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A novel design of pulsating heat pipe with fewer turns applicable to all orientations

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ABSTRACT

This study presents a novel pulsating heat pipe (PHP) concept that is functional even when PHP is with fewer turns and is operated horizontally. Two heat pipes were made of copper capillary tubes with an overall size of 122 mm × 57 mm × 5.5 mm is investigated, one had 16 parallel square channels having a uniform cross-section of 2 mm × 2 mm (uniform CLPHP), and the other had 16 alternative size of parallel square channels (non-uniform CLPHP; a cross-section 2 mm × 2 mm and a cross-section of 1 mm × 2 mm in alternating sequence). Test results showed that the performance of PHP rises with the inclination but the uniform channel CLPHP is not functional at horizontal configuration whereas the proposed non-uniform design is still functional even at horizontal arrangement. The thermal resistance for uniform PHP is relative insensitive to change of inclination when the inclination angle exceeds certain threshold value.

1. Introduction

With the continuous shrinkage in size of the electronic devices, the influence of temperature becomes more and more conspicuous due to the gigantic rise of spreading resistance. Hence effective thermal management has become the prior problem that must be resolved for further shrinkage of electronic devices. Among resolving the enormous spreading resistance caused by the concentrated heat sources, some typical passive methods include heat pipe, oscillating heat pipe, and vapor chamber are proved to be quite effective. Among them, the pulsating (or) oscillating heat pipe (PHP), first introduced by Akachi [1], has demonstrated to be a promising solution for future heat flux management and applications and is especially useful for its comparatively long distance transport ability. Unlike traditional, coaxial heat pipes, the PHP is wickless, featuring a variety of form factors, and is easier to manufacture and possesses fewer operating limitations [2].

It is known that conventional heat pipes with wick structures are widely used to manage thermal problems of electronic products such as laptop computers, servers and power electronic components [3]. The important feature of the heat pipes is its ability to transport a large amount of heat over its length with a small temperature drop. The huge heat transportation in heat pipe is accomplished by liquid evaporation at heat source followed by condensation at the heat sink side, and finally the condensate liquid moves back to evaporator via wick structures using capillary force [4]. Unlike conventional heat pipes, the pulsating heat pipes (PHP) are made from a long continuous capillary tube bent into many turns, and filled more working fluid than conventional heat pipes [5]. One of the outstanding features of PHP is its long distance transporting ability. 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 [6]. The PHP has no internal wick structure and is easier to manufacture with fewer operating limitations [2].

Generally, typical PHPs can be designed at least three ways, namely open loop system, closed loop system, and closed loop pulsating heat pipe with additional flow control check valves [7]. In closed loop PHP, working fluid is circulated to augment heat transfer. Therefore, closed loop PHP has a better heat transfer performance and most research work were associated with this design [8]. However, PHP with fewer turns fails to operate at horizontal orientations for lacking gravity assistance [9]. This is a critical limitation of PHPs for horizontal arrangement is quite common in applications. Although an addition of check valves [10] can make the working fluid move in a specific direction or increased numerous turns [7]. However, it is difficult and expensive to install these valves when miniaturization of the PHP device is concerned. In the meantime, significant rise of turns may lead a considerable rise of size which may not be applicable in practice. Therefore, this study proposes a novel concept by variation of channel diameter of the PHP. With the introduction of alternative diameter size within the PHP, additional un-balancing capillary force may prevail amid adjacent channels, thereby promoting the fluid motion alongside

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Nomenclature

C_p	specific heat (J kg^{-1} K^{-1})
D	diameter (m)
FR	charge ratio (V_{liq}/V_{tot})
ID	inner diameter (m)
$\dot{m}$	flowrate (kg s^{-1})
P	pressure (Pa)
PHP	pulsating heat pipe
Q_{input}	supplied input power (W)
Q_{out}	heat removed from of the condenser (W)
R	thermal resistance (k W^{-1})
T	temperature (°C)
ΔT	$T_e - T_c$ (°C)
V	volume (m^{-3})

Greek symbols

σ	surface tension (N m^{-1})
μ	dynamic viscosity (kg m s^{-1})
ρ	density (kg m^{-3})

Subscript

e	evaporator
c	condenser
$crit$	critical
liq	liquid
sat	saturation
tot	total
vap	vapor

the PHP. The proposed concept is applicable even when the PHP is operated horizontally.

