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Experimental study on the thermal performance of vertical closed-loop oscillating heat pipes and correlation modeling

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HIGHLIGHTS

- ▶ An experimental study is performed on the vertical closed-loop oscillating heat pipes.
- ▶ Inner diameters show different effects on the thermal performance.
- ▶ There is an optimal evaporator length with respect to the lowest thermal resistance.
- ▶ We develop an empirical correlation to predict the thermal performance.

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ABSTRACT

Experimental studies were performed to investigate the thermal performance of three closed-loop oscillating heat pipes (CLOHPs) operating at the vertical bottom heat mode with heating power input in a range of 15–127 W. The tested CLOHPs are all made from copper capillary tubes with inner diameters (IDs) of 1.2, 2, and 2.4 mm. Two working fluids, pure water and ethanol, were used with filling ratios of 40%, 50%, and 60% by volume. The evaporator of each CLOHP was electrically heated with alterable lengths, while the condenser was liquid cooled with a constant length. Experimental results show that the thermal performance of the CLOHPs depends on the conjugation effects of working fluid, filling ratio, inner diameter, evaporator length, and heating power input. The 2 mm ID and 2.4 mm ID CLOHPs had better thermal performance when charged with water as compared with ethanol, while ethanol was preferred for the 1.2 mm ID CLOHP. The thermal performance of these CLOHPs was enhanced at the relatively lower filling ratios (40% and 50%). An optimum evaporator length corresponding to the lowest thermal resistance was proved. Finally, an empirical correlation based on 510 sets of available experimental data both from the present study and other literatures was proposed to predict the thermal performance of vertical CLOHPs. The proposed correlation agreed with the experimental data within a deviation of approximately ±40%.

1. Introduction

Heat pipes are highly effective two-phase heat transfer devices. The ultra-high thermal conductivity of a heat pipe facilitates heat to be transported at high efficiency over considerable distances with minimum heat loss. Consequently, the use of heat pipes is becoming widespread in various energy consumption systems [1–4] and accepted as a good choice for saving energy and preventing ever-increasing global warming [5].

The oscillating heat pipe (OHP), or pulsating heat pipe (PHP), is a relatively new member in the heat pipe family. As compared to conventional heat pipes, there is no additional capillary structure in OHPs, suggesting that countercurrent flow between the liquid and vapor almost not appear inside them. Consequently, the OHPs are not subjected to the entrainment limit affecting the heat transport capability in conventional heat pipes or even relatively new micro-grooved heat pipes [6–9]. After evacuating the serpentine tube and then partially filled with a phase-change working fluid, liquid slugs and vapor plugs could be randomly distributed in an OHP, owing to the very small inner diameter [10]. Although OHPs are passive two-phase heat transfer devices, the unique 'self-excited' thermally-driven oscillating motion of slugs/plugs allows high heat transfer rate from high-density power devices and turns it into the kinetic energy. Due to the simple structure and excellent heat transfer characteristics [11–14], OHPs have received much attention in the last decade and considered as a viable alternative in the area of highly-efficient compact heat exchangers.

The potential applications of OHPs in diverse areas, such as heat recovery [15], drying [16], solar energy collecting [17,18], cooling

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Nomenclature

c_p	specific heat (J kg^{-1} K^{-1})
Bo	Bond number $\left(= D_i\sqrt{g(\rho_l-\rho_v)/\sigma}\right)$
D	tube diameter (mm)
g	gravitational acceleration (m s^{-2})
h_{lv}	latent heat of vaporization (J kg^{-1})
Ja*	modified Jacob number $\left(=\frac{\phi c_p \Delta T_{e-c}}{(1-\phi)h_{fg}}\right)$
k	thermal conductivity (Wm^{-1} K^{-1})
Ku	Kutateladze number $\left(=\frac{q''}{h_{fg}\rho_v\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]^{0.25}}\right)$
L	length (m)
$\dot{m}$	mass flow rate (kg s^{-1})
n	number of experimental data set
Mo	Morton number $\left(=\frac{g\mu^4(\rho_l-\rho_v)}{\rho_l^2\sigma^3}\right)$
p	pressure (Pa)
Pr	Prandtl number (=$\mu c_p/k$)
q''	heat flux (Wm^{-2})
Q_{in}	heat input (W)
Q_{out}	heat output (W)
T	temperature (K)

