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ANALYSIS OF PULSATING HEAT PIPE WITH A DUAL-DIAMETER TUBE

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ABSTRACT

A series of experiments was performed to investigate the effect of a dual-diameter channel on the flow and heat transfer characteristics of single-turn pulsating heat pipes (PHPs). Various types of PHPs were made of glass capillary tubes with various inner diameters, and experiments to evaluate thermal performance of the PHPs were performed with varying input power and inclination angle. Quantitative data obtained by high-speed photography that asymmetric PHPs with a dual-diameter channel promote circulating flow over a wider range of experimental conditions compared to symmetric PHPs with a uniform diameter channel. Circulating flow promoted by a dual-diameter channel helps to enhance the thermal performance of the PHP and reduces thermal resistance by up to 45%. A simplified model was developed to predict thermal characteristics of asymmetric PHPs with circulating flow, and the predicted data matched well with experimental data to within the error of 15%. Experimental and calculated data show that there exists an optimum range of diameter deviation where the thermal performance enhancement is maximized. The optimal range of dimensionless diameter deviation is found to be between 0.25 and 0.4, and this study provides design guidelines to improve thermal performance of the PHP.

NOMENCLATURE

A_c	[m^2]	Cross sectional area
A_s	[m^2]	Surface area
Bo	[-]	Boiling number
Co	[-]	Convection number
D	[m]	Diameter
F_{cap}	[N]	Thermocapillary force
F_f	[N]	Frictional force
F_g	[N]	Gravitational force
F_{sat}	[N]	Force due to saturation pressure difference
f_{TP}	[-]	Friction factor
G	[Kg/sm^2]	Mass flux
g	[m/s^2]	Gravitational acceleration
H	[J/kgK]	Enthalpy
h	[W/m^{2}K]	Heat transfer coefficient
H_{fg}	[J/kg]	Latent heat
h_{lo}	[W/m^{2}K]	Single-phase heat transfer coefficient
I	[A]	Electric current
k	[W/mK]	Thermal conductivity
L	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
P	[N/m^2]	Pressure
Pr	[-]	Prandtl number
q''	[W/m^2]	Heat flux
R_{th}	[K/W]	Thermal resistance
T_f	[°C]	Temperature of working fluid
x	[-]	Vapor quality
z	[m]	distance

Special characters

ρ	[kg/m^3]	Density
θ	[°]	Inclination angle
υ	[m^3/kg]	Specific volume
μ	[Pa· s]	Viscosity

Subscripts

c	Condenser
e	Evaporator
l	Liquid phase
TP	Two-phase
v	Vapor phase

INTRODUCTION

A heat pipe, a type of passive thermal control device, is one of the most efficient and attractive technologies for high heat flux electronic cooling. As the semiconductor industry continues to make rapid advances in processor speed, the power consumption and heat flux of electronic equipment are increasing [1]. As a consequence, the need for efficient cooling systems has increased [2], and the improvement of heat transport technology is a key factor that would help improve cooling efficiency and ensure the reliable system operation of electronics [3]. Thus, evolution in the design of the heat pipe has accelerated in the past decade, and new types of heat pipes, such as capillary pumped loops (CPLs) and loop heat pipes (LHPs), were introduced to overcome the inherent limitations of conventional heat pipes [3].

The pulsating heat pipe (PHP), invented by Akachi in 1990 [4], has shown great promise to manage and transfer heat for microelectronic system cooling. A typical PHP is a small and long meandering tube that is filled with liquid slugs and vapor plugs of a working fluid, and it allows PHPs to be potentially cheaper and more suitable for application in thin and small-sized devices. Operating principle of the PHP is described in the following paragraph.

When heat is applied to the evaporator section, the pressure of vapor bubbles in the evaporator section rapidly increases due to boiling and evaporation. Although the temperature of evaporator section continues to rise, the condenser section is maintained at a low temperature. Therefore, pressure difference between bubbles caused by temperature variation makes the working fluid start to move. When bubbles from the evaporator reach the condenser section, heat is released from the working fluid with bubble shrinkage due to condensation. On the contrary, bubbles which move from the condenser section to the evaporator section absorb heat, and the pressure of the bubbles increases because of rising temperature and evaporation. Thus, the working fluid continues to move, and the applied heat is transported from the evaporator section to the condenser section through the working fluid movement.