2. Experimental apparatus

The schematic of the experimental apparatus and test section is shown in Fig. 1. The experimental setup comprises of an evaporator section, a water loop for condenser, along with measurement devices and a data acquisition system. The evaporator section is made of copper block (15 mm × 40 mm) for heating the work fluid. During the tests, electric power supply provided power input to the heater. The bakelite board is installed beneath the copper block in order to minimizing the heat loss. For the water loop for condenser, the low temperature water flow is maintained via a thermostat with its flowrate being measured by a flow meter.

Thermocouples are used to measure the surface and fluid temperature. A total of nine 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.

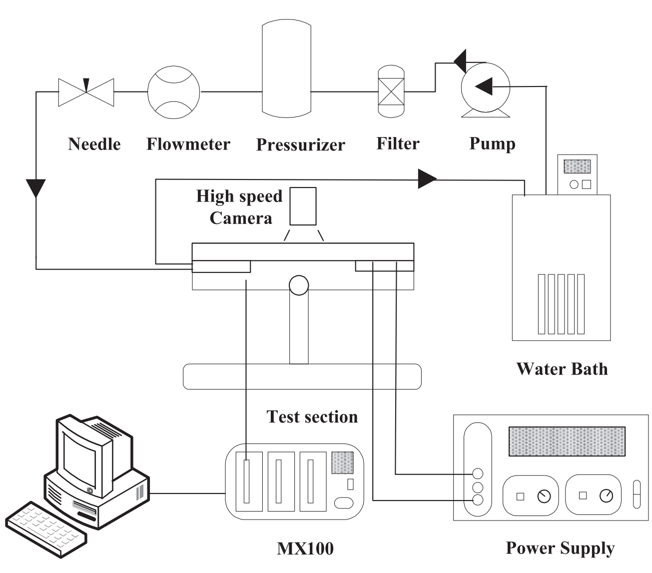


Fig. 1. Experimental set up.

The locations of the thermocouples are distributed below the surface of test section as schematically shown in Fig. 2. 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). The data acquisition system then transmitted the converted signals through Ethernet interface to the host computer for further operation.

In this study, two CLPHPs were made and tested, the corresponding channels patterns are (a) with a uniform diameter; and (b) with a non-uniform diameter. Detailed geometries and dimensions of tested CLPHPs are shown in Fig. 3. The CLPHPs samples are made of copper via precise machining, and the dimension of the CLPHPs samples is 122 × 57 × 5.5 mm^3. The uniform CLPHP had 16 parallel square channels with cross-section 2 mm × 2 mm, and the non-uniform CLPHP had 8 parallel square channels with a cross-section of 2 mm × 2 mm whereas the non-uniform one is with 8 parallel square channels having a cross-section of 1 mm × 2 mm. The above mentioned CLPHPs are located above a well-fitted bakelite. A transparent glass is placed on top of the test section for flow visualization. Observations of flow patterns are obtained from images produced by a high speed camera of IDT X-Stream XS-3. The high-speed camera can be placed at any position above the square channel.