Greek symbols

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

Subscripts

a	adiabatic section
c	condenser section
c, in	inlet at the cooling chamber
c, out	outlet at the cooling chamber
cor	correlation
e	evaporator section
eff	effective
exp	experiment
i	inner
l	liquid
pre	prediction
v	vapor

of electronics [19–21], LED [22] and fuel cell [23], largely depend on the accurate and reliable prediction of thermal performance for different thermophysical/geometrical parameters and boundary conditions. So the theoretical model correctly predicts the performance of an OHP is fundamentally important and helps to design a suitable OHP. Nevertheless, due to the complicated operating mechanism resulted from the inherent highly complex and non-linear thermo-hydrodynamics behaviors [24,25], most of the proposed mathematical models for OHPs have rather limited applicability and, therefore, is still a difficulty issue. Among all the existing theoretical methods, the empirical correlation seems to be one of the most promising solutions by now and some works have been performed recently by researchers [26–29]. However, it is really difficult to establish a recognized and universal correlation available for predicting the thermal performance of OHPs due to the complicated physical mechanism and various experimental conditions. Therefore, more reliable correlations with respect to some specified conditions, such as OHPs placed at horizontal or vertical position, are needed in favor of the design and application currently.

In this study, the effects of some important parameters (working fluid, heating power input, filling ratio by volume, inner diameter, and evaporator length) on the thermal performance of vertical closed-loop OHPs (CLOHPs) were experimentally investigated. Then, with large number of available experimental data sets both from the present study and other literatures, an empirical power-law correlation based on dimensionless groups, such as the Bond number, Morton number, and Prandtl number, describing the flow and heat transfer characteristics, was developed to predict the thermal performance of vertical CLOHPs.

2. Description of the experiment

2.1. Experimental setup

In the experiment, three CLOHPs made by cooper capillary tubes with inner diameters (IDs) of 1.2, 2, and 2.4 mm and outer diameters (ODs) of 2, 3, and 3.2 mm, respectively, were used. The total length of each tube is about 3.1 m. Fig. 1 illustrates the experimental setup, mainly consisting of a CLOHP assembly, a multi-channel data acquisition system, a DC power supply unit and a cold bath. The six-turn CLOHP was fabricated by bending a copper capillary tube back and forth parallel to itself. The two ends of the tube were connected by a T-connector to form a closed loop, dividing into three parts: evaporator, adiabatic and condenser sections. The length of the condenser was kept at a constant of 70 mm, but the lengths both of the evaporator and adiabatic sections varied and maintained a value of 150 mm (i.e., $L_e + L_a$ = 150 mm). The out surface of the evaporator section was electrically heated by a DC power supply, and the condenser section, welded onto a cooling chamber, was cooled by the mixture of 50% glycol–water pumped from the cold bath. Note that both of the evaporator and adiabatic sections were well thermally insulated with aluminum foil enveloped fiberglass insulation to minimize the heat loss from these two sections to the ambience, and then the heat applied to the heat pipe was determined in terms of the Ohm's law. The inlet temperature, outlet temperature, and flow rate of the cooling liquid flowing through the cooling chamber attached to the condenser were measured by two T-type thermocouples (OMEGA) and a flow meter, respectively, from which the heat output from the condenser could be calculated by:

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

where $\dot{m}$ is the mass flow rate of cooling mixture, c_p is the specific heat, and $T_{c,out}$ and $T_{c,in}$ are the temperatures at the outlet and inlet parts of the cooling chamber, respectively.

Strictly speaking, the heating power input at the evaporator is a little larger than the net heat absorbed by cooling mixture at the condenser due to the heat loss. For each CLOHP, the thermal balance analysis made between the heating power input at the evaporator and the heat output at the condenser demonstrated that the maximum relative heat loss from the evaporator and adiabatic sections to the ambience is less than 6.7%.

