Kammuang-Lue, N., *et al.*: Effect of Working Fluids and Internal Diameters on ...
THERMAL SCIENCE: Year 2016, Vol. 20, No. 1, pp. 77-87

77

EFFECT OF WORKING FLUIDS AND INTERNAL DIAMETERS ON THERMAL PERFORMANCE OF VERTICAL AND HORIZONTAL CLOSED-LOOP PULSATING HEAT PIPES WITH MULTIPLE HEAT SOURCES

by

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Some electrical applications have a number of heat sources. The closed-loop pulsating heat pipe (CLPHP) is applied to transfer heat from these devices. Since the CLPHP primarily transfers heat by means of the working fluid's phase change in a capillary tube, the thermal performance of the CLPHP significantly depends on the working fluid type and the tube's internal diameter. In order to provide the fundamental information for manufacturers of heat exchangers, this study on the effect of working fluids and internal diameters has been conducted. Three electrical plate heaters were installed on the CLPHP as the heat sources. The experiments were conducted by varying the working fluid to be R123, ethanol, and water, at the internal diameter of 1.0 mm, 1.5 mm, and 2.0 mm. For each set of the same working fluid and internal diameter, the input heat fluxes of the heat sources were also made to vary within six different patterns. It can be concluded that when the latent heat of evaporation increases – in the case of vertical CLPHP – and when the dynamic viscosity of the liquid increases – in the case of horizontal CLPHP – the thermal performance decreases. Moreover, when the internal diameter increases, the thermal performance increases for both of vertical and horizontal CLPHP.

Key words: *closed-loop pulsating heat pipe, multiple heat sources, thermal performance, working fluid, internal diameter*

Introduction

The closed-loop pulsating heat pipe (CLPHP) is a heat exchanger with a very high thermal conductivity. It was invented by Akachi *et al.* [1]. The CLPHP is made from a copper capillary tube, the internal diameter of which does not exceed the critical value, following the Maezawa's *et al*. criterion [2], bent into an undulating tube, and connected at both ends to form a closed-loop. The tube is evacuated and consequently partially filled with the working fluid. Since the inner diameter of the tube is very small, upon meeting the capillary scale, the working fluid inside forms into liquid slugs alternating with vapor plugs along the entire length of the tube. This internal flow pattern is well known as *slug-train* [3]. Heat can be transferred by means

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of the replacement mechanism [4]. When one end of the CLPHP, called *evaporator section*, is subjected to heat or high temperature, the working fluid, which is in liquid slug form, will evaporate, expand, and move through the no-heat transferring zone, or the *adiabatic section*, toward a cooler section, namely the *condenser section*. Then, the vapor plug will condense, collapse, and release the heat into the environment. The vapor plug evaporating in the evaporator section will, consequently, flow to replace the vapor plug collapsing in the condenser section. Due to this mechanism, the working fluid can circulate and continuously transfer heat in a cycle.

In general application, a single CLPHP is frequently used to be the heat exchanger between one heat source and one heat sink, or *one-to-one* CLPHP. Design and fabrication of such heat exchangers do not come with difficulty since a lot of useful information and knowledge for the design have been discovered and obtained from the studies in the past decade [5-10]. Nevertheless, in the case of more than one heat source, especially, not with identical heat rates, such as a high-end laptop personal computer which has three heat dissipating devices consisting of CPU, GPU, and RAM and only one heat sink which is a fan, or a main distribution board (MDB) which has a number of heat generating devices, if one-to-one CLPHP – that is, one CLPHP with one heat source and all CLPHP releasing heat to the same heat sink – are installed, this configuration causes the heat exchanger to be geometrically larger and the manufacturing cost to increase. Thus, the most suitable method to breakthrough these problems is to devise such a design and manufacturing of CLPHP that can simultaneously receive the heat from multiple heat sources and carry it to a single heat sink.

