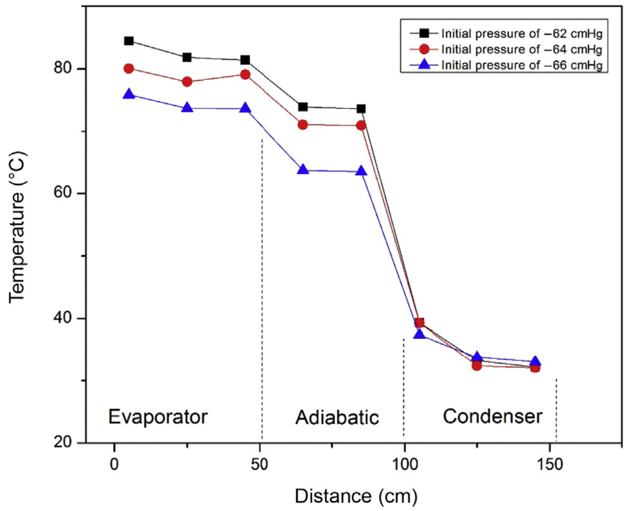

Fig. 6 – Two-phase closed thermosyphon (TPCT) wall temperature distribution at a filling ratio of 60% and heat load of 187 Watts.

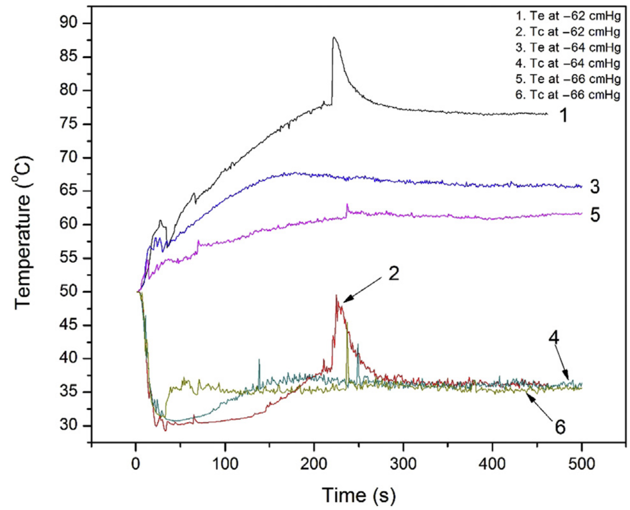

Fig. 7 – RELAP5/MOD 3.2 (developed at Idaho National Laboratory) simulation for transient temperature.

temperature distribution. The smallest difference between the evaporator and the condenser temperature resulted in the lowest thermal resistance. The TPCT will more quickly reach a stable condition at a lower initial pressure. A lower initial pressure decreases the boiling point of the working fluid. Decreasing the working fluid boiling point accelerates the formation of vapor that rises to the condenser and accelerates the establishment of natural circulation.

The initial pressure of the TPCT affects the saturation temperature of the fluid. Decreasing the initial pressure can decrease the amount of noncondensable gas (NCG) that accumulates near the inner wall of the condenser section. In fact, an NCG is a gas that cannot be condensed inside the TPCT and will stay in the gas state during the operation. The most common NCGs are air, oxygen, carbon dioxide, nitrogen, etc. NCGs are mainly created by chemical reactions among impurities, the container wall material, and the working fluid. The existence of an NCG in the TPCT leads to increased evaporator wall temperature, decreased temperature difference between the evaporator and the condenser, and decreased thermal resistance. At higher TPCT initial pressures, the condenser wall temperature was lower, and the evaporator temperature was slightly above the saturation temperature, causing higher thermal resistance in the vertical TPCT.

3.2. *Simulation*

Simulation using RELAP5/MOD 3.2 was conducted after performing the experiment. The simulation model was made using the same geometry as that used in the experiment. Fig. 7 shows the transient temperature for the TPCT initial pressures of −62 cmHg, −64 cmHg, and −66 cmHg. At an initial pressure of −66 cmHg, the boiling point of demineralized water is the lowest. A lower initial pressure results in a lower temperature difference between the evaporator and the condenser. A lower boiling point decreases the evaporator temperature and results in more rapid vapor growth toward the condenser. Achieving natural circulation will occur faster in this state.