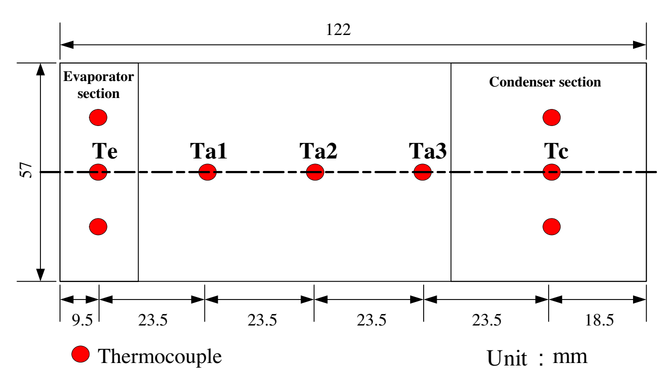
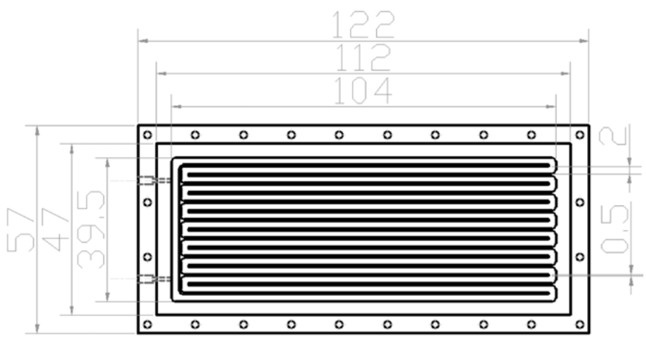
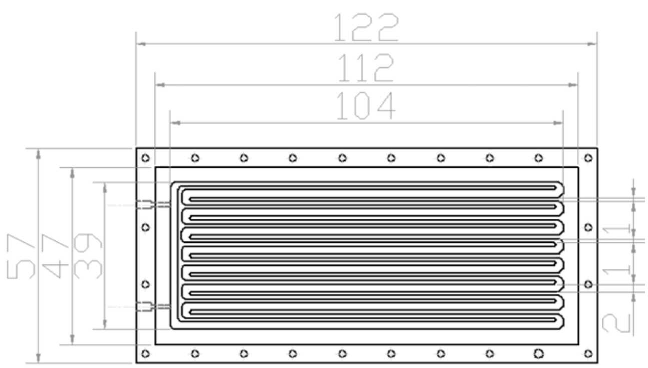


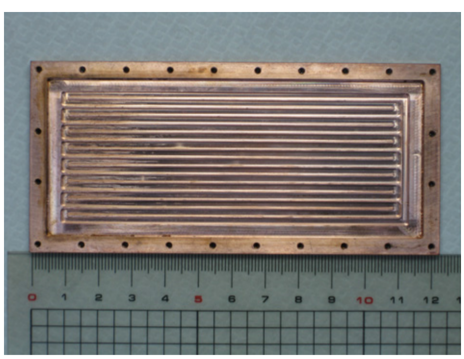
Fig. 2. Schematic of the locations of the thermocouples.



(a) Detailed dimension of the test section of uniform diameter



(b) Detailed dimension of the test section of various diameter



(c) Photo of the test section of uniform diameter

Fig. 3. Configuration of the CLPHP test section.

3. Data reduction

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

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

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

$$R_{total} = \frac{\Delta T}{Q} = \frac{T_e - T_c}{Q_{actual}} \tag{2}$$

where T_e and T_c is the average temperature of evaporator and condenser, and Q_{actual} is represents the average of the heat removal form of the condenser and supplied input power ($Q_{input} = I \times V$). 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 from the single sample analysis suggested by Moffat [11]. The highest uncertainties are 1.86% for the cooling capacity of condenser and 2.26% for total thermal resistance.

4. Results and discussion

For the design of inner diameter of CLPHPs, the best range can be calculated by using the followed equation [12].

$$0.7\sqrt{\frac{\sigma}{(\rho_{liq} - \rho_{vap}) \cdot g}} \leqslant D \leqslant 1.8\sqrt{\frac{\sigma}{(\rho_{liq} - \rho_{vap}) \cdot g}} \tag{3}$$

where σ, g, ρ_{liq}, and ρ_{vap} represent the surface tension, gravitational constant, density of liquid and Density of vapor. For water, the appropriate range of inner diameter of CLPHPs ranges from 1.92 mm to 4.94 mm. In this study, the hydraulic diameter of uniform diameter CLPHP is 2 mm.

In order to examine the effects of operating conditions on CLPHPs, various charge ratio and inclinations was tested. The tested conditions of CLPHPs contain distilled water having charge ratios of 40%, 50%, 60% and 70%, respectively. A total of four inclinations for the CLPHPs are conducted during the experiments, with the inclination angle (θ) being 0° (horizontal), 30°, 60°, and 90° (vertical upward). During the experiments, the power is gradually increased to study the response of the CLPHPs.