Although their structure is simple, PHPs are complex heat transfer devices whose thermal performance is governed by a strong thermo-hydrodynamic coupling [4]. A number of studies have been performed so far to understand the thermo-hydrodynamics of PHPs [4]. It is generally agreed by researchers that heat transfer performance is enhanced when oscillating frequency and amplitude increases, and a PHP with circulating flow instead of oscillating flow has better thermal performance [5, 6]. The reason is that a circulating flow has higher heat transport capability from the evaporator to the condenser than an oscillating flow. Tong et al.[7] found that circulating flow will not result in permanent dry-out at the evaporator, unlike an oscillating flow. Therefore, the development of a PHP which has a circulating flow under a wide range of working conditions would help to improve the thermal performance of PHPs. Some researchers [5, 6, 8, 9] have proposed special configurations of PHPs to promote circulating flow in PHPs. These previous studies found that an asymmetric configuration helps to generate a circulating flow and enhance heat transfer performance. However, they could not explain the reason why the heat transfer of asymmetric PHPs is augmented, and no work has established a proper design guide to effectively generate a circulating flow in a PHP. Therefore, it is necessary to confirm the heat transfer enhancement mechanism of a circulating flow. This study conducted experiments using an asymmetric single-turn PHP with dual-diameter tubes.

The purpose of this study was to develop an asymmetric PHP which has improved thermal performance using the circulating flow of the working fluid. Experiments were carried out to identify the heat transfer and flow characteristics of the PHP. The effects of a dual-diameter channel and the inclination angle on the operating characteristics of PHPs were investigated using several types of PHPs. The flow behaviors of the working fluid in the PHP were observed using a high-speed camera. The flow behaviors and thermal performance of asymmetric PHPs were compared with those of symmetric PHPs made of uniform diameter tube. A simplified model was developed to predict the thermal characteristics of a PHP with a circulating flow, and it provides guidelines for the design of asymmetric PHPs with improved thermal performance.

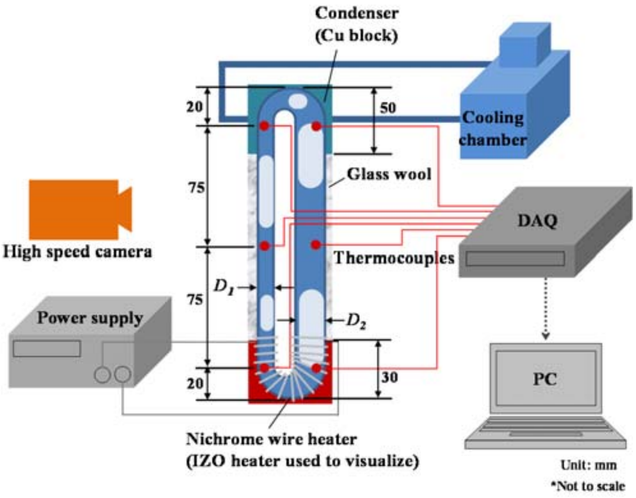
EXPERIMENTAL SETUP

Figure 1 shows a schematic diagram of experimental setup. Experimental apparatus to evaluate thermal performance of glass capillary PHPs is consisted of a PHP, a chrome wire heater, a DC power supply, a condenser made of copper block, a bath circulator (cooling chamber), thermocouples, a data acquisition system (DAQ), and a computer. To verify heat transfer performance of circulating flow, PHPs with alternately varying channel diameter are made of Pyrex glass capillary tubes of three different inner diameters, 1.2, 1.7, and 2.2 mm. It is known that variation in inner diameter is helpful to initiate and sustain reliable circulation of the working fluid [5, 8]. To probe it, six types of PHPs were designed using glass capillaries as listed in Table 1; three of them are symmetric PHPs with uniform inner diameter, and the other three, asymmetric PHPs, are designed to have two adjacent tubes of different inner diameters with length of 190 mm.

Table 1 Experimental parameters

PHP No.	$D_{i,1}$ (mm)	$D_{i,2}$ (mm)	D_o (mm)	L_e (mm)	L_c (mm)	L_a (mm)	Filling ratio (%)	Working fluid	Input heat (W)	Inclination angle (°)
1	1.2	1.2	6	30	50	110	50	Absolute ethanol	~ 25	0 ~ 90
2	1.7	1.7								
3	2.2	2.2								
4	1.2	1.7								
5	1.2	2.2								
6	1.7	2.2								

The heater made of chrome wire of diameter 0.75 mm is used to make uniform heat flux condition in the evaporator section and connected to DC power supply (UP-750). When the DC electric current is applied to the heater from the power supply, an almost uniform wall heat flux boundary condition is ohmically established. The condenser section of the PHP is cooled by the water jacket made of a copper block with high thermal conductivity. The water in the condenser is supplied from the bath circulator (RW-0525G) and the inlet temperature of the water is maintained constant. Temperatures of the PHP are measured using K-type thermocouples on the outer wall of the PHP at six points: two points at the evaporator, two points at the adiabatic section, and two points at the condenser. The lengths of each section, evaporator, adiabatic section, and condenser, are 45 mm, 95 mm, and 50 mm, respectively. Temperature data are obtained by a data acquisition system (34970A) and saved on a computer. Ethanol is used as the working fluid. The filling ratio (FR, defined as volume fraction occupied by the liquid in the channel at room temperature) is fixed at 50%. As shown in Fig. 1(b), experiments to evaluate thermal performance of glass capillary PHPs are performed with varying inclination angle.



