



## Investigation of the performance of pulsating heat pipe subject to uniform/alternating tube diameters



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### ABSTRACT

The present study examines the performance of closed-loop pulsating heat pipes (CLPHPs) with an ID of 2.4 mm. The effect of uniform and alternating tube diameter on the performance is investigated. The working fluids include distilled water, methanol and HFE-7100. Tests are performed with both horizontal and vertical arrangement. For the horizontal arrangement, when compared to uniform design, the alternating channel design can be started at a rather low heat input with a much smaller thermal resistance. Normally the thermal resistance is decreased with the rise of heat input, and reveals a minimum value at a certain heat input followed by shows a marginal rise when the heat input is increased further. Both uniform and alternating design reveals the similar trend. For the vertical arrangement, the thermal resistance is much lower than that in horizontal arrangement. Different from that in horizontal arrangement, the thermal resistance shows a continuous decline against heat input for all the working fluids. For a low input power, CLPHP with HFE-7100 shows the least thermal resistance. By contrast, CLPHP with distilled water shows the smallest thermal resistance when the input power is increased over 60 W.

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### 1. Introduction

The number of transistors on integrated circuits followed the so called Moore's law in which the integrated circuits double approximately every two years in a specific area. This eventually leads to an enormous rise of heat density, placing a major roadblock for further miniaturization. As a consequence, improving the performance of the thermal module is imperative to maintain the junction temperature of the electronic devices below certain threshold. One of the most successful heat removal designs is heat pipe which is now used in every aspect of electronic cooling applications. However, convective heat pipes normally suffers from comparatively lower maximum heat dissipation and shorter operational distance as used in electronic cooling. In this regard, multiple heat pipes are commonly employed to tackle high flux demand. However, the wick structure in the conventional heat pipe limits the transport distance. In this regard, the concept of pulsating heat pipe (PHP) is proposed to tackle the foregoing problems (Akachi [1,2]). Unlike traditional coaxial heat pipes, the pulsating heat pipes are made from a long continuous capillary tube bent into many turns and they are free from wick. The PHPs is also easier to manufacture

with fewer operating limitations [3]. Yet they required more working fluid than conventional heat pipes [4]. The heat transfer of PHP occurs due to self-exciting oscillation which may be driven by fast fluctuating pressure wave engendered from nucleate boiling and subsequent condensation of the working fluid [5]. In fact, as explained by Shafii et al. [5], due to the pulsation of the working fluid in the axial direction of the tube, heat is transported from the evaporator section to the condenser section. The heat input, which is the driving force, increases the pressure of the vapor plug in the evaporator section. In turn, this pressure increase will push the neighboring vapor plugs and liquid slugs toward the condenser where a lower pressure prevails.

There had been quite a few studies in association with the performance of PHPs. Khandekar et al. [6,7] conducted experiments on a PHP made of 2 mm copper capillary tube with three different working fluids – water, ethanol and R-123. The PHP was tested in vertical (bottom heat mode) and horizontal orientation. Their results demonstrated that the effect of input power and volumetric filling ratio of the working fluid on the thermal performance is quite essential. Qu et al. [4] performed tests of a 2 mm copper capillary PHP having 5 turns, and ethanol was selected as the working fluid with filling ratios (FR) of 30%, 50% and 70%. Three types of attractors were identified under different power inputs. Based on their nonlinear analysis and tests, the best fill ratio is found to be at 50%. There were some studies associated the working fluids

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### Nomenclature

$c_p$	specific heat of water ( $\text{J kg}^{-1} \text{K}^{-1}$ )	$T_{c,avg}$	average surface temperature at the condenser (K)
$D$	diameter (m)	$T_{e,avg}$	average surface temperature at the evaporator (K)
$\dot{m}$	mass flow rate of coolant water ( $\text{kg s}^{-1}$ )	$T_{w,in}$	temperature at the chilled water inlet (K)
$Q$	heat transfer rate (W)	$T_{w,out}$	temperature at the chilled water outlet (K)
$R$	thermal resistance ( $\text{K W}^{-1}$ )		
$T$	temperature (K)		

applicable PHP. For instance, Fumoto et al. [8] conducted PHP made of flat aluminum plate. The working fluid is n-butane/water mixture and the supplied power is 10–70 W. It shows that the 30% higher power input can be achieved for 1% n-butane water solution. Similar results for water, ethanol, and methanol were reported by Vassilev et al. [9], and the best filling ratios for water, ethanol, methanol are 30%, 30%, and 20%, respectively. Kammuanglue et al. [10] performed experiments for ethanol, water and R-123. They found that the thermal resistance is related to viscosity during horizontal operation. Groll and Khandekar [11] found the best filling ratio is around 30–50% for ethanol under vertical operation. Qu and Wang [12] reported a best filling ratio of 40% for ethanol, water, and the results are insensitive to change of tube diameter and evaporator size.