Test results of thermal resistance vs. heating power of various inclinations having charge ratio of 70% for uniform and non-uniform channels CLPHP are plotted in Figs. 4 and 5. For the uniform channel CLPHP, the thermal resistance decreases considerably with the rise of heating power. This is mainly attributed to a higher circulation rate of the working fluid which results in heat transfer enhancement. At a lower heat input, gravity adversely affected the performance and only intermittent pulsations (periodic 'start–stop' behavior as described by Yang et al. [3]) were observed (Hemadri et al. [13]). On the other hand, the thermal driving forces could overcome the adverse effect of gravity with further addition of heat, leading to performance improvement and self-sustained pulsations.

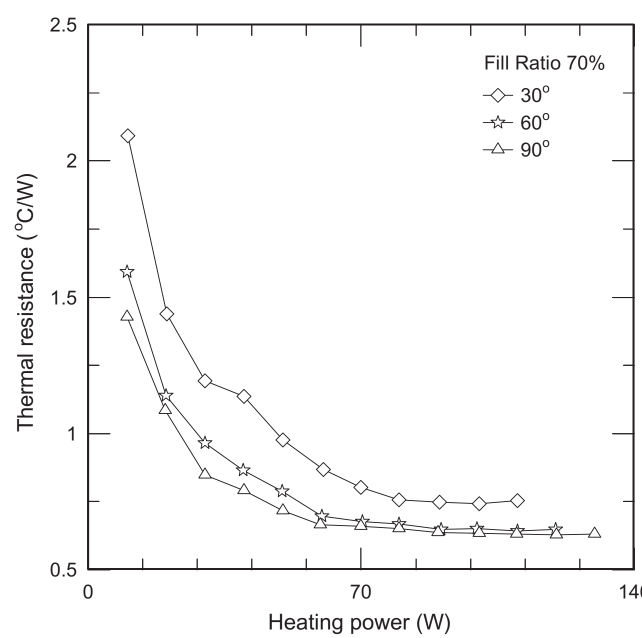


Fig. 4. Thermal resistance vs. heating power of various inclinations for uniform channels CLPHP.

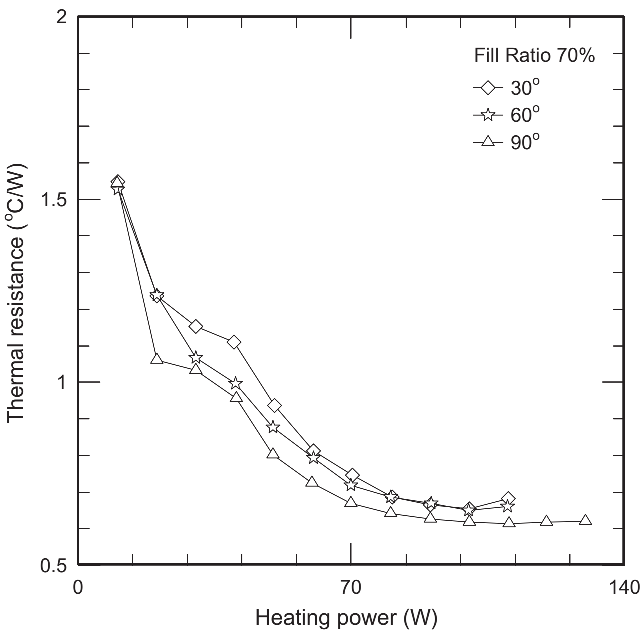


Fig. 5. Thermal resistance vs. heating power of various inclinations for non-uniform channels CLPHP.