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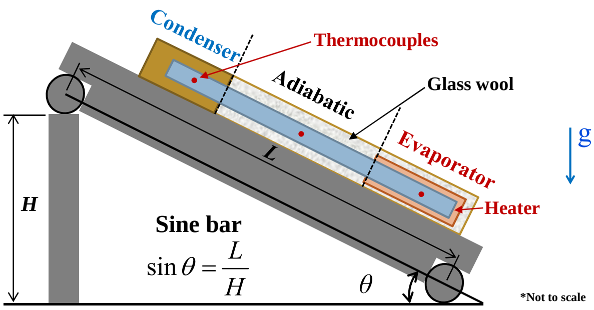
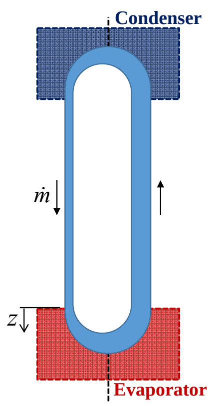


(b)

Figure 1 Experimental apparatus: (a) schematic of experimental setup and (b) sine bar to control inclination angle.

MODEL DESCRIPTION

The model developed in this study was used to predict the flow and thermal characteristics of a single-turn PHP when it has circulating flow. Because the flow and thermal characteristics of the circulating flow is very similar to forced convective flow with boiling/condensation, the PHP could be assumed as straight tube, as shown in the Fig. 2.



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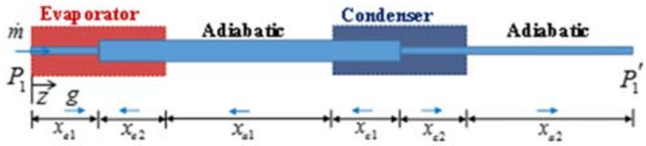


(b)

Figure 2 Schematic illustration of the PHP: (a) actual PHP diagram and (b) simplified PHP diagram

The simplified model was developed based on a previous study by Khandekar *et al*. [10] and modified. The momentum equation for two-phase flow in a duct is expressed as [12]:

$$\left(-\frac{dP}{dz}\right) = \frac{2 f_{TP} G^2}{D}\left[x v_v + (1-x) v_l\right] + G^2 (v_v - v_l)\frac{dx}{dz} + \rho_{TP}\, g \sin\theta \qquad (1)$$

Each term of Eq. (1) on the right hand side represents the pressure gradient due to friction, convective acceleration, and gravity. Because vapor mass quality is related to enthalpy, based on homogeneous flow model, dx/dz of the second term on the right side can be written as:

$$\frac{dx}{dz} = \frac{1}{H_{fg}}\frac{dH}{dz} \qquad (2)$$

Enthalpy changes along the channel in the evaporator and condenser sections because the working fluid adsorbs or releases heat. Therefore, in various sub-sections of the PHP, the enthalpy gradient is assumed as:

$$\frac{dH}{dz} = \begin{cases} \dfrac{4q''}{GD} & \text{in the evaporator and condenser sections} \\ 0 & \text{in adiabatic section} \end{cases} \qquad (3)$$

q'' is the heat flux which contributes to the evaporation and condensation of the working fluid, and it has a positive value in the evaporator section and a negative value in the condenser section. By substituting Eq. (3) to Eq. (2), the vapor mass quality gradient can be shown as

$$\frac{dx}{dz} = \frac{4q''_{latent}}{GDH_{fg}} \qquad (4)$$

where q''_{latent} is the heat flux which causes phase change heat transfer. As mentioned above, the difference between the vapor mass qualities in the left and right tubes results in unbalanced gravitational force and can be obtained by integrating Eq. (4) as follows:

$$x_{right} = x_{left} + \frac{4Q_{latent}}{G\pi D^2 H_{fg}} \qquad (5)$$

where $Q_{latent} = \pi D L_e q''_{latent}$. Based on the experimental data, the latent heat was assumed to be 80% of total input power ($= V \times I$) in the proposed model. The thermal resistance of the PHP could be characterized using the following equation:

$$R_{th} = \frac{1}{h_e A_{s,e}} + \frac{1}{h_c A_{s,c}} \qquad (6)$$

To estimate thermal resistance of a PHP, the heat transfer coefficients in the evaporator and condenser sections have to be accurately evaluated. The heat transfer coefficient in the evaporator section is calculated using the correlation suggested by Kandlikar as follows [12]:

$$h_e = h_{lo}[0.6683Co^{-0.2} + 1058Bo^{0.7}] \quad (7)$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_v}{\rho_l}\right)^{0.5} \quad (8)$$

$$Bo = \left(\frac{q''}{GH_{fg}}\right) \quad (9)$$

Mass flux is nothing but the mass flow rate per unit area. Because fluid flow is in the laminar flow regime, the correlation developed by Shah [13] for the thermally developing laminar flow is applied to evaluate the heat transfer coefficient via liquid only flow given by

$$h_{lo} = \begin{cases} \left(\frac{k_l}{D}\right)\cdot 1.953\cdot\left(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L}\right)^{1/3} & \text{when } \mathrm{Re}\,\mathrm{Pr}\frac{D}{L} \geq 33.3 \\ \left(\frac{k_l}{D}\right)\left(4.364 + 0.0722\,\mathrm{Re}\,\mathrm{Pr}\frac{D}{L}\right) & \mathrm{Re}\,\mathrm{Pr}\frac{D}{L} < 33.3 \end{cases} \quad (10)$$

