MARS-KS Simulation of a Vertical Heat Pipe with Finned Air-cooled Condenser

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1. Introduction

Since the Fukushima nuclear accident, the passive cooling system for the spent fuel pool (SFP) against a station blackout (SBO) accident has got great attention. Typically, heat pipes have been used for cooling of small electronic devices. Therefore, to exploit heat pipes for application to the design of passive cooling systems for nuclear power plants, diverse experimental and numerical analysis studies should be conducted.

Recently, some new passive cooling systems using heat pipes were proposed [1][2]. Kusuma et al. [3]-[5] conducted heat transfer experiments for a large-scale vertical heat pipe and numerical simulations using RELAP/MOD 3.2. Lim and Kim [6] proposed a new concept of a large-scale, air-cooled fork-end heat pipe (FEHP) and experimentally examined the heat transfer performance of a laboratory-scale FEHP sample. In connection to the work of Lim and Kim [6]. Park et al. [7] attempted to numerically simulate the experimental results using MARS-KS 1.4.

In order to design a fully passive cooling system that can continuously remove residual heat from a spent fuel pool without an aid of electrical power, experimental and numerical studies about the development and validation of large-scale, air-cooled heat pipes should be continuously conducted. In this study, heat transfer experiments for a vertical, air-cooling heat pipe with height of 2 m were conducted and thermal performance of the heat pipe was simulated using MARS-KS 1.5. From a comparison of the obtained results from the experiment and simulation, capabilities of MARS-KS 1.5 to simulate the heat transfer characteristics of a laboratory-scale heat pipe that operates at low pressure condition was evaluated.

2. Methodology

2.1 Experiment

A schematic diagram of the vertical heat pipe experimental device is shown in Figure 1. The heat pipe was divided into three parts: evaporator, adiabatic section, condenser. The heat pipe was made from copper tube. A length of the evaporator is 0.85 m, with outer diameter of 19.05 mm and width of 0.8 mm. A length of the adiabatic part is 0.207 m, with outer diameter of 19.05 mm and width of 0.8 mm. A length of the condenser is 1 m, with outer diameter of 19.05 mm and width of 0.8 mm. The fin is 9.225 mm high and 0.5 mm thick, with 217 attached to the condenser tube. An adiabatic part acted as collect condensed working fluid from condenser tubes. For this reason, the working fluid easily returned from the condenser to the evaporator. The working fluid is water. The heat pipe is heated by a ribbon heater wrapped in an area of 85cm of evaporator, and a wind tunnel with a width of 12cm – total length of about 130cm is installed and cooled by air.

This experiment was conducted to measure the cooling capacity of the heat pipe. A sensor is attached to the top of the heat pipe to measure the temperature and pressure inside the heat pipe, and thermocouples are requested at intervals of 17cm and 22.5cm on the surface of the evaporator and condenser, respectively, so that surface temperature can be measured. The cooling capacity of the air-cooled section of the heat pipe was calculated as shown in Equation (1).

$$Q = v_g \rho_g A_{flow} c_{p,g} \Delta T \tag{1}$$

In the above equation, v_g is the flow rate of air in the air-cooling section, ρ_g is the density of air, A_{flow} is the flow area in the air-cooling section, $c_{p,g}$ is the specific heat of the air, and ΔT is the difference of inlet and outlet temperature in air-cooling section. The uncertainty for each measured value is as shown in Table 1.

Table 1: Uncertainties of experimental measurement.

Value	Uncertainty
Air velocity (v_g)	±0.03 m/s
Air density (ρ_g)	±0.001 kg/m^3
Specific heat ($c_{p,g}$)	±0.87 J/kg·℃
Temperature (T_{in} & T_{out})	±1 ℃
Pressure (P)	±0.2585 kPa

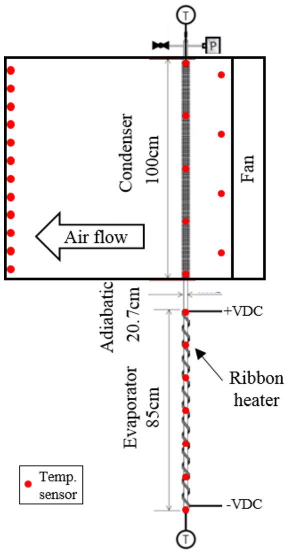

Fig. 1 Schematic of experiment facility.

2.2 Simulation using MARS-KS 1.5

The MARS-KS 1.5 modeling has the same geometric structure as the experiment setup, and the simulation was performed after the experiment. The nodalization of the heat pipe model consists of various pipe models as shown in Figure 2.