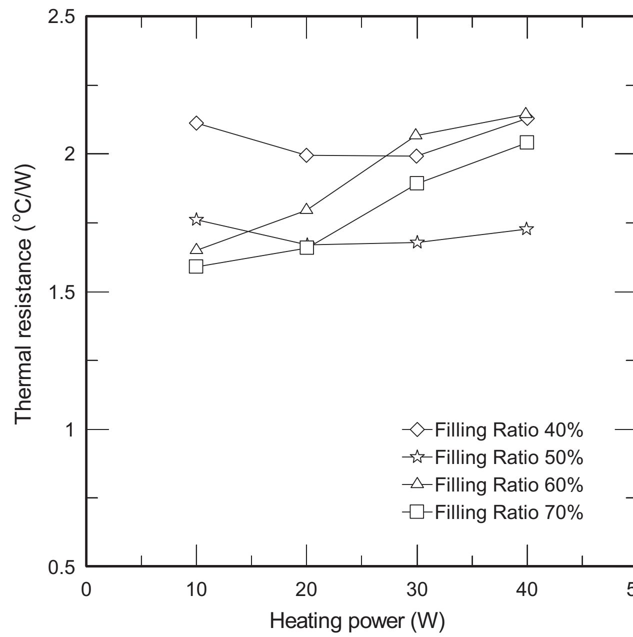
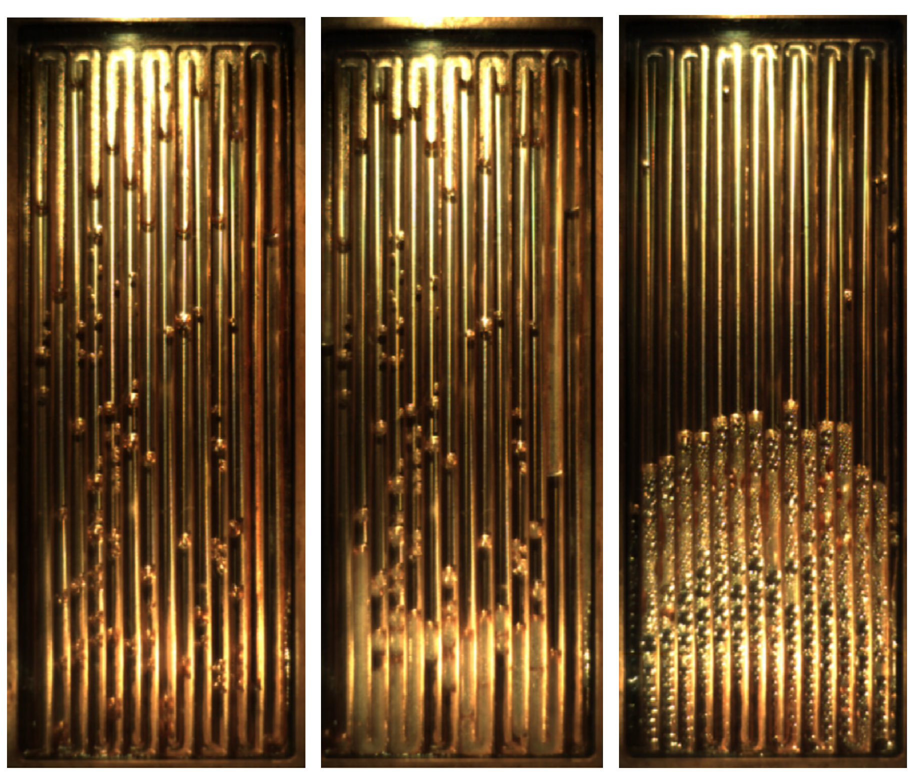


Fig. 6. Thermal resistance vs. heating power of horizontal for uniform channels CLPHP.

On the other hand, vertical arrangement shows the smallest thermal resistance but the 60° inclination shows comparable thermal resistance with that of the vertical arrangement. However, one can see a detectable rise of thermal resistance when the inclination angle is reduced further to 30°. It is well understood that the gravity helps to boost PHPs, thereby reducing the thermal resistance when the PHP is placed vertically. At a moderate mass flux and vapor quality, the major two-phase flow pattern within microchannels is normally elongated bubble, this is also applicable in the present visual observation. As pointed out Bonnecaze et al. [14], the bubble velocity is a combination of two components. The first component is due to the buoyancy force whereas the second component is associated with non-stationary liquid slug which is traveling with no-slip condition prevailing at the surface. Their



(a) (b) (c)

Fig. 7. High speed video picture of (a) initial state (b) 1 min of heating (c) steady state of horizontal orientation for uniform channels CLPHP.

