Experimental investigation on Heat Transfer Performance of Annular Flow Path Heat Pipe

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1. Introduction

Heat pipes have been widely used in various fields like solar energy, CPU cooling, small scale heat transfer device, and so on. Many researches have presented the conceptual design of heat removal systems using heat pipe to apply nuclear safety systems. Jouhara et al. [1] suggested a new concept for nuclear desalination system based on heat pipe technology and the use of heat pipe-based heat recovery systems in desalination plant is expected to improve the overall system performance of the desalination process. Mochizuki et al. [2] was suggested the passive cooling system to spent nuclear fuel pool. Detail analysis of various heat pipe design cases was studied to determine the heat pipes cooling performance. Wang et al. [3] suggested the concept PRHRS of MSR using sodium heat pipes, and the transient performance of high temperature sodium heat pipe was numerically simulated in the case of MSR accident.

The meltdown at the Fukushima Daiichi nuclear power plants alarmed to the dangers of station blackout (SBO) accident. After the SBO accident, passive decay heat removal systems have been investigated to prevent the severe accidents. Mochizuki et al. [4] suggested the heat pipes cooling system using loop heat pipes for decay heat removal cooling and analysis of heat pipe thermal resistance for boiling water reactor (BWR). The decay heat removal systems for pressurized water reactor (PWR) were suggested using natural convection mechanisms and modification of PWR design.

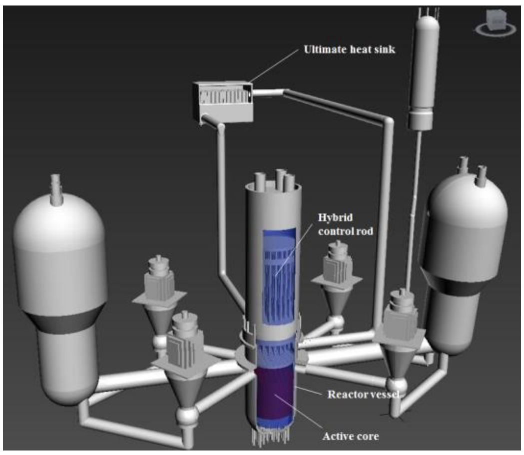


Fig 1. A design of Passive In-core Cooling system (PINCs)

Our group suggested the concept of a hybrid heat pipe with control rod as Passive IN-core Cooling System (PINCs) for decay heat removal for advanced nuclear power plant. Hybrid heat pipe is the combination of the heat pipe and control rod. It is necessary for PINCs to contain a neutron absorber (B_4C) to have ability of reactivity control. It has annular vapor space and it might cause the heat transfer degradation to the thermal performance of the heat pipes. In the present research, the main objective is to investigate the effect of the inner structure to the heat transfer performance of heat pipe containing neutron absorber material, B_4C.

2. Experimental Setup and Procedure

In the present work, the heat pipe has two layers of stainless steel screen wire mesh as the wick structure and distilled water as the working fluid. Thermal performance of heat pipe was tested horizontally and vertically with various heat loads. Stainless steel 316L test sections have outer diameter of 12.7 mm (11.7 mm inner diameter) and length of 300 mm. Test section had evaporation region of 100 mm that heated by copper electrodes. The adiabatic region of 100 mm was insulated by the glass wool. Condenser section was 100 mm in length. The condenser section cooled working fluid maintaining a constant temperature. Six thermocouples were installed to measure the wall temperature along the test section. Two thermocouples were attached to the outer wall of evaporation region.

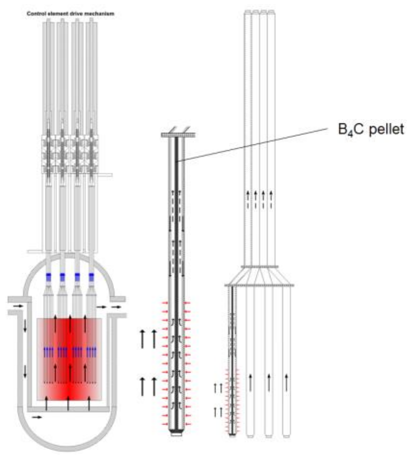


Fig 2. Schematic design of the hybrid heat pipe with control rod as PINCs

Other thermocouples were attached to the adiabatic region. Two thermocouples measured the outer wall of condensation region. Thermocouple locations and a schematic view of the experimental system are shown in Fig. 3.