Though PHP has been experimentally tested in some high power electronic cooling applications, large-scale commercialization of the PHP is still not available due to some practical concerns. For instance, PHP with fewer turn is unable to function properly subject to horizontal arrangement [13]. The problem can be remedied by adding a check valve or more turns. However, introducing a check valve inevitably complicates the system operation and adding more turns will increase the size of PHPs. Hence Chien et al. [14] introduce a novel concept by having different sizes of capillary tubes rather than conventional uniform tube diameters in a PHP. Their design made use the uneven capillary force and was proven to operate successfully under horizontal arrangement. The original design of [14] was applicable in a flat panel which shows less flexibility when the space constraint is conspicuous. Also, it may not directly apply to the conventional corrugated PHP configuration. Hence, the present study aims to extend the applicability of the previous concept to the commonly corrugated PHPs. Moreover, the effect of working fluids on the performance of the PHP is also investigated. It will be found in subsequent discussion that the selection of working fluid casts a significant impact on the performance of PHPs. The proposed design can be used in a larger PHPs subject to long distance transportation and also overcome the influence of gravity.

## 2. Experimental apparatus

The schematic of the present apparatus for the closed loop PHP (CLPHP) experiment is shown in Fig. 1. Besides the tested CLPHP samples, the experimental setup comprises an evaporator section, a water loop served as a condenser, and the measurement devices as well as a data acquisition system. The evaporator section is made of a copper block (70 mm × 230 mm × 20 mm) for heating the CLPHP by providing electric power from a power supply to the heater which is embedded in the copper block. In order to minimize the heat loss from the heater to the environment, a bakelite board having a rather low thermal conductivity is installed beneath the copper block. As for the water loop, the inlet water temperature for cooling the CLPHP is controlled by a precision water thermostat and the water flow rate is measured by a magnetic flow meter.

In this study, two CLPHPs having either uniform or alternating tube diameter were tested. Detailed geometries and dimensions of the tested CLPHPs are shown in Fig. 2(a). The CLPHPs were made of copper tube with an overall size of 200 mm × 210 mm × 3 mm having 8 parallel channels. The outer diameter of the uniform CLPHP is 3 mm having a wall thickness of 0.3 mm. For the alternating CLPHP, half of the tubes are the same with that uniform CLPHP but the rest of tubes using the original 3 mm tube were compressed to an oval like tube with the minor diameter being 1.5 mm as depicted in Fig. 2(b). The working fluids tested in this study includes water, methanol and HFE-7100. After filling working fluid and sealing, the CLPHP was tested with power inputs ranging from 20 W to 140 W, and tests were performed in both horizontal and vertical orientation. The above-mentioned CLPHP is seated on a well-fitted bakelite.

Thermocouples are used to measure the surface and fluid temperature. A total of twenty-four T-type thermocouples are placed underneath the test section for measuring the average surface temperature whereas two thermocouples are used to record the inlet and outlet temperature of cooling water across the condenser. The schematic of the condenser is shown in Fig. 2(c). The condenser is a cold plate made by copper block (70 mm × 230 mm × 20 mm). In order to minimize the heat loss from the CLPHP to the environment, a U-shaped groove was made by precision machining to locate the CLPHP. The vacuum, fluid degassing, filling systems are shown in Fig. 2(d) and (e). Firstly, the working fluid is charged into the filling tube which is attached to the connecting valve (2) as shown in Fig. 2(d). The filling tube and its dimension is shown in Fig. 2(e). The degassing of the system is carried out by a rotary vane pump (Alcatel Pascal 2015SD with the vacuum pressure of the system being lowered to  $10^{-4}$  torr and remains unchanged for over six hours). Then the PHP system is connected to (3) shown in Fig. 2(d) with the filling tube for charging the working. Then the working fluid is charged into the system once the valves connected to (2) and (3) are opened. The locations of the thermocouples on the test section are as schematically shown in Fig. 2(a). These data signals were individually recorded and then averaged. During the isothermal test, the variation of these thermocouples was within 0.2 °C. In addition, all the thermocouples were pre-calibrated by a quartz thermometer having 0.01 °C precision. The accuracies of the calibrated thermocouples are of 0.1 °C. All the data signals are collected and converted by a data acquisition system (a hybrid recorder, YOKOGAWA MX-100). The shortest measurement interval of data acquisition system is 100 ms. The data acquisition system then transmitted the converted signals through Ethernet interface to the host computer for further operation.

