INVESTIGATION AND PREDICTION OF OPTIMUM MEANDERING TURN NUMBER OF VERTICAL AND HORIZONTAL CLOSED-LOOP PULSATING HEAT PIPES

by

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The objectives of this study are to experimentally investigate the effect of meandering turn numbers on thermal performance and to predict the optimum meandering turn number of vertical and horizontal closed-loop pulsating heat pipes. The closed-loop pulsating heat pipes were made from a Cu capillary tube with internal diameter of 2.0 mm. The closed-loop pulsating heat pipes were bent into undulating tubes with various meandering turn numbers, such as 5, 7, 10, 16, and 30. Each set of the closed-loop pulsating heat pipes had different evaporator section lengths of 50 mm and 150 mm. Heat input was supplied to the closed-loop pulsating heat pipe by allowing the flow of distilled water as the heating medium through the evaporator section. The adiabatic section temperature was constantly controlled at 50 °C. It could be concluded that the optimum meandering turn number of vertical closed-loop pulsating heat pipes with an evaporator section length of 50 mm is 10 for both R123 and water, and the optimum meandering turn numbers of vertical closed-loop pulsating heat pipes with an evaporator section length of 150 mm are 5 and 10, respectively, for R123 and water. However, the optimum meandering turn number of the horizontal closed-loop pulsating heat pipes could not be found since the heat flux directly varies with the turn number. In addition, the correlation to predict the optimum meandering turn number of the vertical closed-loop pulsating heat pipes was successfully established.

Key words: *closed-loop pulsating heat pipe, optimum meandering turn number, normal operation, thermal performance, correlation*

Introduction

The closed-loop pulsating heat pipe (CLPHP) was firstly introduced in [1]. It is a type of small heat transfer device with a very high thermal conductivity. It was invented to meet the requirement for smaller heat transfer devices. It can transfer sufficient heat for heat dissipation applications in modern electronic devices. The CLPHP was made of a long Cu capillary tube, bent into an undulating tube and connected at the ends to form a closed-loop with no internal wick structure. Working fluid is partially filled in the evacuated tube. The inner diameter of the tube is very small, and with the tube meeting a capillary scale, the inside working fluid forms into liquid slugs alternating with vapor plugs along the entire length of the tube.

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This internal flow pattern is well known as *slug-train*. However, the internal diameter of the CLPHP must not exceed the critical value following the Maezawa's criterion [2] because the working fluid will not form itself into slug-train and the heat pipe will not become the CLPHP at all. Heat can be transferred by means of the replacement mechanism [3]. 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 *adiabatic section*, toward a cooler section, *condenser section*, namely. Then, the vapor plug will condense, collapse, and release the heat into the environment. Therefore, 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. Generally, if the liquid quantity in the evaporator section is sufficient to transfer the heat input, the CLPHP can be normally transfer the heat, and an operation with this condition is defined as a *normal operation*. However, when the heat input gradually increases until the liquid quantity in the evaporator section is not sufficient to transfer that heat, dry-out of the liquid film will occur at a tube surface inside the evaporator section [4-7]. This may lead to damage of the tube and the environment if the heat input is further increased. The state in which the CLPHP can not normally operate is defined as the *maximum heat flux-state*, or the *critical-state*. Since the CLPHP must be designed to operate within the normal operation for safety reasons, most of the past studies on CLPHP have frequently laid emphasis on the thermal performances and the heat transfer characteristics of CLPHP operating in this state, as it has been in this study.

Apart from the capillary tube structure which is different from ordinary heat pipes, the meandering turns are the obvious difference. The meandering turn is a part of the capillary tube which is bent into a U-shape. It connects between adjacent capillary tubes that are arranged in parallel. The meandering turns are in the evaporator and condenser sections. The quantity of the meandering turns is called as the *meandering turn number*, which is counted from the meandering turns in the evaporator section. Since the meandering turn number defines the geometrical characteristics of the CLPHP, the thermal performance and the working fluid's circulation in the CLPHP strongly depend on the meandering turn number as well as other parameters. Besides the effect of the meandering turn number, a number of past studies have focused also on the effect of various other parameters, such as the evaporator section lengths [8, 9], internal diameters [9, 10], working fluids [8-10], filling ratios [9, 11, 12], working orientations [10-12], working temperatures [3, 13], and heat inputs [12, 14].