From the past studies, the suitable arrangement of heat sources in the case of not identical heat flux has been archived. It was found that maximum thermal performance is obtained when heat sources are placed in consecutive order from the lowest to the highest heat flux, beginning from the inlet of the evaporator section in the case of the vertical CLPHP, and *vice versa* for the horizontal CLPHP [11]. However, these data obtained are not sufficient for a heat pipe heat exchanger design. Since the CLPHP primarily transfers the heat by means of the working fluid's phase change in a capillary tube, as previously mentioned, the working fluid type, which is chosen to fill in the heat pipe, and the internal diameter of the copper tube play very important roles in the thermal performance of the CLPHP. As long as fundamental knowledge on suitable working fluids and internal diameters are limited, as it is in the present, heat pipe heat exchanger manufacturers cannot choose the suitable working fluid to fill in their heat pipes with suitable internal diameters, especially in the case of CLPHP with a number of heat sources. From this point of view, the significance of this study is its objective which is to investigate the effect of working fluids and internal diameters on the thermal performance of a CLPHP used to release heat from multiple heat sources in both vertical and horizontal orientations. Results obtained from this study will be a great advantage for not only the manufacturers of heat pipe heat exchangers but also researchers interested in heat release applications.

Experimental set-up and procedure

The CLPHP used in the experiment were made of long copper capillary tubes with inner diameters of 1.0 mm, 1.5 mm, and 2.0 mm, and bent into 32 turns. Both the ends of the capillary tubes were connected together to form a loop. The R123, ethanol, and water were selected as the working fluids. The filling ratio was 50% by total volume. Three electrical plate heaters (400 W, 220 VAC) with sizing of 25 mm width, 340 mm length, and 3.5 mm thickness represented the three heat sources. The heaters were attached with copper bus bars, which were machined to have semi-circular grooves to fit the outer walls of the tube of the CLPHP. Thermal grease was fully filled in the gap between the heaters, copper bus bars, and tubes to ensure per-

fect thermal contact. The CLPHP was installed in a test rig that could be oriented in the vertical plane, with the evaporator section located at the lowest part, called as the bottom heat mode (BHM), or in the horizontal plane. The input heat flux of each heat source could be independently controlled by using three power controllers (Shimax, MAC3D, accuracy ±0.25% full scale) and simultaneously monitored for validation of the heat flux quantity by using the watt meters (Axe, MMP, accuracy ±0.25% full scale). Each heat source was defined to alternately change within three different input heat fluxes corresponding to the inner area of the copper tube, which were 1 kW/m^2, 3 kW/m^2, and 5 kW/m^2. Therefore, the heat source arrangement could be permuted into six unduplicated sets. The number of each set was defined as shown in tab. 1.

Table 1. Heat source arrangement used in this study

Heat source position	Heat flux of each heat source [kWm^{-2}]					
	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6
Upper	1	1	3	3	5	5
Middle	3	5	1	5	1	3
Lower	5	3	5	1	3	1

All of the condenser sections were placed in a zinc cooling jacket with inlet and outlet tubes. The cooling medium was a solution of water and ethylene glycol with 1:1 volume ratio. It was circulated to transfer heat from the condenser section to the heat sink, which was the cold bath (Bitzer, D7032, accuracy ±1 °C). The temperature and the flow rate of the cooling medium were directly controlled by this cold bath. The mass flow rate of the cooling medium, which flowed for a specified time, was weighed by the digital scale (Ohaus, Adventurer, accuracy ±0.01 g). The time was monitored using a high precision stopwatch (Casio, HS70W-1D, accuracy ±0.001 s). All the sections, including the cooling medium hoses were well insulated by using a thermal-insulated sheet (Aeroflex, 3/8 in. thickness). At specified points, the temperature was monitored using a data logger (Brainchild, VR18, accuracy ±0.1 °C). Fourteen Chromel-Alumel thermocouples (Omega, Type K, accuracy ±0.5 °C) were installed on the outer surface of the capillary tube to measure the variations in the temperature at every part of the CLPHP. There were totally six points for measuring the evaporator section temperatures consisted of two points on each of the upper (T_{e1} and T_{e2}), middle (T_{e3} and T_{e4}), and lower (T_{e5} and T_{e6}) heater. It could be noticed that each groups of thermocouples was attached in parallel between the adjacent tubes in the same meandering turn. The thermocouples in the evaporator section were installed on the 16th turn, which was a center of the heat pipe. There were four points in the middle of the adiabatic section (T_{a1}-T_{a4}), and four points in the middle of the condenser section (T_{c1}-T_{c4}). The T_{a1}, T_{a2}, T_{c1}, and T_{c2} were belonged to the 16th turn and the T_{a3}, T_{a4}, T_{c3}, and T_{c4} were installed on the 24th turn. Two thermocouples were also placed on each inlet tube and outlet tube of the cooling jacket to measure the variations of the cooling medium temperature. The CLPHP and the experimental setup were schematically executed, as illustrated in fig. 1.