The evaporator of the CLOHP was placed vertically and heated from the bottom, i.e., in the vertical bottom heat mode. To measure the wall temperatures of the CLOHP, another fourteen T-type thermocouples were soldered at different locations along the tube, with six being placed at the evaporator, six being placed at the

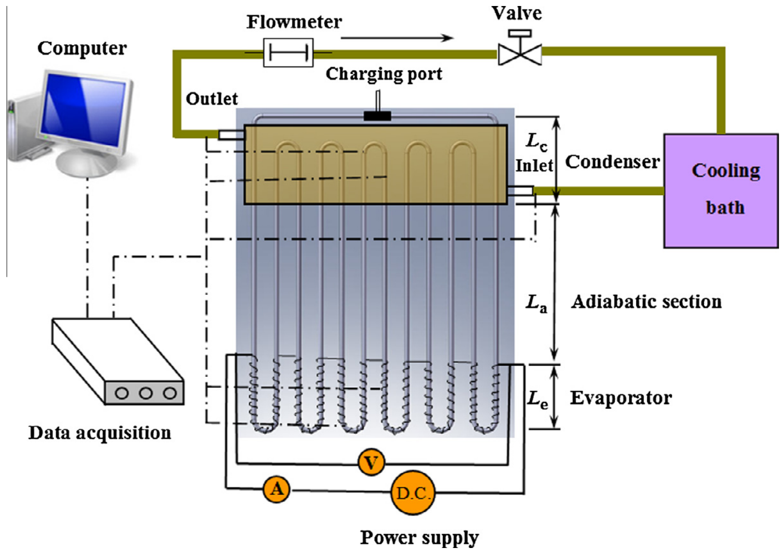

Fig. 1. Schematic of experimental setup.

condenser, and other two being placed at the adiabatic section. All temperature measurements (including wall temperatures of the CLOHP and inlet/outlet temperatures of the cooling chamber) were collected by a data acquisition system (34970A, Agilent). The average temperatures of the evaporator and condenser could be calculated subsequently.

Before the experiment, each CLOHP was firstly evacuated by a vacuum pump and then partially charged with a degassed liquid. Two working fluids, pure water and ethanol, were used, respectively, at moderate filling ratios by volume ranging from 40% to 60%. During the experiment, the heating power input was stepwise increased from a low value to relatively higher levels until the evaporator temperature exceeded 100 °C in accordance with the practical operation for medium temperature applications (75–120 °C) such as solar cooling or industrial process heat and upper limit temperature applications (80–125 °C) such as cooling for electronics, LED, or PEM fuel cell. The temperature data at the evaporator and condenser was recorded while reaching a quasi-steady state. All tests were conducted at an ambient temperature of 25 °C.

2.2. Data reduction and uncertainty analysis

The overall thermal resistance, defined as a ratio of the temperature difference between the evaporator and condenser to a heating power input, is widely applied to evaluate the thermal performance of a CLOHP, and a lower value of thermal resistance usually means higher thermal performance. The overall thermal resistance of a CLOHP is defined as

$$R = \frac{T_e - T_c}{Q_{in}} \tag{2}$$

where T_e and T_c are the average wall temperatures of the evaporator and condenser, respectively. Q_{in} in Eq. (2) is the heating power input applied to the evaporator, which can be calculated by follows:

$$Q_{in} = UI \tag{3}$$

As mentioned in Section 2.1, the heat loss between the evaporator and the condenser of each CLOHP was less than 6.7%. Thus, the uncertainty of Eq. (3) due to heat loss is 6.7%.

Table 1 gives the maximum uncertainties of the main parameters in this study. The uncertainties of the direct measurement parameters such as T, U, and I were synthesized by the system

Table 1
Uncertainties of the main parameters.

Parameters	T_e	T_c	U	I	Q	R
Uncertainties (±)	0.21%	0.35%	0.43%	0.46%	7.59%	8.15%

uncertainty e_s from the precision of instruments (thermocouple, multimeter) and the random uncertainty e_r from the repeatability of data as follows:

$$e = \sqrt{e_s^2 + e_r^2} \tag{4}$$

The uncertainties of the indirect measurement parameters such as T_e, T_c, Q and R were obtained with the corresponding definitions as aforementioned according to error propagation principle. Note that the uncertainty induced by the heat loss (6.7%) was also included in the uncertainty estimation of Q and R.

3. Results and discussion

In the following, the effects of working fluid, filling ratio, inner diameter and evaporator length on the thermal performance of CLOHPs at different heating power inputs will be discussed, and then an empirical correlation is developed to predict the performance of CLOHPs based on a series of available experimental data sets both from the present study and other literatures.