## 3. Data reduction

The cooling capacity of condenser is calculated from the following equation:

$$Q_{out} = \dot{m}c_p(T_{w,out} - T_{w,in}) \quad (1)$$

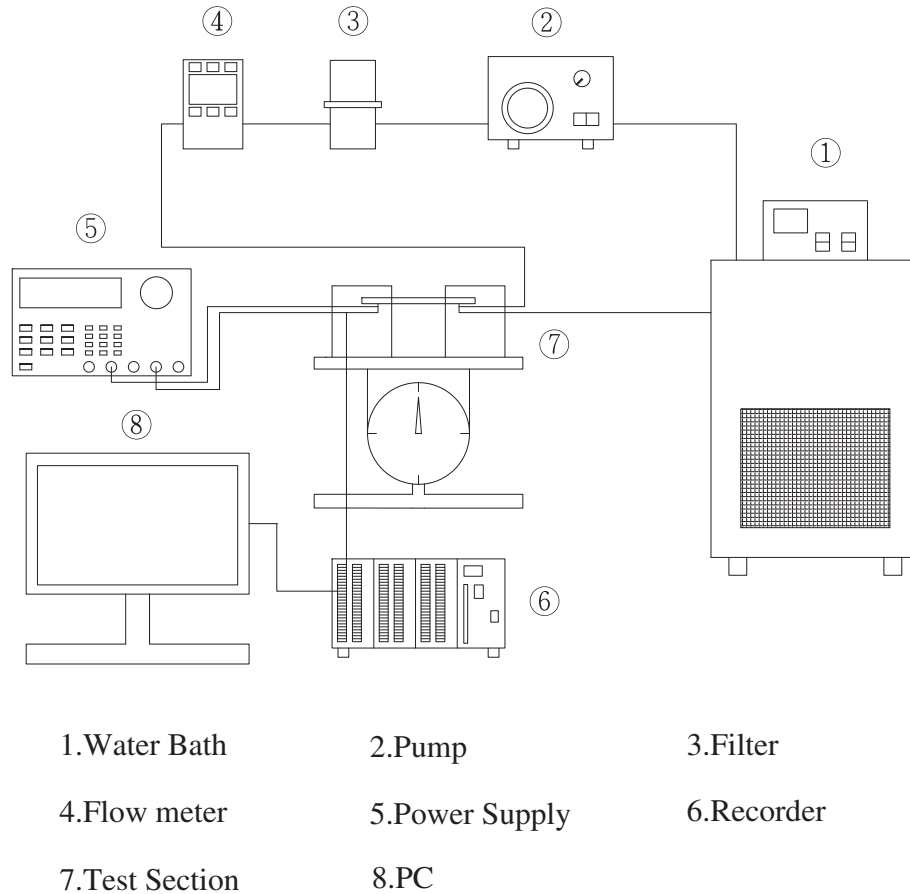


Fig. 1. Schematic of the experimental setup.

where  $\dot{m}$ ,  $c_p$ ,  $T_{w,out}$  and  $T_{w,in}$  represent the mass flow rate, specific heat at constant pressure, outlet temperature, and inlet temperature of chilled water, respectively. The total thermal resistance is obtained from the following equation:

$$R = \frac{T_{e,avg} - T_{c,avg}}{Q_{out}} \quad (2)$$

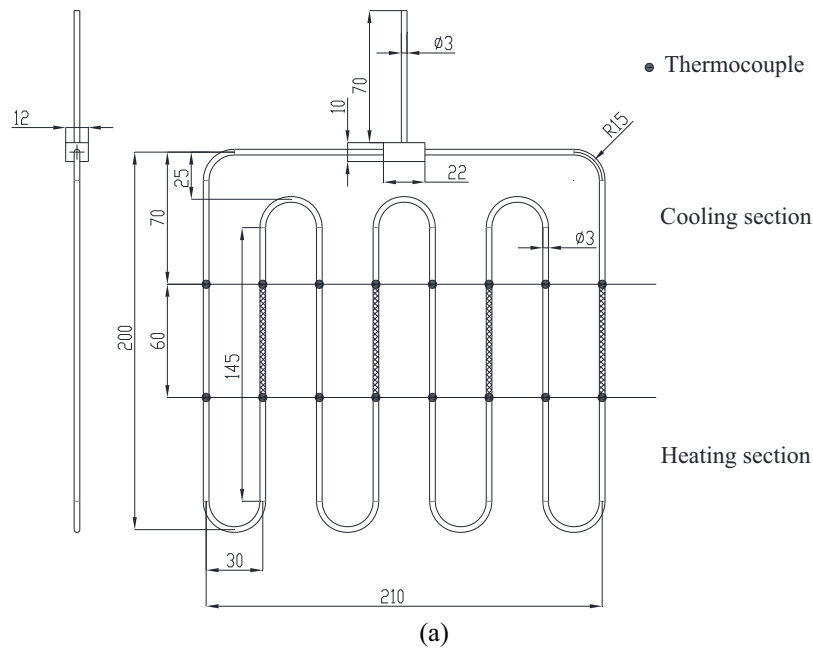
where  $T_{e,avg}$  and  $T_{c,avg}$  is the average temperature of evaporator and condenser, and  $Q_{out}$  is cooling capacity of condenser. Normally the difference amid the heat removal from condenser and the supplied power is less than 5%.