In the condenser section, the heat transfer coefficient by condensation was estimated using the correlation suggested by Collier et al. [14] as follows:

$$h_c = \frac{k_l}{D} C\,\mathrm{Re}_{TP}^{n}\,\mathrm{Pr}_l^{1/3} \quad (11)$$

$$\mathrm{Re}_{TP} = \frac{G\left[(1-x) + x(\rho_l/\rho_v)^{1/2}\right]D}{\mu_l} \quad (12)$$

$$\text{where } C = \begin{cases} 0.0265,\ n = 0.8 & \text{for } \mathrm{Re}_{TP} > 50{,}000, \\ 5.03,\ n = 1/3 & \text{for } \mathrm{Re}_{TP} < 50{,}000. \end{cases} \quad (13)$$

Heat transfer coefficients in the evaporator and condenser sections are functions of many parameters related to flow characteristics including fluid thermophysical properties, mass flow rate, and vapor mass quality. Therefore, they have to be simultaneously solved with the momentum equation. In this study, an iterative method was used, and the temperature dependent properties of working fluid were updated after each iteration until convergence. The numerical procedure of the proposed model is outlined, as shown in Fig. 3.

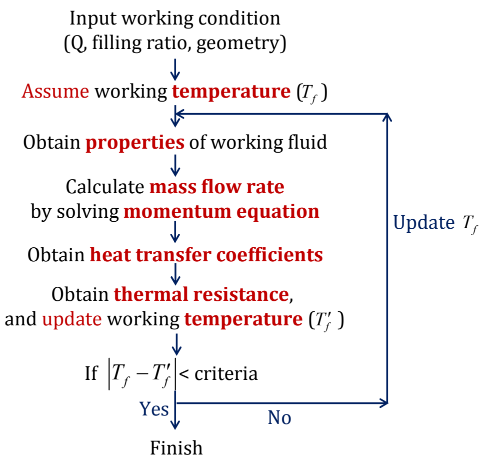


Figure 3 Numerical procedure

The predicted thermal resistances were validated using the experimentally-determined thermal resistances of six types of PHPs in the range of $10 \leq Q_{in} < 20$ W (Fig. 4). The predicted data matched well with the experimental data within $\pm 15\%$.

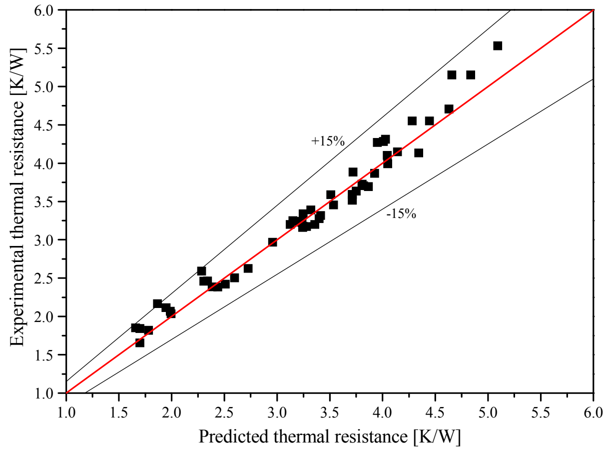


Figure 4 Comparison of predicted thermal resistance with the experimental data.

RESULTS AND DISSCUSSION

Flow behaviours

To investigate effect of channel configuration on flow patterns, flow patterns of PHPs are observed experimentally as heating power (Q) increases. According to previous studies, variation in channel inner diameter helps to generate circulation of the working fluid in the PHP. The temperature of the copper block condenser is maintained at 25°C, and the heating power supplied to the heater ranges from 10 to 45 W. The observed flow patterns are recorded and summarized in Table 2. S, C, O, and AC indicate stationary, circulating, oscillating, and arbitrary circulating flows, respectively. Flow patterns mostly observed are presented in capital letters. Input power Q in Table 2 indicates the power provided by power supply. Because the experiments explained in this section are performed to observe flow patterns in the PHP, thermal insulator to prevent heat loss to environment is not used. As a result, actual heat input supplied into the PHPs is smaller than power provided by power supply. The results show that the working fluid starts to oscillate above a certain value of heating power, and the flow pattern changes from oscillating flow to circulating flow as input power increases. The transition value of input power to circulatory flow depends on the type of PHP and is lower in case of asymmetric PHSs. Especially, two PHPs, PHPs of inner diameters 1.2-1.7 mm and 1.7-2.2 mm, have circulation flow under wide range of input power. However, the PHP of inner diameters 1.2-2.2 mm does not show different flow patterns with that of uniform PHPs. The reason is that too large difference between inner diameters causes side effects for generating circulatory flow, and there exists optimum range of deviation in diameter which leads to make better flow characteristics. This is consistent with the results obtained by Holley *et al.* [9]. The effect of variation in diameter on flow patterns is investigated, and it is verified that variation in diameter is helpful to induce circulation of working fluid in the PHP.