3.1. Thermal performance of the CLOHPs

3.1.1. Effect of filling ratio and working fluid

Fig. 2 shows the thermal resistance of the 2 mm ID CLOHP charged with water and ethanol, respectively, at filling ratios ranging from 40% to 60%. The evaporator length is 5 cm. It is seen that (1) the thermal resistance of the CLOHP either charged with water or ethanol increased with increasing the filling ratio at almost each same heating power input, which was consistent with the reported experimental results [10]; (2) compared to the CLOHP charged with ethanol, the water-charged heat pipe had lower thermal resistances, indicating higher thermal performance. Similar experimental results were obtained for this CLOHP worked at the same filing ratio range when having an evaporator length of 3 or 7 cm. Thus,

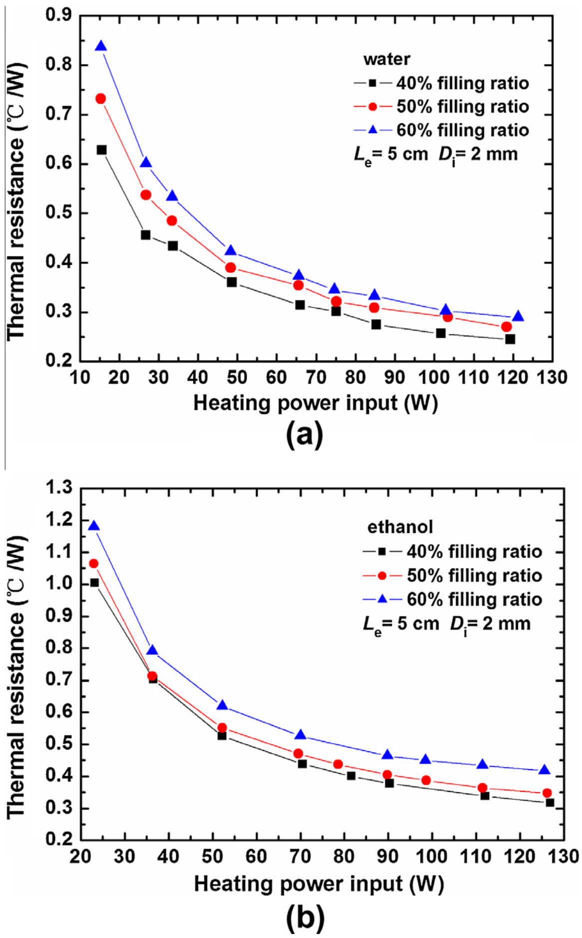

Fig. 2. Thermal performance of the 2 mm ID CLOHP charged with (a) water and (b) ethanol at different filling ratios (L_e = 5 cm).

40% is the optimum filling ratio for this CLOHP charged with water or ethanol. Table 2 gives the optimum filling ratios for all of the three CLOHPs at different working conditions, and it can be seen that the optimum value depends on the combined effects of working fluid, inner diameter and evaporator length. Besides, except for the 1.2 mm ID CLOHP charged with water, most of the optimum values are 40% at the present heating power input range. In conclusion, the heat transfer performance of a CLOHP could be enhanced when charged both with water or ethanol at a relatively lower filling ratio.

3.1.2. Effect of inner diameter

It is acknowledged that the surface tension predominate the two-phase flow in a CLOHP with the inner diameter satisfying such an inequality of $D_i \leqslant 2\sqrt{\sigma/g(\rho_l - \rho_v)}$ [10]. As the inner diameter decreases, the effect of surface tension on the slug/plug flow and hence thermal performance increases, while the effect of gravity becomes relatively small and even insignificant. Also, the effect of the inner diameter is related to the working fluid, which has great impact on the thermal performance of a CLOHP with a specified inner diameter as aforementioned.

Fig. 3 presents a comparison of the thermal resistance of the three CLOHPs characterized by different inner diameters. It is clear that, at the filling ratio of 50%, the thermal resistance decreases with the increase of inner diameter for the water-charged CLOHPs (shown in Fig. 3a). In contrast, the ethanol-charged CLOHPs (shown in Fig. 3b) were subjected to a reverse trend. Similar results were found for these CLOHPs worked at other filling ratios and evaporator lengths. Therefore, the choice of the inner diameter for each working fluid was not only less than the maximum value proposed by Khandekar et al. [10], but a relatively small value available to properly coordinate the relationship between the surface tension and gravity, contributing to high thermal performance, should also be took into account. The Bond number that indicates the relative importance between the surface tension and gravitational force is introduced to quantify the value appropriately, which is an area that warrants further study.