Uncertainties in the reported experimental values were estimated by the method suggested by Moffat [15]. The variables of  $\dot{m}$ ,  $T_{e,avg}$ ,  $T_{c,avg}$ ,  $T_{w,out}$  and  $T_{w,in}$  are used to derive the thermal resistance in Eq. (2). Hence the individual terms are used and combined by a root-sum-square method to obtain the final thermal resistance. The uncertainties range from 5.2% to 18.3% for the cooling capacity of condenser and are from 7.36% to 18.5% for total thermal resistance. Table 1 denotes a list of all the performed tests in this study, the corresponding experimental conditions such as working fluids, heat input, filling ratio and orientation are tabulated.

#### 4. Results and discussion

In order to examine the effects of alternating design on the performance of CLPHPs, about 60% charge ratio of methanol pertaining to the horizontal orientation was first tested. Test results of the thermal resistance vs. heating power for uniform and alternating channels CLPHP are depicted in Fig. 3(a). For the uniform channel CLPHP, the thermal resistance drops considerably provided that

the heating power can stably sustains the circulation of oscillation motion of the vapor slug within the CLPHP. Thus, a sharp decline of the thermal resistance is observed. The results can be made clear from the measured surface temperature variation at the evaporator and condenser as shown in Fig. 3(b) where the corresponding input power is 40 W. As seen in the figure, the alternating design reveals considerable oscillation of the surface temperatures at the evaporator and condenser, suggesting a self-sustaining periodic motion of the vapor slug. On the other hand, the surface temperatures of the uniform design remains about the same. Hence, one can see a dramatic difference in thermal resistance between the uniform and alternating design at  $Q_{in} = 40$  W. The thermal resistances of the uniform CLPHP are between  $0.93 \text{ K W}^{-1}$  and  $2.82 \text{ K W}^{-1}$ , and the alternating design can effectively lower the thermal resistance to  $0.325 \text{ K W}^{-1}$ . For all input powers, it is found that the thermal resistance of the alternating design is lower than those of uniform design. The results are actually associated with the additional unbalanced capillary force for the alternating design as suggested by Chien et al. [14]. It is also interesting to know that the thermal resistance is decreased to a minimum value, and followed by a marginal increase. Notice that despite the alternating design can promote the start-up operation of the CLPHP under horizontal arrangement, the flow circulation caused by the liquid/vapor slug is actually less smooth than that in vertical arrangement. In this sense, without the help of gravity, the generated vapor/liquid slug motion may easily interact with each other and becomes comparatively hindered as input power is further increased. As a consequence, a small rise of thermal resistance emerges, and this is applicable for both uniform and alternating design. A theoretical study carried out by Chiang et al. [16] also unveils the same results.



(a)

(b)

**Fig. 2.** Schematic of the test CLPHP and accessory system; (a) dimensions and the locations of the thermocouples; (b) photos of CLPHPs; (c) dimensions of condenser; (d) the vacuum, fluid degassing, and filling system; and (e) filling system. (Unit: mm).

The foregoing results are applicable for methanol with the horizontal arrangement. For further examination of the working fluids on the performance of the CLPHPs. Various working fluids, including distilled water, methanol and HFE-7100 having a charge ratio of 60% with a vertical upward arrangement was tested and compared. Test results of the thermal resistance vs. heating power subject to various working fluids for the alternating design are plotted in Fig. 4. As expected, the thermal resistance decreases considerably with the rise of heating power and the thermal resistance in vertical operation is much lower than that of horizontal operation when compared to that in Fig. 3(a). This is expected for the imposed gravity force appreciably strengthens the circulation of the liquid/vapor slug. Moreover, with the help of gravity force, the engendered motion of the liquid/vapor slug can operate quite smoothly even at a high input power. As a result, the thermal resistance reveals a continuous decline with the heat input. However,