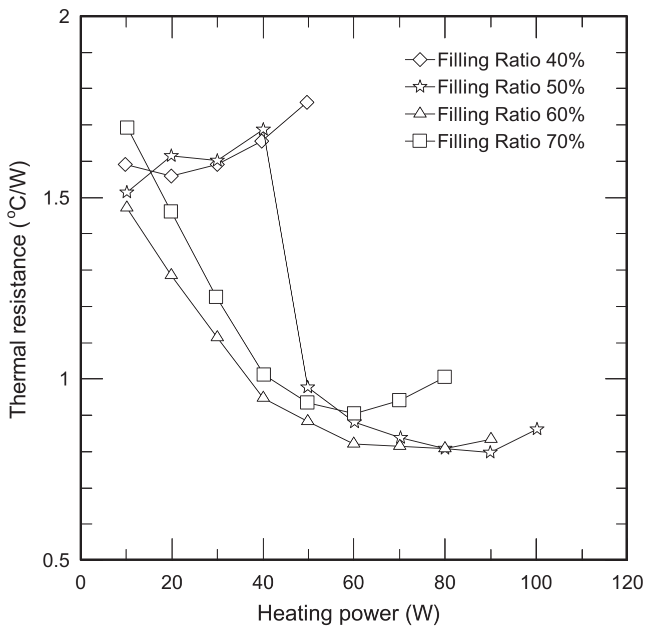


Fig. 8. Thermal resistance vs. heating power of horizontal for non-uniform channels CLPHP.

theoretical calculation indicated that an increase of bubble rise with respect to inclination angle, and a peak value occurred at an inclination angle amid vertical and horizontal arrangement. In fact, the role of the buoyancy plays a role to strength the length of vapor slug and increases the bubble velocity accordingly. At the same time, the flow inertia for upward arrangement is concurrent with the buoyancy force, thereby reinforcing the enhancement effect. The results had been confirmed with some visual observations. For example, Cheng and Lin [15] and a recent study by Wang et al. [16], a maximum heat transfer performance occurs at a specific inclination rather than vertical or horizontal configuration. In this sense, one can find a relative small difference in terms of thermal resistance between 60° and vertical arrangement while a much larger deterioration is encountered for a 30° arrangement. For the non-uniform CLPHP, the effect of inclination shows somehow a less impact on the thermal resistance. This is expected, for the main purpose of the non-uniform tube diameter design is to boost the PHPs under horizontal arrangement. By introducing additional un-balancing surface tension to amend the loss of gravity contribution, the originally strong influence caused by gravity is thereby reduced. As a consequence, the effect of inclination is lessened. However, it should be mentioned that the thermal resistance for the non-uniform CLHP is slightly increased as compared to that of uniform configuration. This is associated with the larger viscous force inherited with the small size tube.

However, when the arrangement is horizontal, variation of the thermal resistance for uniform CLHP is significant different from other inclinations as shown in Fig. 6. Firstly, the thermal resistance is rather large, implying PHP is not functioned. Yet the thermal resistances are comparatively insensitive to change of heating power, and are especially pronounced at a lower charge ratio like 40% or 50%. For a further explanation of the test results, typical photos showing the progress of internal flow are shown in the Fig. 7. The photos are taken from the initial state, one minute of heating, and final steady state of horizontal for uniform channel CLPHP at a heating power of 30 W and a filling rate of 70%. As shown in pictures, the oscillations occurred only at some short time periods and the oscillation is completely stopped at steady state. Without the help of flow oscillations, heat transferred takes place merely by conduction from evaporator section to condenser section. Therefore, the uniform channel CLPHP shows a very poor heat transfer performance at horizontal orientation.