The pipes of P-100 and P-120, respectively, represented the evaporator and condenser. Each pipe consists of 10 nodes. SV-110, SV-210, and SV-220, respectively, represent adiabatic section and air cooling section. In pipe components, the boundary conditions are set by pressure, temperature. Meanwhile, the Single junction components of SJ-105, SJ-115, SJ-215, SJ-215 and SJ-225, were used to connect pipe. SJ-105 was used to connect the evaporator with the adiabatic part, and SJ-115 was used to connect the adiabatic section with condenser. SJ-225 was used to set the air-cooling section to the same conditions as the ambient.

Time dependent volume TDV-200 and TDV-230 represented the atmospheric environment of the air-cooled section. TDV was used to set boundary conditions related to pressure and temperature of air flow condition. Time dependent junction TDJ-205 is used to set the air flow rate from the blower. The heat structures HS-100 and HS-120 were applied to the simulation to capture the heat transfer phenomenon of the heat pipe system. The boundary condition of the heat structure is related to the heat flux. The thermal structure HS-100 is implemented to simulate the heat load (heat flux) of the evaporator applied in the ribbon heater. In addition, the heat structure HS-120 was used to indicate the heat transfer between the air-cooling section and condenser.

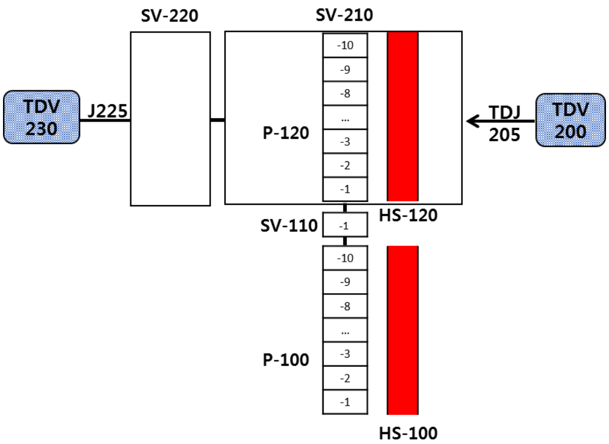

Fig. 2 MARS-KS 1.5 nodalization of a vertical heat pipe.

2.3 Air-cooling modeling

The heat transfer coefficient of a single bare tube in the condition of cross flow can be calculated using the Zukauskas's correlation in Eq. (2). The constants to account for the effects of flow velocity in Eq. (2) are summarized in Table 2.

$$h = C\frac{k}{D}Re^{m}Pr^{0.36}\left(\frac{Pr}{Pr_w}\right)^{1/4} \quad (2)$$

Table 2: Empirical constants.

Re	C	M
10 ~ 100	0.8	0.4
10 ~ $2{\cdot}10^5$	0.27	0.63
>$2{\cdot}10^5$	0.021	0.84

In addition, the effects of the increased heat transfer area due to fins can be reflected into the heat transfer coefficient correlation using the fouling factor. The increased heat transfer area due to fins, as seen in Figure 3, was calculated using Eqs. (3) – (6).

$$A_f = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 \tau \quad (3)$$
$$A_b = 2\pi r_1 L \quad (4)$$
$$A_b' = 2\pi r_1 (L - N_f \tau) \quad (5)$$
$$A^* = \frac{N_f A_f + A_b'}{A_b} \quad (6)$$

Here A_f is the surface area of the pin, τ is the thickness of the fin, and L is the length of the tube. A_b is the surface area of the bore tube without fin, A_b' is the surface area of the tube with fin, and N_f is the number of fins per tube. A^* is the fin factor which represents the increment of heat transfer area by fin. The fin factor calculated using the above equations was 6.98.

Internal vapor pressure is one of major physical parameters that represent the heat transfer characteristics of operating heat pipes. Thus, if the heat transfer modeling in the part of air-cooling is appropriate, the calculated internal pressure in simulation should be in reasonable agreement with experimental data. However, when the calculated fin factor of 6.98 was applied for the MARS-KS simulation, tremendously large deviations in vapor pressure between simulation and experiment were observed. Instead, it was attempted in the present simulation to find the optimal fin factor that best predicts the experimentally measured vapor pressures of the heat pipe at various heat flux conditions. It was found that the MARS-KS simulation with the fin factor of 20.98 accurately predict the internal pressure of the heat pipe measured in the experiment. It is unclear why the fin factor of 20.98 approximately three times higher than the calculated value of 6.98 shows such a good prediction. It can be supposed that the addition of thin fins to the bare tube improves the convection heat transfer by enhancing turbulent agitation near the heat transfer surface. However, further works are required to clarify the observed discrepancy related to the fin factor.