the variation of the thermal resistance against the heat input for the three working fluids are not the same. As seen in Fig. 4, the HFE-7100 shows the least thermal resistance when the input power is less than 60 W, followed by methanol and water. However, the trend is reversed when the heat input is raised above 60 W. In fact, the difference of the thermal resistance of water with other fluids shows a more pronounced increase when the power is increased further. To explain the different characteristics of the working fluids, the average temperature difference between evaporator and condenser vs. elapsed time with the heating power of 40 W, 100 W and 140 W are plotted in Fig. 5(a) for distilled water and in Fig. 5(b) for HFE-7100. As clearly seen in Fig. 5(a) for distilled water, apparent ‘start–stop’ of the CLPHP operation is seen for a heating power being lower than 40 W, suggesting an unsustainably oscillatory motion and accordingly a higher thermal resistance. On the other hand, with the rise of input power to 140 W,

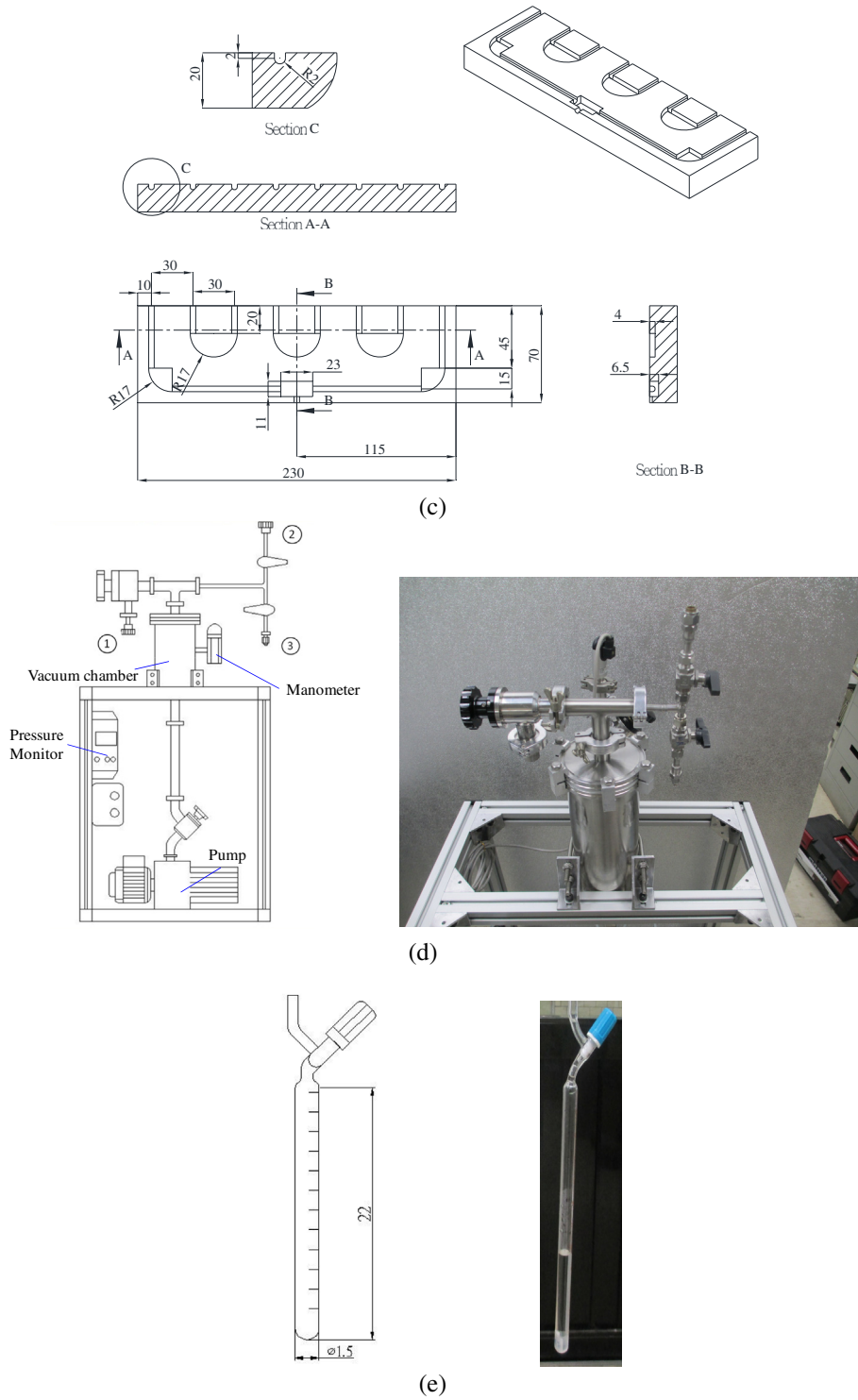
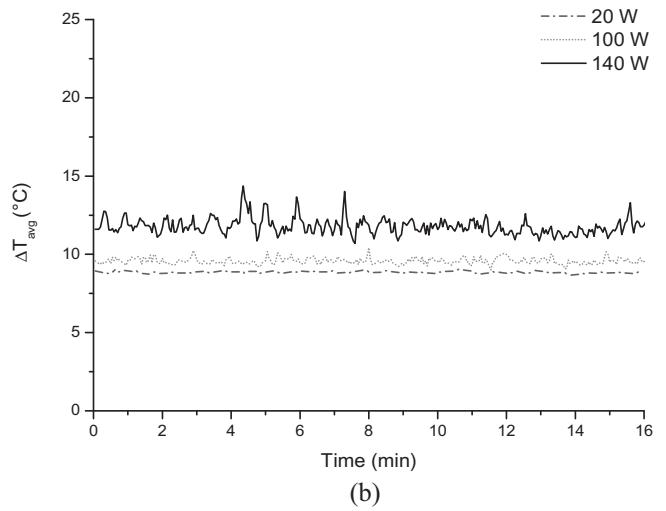
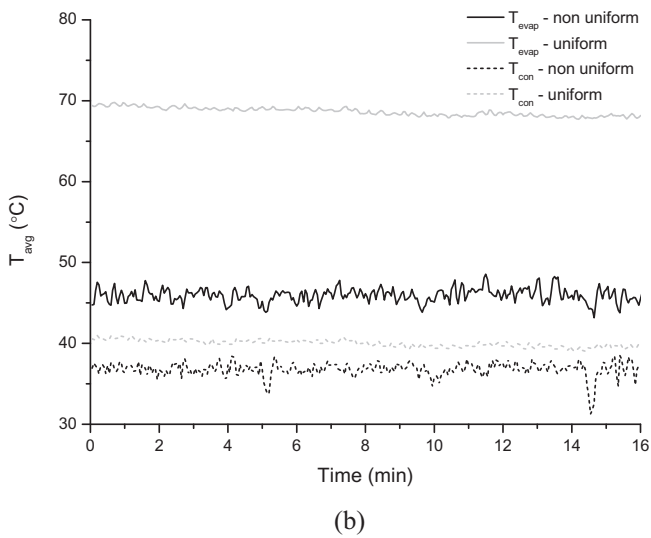
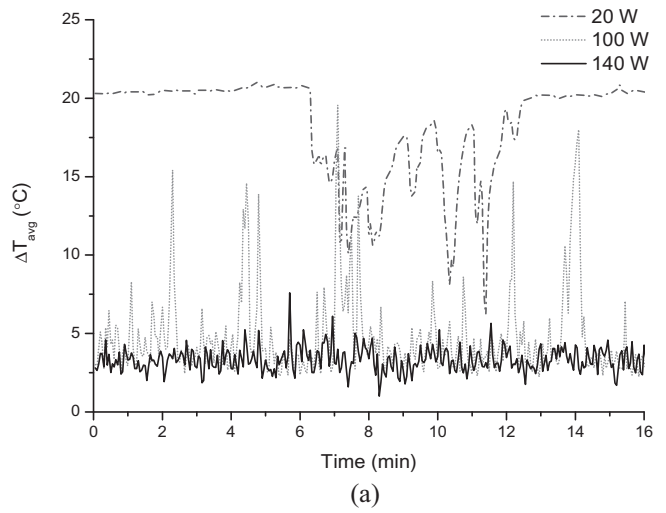
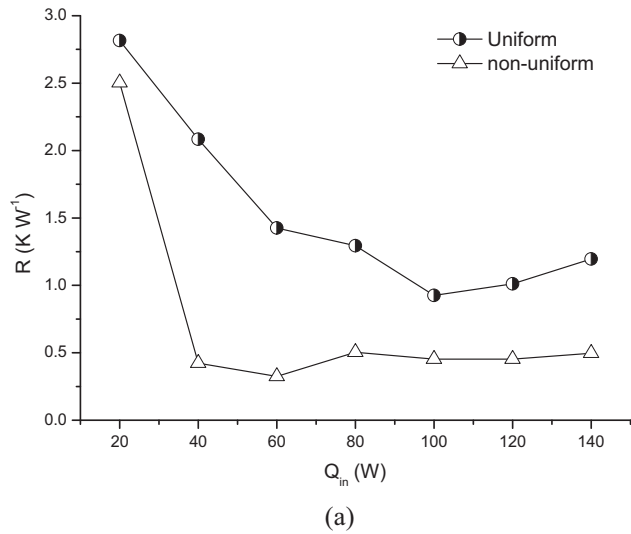