Before filling working fluid, non-condensable gas was removed by vacuum pump. The fluid charge was rated based on the void volume in the wick structure. Distilled water was charged to evaporation section with 100% fill ratio. A pressure gage placed at the top of the condensation section was used to measure the initial saturation pressure and operation pressure of the steam in the heat pipe. Uncertainty of water level due to instrumental error was less than ± 5 %. The inlet temperature of coolant was maintained constant by chiller. The heat load range was 45W - 105W.

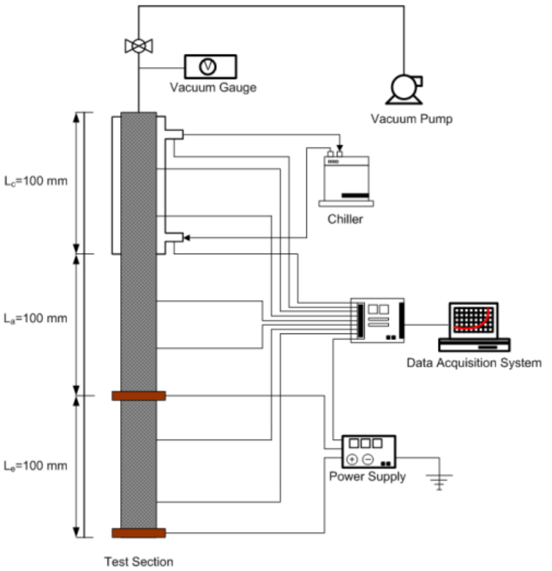


Fig 3. Schematic diagram for the experimental apparatus

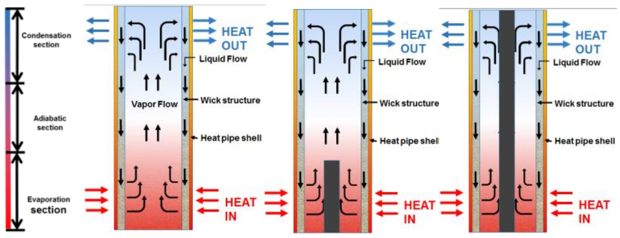
3. Results and Discussion

3.1 Test Section Geometry

The test section consists of the tube that was made of a stainless steel 316L and stainless steel 100 mesh wick inserted inside. The length, outer diameter and thickness of the heat pipe were 300 mm, 12.7 mm and 1.0 mm, respectively. Figure 4 shows the geometry of heat pipe test sections with flow direction. Figure 4 (a) shows Concentric Heat Pipe (CHP) that is conventional heat pipe having evaporator and condenser section. Both (b) Annular Evaporation section Heat Pipe (AEHP) and (c) Annular flow path Heat Pipe (AHP) included cylindrical structure that made by Acrylonitrile Butadiene Styrene (ABS) material using fused deposition modeling (FDM). The length and diameter of cylindrical structure in AEHP are 100 mm and 6.05 mm, respectively. The length and diameter of cylindrical structure in AHP are 300 mm and 6.05 mm, respectively. The conditions of heat pipes were summarized in Table I.

Table I: Initial Conditions

Filling ratio	100 % in wick
Wick size	100 mesh
Porosity	0.62
Initial pressure	8.0 kPa
Cooling temperature	5 °C

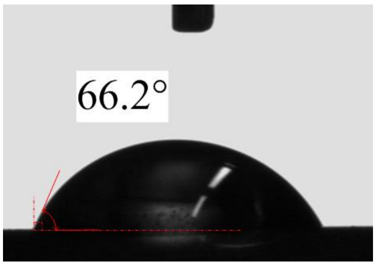


(a) (b) (c)

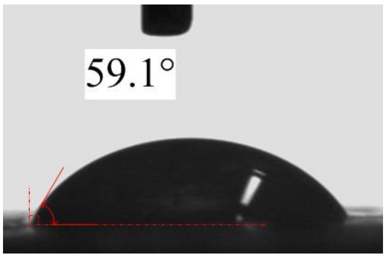
Fig 4. Test sections (a) Concentric heat pipe, (b) Annular evaporation section heat pipe, (c) Annular flow path heat pipe