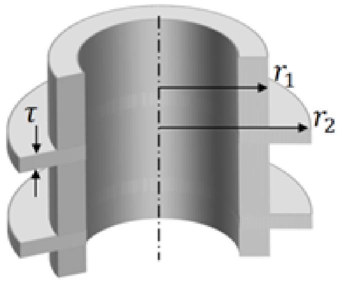

Fig. 3 Schematic of fin tube geometry.

3. Results and discussions

3.1 Thermal performance

Figure 4 shows heat transfer rate at condenser according to the heat load. Within the range of measurement uncertainty, it was found that the experimental results and the simulation results were consistent. This confirmed that most of the heat were condensed through heat pipes without heat loss.

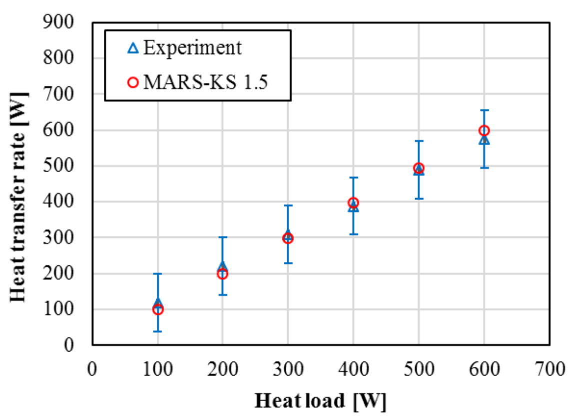

Fig. 4 Heat transfer rate at condenser with heat load.

3.2 Pressure

Figure 5 showed comparison of the heat pipe operating pressure measured in the experiment with the results of the MARS-KS 1.5 simulation. The repeated calculation of the MARS-KS 1.5 code confirmed that the calculated pressure of the heat pipe was well converged under the experiment pressure condition. As shown in Figure 5, the heat pipe pressures of the MARS-KS analysis and experiment are almost identical in all heat load conditions. As the heat load increases, the amount of evaporation of the working fluid increases, which lowers the water level, thereby increasing the pressure throughout the heat pipe.

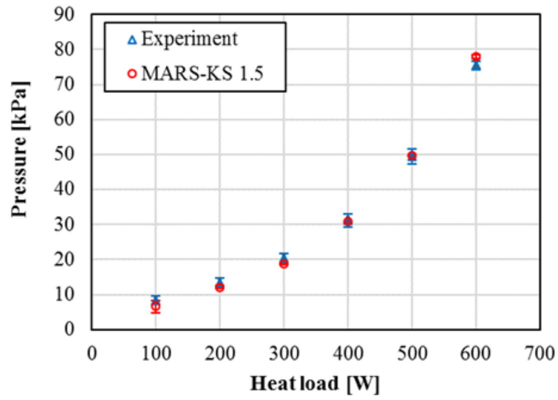

Fig 5. Pressure at the adiabatic section with heat load.

3.3 Temperature

Figure 6 showed comparison results the temperature of the adiabatic section measured in the experiment with that calculated by MARS-KS code according to heat load. The adiabatic section converged at the saturation temperature corresponding to the operating pressure as the working fluid is present at saturated condition. Comparing the experimental values and simulation results for temperature of adiabatic section to relative error, the results were almost converging within the margin of error (±4°C-1.2%).

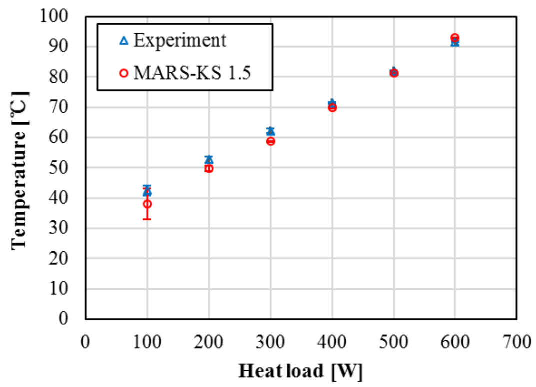

Fig 6. Working fluid temperature at the adiabatic section.

4. Conclusions

Heat transfer performance of a vertical finned tube heat pipe was analyzed using the MARS-KS code. The key findings from the present study are following:

1. In the experiment results, a vertical finned tube heat pipe removed heat stably under the air cooling natural convection conditions. Thus, it demonstrated the possibility of application to the design of large-scaled passive cooling system.

2. The operation conditions of the heat pipe were analyzed using MARS-KS 1.5. This confirmed that the analysis results well predict the experimental results, such as working fluid temperature, operation pressure, etc., when additional factors are considered in the correlation of the wall heat transfer coefficient for the air cooling through orthogonal flow of the fin tube.

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