Table 2 Flow patterns of glass capillary PHPs

Type	Q(W)							
	10	15	20	25	30	35	40	45
1	S+c	S+o	O+c	o+C	o+C	**o+C**	**C**	C
2	S	s+o+c	s+O	O+c	O+C	O+C	AC	AC
3	S+o+c	s+o+C	o+C	o+C	C	C	C	C
4	S+c	s+C	C	C	C	C	C	C
5	S+c	S+c	s+O+c	o+C	C	C	C	C
6	S	S+o+c	s+O	s+O+c	O+c	O+C	AC	AC

Effect of gravity

The effect of gravity on thermal resistance of PHPs is also studied with varying inclination angle in Fig. 5. 90° of inclination angle means vertical position of a PHP, and 0° is horizontal position. Thermal resistance decreases with increasing inclination angle, so thermal performance of vertical position is the best. The relation between thermal resistance and inclination angle depends on channel configuration and input power. Over the whole range of graph, the PHP of diameters of 1.7 and 2.2 mm is better than the other PHP under the same input power condition. The range without data is where a PHP does not function well. When inclination angle is smaller than 30°, the uniform PHP does not work under any condition. To clearly figure out effect of inclination angle on thermal performance, data is represented as shown in Fig. 6. Y axis indicates the ratio of thermal resistance to thermal resistance in a vertical position where inclination angle is 90°. The results show that sensitivity to inclination angle slightly decreases as input power increases. The reason is that velocity of circulation motion and momentum of working fluid is high when input power is large. Therefore, effect of gravity on the performance becomes smaller. A rising tendency of asymmetric PHP with decreasing inclination angle is a little bit higher than that of symmetric PHP, but the performance of symmetric PHP suddenly get worse. Thus, working range of inclination angle of symmetric PHP is relatively limited.

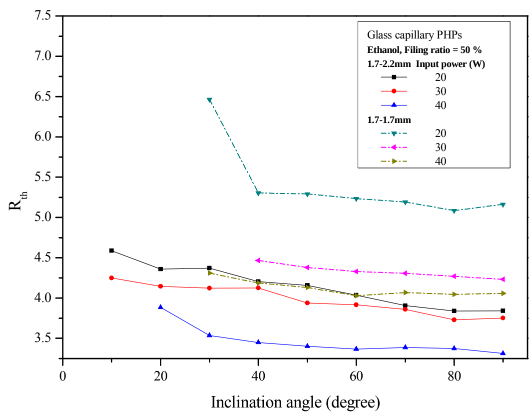


Figure 5 Thermal resistance with varying inclination angle.

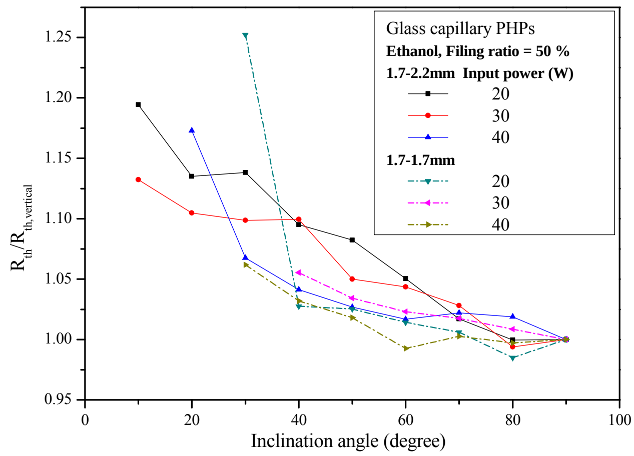


Figure 6 Ratio of thermal resistance to thermal resistance.