3.2 Simulant material

ABS material takes advantages of easy to use and fabricate. ABS and B_4C have the similarity of the contract angle at the surface and specific heat. And both materials are insoluble in water. Structures in heat pipe have the rule of obstacle of vapor flow. It is the reasons that cylindrical structure manufactured by ABS material. Figure 5 shows the water-wetting surface with contact angle. B_4C and ABS pellet have the contact angle about 66.2° and 59.1°, respectively. The properties of two materials were summarized in Table II.



(a)



(b)

Fig 5. Contact angles (a) Boron cabide (B_4C) pellet and (b) ABS pellet

Table II: Pellet properties of B_4C and ABS pellet

	B4C pellet	ABS pellet
Density, g/cm^3	1.84	1.06 – 1.25
Thermal conductivity, W/mK	3.30 (irradiated)	0.6
Specific heat, J/kgK	950 - 1288	1100 - 1486
Solubility in water	Insoluble	
Contact angle, °C	66.2	59.1

3.3 Temperature Distribution

The heat transport performance of the CHP with a 100 % filling ratio is shown in figure 6. The heat input power at evaporation section is changed from 45 to 105 by step 20 W. The temperatures of evaporator and condenser have stable state each heat loads. When heat power of 110 W is loaded, the oscillations are started and the peak temperature was observed. The point of peak temperature was expected capillary limit of heat pipe. Adiabatic temperatures are slightly increasing due to internal pressure change of heat pipe.

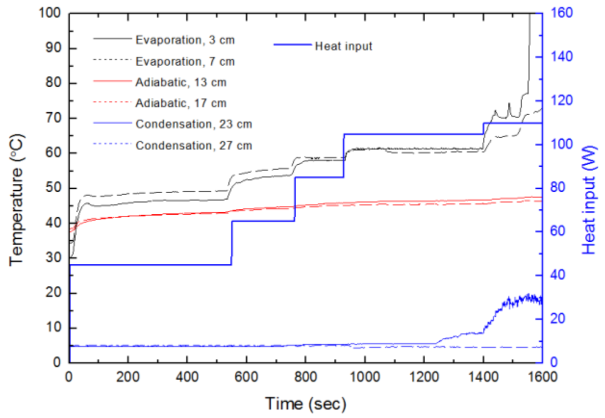
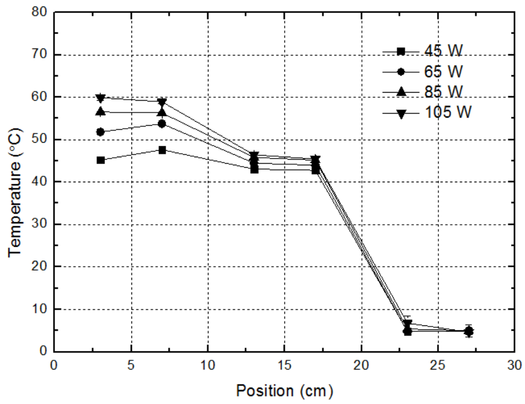
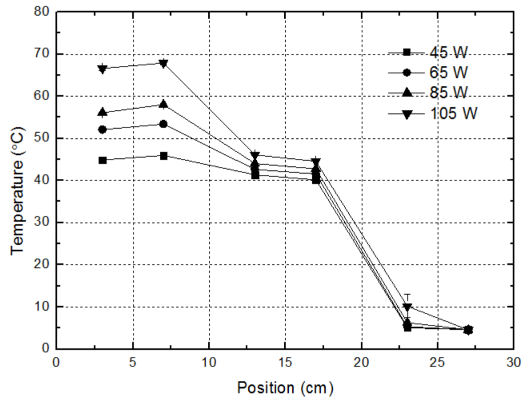


