Study on the Capillary Limitation in Copper-Water Heat Pipes with Screen Wicks

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ABSTRACT: This paper is to study the heat transfer performance of the copper-water heat pipe with screen wicks. Recently, the semiconductor capacity of an electronic unit becomes larger, but its size becomes much smaller. As a result, a high- performance cooling system is needed. Experimental variables are inclination angles, temperatures of cooling waters and the mesh number of screen wicks. The distilled water was used as a working fluid. Based on the experimental results, when the copper-water heat pipe of 6 mm diameter is used at the top heat mode, the heat transfer performance of 100 mesh 2 layers heat pipe is better than that of 150 and 200 mesh. The thermal resistance of the two layers with the 100-mesh screen was 0.7−0.8℃/W.

Nomenclature

A_v	: vapor core cross-sectional area [m^2]
A_w	: wick cross-sectional area [m^2]
d	: screen wire diameter [m]
d_v	: vapor core diameter [m]
F_l	: liquid frictional coefficient [Pa/mW]
F_v	: vapor frictional coefficient [Pa/mW]
g	: gravitational acceleration [m/sec^2]
h	: heat transfer coefficient [W/m^2℃]
h_{fg}	: latent heat of vaporization [J/kg]
K	: permeability [m^2]
k_{eff}	: effective thermal conductivity [W/m℃]
k_l	: liquid thermal conductivity [W/m℃]
k_P	: pipe thermal conductivity [W/m℃]
k_w	: wick thermal conductivity [W/m℃]
L_{eff}	: effective pipe length [m]
L_t	: pipe length [m]
N	: screen mesh number
P_{cm}	: maximum capillary pressure [Pa]
P_{pm}	: maximum pump pressure [Pa]
$Q_{c,\max}$	: heat flow rate [W]
r_c	: capillary radius [m]
r_i	: inside radius of pipe [m]
r_o	: outside radius of pipe [m]
r_v	: vapor core radius [m]
Re_v	: Reynolds number
R_{HP}	: thermal resistance of heat pipe [℃/W]
R_p	: thermal resistance of pipe wall [℃/W]
R_{we}	: thermal resistance of pipe wick [℃/W]
S	: crimping factor
T_c	: temperature of condenser [℃]
T_e	: temperature of evaporator [℃]
W	: heating quantity [Watt]

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Greek symbols

θ : inclination angle
μ : viscosity [kg/m-sec]
ε : porosity
ρ_l : density of liquid [kg/m^3]
ρ_v : density of vapor [kg/m^3]

1. Introduction

The heat pipe, a sealed slender tube, transfers a large amount of heat by latent heat, in which the working fluid is vaporized under low temperature in vacuum, and operated with small temperature difference. Especially, since it is easy to attach fins in the evaporator and the condenser sections for the enlargement of heat transfer surface area, efficiency of cooling system and heat recovery technologies could be highly effective. The heat pipe is very important technology for cooling system of advanced electronic equipment and semiconductor. In case of notebook PC, the thermal dissipation power has been increased dramatically with the process speed of CPU increasing.

The process speed of CPU increased 450 MHz in 1999, 800 MHz in 2000, 1.0 GHz in 2001, 1.5~1.8 GHz in 2002, and 2.0 GHz in 2003, and the thermal dissipation power(1) increased 20 W, 27 W, 30 W, 60~70 W, and 75.3 W, respectively. If the development pace of CPU is remained constant, the growth of cooling system may not follow the speed. Therefore, it will cause that the speed of the development of CPU is slowed. Now several cooling methods are introduced for solving this problem, and the composition cooling system using heat pipe is remarked as a suitable cooling way. Heat pipe of 3~4 mm diameter installed at CPU of Mobile PC, and mini fan installed at condenser section cool the system. The thermal dissipation power is rejected by using two heat pipes which are a primary heat pipe from CPU to hinge and a secondary heat pipe from hinge to the back of LCD panel.(2)

Heat pipes is used in various applications such as composition heat pipe module cooling device of flat type, semiconductor cooling in subway vehicles and the outdoor PCS base station transceiver system cooling.(3) The return of working fluid is affected by gravity or capillary force of the wick in a heat pipe, so the capillary limitation is the most influential factor for maximum heat transfer among operation limits. It is influenced by pumping pressure of capillary structure, friction loss of length direction and inclination angle of heat pipe.