The experimental procedure was: The power controllers were started and the heat flux of each heat source was adjusted following Set 1 in tab. 1. The adiabatic temperature was controlled at 50 ± 3 °C, as this was a suitable working temperature and did not exceed the critical point of all the chosen working fluids. This was achieved by adjusting either the temperature or the flow rate of the cooling medium. When the system reached the steady-state, the temperature
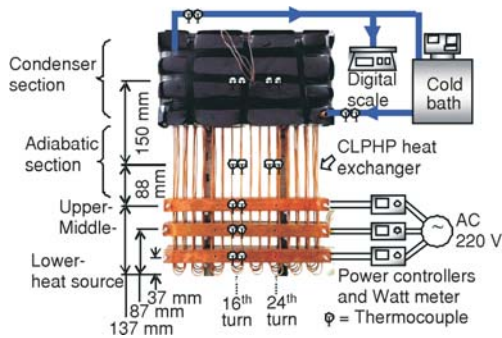
Kammuang-Lue, N., *et al.*: Effect of Working Fluids and Internal Diameters on ...

80   THERMAL SCIENCE: Year 2016, Vol. 20, No. 1, pp. 77-87

Figure 1. The CLPHP and experimental set-up

at each point in the evaporator, adiabatic, and condenser sections was recorded, while the variations in the inlet and outlet temperatures and the flow rates of the cooling medium were simultaneously measured in order to calculate the heat flux at specified times by means of the calorific as given in eq. (1). The advantage of this way of measuring is that the actual through-put heat along the CLPHP can be obtained. The experiment for Set 1 was completely done after the test was repeatedly conducted for five times. The whole procedure, right from the beginning, was condu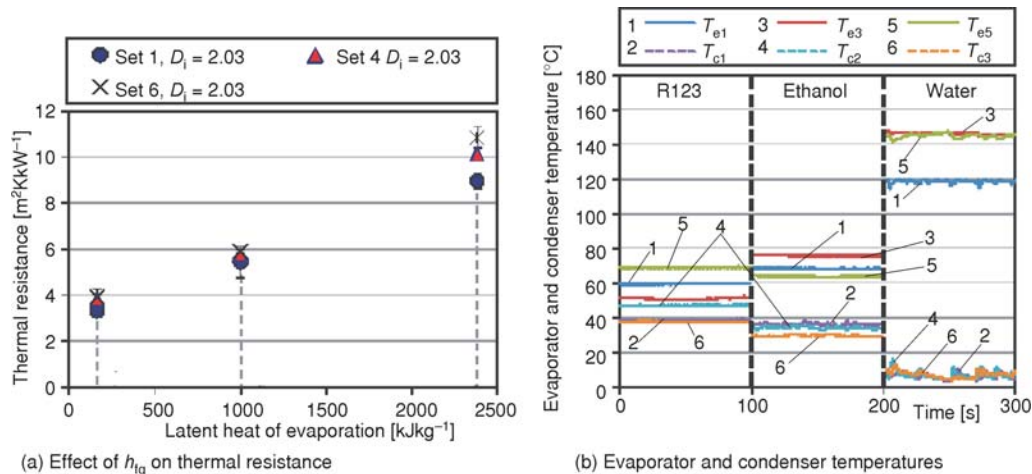cted again by changing the heat flux of each heat source until all the sets, as shown in tab. 1, were successfully investigated.

$$\dot{q}_c = \frac{\dot{m}_{cl}\, c_{pcl}\, (T_{out} - T_{in})_{cl}}{A_c} \tag{1}$$

The values of heat flux obtained from eq. (1) were analyzed in the next step and subsequently brought into an account to determine the thermal performance of the CLPHP through the *thermal resistance per unit area* (z) which can be calculated from eq. (2). However, from now on, it will be called as *thermal resistance*, for short, in order to shorten sentences:

$$z = \frac{T_{e,avg} - T_c}{\dot{q}_c} \tag{2}$$

Results and discussions

Effect of working fluids in the case of vertical CLPHP

In a study on the effect of working fluids, it is necessary to identify each of the working fluids by quantitative parameters, which generally are the thermodynamic properties of the working fluid, for example, latent heat of evaporation, specific heat, surface tension, viscosity, thermal conductivity, *etc*. Since the CLPHP transfers heat by the evaporative and condensing mechanism of the working fluid [4], the most suitable parameter to identify the differences between the working fluids in this topic is the latent heat of evaporation (h_{fg}). The R123, ethanol, and water were chosen to be the variable parameters, with the latent heat of evaporation values of 161, 1000, and 2382 kJ/kg, respectively. The study on the effect of the working fluid on the thermal resistance of the CLPHP with internal diameter of 2.03 mm showed that when the working fluid changed from R123 to ethanol and water, or when the latent heat of evaporation increased, the thermal resistance increased, for every set of the heat source arrangement, as shown in fig. 2(a). It can be seen that when the latent heat of evaporation increased from 161 to 1000 and 2382 kJ/kg, the thermal resistance increased from 3.3 to 5.4 and 9.0 m²K/kW, respectively, for Set 1, from 3.8 to 5.7 and 10.1 m²K/kW, respectively, for Set 4, and from 3.9 to 5.9 and 10.8 m²K/kW, respectively, for Set 6.

According to results obtained in the past studies that the internal flow pattern in the vertical CLPHP is slug flow mixing with churn flow inside the hot tube (the tube that vapor flowing out of the evaporator) [6, 12] and the working fluid circulates in one direction [13, 14]. Since the churn flow has a portion of the vapor greater than that of the liquid, this basically shows that the CLPHP transfers the heat mainly by latent heat rather than the sensible heat. The

Kammuang-Lue, N., *et al*.: Effect of Working Fluids and Internal Diameters on ...
THERMAL SCIENCE: Year 2016, Vol. 20, No. 1, pp. 77-87

81

(a) Effect of h_{fg} on thermal resistance

(b) Evaporator and condenser temperatures

Figure 2. The effect of the working fluids in the case of the vertical CLPHP

heat transfer is higher compared to the only single phase forced convection [14]. From this evidence, the physical reason that describes why the thermal resistance increases as the latent heat of evaporation increases with the same trend in all arrangements of the heat sources, can be analyzed. In general, the working fluid with lower latent heat of evaporation, such as R123, requires lower heat quantity used in the complete evaporation of certain-mass liquid working fluid than that of the working fluid with higher latent heat of evaporation, that is ethanol or water. When R123 is used, therefore, the liquid working fluid will evaporate, carry the heat out from the evaporator section, and then flow to the condenser section rapidly. This causes the evaporator section temperature to be low, and the temperature difference between the evaporator section and the condenser section, subsequently, becomes low, and, in turn, the thermal resistance becomes low. Thus, the highest thermal performance of the CLPHP is achieved. These phenomena of the working fluid with lower latent heat of evaporation also correspond with the results of Charoensawan *et al.* study [15] which found that when the working fluid changes from R123 to R141b, or when the latent heat of evaporation increases, the vapor length increases and the vapor collapsing ratio in the condenser section decreases. These cause a decrease in the transferred heat flux of the CLPHP. In the case of water, the liquid working fluid spends longer time receiving heat, compared to R123, until it can evaporate completely, and then it flows to the condenser section. The longer duration of evaporation causes higher evaporator section temperature. Thus, the temperature difference between the evaporator section and the condenser section is higher than that for R123. Consequently, the thermal resistance is high, or the thermal performance of the CLPHP is low. The temperature differences between the evaporator section and the condenser section of the CLPHP when R123, ethanol, and water are used as working fluids are shown in fig. 2(b). It can be seen that the maximum temperature difference occurs in the case of water.

Effect of working fluids in the case of horizontal CLPHP

The study on the effect of working fluids on the thermal resistance of the CLPHP with internal diameter of 2.0 mm showed that when the working fluid changed from R123 to ethanol and water, or when the latent heat of evaporation increased, the thermal resistance increased and

82

Kammuang-Lue, N., *et al.*: Effect of Working Fluids and Internal Diameters on ...
THERMAL SCIENCE: Year 2016, Vol. 20, No. 1, pp. 77-87

then decreased, for every set of the heat source arrangement, as illustrated in fig. 3(a). It can be seen that when the latent heat of evaporation increased from 161 to 1000 and 2382 kJ/kg, the thermal resistance increased from 20.7 to 39.4 and decreased to 23.0 m^2K/kW, respectively, for Set 1, increased from 11.8 to 15.3 and decreased to 15.2 m^2K/kW, respectively, for Set 4, and increased from 9.2 to 15.3 and decreased to 14.5 m^2K/kW, respectively, for Set 6.