Table 2
Optimum filling ratios of the CLOHPs charged with water and ethanol, respectively.

CLOHPs	Water (%)			Ethanol (%)		
	L_e = 3 cm	L_e = 5 cm	L_e = 7 cm	L_e = 3 cm	L_e = 5 cm	L_e = 7 cm
1.2 mm ID	50	50	50	40	40	50
2 mm ID	40	40	40	40	40	40
2.4 mm ID	40	40	40	40	40	40

3.1.3. Effect of evaporator length

In general, the condenser area of a heat pipe is supposed to be larger than the evaporator area to ensure normal operating conditions, especially when high heat loads are imposed on the evaporator and thus prone to the dry-out. Although a CLOHP is not subjected to the entrainment limit due to its wickless structure, the boiling limit might occur and manifest itself by an unacceptable overheating of the evaporator due to the lack of cooling working fluid, tending to a dry-out eventually. To enhance the heat transfer rate and therefore sustaining higher heat loads without dry-out, the evaporator length of a CLOHP should be not larger than that of the condenser in principle.

Table 3 gives the different length distributions of the evaporator, adiabatic section, and condenser of these CLOHPs in the present study. The ratios of the evaporator length to the condenser length at three different conditions were 3/7, 5/7, and 1, respectively, corresponding to a decreasing trend of the effective heat transfer length.

Fig. 4 shows the thermal resistance of the 2 mm ID CLOHP for different filling ratios. The ratios of the evaporator length to the condenser length were varied in line with that in Table 3. It is clear that the water-charged CLOHP worked at lower thermal resistances than that charged with ethanol as aforementioned in Section 3.1.1. In addition, there is an optimum evaporator length, which contributes to the lower thermal resistance and thus better thermal performance of the CLOHP. The optimum evaporator length is largely dependent on the working fluid, filling ratio, and heating power input. For the ethanol-charged CLOHP, the optimum values are 5, 5, and 7 cm at the filling ratios of 40%, 50% and 60%, respectively, within almost all of the heating power input range. However, for the water-charged CLOHP at the corresponding filling ratios, the heating power input also partially influences on the optimum evaporator length and should be took into account. The curves of thermal resistance for the water-charged CLOHP with an evaporator length of 5 cm started to cross with that of 7 cm at medium power inputs (about 60 and 65 W in Fig. 4b and c, respectively). In general, it can be seen that the optimum evaporator lengths are 5, 7, and 7 cm for the filling ratios of 40%, 50% and 60%, respectively, at relatively higher power inputs. Among all of the evaporator lengths, the largest thermal resistances of the CLOHP appeared at the 3 cm length, indicating that the over-short evaporator length makes the device work at a low efficiency. It seems to be at odds with the result as reported by Charoensawan and Terdtoon [30], however their conclusion was derived from