Fig. 2 (continued)

gigantic oscillation of the temperature difference is encountered. This eventually leads to tremendous fluctuations of the flow field and a stable operation of CLPHP and therefore a much smaller thermal resistance. Conversely, sustainable oscillating of temperature difference for the HFE-7100 is observed even for the heat input is as low as 40 W. Thus, one can experience a smaller resistance at a low heat input. However, despite the oscillatory phenomenon remains when the power is further increased, the decline of its magnitude is very small. This is because the flow has already sustained a stable operation and though further increase of heat input will

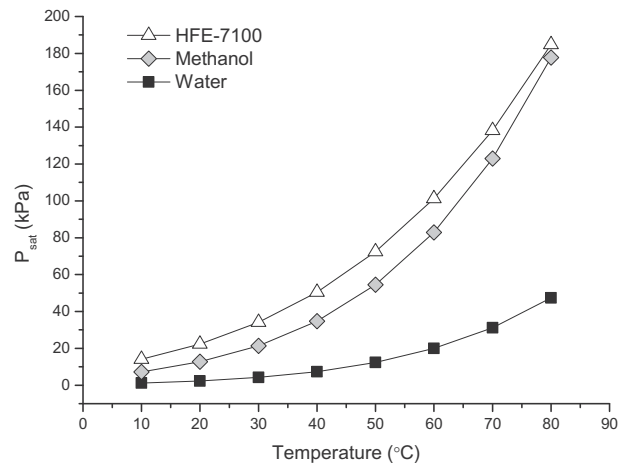
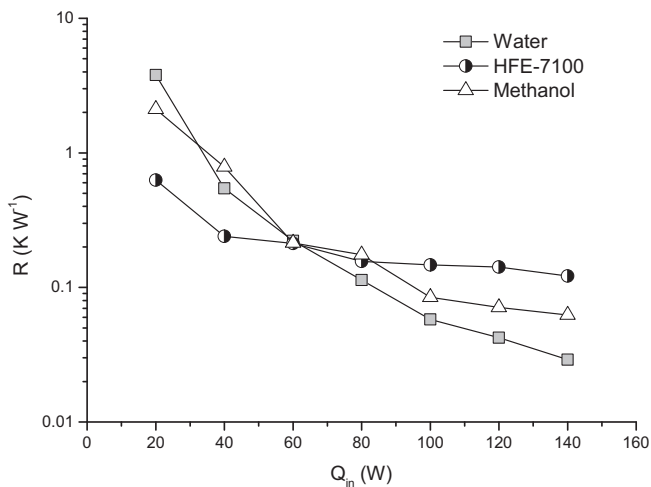
Table 1  
Experimental matrix.

Working fluid	Filling rate (%)	Orientation	Type of tubes	Input power (W)
Methanol	57.6	Horizontal	Uniform	20–140
Methanol	59.8	Vertical	Non-uniform	20–140
Water	59.7	Vertical	Non-uniform	20–140
HFE-7100	62.4	Vertical	Non-uniform	20–140



**Fig. 3.** Performance of the CLPHP at horizontal arrangement subject to uniform and alternating design; (a) thermal resistance vs. heating power for uniform and alternating channels CLPHP. (b) Average evaporator and condenser temperature vs. elapsed time for a heating power of 40 W.

**Fig. 5.** Variation of the average temperature difference of evaporator and condenser vs. elapsed time with alternating design having vertical arrangement for (a) distilled water (b) HFE-7100.



**Fig. 4.** Thermal resistance vs. heating power of various working fluids for the CLPHP having alternating design and vertical arrangement.

**Fig. 6a.** Relation of the saturation temperature with temperature subject to various working fluids.

assist the flow circulation but the corresponding viscous drag also increase notably. Therefore further decline of the thermal resistance is rather small.