Inclination angle significantly affects the thermal performance of PHPs as shown in Fig. 5 and Fig. 6, and thermal resistance increases as the inclination angle decreases. The simplified design equation for a PHP can be expressed as [15]

$$F_{cap} + F_{sat} + F_g \geq F_f \qquad (14)$$

where F_{cap}, F_{sat}, F_g, and F_f indicate the capillary force, force due to saturation pressure difference between bubbles, gravity force, and the frictional force. Three terms of the LHS of Eq. (14) forces drive and cause fluid flow in the PHP. However, with channel diameter range used in this study, capillary force is smaller than other forces and can be negligible. Here, F_{sat} is related to rapid bubble growth, collapse and a sudden temperature change of bubbles. Although it is hard to measure or estimate this term, it is well known that this force leads to rapid momentum change because of spatial and temporal variation [16]. When there is a circulating flow with an input power lower than 20 W, however, vapor bubbles gradually expand in the evaporator section and shrink in the condenser section, and bubble nucleation is occasionally observed. Therefore, this force is smaller than other interacting forces including gravitational and viscous forces, and F_g plays an important role as a driving force. Eq. (14) is simplified, and gravitational force can be described as follows:

$$F_g \approx F_f \qquad (15)$$

$$F_g = \left[A_{c,left} L_{left} \rho_{left} - A_{c,right} L_{right} \rho_{right} \right] \cdot g \cdot \sin\theta \qquad (16)$$

The net gravitational force is caused by the difference between vapor mass qualities (x) in the left and right tubes due to not only phase change but also thermal expansion and shrinkage of bubbles. The density of two-phase flow including liquid slugs and vapor plugs is determined using vapor mass quality as follows:

$$\rho_{TP} = \frac{1}{x v_v + (1-x) v_l} \qquad (17)$$

Therefore, the vapor mass quality difference makes variation between the densities of the working fluid in the left and right tubes, and the net gravitational force obtained by Eq. (16) becomes non-zero. The difference between the vapor mass qualities of the left and right tubes is easily observed when there is a circulating flow. The working fluid circulates and goes up along one tube after passing through the evaporator section, and the meniscus in the tube is placed at higher position due to the increased vapor mass.

Pressure loss due to vapor flow is much smaller than that due to liquid flow because liquid has a higher viscosity at least 100 times higher than that of vapor. Accordingly, frictional pressure drop is mainly caused by fluid flow in liquid phase. The friction factor is evaluated using the Poiseuille flow friction factor because the maximum Reynolds number of fluid flow based on visualized data is less than 1000. Therefore, frictional force can be roughly predicted using mass flow rate ($\dot{m}$) as follows:

$$F_f \approx \frac{32\mu_l}{\rho_l}\left(\frac{L_{l,left}}{D_{left}^2}+\frac{L_{l,right}}{D_{right}^2}\right)\dot{m} \tag{18}$$

If the input power is constant, the total liquid length and vapor quality on the left and right sides barely vary with varying inclination angle. Therefore, the mass flow rate is roughly proportional to $\sin\theta$ when the working fluid circulates inside of a PHP as shown in Fig. 7. The vertical axis shows the ratio of the mass flow rate at a given angle to the vertical position. As expected, the flow rate decreases with decreasing inclination angle, and it agrees well with a value of $\sin\theta/\sin 90°$. In Fig. 7, only experimental data obtained from using asymmetric PHPs are shown because, in the case of the symmetric PHP, the total lengths of liquid slugs on the left and right sides varies with time, and this cause the gravitational force, a dominant driving force, to fluctuate.

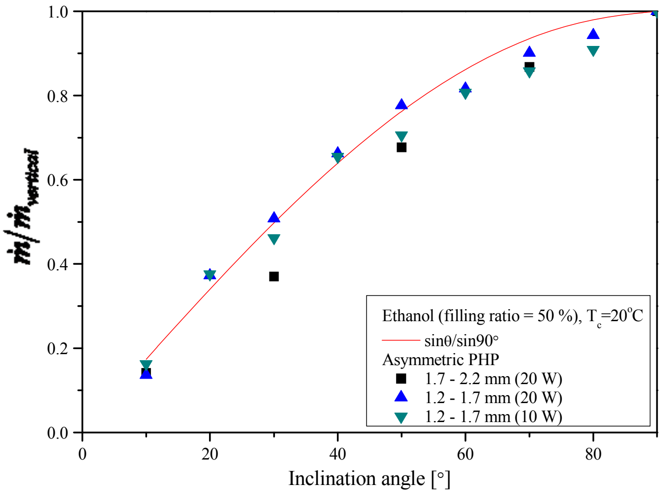


Figure 7 The effect of inclination angle on mass flow rate.

Figure 8 shows a comparison of the measured thermal resistance and mass flow rate with predictions of the proposed model. The model accurately estimates experimental data with a maximum error less than 10%. As mentioned above, the mass flow rate decreases with decreasing inclination angle, and it makes the thermal performance of the PHP worse because the heat transfer coefficients become generally lower as the mass flow rate decreases. These results provide the evidence that gravitational force is the main driving force in a single-turn PHP with a circulating flow.