The simulation results show smooth fluctuations at steady state. The transient temperature profiles in the evaporator, adiabatic, and condenser sections fluctuate following a pattern and trend line similar to those obtained from the experiment. The patterns that were obtained in this simulation showed the same phenomena as the experiment, namely, overshoot, zigzag, and steady phenomena. The different values of transient temperature between the experiment and the simulation may be due to the insulation. Perfect insulation was easy to achieve in the simulation model, but it was difficult to achieve for the experimental test section. Another reason could be the number of defined model nodes. If more nodes were added to each section, the simulation results would be better, but the simulation running time would be longer.

The TPCT model of the RELAP5/MOD 3.2 simulation was validated with the experiment, and it can therefore be used to simulate TPCTs with different inputs.

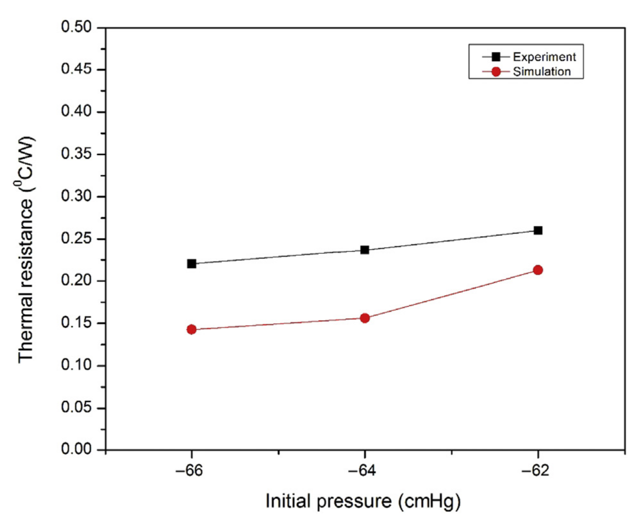

Fig. 8 – Thermal resistances of experiment and simulation.

3.3. Thermal performance

The thermal performance of the TPCT can be represented by the thermal resistance. Fig. 8 shows the thermal resistances of the experiment and the simulation. The best thermal performance of the experiment and simulation were 0.22°C/W and 0.14°C/W, respectively. The best thermal performance was obtained at an initial pressure of −72 cmHg.

It can be observed that the simulation has better thermal performance than the experiment. This can be explained as follows: in the simulation, no heat loss occurs from the adiabatic system, and the insulation can perform perfectly as an ideal system; also, the simulation did not consider the existence of NCG in the TPCT.

4. Conclusions

The thermal performance of a prototype model of a large-scale vertical TPCT applied as a passive cooling system for a nuclear reactor SFSP was investigated. The experimental investigation showed that the best thermal performance was obtained at a thermal resistance of 0.22°C/W. The best thermal performance was achieved at the lowest initial pressure, at a filling ratio of 60%, and at a high evaporator heat load. The simulation model showed a pattern and trend line similar to those of the experiment, and thus the simulation can be used to simulate heat transfer phenomena with varying TPCT inputs.

The vertical TPCT used as passive cooling for a spent fuel pool is large due to the spent fuel pool size. The future plan is to propose a large-scale vertical TPCT and to improve the vertical TPCT design for SFSP in order to determine the TPCT's heat transfer characteristics.

Conflicts of interest

All authors have no conflicts of interest to declare.

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Nomenclature

A_s	Surface area of evaporator insulation wall (m^2)
α	Thermal diffusivity of the air (m^2/s)
β	Volumetric thermal expansion coefficient (1/K)
g	Local acceleration due to gravity (m/s^2)
$\overline{h}$	Heat transfer coefficient ($W/m^2 \cdot K$)
I	Electrical current (A)
k	Thermal conductivity of the ambient air ($W/m \cdot K$)
L	Insulation height of the evaporator (m)
$\overline{Nu_L}$	Nusselt number
Pr	Prandl number
Q_{in}	Amount of heat load at the evaporator (W)
q_{loss}	Amount of heat loss through the evaporator insulation wall (W)
Ra_L	Rayleigh number
R_T	Thermal resistance (°C/W)
T_∞	Ambient temperature (°C)
T_c	Average wall temperature of the condenser (°C)
T_e	Average wall temperature of the evaporator (°C)
T_s	Insulation temperature on the evaporator (°C)
V	Electric voltage (V)
ν	Kinematic viscosity of the air (m^2/s)

Abbreviations

SFSP	Spent fuel storage pool
TPCT	Two-phase closed thermosyphon
SBO	Station blackout

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