Gupta and Upadhya investigated heat transfer performances such as limit power and thermal resistance by screen mesh number and screen mesh layers, and studied heat performance among to inclination angle in 3, 4 and 5 mm diameter heat pipes.(4) Since the small size heat pipes, 3~6 mm, are usually installed horizontal mode than vertical mode, they have the capillary structures such as wick. In this case, the optimum working fluid was found by the previous experiment.

Tested model was the heat pipe that charged the optimum working fluid quantity determined by previous experiment.(5) In this study, 6 mm copper-water heat pipe with inserting screen mesh wick for cooling electronic equipment is fabricated, and compared heat transfer performance according to various cooling water temperatures, inclination angle, and screen wick layers by the previous experiment.

2. Experimental apparatus and method

The heat pipe for cooling electronic equipment usually has about 3~12.7 mm diameter, and it is applied to various parts such as cooling of the semiconductor and the amplifier in audio. In this study, we manufactured copper-water heat pipe of diameter 6 mm with screen mesh wick and tested it.

Table 1 Experimental conditions

Parameters	Specification
Container	
material	Copper
pipe diameter	6 mm
total length	300 mm
length of evaporator zone	50 mm
length of adiabatic zone	100 mm
length of condenser zone	150 mm
Working fluid	Distilled water
Inclination angle	0°, 3°, 6°
Cooling water temperature	20℃, 30℃, 40℃

Table 2 Working fluid quantity

Mesh number	Layer	Working fluid quantity (cc)
100	1	1.23
	2	1.91
	3	2.51
150	1	0.94
	2	1.37
	3	1.77
200	1	0.91
	2	1.38
	3	1.79

The heat pipe is made of copper with an effective length (L_t) of 300 mm, an evaporator section (L_e) of 50 mm, an adiabatic section (L_a) of 100 mm, and a condenser section (L_c) of 150 mm. The ratio of condenser length to evaporator length is 3 to 1.

Distilled water is used as the working fluid, and its amount is determined by the previous experiment. 140%, 120%, and 110% of working fluid are injected into 1-layer, 2-layer, and 3-layer. Here, the porosity is expressed wick's pore volume over wick's whole volume. The heat pipe performances are influenced by the porosity which is affected the permeability of a heat pipe.

Table 1 shows the specification of the heat pipe and experimental conditions used in this study. Table 2 shows the quantity of the working fluid charged to the heat pipe.

Figure 1 is schematic of experimental apparatus for the performances of heat pipe element. Experimental apparatus consists of a heat pipe and a inclination angle adjuster, data acquisition unit, a power supply and constant temperature bath.

Electricity is supplied to nichrome wire (4.5 Ω/m), which is wrapped around evaporator section, by the variable voltage adjuster. It is insulated with the cerak wool (k=0.075 W/m℃) and then with urethane foam in order to min-

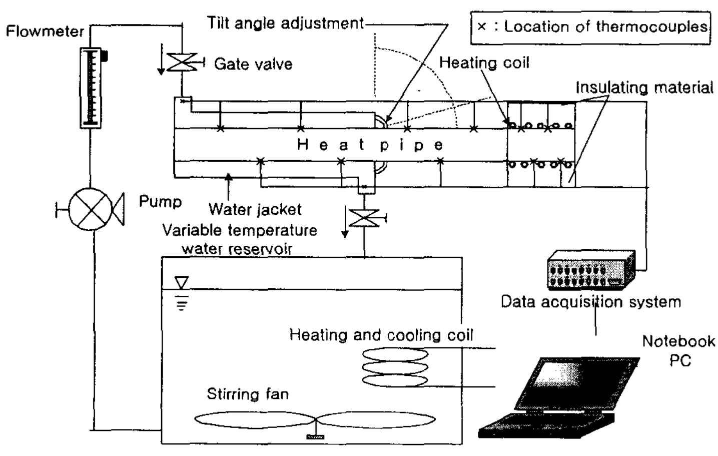

Fig. 1 Experimental apparatus.