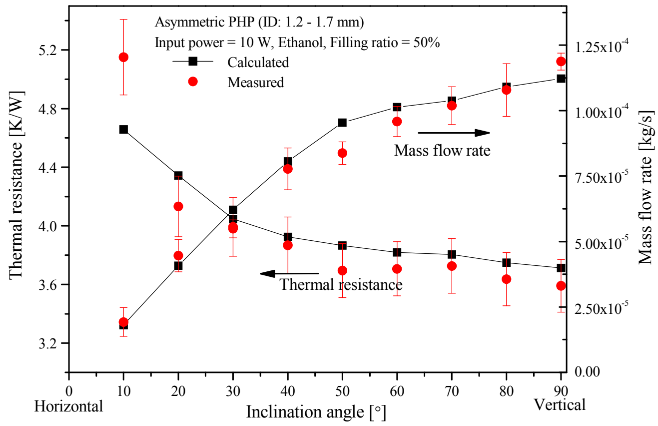


Figure 8 Comparison of measured thermal resistance and mass flow rate with predictions of the proposed model.

Effect of asymmetric configuration

Asymmetric configuration significantly affects flow characteristics, and Geng et al. introduced bubble-based micropump which has asymmetric configuration [17]. This pump uses the periodic generation and collapse of a single vapor bubble in a diverging channel, and the channel shape creates an asymmetry in the surface tension forces which result in a pumping effect. A PC controls a function generator which powers the pump via an amplifier. Figure 9 is a sequence of CCD camera images showing the bubble cycle. When the liquid starts to heat up, bubble rapidly grows, especially along the bigger tube. When the voltage is brought to zero at 100 ms, the liquid rapidly cools, and the bubble collapses to a very small residual size. Due to this motion, upward flow is generated.

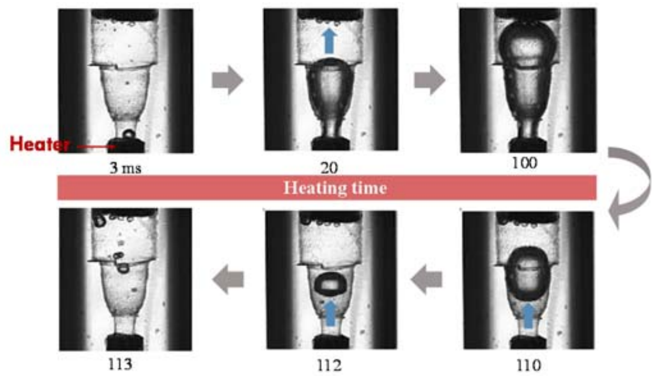
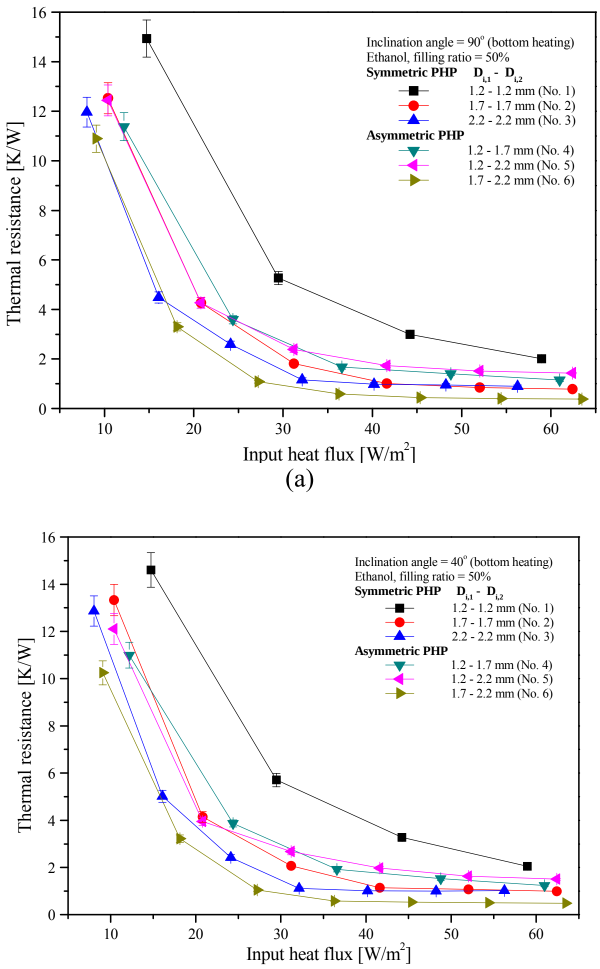


Figure 9 Bubble-based micropump [18].

Our experimental results demonstrate that these characteristics caused by the dual-diameter tube help to enhance the thermal performance of PHPs because a circulating flow of the working fluid is promoted by an asymmetric configuration.