(a) Effect of h_{fg} on thermal resistance

(b) Effect of μ_l on thermal resistance
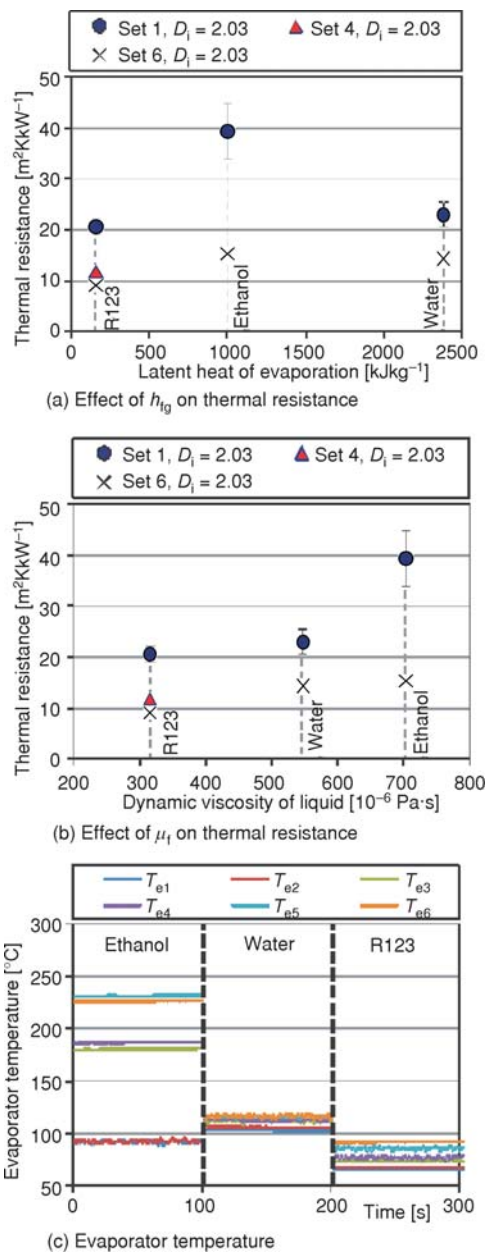

(c) Evaporator temperature

Figure 3. The effect of the working fluids in the case of the horizontal CLPHP

It can be seen from fig. 3(a) that thermal resistance has no relation with latent heat of evaporation, as in the case of the vertical CLPHP. This is because the working fluid's flow pattern inside the horizontal CLPHP is oscillating flow, rapidly changing its direction all the time [6, 16]. While the liquid working fluid is receiving heat input in the evaporator section, the flow direction suddenly changes. The working fluid is forced to flow out to the condenser section to release the heat. Because the working fluid has a very short duration of stay in the evaporator section, the liquid has insufficient heat quantity to completely evaporate. From this point, the CLPHP transfers the heat by means of the latent heat of evaporation and the sensible heat, simultaneously, as discussed in [15]. However, since a proportion of the transferred heat between the latent heat of evaporation and the sensible heat cannot be experimentally investigated [17], and also since there is no other study which can be said to have a reasonable conclusion on this topic, the latent heat of evaporation, the sensible heat, or the proportion between these two properties cannot be used to identify the working fluid type in the case of the horizontal CLPHP. For this reason, analysis on the working fluid's flow to find a suitable thermodynamic property, which can define the working fluid type and has a relation with the thermal resistance, must be necessarily conducted.

According to very active oscillating flow inside the horizontal CLPHP found in [15], the coalescence between vapor plugs flowing out from the evaporator section can take place frequently. This causes vigorous motion in the surrounding liquid and has a dominating effect on heat and mass transfer in the liquid flow [14]. From this mechanism, the thermal performance of the horizontal CLPHP has a significant relation with the liquid flow between the