imize heat loss. The temperature of vicinity nichrome wire and the outside temperature are measured to calculate heat loss. Once the adiabatic section is wrapped by insulation tape, and it is connected to the inclination angle adjuster made of insulation material of teflon. Water in constant temperature bath (heating capacity: 750 W, cooling capacity: 250 W) is supplied to the condenser section for sufficient cooling. For cooling, the condenser section is covered with a water jacket, which is a pyrex tube of 150 mm length and 15 mm diameter, and the tube is insulated with 10 mm thick urethane. Heating rate, surface temperatures at each section, saturation temperature and inside pressure are measured to investigate the characteristic. In order to measure them, thermocouples are sticked to each section. In evaporation section eight thermocouples are arranged at a 10 mm interval. Four thermocouples are also arranged at a 20 mm interval in adiabatic section, and six thermocouples in condenser section are arranged at a 30 mm interval. Data from these thermocouples are collected in PC through the data acquisition system (DBK board 42 channel, IO Tech.). Moreover, heat transfer rate at the condenser section is estimated by the temperature difference between the inlet and outlet of cooling water. Experiment is performed at the top heat mode and inclination angles of 0°, 3° and 6° at horizon on 6 mm copper-water heat pipe.

The heat pipe for electronic equipment cooling system is usually installed at inclination angle of horizon or approximately within 10°. Therefore experiment was performed to investigate on effect of inclination angle at interval 3° from 0° to 6°, and the input heat load and thermal resistance.

The experimental model is the heat pipe injected 140%, 120% and 110% with 1, 2 and 3 layers screen mesh wick. Then, the input power is gradually increased from 10 W to the estimated capillary limitation power by 10 W. Experiment is stopped at about 110℃ on the evaporator section. Although temperature does not rise steeply, it is also stopped at dry-out condition regardless of temperature.

3. Capillary limitation theory

Figure 2 shows the recirculation of working fluid and heat flow in a heat pipe.[6] In this figure L_c, L_a and L_e are length of condenser section, adiabatic section, and evaporator section, respectively. r_v, r_i and r_o are radius of vapor region, inside radius of pipe, and outside radius of pipe, respectively.

In general, if a heat pipe is operating in a gravitational field, maximum effective pumping pressure, maximum effective capillary pressure

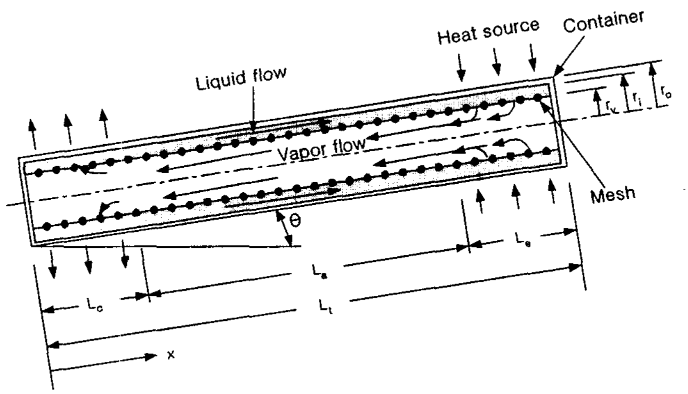

Fig. 2 Schematic diagram of a heat pipe.

and capillary limitation on the heat transport rate can be expressed as:

$$P_{pm} = P_{cm} - \rho_l g d_v \cos\theta - \rho_l g L t \sin\theta \quad (1)$$

$$P_{cm} = \frac{2\sigma}{r_c} \quad (2)$$

$$Q_{c,\max} = \frac{P_{pm}}{(F_l + F_v) L_{eff}} \quad (3)$$

$$F_l = \frac{\mu_l}{K A_w \rho_l h_{fg}} \quad (4)$$

$$L_{eff} = (0.5 L_c + L_a + 0.5 L_e) \quad (5)$$

$$K = \frac{d^2 \varepsilon^3}{122(1-\varepsilon)^2} \quad (6)$$

$$\varepsilon = 1 - \frac{\pi S N d}{4} \quad (7)$$

$$F_v = \frac{(f_v \mathrm{Re}_v)\mu_v}{2 r_{h,v}^2 A_v \rho_v h_{fg}} \quad (8)$$

where σ is the surface tension at the liquid vapor interface, r_c is the capillary radius, d is the screen wire diameter, ε is the wick porosity, K is the wick permeability, L_{eff} is the effective heat pipe length, ρ_l is the liquid density, abd F_l, F_v are the friction coefficient for liquid flow and vapor flow, respectively.