As shown in Fig. 10, the thermal resistances of the asymmetric PHP (no.6) were generally smaller than those of symmetric PHPs under all experimental conditions. In particular, there is noticeable improvement in the thermal performance of asymmetric PHPs at lower heating power. It is because asymmetric PHPs have a circulating flow with better heat transfer characteristics while symmetric PHPs have an oscillating flow at low heating power. At high input power (≥15 W), the symmetric PHPs (no. 2 and no. 3) made of large tubes was observed to have better thermal performance than some of the asymmetric PHPs (no. 4 and no. 5). The reason is that circulating flow was promoted as well as in the symmetric PHPs at high input power and results in thermal performance enhancement of the symmetric PHPs. Meanwhile, asymmetric PHPs (no. 4 and no. 5) have the smallest tube of inner diameter of 1.2 mm. This small tube makes large friction which increases with decreasing tube diameter and reduces mass flow rate of working fluid. Especially, among the asymmetric PHPs, the asymmetric PHP (no. 5) that had the largest difference between tube diameters has the worst performance.



(b)

Figure 10 Thermal performance with varying input heat flux: inclination angle of (a) 90° and (b) 40°.

The effect of the dual-diameter tube on thermal performance of PHPs was investigated with respect to deviation in tube diameter. Figure 11 presents data which is obtained from experiments with varying input power and inclination angle. The experimental data show that when diameter difference is 0.5, thermal resistance is generally lower, but thermal resistance becomes larger again as diameter deviation increases. From these results, we can guess there exits the optimum range of diameter deviation where the thermal performance enhancement is maximized. However, it is hard to find exact optimal diameter deviation by performing experiments.

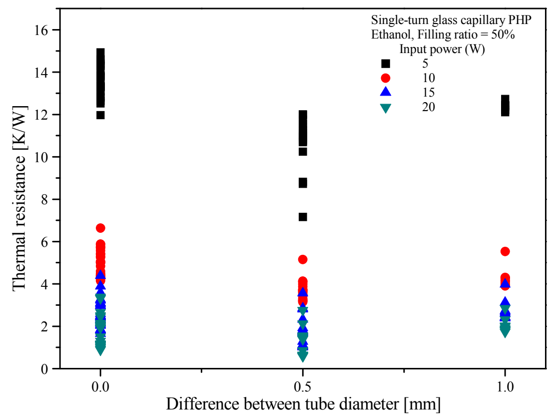


Figure 11 Thermal resistance vs. difference between tube diameters

Therefore, to figure out the effect of asymmetry on thermal performance of a PHP, the model was developed, and two types of modeling constraints were used, as shown in Fig. 12. First, the internal volume of a PHP was constant, and second, the inner diameter of the left tube was fixed. As we anticipated, the dual-diameter channel helped to improve thermal performance for not only vertical position but also inclined positions. Although the exact optimum deviation of diameters varies depending on modeling constraints and working conditions, there exists optimal range.

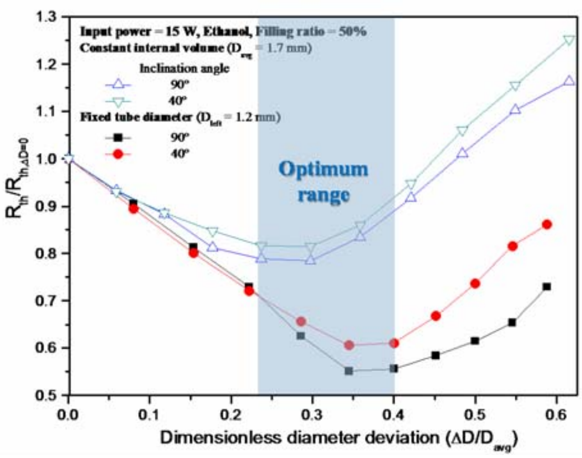


Figure 12 Effect of a dual-diameter tube on the thermal performance of the PHP

CONCLUSION

Experiments using symmetric PHPs and asymmetric PHPs were performed to verify whether a dual-diameter channel improves thermal performance. Glass capillary PHPs were charged with ethanol at a filling ratio of 50% and evaluated with varying input power and inclination angle. It was found that a dual-diameter channel helps to promote and enhance a circulating flow in the PHP, and it causes a circulating flow to start at a lower input power. This is advantageous for PHP functionality as the PHP thermal resistance was reduced by more than 15%. A simplified model was also implemented to predict the thermal performance of PHPs with a circulating flow. Although the proper diameter difference results in effectively generating a circulating flow and pumping more working fluid from the evaporator to the condenser, too large diameter deviation causes less driving force and larger frictional loss. Therefore, there is an optimum range of diameter deviation in which the thermal performance enhancement is maximized, as was found in the experimental results. It was proved that variation in diameter can be an effective way of improving PHP performance based on not only measured and visualized data but also numerical results. However, many parameters which were not counted in this study, such as types of working fluid, filling ratio, and geometric characteristics, affect flow and thermal characteristics. Therefore, further research on asymmetric configurations is needed to examine the effects of design parameters, and a PHP with optimized design would provide better thermal performance.

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