To tackle this problem, we have proposed a non-uniform channel CLPHP to circulate working fluid. Zhang and Faghri [8] had

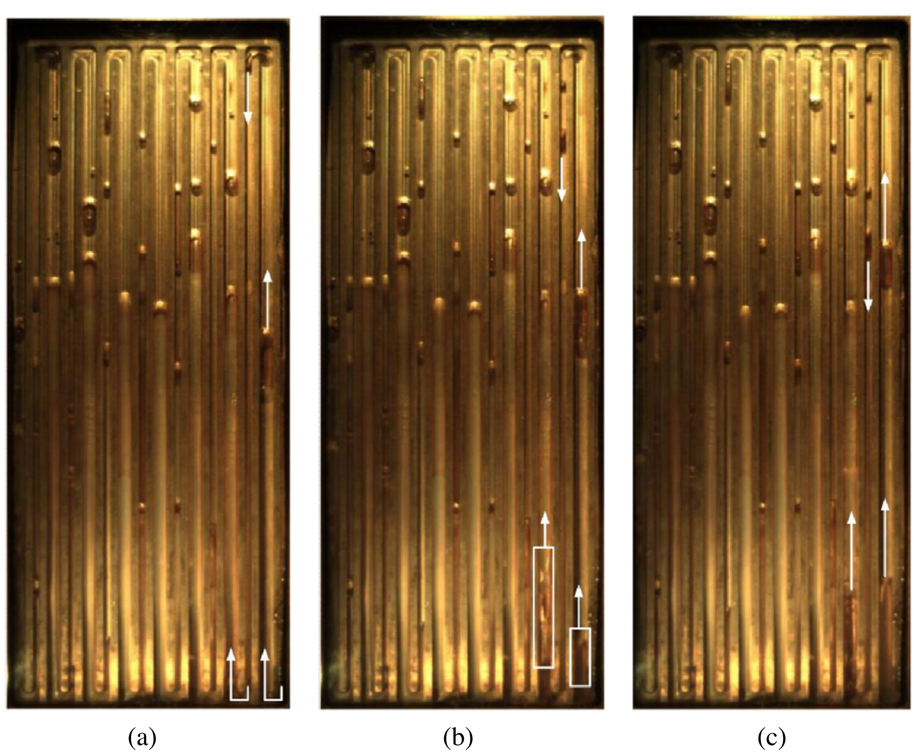


Fig. 9. High speed video picture of (a) 0 s (b) 0.4 s (c) 0.8 s of horizontal orientation for non-uniform channels CLPHP of FR = 70%.