Although there are some past studies on the effect of meandering turn numbers on the thermal performance of CLPHP, the relationships between the meandering turn number and thermal performance obtained from the different studies are not in agreement with each other: In a case pertaining to vertical CLPHP, Charoensawan *et al.* [8] found that when the meandering turn number of the CLPHP with R123 as working fluid was increased from 10 to 28, the heat flux decreased from 8.313 W/m^2 to 7.603 W/m^2, respectively. In contrast, Charoensawan *et al.* [10] reported that when the meandering turn number was increased from 7 to 16, and then to 28, the heat transfer rate increased for the CLPHP with R123, ethanol, and water. Disagreements on the relationship between the meandering turn number and thermal performance are also found among mathematical studies. Khandekar *et al.* [15] proposed the semi-empirical correlation to predict the heat flux of the CLPHP oriented in any angles above the horizon. This correlation demonstrated that when the meandering turn number increased, the heat flux decreased in every inclination angle. Later, Soponpongpipat *et al.* [16] established the prediction model of the average heat transfer capacity of CLPHP. This model was established by considering the thermal resistance of the CLPHP together with the phase change damping coefficient obtained from simultaneous analyses of quantitative and qualitative results. This model showed that when the meandering turn number increased, the heat flux increased. The trend is totally contrary to the one predicted by an earlier correlation. Additionally, disagreements are found in the case of the horizontal CLPHP as well. Charoensawan and Terdtoon [17] proposed the empirical correlation to predict the thermal performance of the horizontal CLPHP in terms of the Kutateladze number, Ku. It was found in the correlation that the meandering turn number was absent. This can be understood as implying that the meandering turn number has no effect on thermal performance, which in turn, suggests that the relationship can not be exactly concluded. A year later, the same researchers [9] presented the quantitative results, which were that when the meandering turn number increased from 11 to 16 and then to 26, the thermal resistance decreased, or the thermal performance increased.

Due to the disagreement in the relationship between the meandering turn number and thermal performance of the vertical and horizontal CLPHP, as previously described, significant negative impact is caused on heat pipe designers in both research and industrial fields. Since it can not be exactly specified whether the meandering turn number should be increased or decreased to archive an increase in the thermal performance of CLPHP, a case-by-case quantitative experiment must be conducted to determine the relationship for individual CLPHP as regards their condition. This procedure highly consumes time and budget. Additionally, another big problem has been identified in the design of the meandering turn number of CLPHP which are for use in real applications. The question is whether, after the CLPHP heat exchanger is designed and the target meandering turn number is set, the CLPHP should be constructed in one set with a meandering turn number that equals the target meandering turn number [6, 18], or it should be constructed in a few sets while the total meandering turn number from a combination of the different sets equals the target meandering turn number [19, 20]. The question as to which design scheme is better still remains unsolved among CLPHP designers. From this point of view, issues regarding the effect of the meandering turn number, as previously described, become the significant point of focus of this study. The hypothesis is that when the meandering turn number of a certain CLPHP increases, the thermal performance can either increase or decrease. This implies that there should be an optimum meandering turn number for individual CLPHP. The optimum meandering turn number is defined to be the meandering turn number that causes the CLPHP to have the maximum heat flux. This hypothesis is in complete agreement with the suggestion in the previous review article, which states that there might exist an optimum number of turns that would achieve the maximum heat flux [21]. In order to examine and elucidate the issues surrounding the meandering turn number, the objectives of this study have been set: to experimentally investigate the effect of meandering turn numbers on thermal performance and to predict the optimum meandering turn number of vertical and horizontal CLPHP. The results and correlation obtained from this study will be highly valuable for heat pipe designers since, as a consequence, CLPHP with proper meandering turn numbers will be designed and constructed and the maximum thermal performance will subsequently be reached easily. In addition, fundamental information about CLPHP from a new point of view which has never been proposed will be archived.