Thermal resistance, R_{HP}, is calculated dividing average temperature difference in the evaporator and condenser section by Q_{in}.

$$R_{HP} = \frac{T_e - T_c}{Q_{in}} \quad (9)$$

where Q_{in} is the heat load input power (W) minus heat loss.

The heat loss was calculated approximately 5~7% by measuring the temperature of insulation outside surface.

The heat transfer coefficient h can be expressed by Eq. (10).

$$h = \frac{1}{R_{pe} + R_{we} + R_v + R_{wc} + R_{pc}} \quad (10)$$

$$R_{pe} = \frac{r_o / t_p}{2 k_p L_e} \quad (11)$$

$$R_{we} = \frac{r_o^2 / t_w}{2 k_{eff} r_i L_e} \quad (12)$$

$$k_{eff} = \frac{k_l[(k_l + k_w) - (1-\varepsilon)(k_l - k_w)]}{[(k_l + k_w) + (1-\varepsilon)(k_l - k_w)]} \quad (13)$$

where R_{pe} is the thermal resistance for the heat pipe wall at evaporator, R_{we} is the thermal resistance of the wick at evaporator, k_l and k_w are thermal conductivity of the liquid and the wick, r_o, r_i and r_v are outside radius of heat pipe, inside radius of heat pipe and vapor core radius, respectively. L_e is the length of evaporation section, and k_{eff} is the effective thermal conductivity of the liquid-wick material.

4. Results and discussions

4.1 Analytical results for the effect of mesh

Figure 3 shows the effect of mesh number on the capillary limitation of a heat pipe by using Eqs. (1)~(8). For the vapor saturation temperature, 40℃ and inclination angle, 6°, the

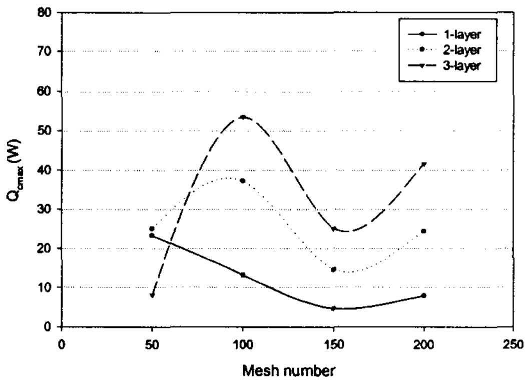

Fig. 3 Capillary limitation of heat pipe with mesh numbers (6 degree).

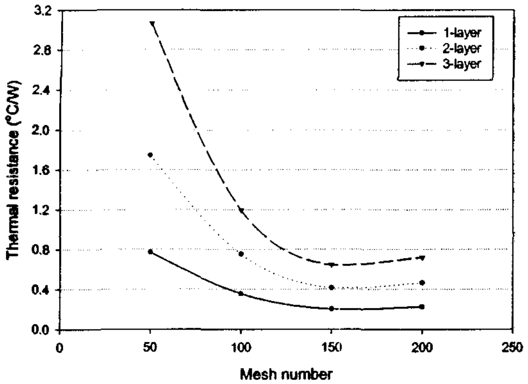

Fig. 4 Thermal resistance of heat pipe with mesh numbers (6 degree).

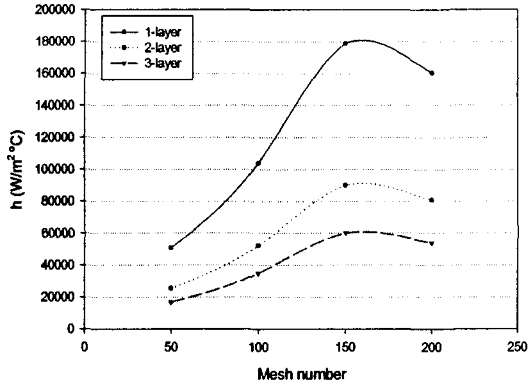

Fig. 5 Heat transfer coefficient of heat pipe with mesh numbers (6 degree).