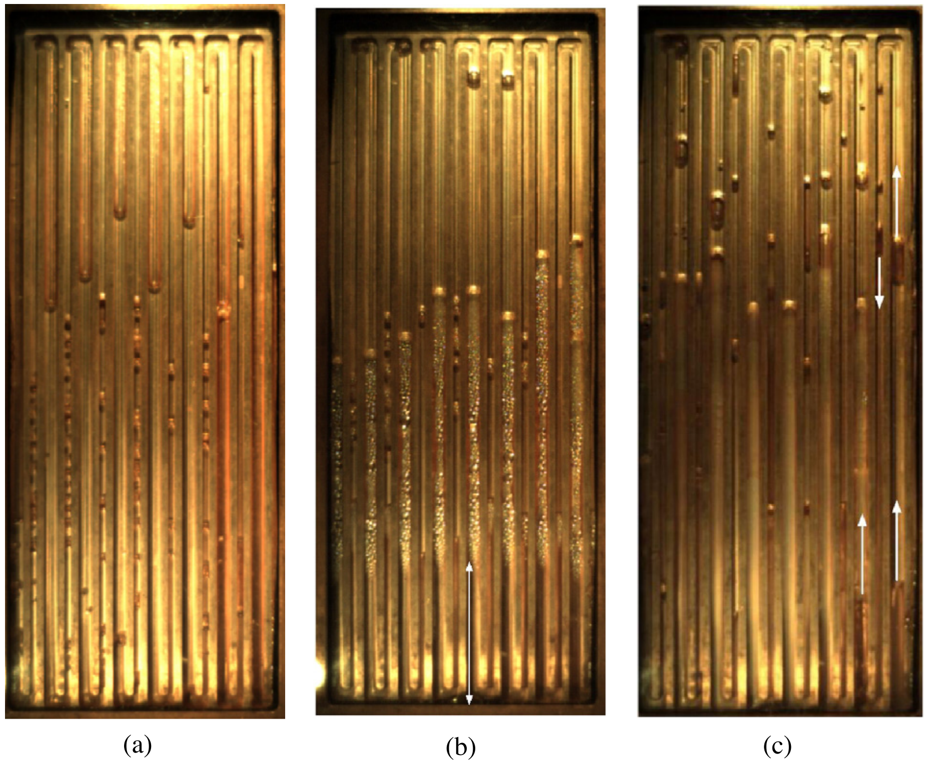


Fig. 10. High speed video picture of (a) initial state (b) steady state of 20 W (c) oscillations of 50 W for non-uniform channels CLPHP at horizontal orientation of FR = 50%.

made a thorough review about PHP, one of their findings is that in horizontal or top-heating mode the PHP is not workable with only fewer turns. Hence the proposed idea is to introduce additional un-balancing capillary force alongside the PHP. On the other hand, the un-balancing frictional resistance also prevails pertaining to the unequal channel diameter, and it also helps promoting the fluid flow motion. As a consequence, the resultant force recovers the loss gravity force and the PHP is found still functional even when the PHP is placed horizontally. Test results of thermal resistance vs. heating power of various inclinations having a charge ratio of 70% for non-uniform channels CLPHP are plotted in Fig. 8. The results show that the thermal resistance decreases with a rise of higher charging rate and heating power. And the corresponding high speed flow visualizations of CLPHP are shown in Fig. 9. The pictures show that the observations at a reference time of 0, 0.4 and 0.8 second of heating with horizontal configuration for uniform channels CLPHP at heating power of 50 W and a charge ratio of 50%. As shown in picture, the oscillations can take place and sustain itself successfully at the horizontal orientation. The working fluid of non-uniform channel CLPHP can continuously circulate and results in an effective heat transfer by un-balanced capillary force and flow resistance amid various channels.

However, the experimental results also show that even with the help of un-balancing capillary force the non-uniform CLPHP is still sensitive to the charge ratio and input power. As shown in Fig. 8, the PHP with a charge ratio of 40% is not functional even at an elevated input and the PHP can be initiated only when the input power is as high as 50 W at a charge ratio of 50%. The results can be made clear from the visual observation shown in Fig. 10(b). At a lower heat input of 30 W, the generated vapor condenses subsequently to form small droplet adjacent to the heating section. The small droplets did not show apparent conglomeration due to the comparatively low charge ratio. Hence some dry out portion is seen at the heating section. As a result, no stable oscillations can be maintained at a heating power of 30 W. Conversely, a stable oscillation is encountered at a heating power of 50 W. This is because sufficient droplets conglomerate to form liquid slug to maintain a stable liquid circulation. Analogous results about the influence of charge ratio are also encountered for the uniform arrangement.

5. Conclusions

This study proposes a novel design of PHP having a non-uniform channel configuration. The concept is to introduce the additional unbalancing capillary force to resolve the problems of fewer turns of PHPs subject to horizontal configuration. Both visual observation and heat transfer measurements are conducted for flat-plate closed-loop pulsating heat pipes (CLPHPs). Two heat pipes were made of copper capillary tubes with an overall size of 122 mm×57 mm×5.5 mm, one had 16 parallel square channels with cross-section 2 mm×2 mm, called uniform CLPHP, and the other one, called non-uniform CLPHP, had 8 parallel square channels with a cross-section of 2 mm × 2 mm and 8 parallel square channels having a cross-section of 1 mm × 2 mm. The working fluid is distilled water.

Test results showed that the thermal resistance decreases with the rise of heating power due to the rise of circulating speed. For the increase of inclinations among the test CLPHPs, the thermal resistance decrease with the rise of inclinations due to gravity effect. However, it appears that the uniform channel PHP is more sensitive to inclination especially when the inclination angle is small and it is not functional at a horizontal configuration. On the other hand, the proposed non-uniform channel CLPHP is functional to all inclinations provided the charge ratio is sufficient (above 50%).

Acknowledgements

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