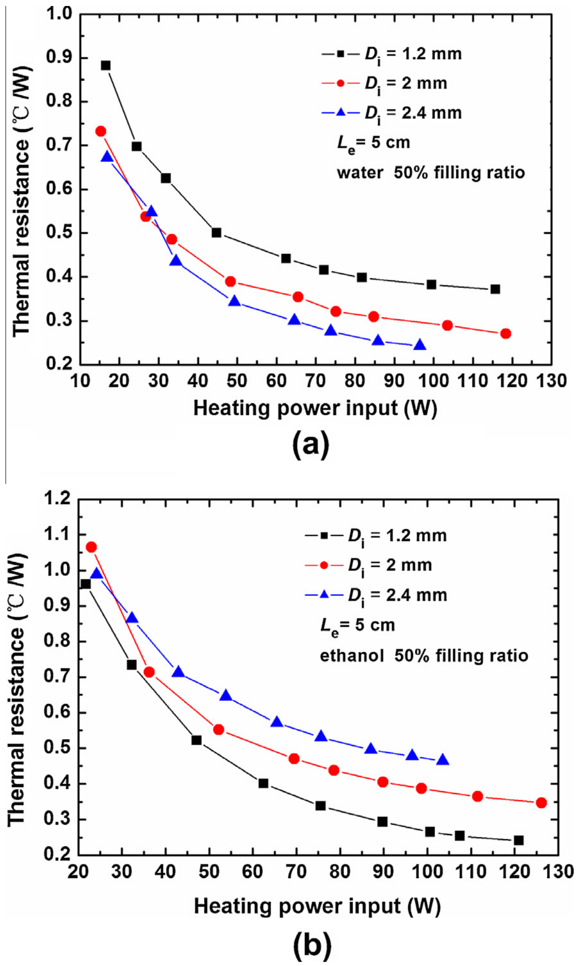

Fig. 3. Thermal performance of the CLOHPs charged with (a) water and (b) ethanol at different inner diameters (L_e = 5 cm, 50% filling ratio).

Table 3
Length distributions of the CLOHPs.

Case	L_e (cm)	L_c (cm)	L_a (cm)	L_{eff} (cm)
1	3	7	12	17
2	5	7	10	16
3	7	7	8	15

$^*L_{eff} = \frac{1}{2}(L_e + L_c) + L_a$ [26].

the comparison of evaporator length among CLOHPs with different total heat pipe lengths (or $L_e + L_a + L_c$), rather than a CLOHP characterized by a constant length herein. When a CLOHP is imposed to a same heat load but maintaining the cooling condition at the condenser unchanged, a shorter evaporator length means a higher heat flux and then a higher evaporator temperature, increasing the temperature difference between the evaporator and condenser [31]. In addition, a shorter evaporator length means a larger effective heat transfer length as given in Table 3, which increases the frictional pressure drop of fluid flow and hence hindering oscillating motion from the evaporator to the condenser and vice versa. Then, the optimum length appeared. Similar results were obtained for these CLOHPs worked at other inner diameters.

3.2. Correlation prediction of thermal performance

To facilitate the applications of CLOHPs in heat exchanger domain, it is hoped that the thermal performance could be effectively