evaporator and the condenser section. Generally, the liquid flow rate in the confined channel is directly proportional to the pressure difference between the evaporator and condenser section and is reversely proportional to the dynamic viscosity of a liquid (μ_l). Due to the replacement mechanism in the CLPHP, the working fluid's circulation is activated by expanding and collapsing of the vapor plugs in the evaporator and the condenser section, respectively. It can be said that the working fluid's circulation is corresponding to a difference in pressure between the vapor plug in the evaporator and the condenser section. If consideration is made only to the liquid phase, it has a similar pressure along the entire CLPHP [4]. Therefore, the thermal performance of the horizontal CLPHP can be obviously enhanced by decreasing the liquid phase's dynamic viscosity. This statement is very well agreed with the experimental study on the effect of working fluids on the thermal resistance of the horizontal CLPHP with internal diameter of 2.0 mm, which was found that, when the working fluid changed from R123 to water and ethanol, or when the dynamic viscosity of the liquid increased from 316 to 547 and 703 µPa·s, respectively, the thermal resistance increased from 20.7 to 23.0 and 39.4 m^{2}K/kW, respectively, for Set 1, from 11.8 to 15.2 and 15.3 m^{2}K/kW, respectively, for Set 4, and from 9.2 to 14.5 and 15.3 m^{2}K/kW, respectively, for Set 6. The results are illustrated in fig. 3(b).

Upon taking into consideration fig. 3(b), it becomes evident that the dynamic viscosity of the liquid has obvious effect on the increase in the thermal resistance more in the case of Set 1 than in the case of Set 4 and Set 6. The physical reason for the same can be described. In general, the working fluid can flow in a capillary tube by *driving force* in order to overcome the frictional force against the working fluid's circulation. The driving force that activates the working fluid's circulation in the CLPHP originates from two phenomena, according to the replacement mechanism, which are: (1) the vapor's collapsing volume, which is with respect to the driving force occurring in the condenser section and (2) the vapor's expanding volume, which is located in the evaporator section [4]. In the case of the heat source arrangement in Set 1, the working fluid flows in a pulsating motion with intermission-stops, since the working fluid continuously evaporates until the cross-sectional area of the tube in the evaporator section is totally occupied by a vigorous vapor plug. This causes the condensate from the condenser section to be not able to flow into the evaporator section. Because of this situation, an intermittent stop in the working fluid's circulation is observed [11]. In order to activate the working fluid to flow again, a driving force (or called as a restoring force) from the vapor's collapsing volume in the condenser section is alternatively required [14]. For this reason, the time duration of the intermission-stop, when working fluid with high dynamic viscosity of liquid, such as ethanol, is used, is obviously longer than that in the case of working fluid with low dynamic viscosity of liquid, for example, R123 or water. Moreover, since the system has to wait until the collapsing volume in the condenser section is large enough to generate sufficient driving force for re-circulation, heat cannot be continuously transferred. This causes the evaporator section temperature of the CLPHP with ethanol to be apparently higher than that of the CLPHP with R123 and water, as shown in fig. 3(c). Consequently, since the temperature difference between the evaporator section and the condenser section is high, the thermal resistance of the CLPHP in Set 1 with ethanol is obviously higher than that of the CLPHP with R123 and water. In contrast to the case of the CLPHP in Set 4 and Set 6, it was found from a previous study that although the working fluid flows in a pulsating motion same as that in the CLPHP with the heat source arrangement in Set 1, as mentioned before, the intermission-stop does not occur at all [11]. This causes the heat to be transferred continuously and, thus, this causes the temperature of the evaporator section of the CLPHP with ethanol to be nearly the same as that of the CLPHP with R123 and water. Thus, the thermal resistance of the

CLPHP in Set 4 and Set 6 with ethanol is rarely higher than that of the CLPHP with R123 and water.

Effect of internal diameters in the case of vertical CLPHP

The study on the effect of internal diameters on the thermal resistance of the vertical CLPHP with the heat source arrangement in Set 1, which is the best arrangement for the vertical CLPHP [11], showed that when the internal diameter increases, the thermal resistance decreases, as illustrated in fig. 4(a). It can be seen that when the internal diameter increases from 1.0 to 1.5 and 2.0 mm, the thermal resistance tends to decrease slightly from 4.0 to 3.0 and 3.2 m^2K/kW, respectively, for the CLPHP with R123, obviously from 21.2 to 5.6 and 5.4 m^2K/kW, respectively, for the CLPHP with ethanol, and dramatically from 70.1 to 38.0 and 9.0 m^2K/kW, respectively, for the CLPHP with water.