capillary limitation of 50 mesh wick heat pipe was approximately 8~25 W at 1, 2 and 3 layers. For the 100 mesh wick, capillary limitation was calculated 13 W, 37 W and 53 W at 1, 2 and 3 layers, respectively. For the 150 mesh wick, capillary limitation was calculated 4.6 W, 14.5 W and 25.1 W at 1, 2 and 3 layers, respectively, and for the 200 mesh wick, capillary limitation was calculated 8.0 W, 24.3 W and 41.5 W at 1, 2 and 3 layers, respectively. Therefore, the 100 mesh wick has the better heat transfer performances than the others.

The thermal resistance of heat pipe according to mesh number is shown in Fig. 4. For the vapor saturation temperature of 40℃ and inclination angle of 6°, the thermal resistance of 50 mesh wick heat pipe was 0.77℃/W, 1.74 ℃/W and 3.06℃/W at 1, 2 and 3 layers, respectively. For the 100 mesh wick, the thermal resistances of 1, 2 and 3 layers was calculated 0.35℃/W, 0.75℃/W and 1.19℃/W, respectively. The resistance of 100 mesh was a little higher than that of 150 and 200 mesh numbers.

Figure 5 shows the heat transfer coefficient of heat pipe according to mesh number. For the vapor saturation temperature of 40℃ and of inclination angle of 6°, the heat transfer coefficient of 100 mesh heat pipe was 104 kW/m^2℃, 52 kW/m^2℃ and 35 kW/m^2℃ at 1, 2 and 3 layers, respectively. Therefore, the heat transfer coefficient of 100 mesh was lower than that of 150 and 200 mesh sizes.

4.2 Experimental results for thermal resi -stance and limiting power

For the 200 mesh wick, 2-layer, the cooling water temperature of 40℃ and inclination angle of 3°, the temperature distribution on the outside wall surface of heat pipe for input heat load is shown in Fig. 6. The temperature at the end of evaporator section was lower than that of middle section at input heat load of 28.8 W, but the temperature at the end of evaporator section was higher than that of middle section at input heat load of 38.2 W.

When the temperature at the end of evapo-

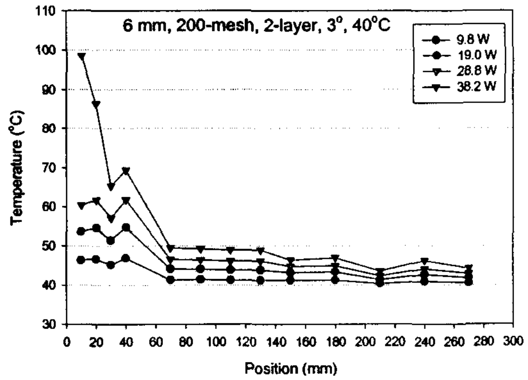

Fig. 6 Wall temperature distribution (3 degree).

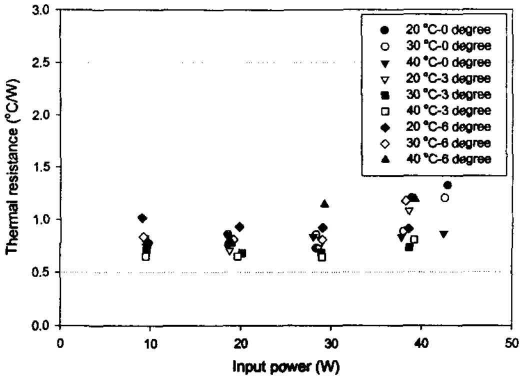

Fig. 7 Variation of thermal resistance at 200 mesh, 2 layers.

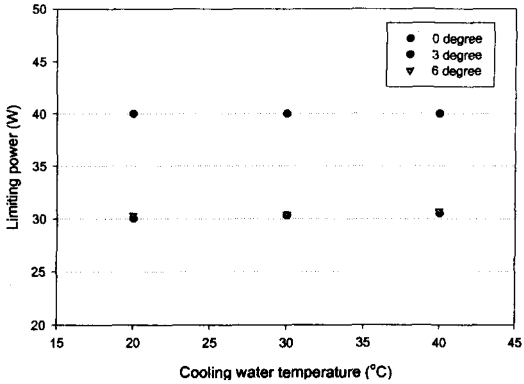

Fig. 9 Variation of limiting power.

rator section is approximately 100℃ input heat load of 38.2 W, dry-out state is appeared at this input heat load. So, it is seen that the amount of working fluid is suitable at input heat load of 30 W.