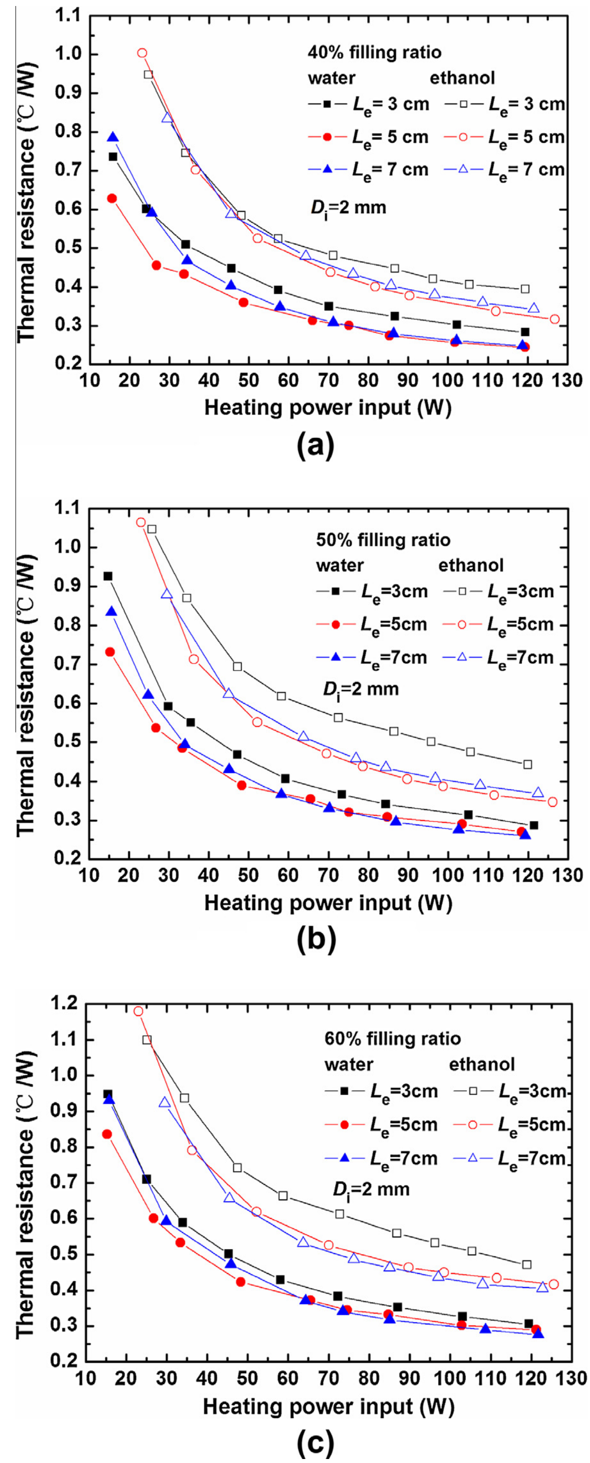

Fig. 4. Thermal performance of the 2 mm ID CLOHP at filling ratios of (a) 40%, (b) 50%, and (c) 60% with evaporator lengths varied from 3 to 7 cm.

evaluated by predicting correlations derived from non-dimensional groups containing primary thermo-physical properties of working fluids and geometrical parameters of heat pipes. Consequently, it is necessary to identify these parameters that affect the performance of CLOHPs, from which appropriate dimensionless numbers can be defined.

For vertical CLOHPs, it is found that the thermophysical properties of working fluid, filling ratio, inner diameter, evaporator/condenser length and temperature difference between the evaporator and condenser are the main factors affecting the heat transfer rate, and the Kutateladze number (Ku) is considered as a non-dimensional measure of the thermal performance [27].

Ku is a ratio of the input heat flux of a CLOHP to the critical heat flux in pool boiling:

$$\mathrm{Ku} = \frac{q''}{h_{fg}\rho_v\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]^{0.25}} \tag{5}$$

This dimensionless number represents the flow boiling phenomena of working fluid in the evaporator, and it could be expressed as a function of the Bond number (Bo), Morton number (Mo), Prandtl number of liquid (Pr), modified Jacob number (Ja*) which adds the influence of filling ratio, and ratios of D_i/L_e and L_e/L_c:

$$\mathrm{Ku} = f(\mathrm{Bo}, \mathrm{Mo}, \mathrm{Pr}, \mathrm{Ja}^*, D_i/L_e, L_e/L_c) \tag{6}$$

The other four dimensionless numbers of interest that manifest the thermophysical properties of working fluids in Eq. (6), Bo, Mo, Pr, and Ja*, are defined respectively as follows:

$$\mathrm{Bo} = D_i\sqrt{\frac{g(\rho_l-\rho_v)}{\sigma}} \tag{7}$$

$$\mathrm{Mo} = \frac{g\mu^4(\rho_l-\rho_v)}{\rho_l^2\sigma^3} \tag{8}$$

$$\mathrm{Pr} = \frac{\mu c_p}{k} \tag{9}$$

$$\mathrm{Ja}^* = \frac{\varphi c_p \Delta T_{e-c}}{(1-\varphi)h_{fg}} \tag{10}$$

where Bo is used to measure the relative importance between the surface tension force and the gravitation force, Mo is used together with Bond number to characterize the shape of bubbles/slugs moving in the surrounding fluid during flow boiling. Note that, according to the definition of Mo, the surrounding fluid herein is liquid, Pr is used to show the single-phase convective effect on heat transfer, Ja* is a ratio of the sensible energy to the latent heat absorbed during liquid–vapor phase change at a relatively small range of filling ratios around 50% relating to the best thermal performance [13].

All the thermo-physical properties of working fluids used in these four dimensionless numbers are calculated at an average temperature of the CLOHP ($T = (L_eT_e + L_cT_c)/(L_e + L_c)$).

In addition, D_i/L_e and L_e/L_c are ratios of the inner diameter to the evaporator length and the evaporator length to condenser length, respectively, representing the geometry of a CLOHP.

Then, an empirical correlation in terms of the basic form of Eq. (6) is calculated by using the curve fitting of power function with 510 sets of available experimental data from various sources as shown in Table 4, and the following exponential relationship has been established:

$$\mathrm{Ku_{pre}} = 8.3\mathrm{Bo}^{-1.598}\mathrm{Mo}^{0.026}\mathrm{Pr}^{-3.458}\mathrm{Ja}^{*-0.157}\left(\frac{D_i}{L_e}\right)^{1.21}\left(\frac{L_e}{L_c}\right)^{-0.232} \tag{11}$$

In Eq. (11), the exponents of various dimensionless numbers are quantitatively specified. Although the absolute value of exponent for Mo is much smaller than that for other dimensionless numbers

Table 4
Data source from experimental studies of CLOHPs.