Upon taking into consideration the physical structure of the CLPHP, it was found that an increase in the internal diameter not only causes the heat transferring area between the heat pipe and the working fluid to increase but also causes the cross-sectional area of the working fluid's flow inside the CLPHP to increase. When the CLPHP has a larger internal diameter or a wider cross-sectional area of the flow passage, the vapor plugs evaporating in the evaporator section consequently flow toward the condenser section more continuously with higher quantity of the working fluid. For this physical reason, the CLPHP can transfer more heat, and, thus, the thermal resistance subsequently decreases, or the thermal performance increases. This is especially true for the CLPHP with Set 1 in which the heat sources are placed in consecutive order, from the lowest to the highest heat flux, beginning from the inlet of the evaporator section. The heat source arrangement in Set 1 originally promotes the working fluid to circulate in complete one-direction, as found in [11]. Therefore, when the cross-sectional area of the working fluid's flow passage further increases, the working fluid is encouraged to circulate from the evaporator section to the condenser section more actively. The superposition in the effect of the heat source arrangements and the effect of the internal diameters finally causes the CLPHP with Set 1, and with the internal diameter of 2.0 mm, to have the highest thermal performance for every working fluid type, in this study.

It can be additionally noticed from the experimental results, which are presented in fig. 4(a), that the thermal resistance of the CLPHP filled with water dramatically decreases as the internal diameter increases. The physical reasons for this can be explained. In the case of
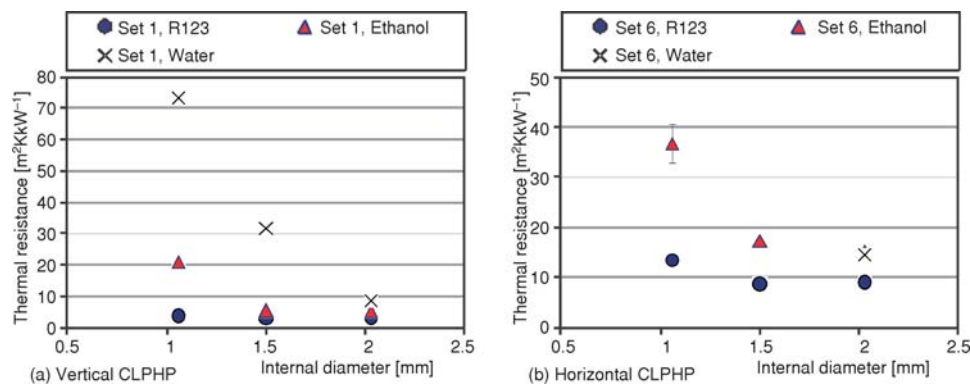

Figure 4. The effect of the internal diameters

the 1.0 mm CLPHP, the thermal resistance was 70.1 kW/m^2. This quantity is considered to be relatively high. Very high thermal resistance occurs when the temperature of the evaporator section is excessively higher than the temperature of the condenser section ($T_{e,avg}$ = 149 °C and $T_{cond,avg}$ = –1.5 °C in this case). This is because the quantity of liquid working fluid is insufficient to transfer the heat out from the evaporator section and, therefore, a dry-out on the inner tube's surface is observed and the temperature of the evaporator section consequently increases. These phenomena are evidence to confirm that the CLPHP in this case is operating in the critical state [18]. In addition, a previous study has found that the vertical CLPHP operates in the critical state when the internal working fluid's flow pattern changes from the slug train to the co-current annular flow since the working fluid's flow velocity increases beyond the critical value [12]. Consequently, when the internal diameter increases to be 1.5 mm and 2.0 mm, respectively, the cross-sectional area of the flow passage also increases dependently. Although the input heat flux from the heat sources is constant, the quantity of the evaporating vapors, which flow toward the condenser section, is almost the same. However, according to the increase in the cross-sectional area, the working fluid's flow velocity obviously decreases until it is lower than the critical value. The operation of the CLPHP consequently changes from the critical state to the normal operating state. After the CLPHP transfers the heat normally, the thermal resistance dramatically decreases and, finally, the thermal performance increases. From these observations, it can be concluded as another issue that a change in the internal diameter has greater significant effect on the variation in the thermal resistance when the CLPHP originally operates closer to the critical state.

Effect of internal diameters in the case of horizontal CLPHP

The study on the effect of internal diameters on the thermal resistance of the horizontal CLPHP with the heat source arrangement in Set 6, which is the best arrangement for the horizontal CLPHP [11], showed that when the internal diameter increases, the thermal resistance decreases, as illustrated in fig. 4(b). It can be seen that when the internal diameter increases from 1.0 to 1.5 and 2.0 mm, the thermal resistance tends to decrease slightly from 13.5 to 8.8 and 9.2 m^{2}K/kW, respectively, for the CLPHP with R123, obviously from 36.7 to 17.4 and 15.3 m^{2}K/kW, respectively, for the CLPHP with ethanol, and dramatically from 232.1 to 166.7 and 14.5 m^{2}K/kW, respectively, for the CLPHP with water. However, the thermal resistances of the CLPHP with water and internal diameters of 1.0 mm and 1.5 mm were excessively high for ordinary heat transferring applications. Therefore, both the data were reasonably neglected from fig. 4(b).