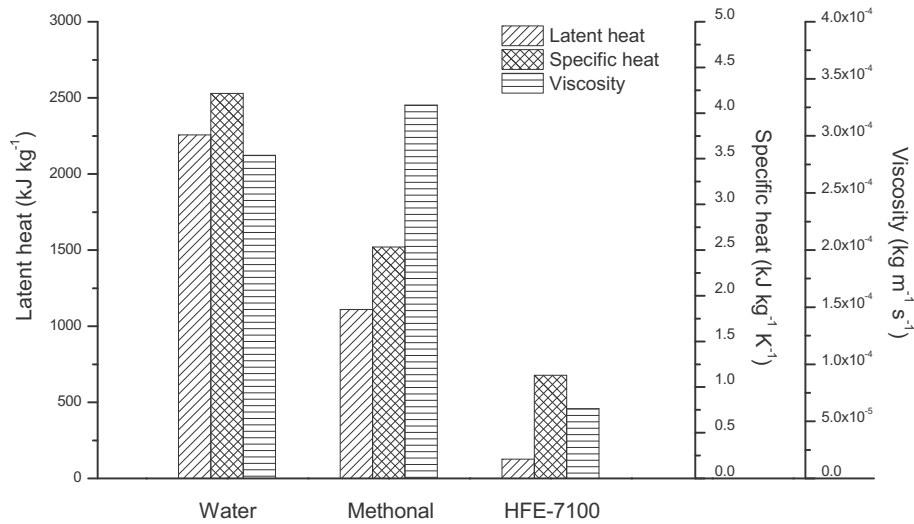


Fig. 6b. Comparison of the thermophysical properties of the various working fluids.

For a further explanation of the variation of thermal resistance subject to the various work fluids,  $(dP/dT)_{sat}$  of various working fluids was plotted in Fig. 6a, and the vaporization latent heat, specific heat and viscosity were plotted in Fig. 6b. Explanations of the foregoing results are resorted to the difference at a lower or at a higher heating power region. Firstly, the key parameter of CLPHPs is to enable sustainably oscillatory flow motion. At a lower heating power region, the highest value of  $(dP/dT)_{sat}$  of HFE-7100 ensures that a small change of evaporator temperature gives rise to a large pressure difference inside the generated bubbles that make way for continuous pumping action of the CLPHP. Secondly, Khandekar et al. [7] suggested that a low latent heat may help the formation and deformation of the vapor slug. From Fig. 6b, one can see that HFE-7100 has the lowest latent heat. Thirdly, the lowest viscosity of HFE-7100 also contributes to a lower frictional pressure drop. In summary of the aforementioned three effects, HFE-7100 shows the prominent thermal characteristics at a lower heat input (<60 W) due to its easier start-up behavior. On the other hand, at a higher heating power, all of the CLPHPs can be started up without difficulty. In this sense, the thermal property is in control of the overall performance. Hence the heat transfer of CLPHPs is primarily due to sensible heat transfer by liquid and latent transport due to evaporation. Consequently, the lowest specific heat of HFE-7100 shown in Fig. 6b will jeopardize the amount of sensible heat transfer [5] and its lower latent heat also deteriorates the overall heat transfer characteristics. In summary of the two results, one can see that water depicts the lowest thermal resistance. Note that the thermal resistances of the working fluids show a continuous drop against heat input. This is because that no apparently dry out is taken place at the present test range.

## 5. Conclusion

In this study, the performance of closed-loop pulsating heat pipes (CLPHPs) subject to the influence of uniform and alternating diameter is reported. The CLPHPs made of copper capillary tubes with 4 turns and 8 straight sections are used for testing, one of the CLPHPs incorporated with a constant tube diameter while the other CLPHP has an alternating tube diameter. The uniform CLPHP is equipped with 2.4 mm ID tube while half of the test tubes of the alternating design is identical to that of the uniform design but the rest of the tubes are compressed to an oval-like configuration with a minor diameter being 1.5 mm. The working fluids include

distilled water, methanol and HFE-7100. Tests are performed with both horizontal and vertical arrangement. Based on the foregoing discussions, the following conclusions are made:

- (1) For the horizontal arrangement, it is found that the alternating design can be started at a rather low heat input, thereby leading to much smaller thermal resistance when compared to that of the uniform design. The thermal resistance reveals a sharp decline once the threshold input power is exceeded. Normally the thermal resistance is decreased with the rise of heat input, and it shows a minimum value at a certain heat input. However, the thermal resistance shows a marginal rise when the heat input is increased further. Both uniform and alternating design reveals similar trend.
- (2) For the vertical arrangement, the thermal resistance is much lower than that in horizontal arrangement. Unlike those in horizontal arrangement, the thermal resistance shows a continuous decline against heat input for all the working fluids provided that no dry-out occurs.
- (3) It is found that the thermal resistance subject to the working fluids for the CLPHP depends on the input power and working fluids. For a low input power (<60 W), the CLPHP with HFE-7100 can be easily starting up with a sustainable operation, thereby resulting in the least thermal resistance. For a high input power, the CLPHP is functional with all the working fluids. In this region, sensible heat along with the latent heat transport plays essential role, therefore distilled water with the highest specific heat and latent heat gives the smallest thermal resistance.

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