Fig 6. Temperature response to heat load of concentric heat pipe

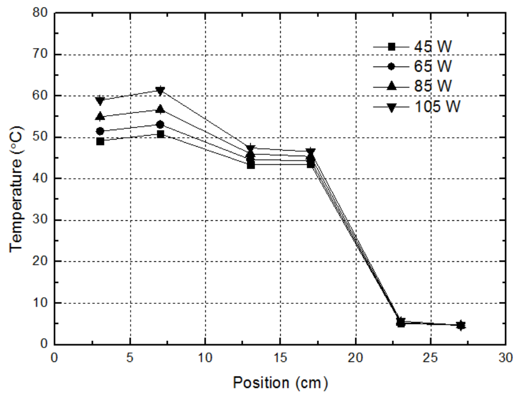
Shown in figure 7, the temperature difference between evaporation and condensation section of each heat pipes was increased by inserting the cylindrical structure at the same cooling condition. CHP had the smallest temperature difference and AEHP had the largest temperature difference, because AEHP has the narrow vapor space at evaporation section. AHP also had annular vapor space , but cooling surface per unit volume of vapor was increased. Temperature differences of AHP ware small, compared with that of AEHP.



(a) CHP



(b) AEHP



(c) AHP

Fig 7. Temperature distribution of tested heat pipes according to position

3.4 Thermal Resistances and Heat Transfer Coefficients

Figure 8 and 9 show the overall thermal resistance and heat transfer coefficient of heat pipes according to the heat load. It presents the heat transfer performance of heat pipes when condensation section has sufficient cooling performance.

Thermal resistance and heat transfer coefficient can be presented by

$$R_{overall}\left(°C/W\right)=\frac{\overline{T}_e-\overline{T}_c}{Q_e} \qquad (1)$$

$$h_{overall}\left(W/m^2{}^\circ C\right)=\frac{q_e''}{\overline{T}_e-\overline{T}_c} \quad (2)$$

where, $\overline{T_e}$ is the average temperature of evaporation section (℃), $\overline{T_c}$ is the average temperature of condensation section, Q_e is the evaporator heat load and q_e'' is the heat flux at the evaporation section. From the experimental results and equation 1 and 2, thermal resistance and heat transfer coefficient was calculated.

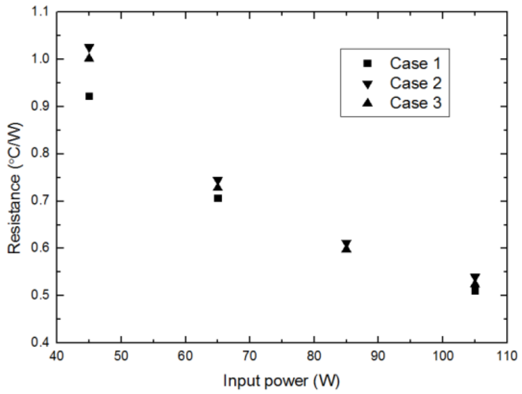


Fig 8. Overall heat resistance of heat pipes according to the heat load.

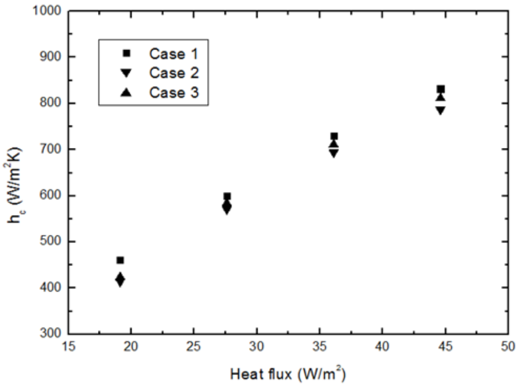


Fig. 9 Overall heat transfer coefficient of heat pipes according to the heat flux.

4. Summary and Future Works

The main objective is to investigate the effect of the inner structure in heat pipe to the heat transfer performance with annular flow path. ABS pellet was used instead of B_4C pellet as cylindrical structures. The thermal performances of each heat pipes were measured experimentally. Among them, concentric heat pipe showed the best performance compared with others.

1. Annular evaporation section heat pipe and annular flow path heat pipe showed heat transfer degradation.
2. AHP also had annular vapor space and contact cooling surface per unit volume of vapor was increased. Heat transfer coefficient of AEHP had relatively small compared with that of AHP according to the heat flux.

Next plan is a sensitivity study of the heat transfer of heat pipes with filling ratios and angle effects.

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