Working fluid	Filling ratio (%)	L_e (cm)	D_i (mm)	n	Investigator
Ethanol	40–60	3–7	1.2, 2, 2.4	6	Present study
Water	40–60	3–7	1.2, 2, 2.4	6	Present study
Ethanol	40, 50	6	1.8	5	Shafii et al. [28]
Ethanol	55	3	2	5	Khandekar et al. [10]

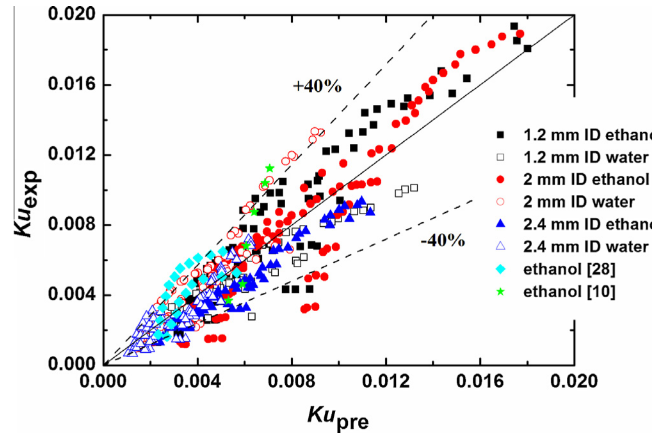

Fig. 5. Experimental vs. predicted values of Kutateladze number.

(especially Bo, Pr, and D_i/L_e), the ultra-low value of Mo make the term of $\mathrm{Mo}^{0.026}$ on the order of 0.1 (i.e., $\mathrm{Mo}^{0.026} \sim 0.1$) and thus it cannot be ignored.

Fig. 5 shows the relationship between the experimental data and theoretical results obtained from Eq. (11). It is found that the numerical prediction of Kutateladze number (Ku) is basically in good agreement with the experimental data both from the present study and other literature [10,28]. With the exception of the relatively lower Kutateladze numbers, the data scatter is centered about the mean of the predictions for all cases. The average standard deviation (ASD) between the experimental data and the empirical correlation of Eq. (11) is about 27.6% with approximately 85.5% of deviations being within ±40%, where ASD is defined as

$$\mathrm{ASD}|_{\mathrm{Ku}} = \frac{1}{n}\sum_{i=1}^{n}\frac{|\mathrm{Ku_{exp}} - \mathrm{Ku_{cor}}|}{\mathrm{Ku_{exp}}} \tag{12}$$

Therefore, it seems that the empirical correlation can reasonably predict the Kutateladze number and then the thermal performance of vertical CLOHPs at a filling ratio around 50%.

4. Conclusions

In this study, three closed-loop oscillating heat pipes fabricated by copper capillaries with 1.2, 2, and 2.4 mm IDs were manufactured, and the effects of working fluid, filling ratio, inner diameter, and evaporator length on the thermal performance of the CLOHPs were experimental investigated. The main conclusions are drawn as follows:

(1) The thermal resistances of the 2 mm ID and 2.4 mm ID CLOHPs charged with water were lower than that of ethanol, indicating higher thermal performance; whereas ethanol rather than water was more beneficial to improve the thermal performance of the 1.2 mm ID CLOHP.
(2) As to a filling ratio in the range of 40–60%, the optimum filling ratios with respect to the higher thermal performance for the 1.2 mm ID, 2 mm ID and 2.4 mm ID CLOHPs charged with water were 50%, 40% and 40%, respectively; for the ethanol-charged CLOHPs, the corresponding optimum filling ratios were almost all 40% at the heating power input range.
(3) Optimum evaporator lengths were confirmed for the CLOHPs that contribute to better thermal performance, and the optimum value for a CLOHP with specific inner diameter depends on the working fluid, filling ratio, and heating power input.
(4) An empirical power-law correlation based on non-dimensional groups, including Bond number (Bo), Morton number

(Mo), Prandtl number of liquid (Pr), modified Jacob number (Ja*), and ratios of D_i/L_e and L_e/L_c, was established to predict the Kutateladze number (Ku) and then thermal performance of CLOHPs worked at the vertical bottom heat mode. The agreement between the experimental data and the empirical correlation was generally good.

Acknowledgments

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