Figures 7 and 8 show thermal resistance at 2 and 3 layers of 200 mesh heat pipe according to inclination angle, cooling water temperature. As can be seen in figures, the thermal resistance at the 2 layers heat pipe was the lower than that of 3 layers. Comparing the difference of thermal resistance according to input heat load, case of 2 layers heat pipe has smaller value than case of 3 layers.

In the 200 mesh screen wick heat pipe, 2 or 3 layers, the screen wick becomes thicker according to increasing of the layer of wick to 2 or 3 layers. This wick thickness also influenced the working fluid quantity, the effective thermal conductivity and the diameter of vapor core to affect thermal resistance. The diameter of vapor core influenced the vapor and liquid frictional coefficient. Therefore, heat transfer limiting power was affected vapor and liquid frictional coefficient. Figure 9 shows the limiting power at 200 mesh heat pipe with 2 layers according to inclination angle and cooling water temperature. As can be seen in Fig. 9, the difference of limiting power was quite small according to cooling water temperature. Therefore, it is seen that the cooling water temperature have a little influences on the capillary

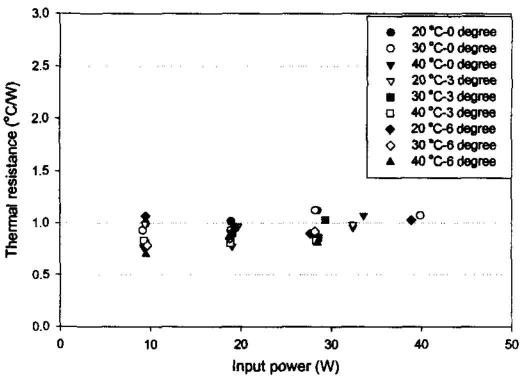

Fig. 8 Variation of thermal resistance at 200 mesh, 3 layers.

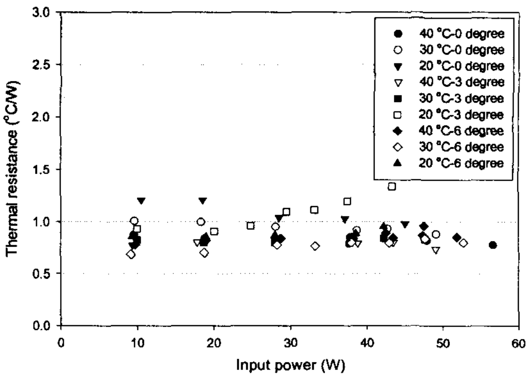

Fig. 10 Variation of thermal resistance at 100 mesh, 2 layers.

limitation power.

Comparing the inclination angle 0° with 3° and 6°, the limiting power decreased approximately 30% due to the effect of maximum effective capillary pressure given Eq. (1). Comparing the tilt angle 3° with 6°, the difference of limiting power was quite small. So, it can also be noticed that there is no difference of the effective capillary pressure between inclination angle 3° and 6°.

Figure 10 shows the thermal resistance of 100 mesh screen wick heat pipe with 2 layers. Comparing the case of Fig. 10 with the case of 2 layers of 200 mesh in Fig. 7, there is no difference of thermal resistance, but the limiting power of 100 mesh was a little higher than the case of 200 mesh. For the screen wick 2 layers, inclination angle of 6° and cooling water temperature of 40℃, thermal resistances according to input heat load and mesh number are shown in Fig. 11. For the 100 mesh heat pipe when the input heat load were 9.8 W, 38.2 W and 51.9 W, thermal resistances were 0.77℃/W, 0.83 ℃/W and 0.85℃/W, respectively.

For the 150 mesh wick, thermal resistance was 1.07℃/W at the input heat load of 36.6 W, and for the 200 mesh, thermal resistance was 1.19℃/W at the input heat load of 39.2 W. So, thermal resistance and limiting power of heat pipe with 100 mesh were better than these of 150 and 200 mesh.