The physical reason describing the decrease in the thermal resistance with an increase in the internal diameter in the case of the horizontal CLPHP is identical to the reason expounded in the case of the vertical CLPHP, as discussed in previous section. The effect of the internal diameter was more distinguished when the CLPHP had the heat source arrangement as in Set 6 in which the heat sources are placed in consecutive order, from the highest to the lowest heat flux, beginning from the inlet of the evaporator section. The heat source arrangement in Set 6 originally promotes the working fluid to circulate in a pulsating flow without any intermission-stop, which means that the working fluid's flow direction changes all the time, and this causes the CLPHP to have the highest thermal performance, as found in the past studies [11]. Therefore, when the cross-sectional area of the working fluid's flow passage further increases, the working fluid is encouraged to have a pulsating flow between the evaporator section and the condenser section more frequently.

In addition, it can be noticed from the experimental results that the CLPHP with water and internal diameters of 1.0 mm and 1.5 mm had unusually high thermal resistance, which were

232.1 and 166.7 m^2K/kW, respectively. It is well known that the thermal resistance of the CLPHP occurs in this range when the CLPHP operates in the critical or post-critical state. The dry patch instantly expands and almost covers the entire evaporator section in these extreme states [16, 19]. Nevertheless, it was observed that when the internal diameter was increased to be 2.0 mm, the thermal resistance dramatically decreased to an ordinary level of 14.5 m^2K/kW. This value shows that the CLPHP was then operating in the normal state. The physical reason why the CLPHP which has been in the critical state can change to operate in the normal state by increasing the internal diameter can be described. When the internal diameter or the cross-sectional area of the flow passage increases, the liquid working fluid requires a higher heat input quantity from the heat sources until it can evaporate to be the vapor plug that has enough volume to totally occupy the cross-sectional area of the tube. Since the heat input is constant, the vapor plug cannot expand to completely cover the evaporator section, and the liquid working fluid remains in the inner surface of the tube. From this observation, it can be concluded that the possibilities that the vapor plug might obstruct the working fluid's circulation and that the dry patch can occur obviously diminish. This causes the working fluid to circulate without any intermission-stops. Subsequently, the heat can be transferred continuously, and the thermal resistance, obviously, decreases.

Conclusions

The effect of working fluids and internal diameters on the thermal performance of a closed-loop pulsating heat pipe used to release the heat from three heat sources has been thoroughly investigated. The itemized conclusions are:
- In the case of the vertical CLPHP, when the latent heat of evaporation increases, the thermal performance decreases since the working fluid with higher latent heat requires longer time duration of heat transfer to completely evaporate.
- In the case of the horizontal CLPHP, when the dynamic viscosity of the liquid increases, the thermal performance decreases since the working fluid with higher liquid viscosity requires higher driving force to overcome the frictional force.
- In the case of the vertical and horizontal CLPHP, when the internal diameter increases, the thermal performance increases since the working fluid flows toward the condenser section more continuously with higher quantities. In addition to the case of the horizontal CLPHP, the possibilities that the vapor plug might obstruct the working fluid's circulation and that the dry patch can occur obviously diminish according to the increase in the internal diameter.
- Although the results show that the greater the internal diameter, the higher the thermal performance, the CLPHP's internal diameter must not exceed the critical internal diameter since a change in the thermal performance beyond this point is still in the black box.

Acknowledgment

Nomenclature

A	– area, [m^2]	D_i	– internal diameter, [mm]
c_p	– specific heat, [$kJkg^{-1}K^{-1}$]	h_{fg}	– latent heat of evaporation, [$kJkg^{-1}$]

$\dot{m}$	– mass flow rate, [kgs^{-1}]		c	– condenser
$\dot{q}$	– heat flux, [kWm^{-2}]		cl	– cooling medium
T	– temperature, [K]		e	– evaporator
z	– thermal resistance per unit area, [m^{2}KkW^{-1}]		l	– liquid
μ	– dynamic viscosity, [Pa·s]		in	– inlet
			out	– outlet

Subscripts

a   – adiabatic

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