For the screen wick 3 layers, inclination angle of 6° and cooling water temperature of 40℃, Fig. 12 shows thermal resistances according to input heat load and mesh number.

For the 200 mesh wick, thermal resistances were 0.7℃/W at 9.5 W, 0.81℃/W at the input heat load of 28.5 W. For the 150 mesh wick, thermal resistance was 1.16℃/W at the input heat load of 9.7 W, 0.93℃/W at 42.6 W. For the 100 mesh wick, thermal resistance was 1.0℃/W at the input heat load of 9.2 W and 1.03℃/W at the input heat load of 47.6 W. So, thermal resistance showed high in the order of 200 mesh, 150 mesh, 100 mesh. 200 mesh was topped the list with the limiting power of 28.5 W, followed by 150 mesh of 42.6 W, 100 mesh placed third with the limiting power of 47.6 W. For the 100 mesh wick, 2 layers heat pipe was shown lower thermal resistance and higher limiting power than 3 layers heat pipe.

For the 100 mesh wick, the experimental limiting power was 1.5 times higher than the analytical value. For the 150 and 200 mesh wick, the experimental value was 2.5 times higher than the analytical one. There is a wick thickness difference between analytical model and experimental model. Since the wick of experimental model was thickened during the heat pipe manufacturing, this result is supposed to influence

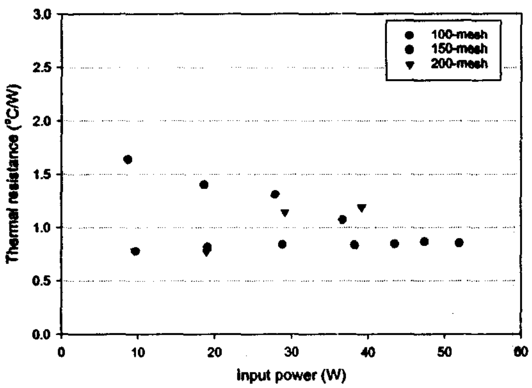

Fig. 11 Variation of thermal resistance at 2 layers, 6 degree.

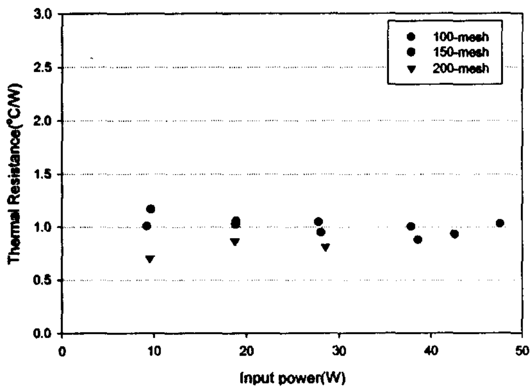

Fig. 12 Variation of thermal resistance at 3 layers, 6 degree.

to the limiting power. As the result of investigation, the 100 mesh wick heat pipe is supposed to be best for electronic equipment cooling system.

5. Conclusions

In this study, 6 mm copper-water heat pipe with the mesh screen wick for small electronic device cooling is fabricated. Heat transfer performance is investigated analytically and experimentally according to layers of screen wick, inclination angles of heat pipe, saturated vapor temperature, and cooling water temperature. The followings are results.

(1) Based on the simulation results, when copper-water heat pipe of 6 mm diameter is used at the top heat mode, the 100 mesh wick heat pipe with 3 layers screen wick has the better limiting power than the others.

(2) Based on the experimental results, when copper-water heat pipe of 6 mm diameter is used at the top heat mode, the heat transfer performance of 100 mesh, 2 layer heat pipe is better than that of 150 and 200 mesh.

(3) In case of the 200 mesh, 2 layer heat pipe, there is no difference of the limiting power between inclination angle 3° and 6° according to cooling water temperature and inclination angle. Comparing the case of the inclination angle, 6° with the case of 0°, the limiting power of inclination angle, 6° is reduced about 30% than that of inclination, 0°.

(4) Thermal resistance is 0.7~0.8℃/W at the two layers of the 100 mesh screen wick.

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