Experimental set-up and procedure

The CLPHP in this study were made from a Cu capillary tube with internal and external diameters of 2.0 mm and 2.9 mm, respectively. The CLPHP were bent into an undulating tube with various meandering turn numbers, N, such as 5, 7, 10, 16, and 30. There were two sets of CLPHP with the same meandering turn number but different evaporator section lengths, L_e, that is, 50 mm and 150 mm. The adiabatic and condenser section length in each CLPHP was restricted to be identical with the evaporator section length in order to prevent heat flux transformation in the CLPHP. An example of structure of the CLPHP with the evaporator section length of 50 mm and actual tested CLPHP are presented in figs. 1 and 2, respectively. Consequently, the evaporator section and the condenser section were individually enveloped by zinc jackets. Each jacket had an inlet tube and an outlet tube for the allowing the flow of the heating medium and the cooling medium, which were hot distilled water and cool distilled water mixed with ethylene glycol (1:1 by volume), through the evaporator section and the condenser section, respectively. After the CLPHP was evacuated, R123 and water were individually selected to be filled in as the working fluid because of the obvious difference in their thermodynamic properties. The filling ratio was maintained at 50% of the total volume since this was the optimum filling ratio [2, 11]. The filling tube of the CLPHP was finally closed and investigated for leakage. The CLPHP was installed on a test rig that could be tilted in order to adjust the working orientation of the CLPHP. The working orientation in this study consisted of the vertical and horizontal planes. The heating medium temperature and mass-flow rate could be constantly controlled by a hot bath (Haake, N6-C41, accuracy ±0.5 °C). A cold bath (Bitzer, D7032, accuracy ±1 °C) was used for adjusting the cooling mediums temperature and mass-flow rate. The heating medium and the cooling medium were circulated through rubber hoses between the jackets and the baths. All the sections, including the heating medium and the cooling medium hoses, were well insulated by using a thermal-insulation sheet (Aeroflex, 3/8 in. thickness). The 26 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 temperature at every part of the CLPHP. These thermocouples consisted of 10 points in the middle of each tube in the evaporator section, 8 points in the adiabatic section, and 8 points in the condenser section. Eight thermocouples were also placed on each of the inlet and outlet tubes of the heating and cooling jackets to measure the variations in the heating medium and cooling medium temperatures. The CLPHP and the experimental set-up are schematically shown in fig. 3.

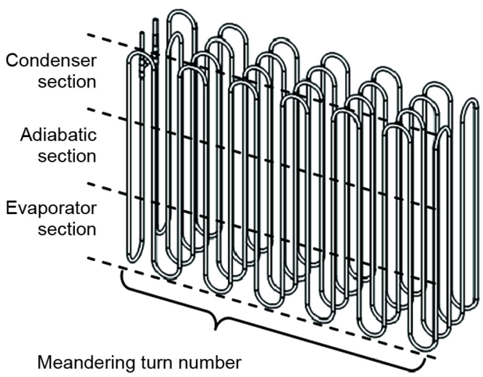

Figure 1. The structure of the CLPHP

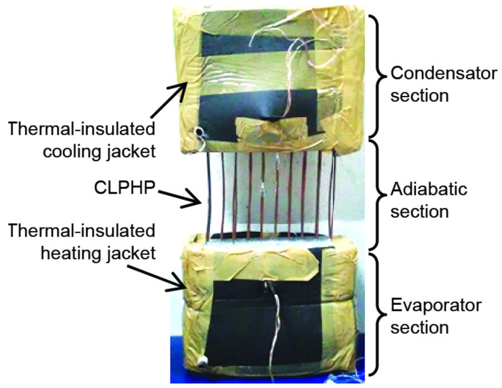

Figure 2. The tested CLPHP

The experimental procedure was: The hot bath and the cold bath were initially started. The heating medium was pumped into the heating jacket while the inlet temperature and the mass flow rate of the heating medium were constantly controlled at 80 °C and 0.1 kg/s, respectively. The cooling medium was circulated into the cooling jacket and its inlet temperature and mass-flow rate were adjusted in order to maintain the working temperature of the CLPHP, or the temperature measured at the adiabatic section, constantly at 50 ± 3 °C throughout the experiment. When the steady-state was reached, the temperature variation at every point was monitored and recorded using a data logger (Brainchild, VR18, accuracy ±0.1 °C) as exemplified in fig. 4. 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 by a high precision stopwatch (Casio, HS70W-1D, accuracy ±0.001 s). The heat flux at specified times was calculated from the variations in the temperature and the mass-flow rate of the cooling medium by means of the calorific as given in eq. (1). The advantage of this way of measuring is that the actual throughput heat along the CLPHP can be obtained.

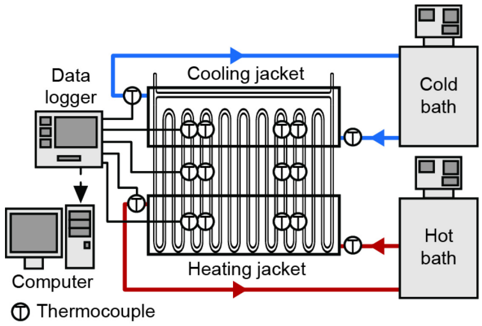

Figure 3. The experimental set-up

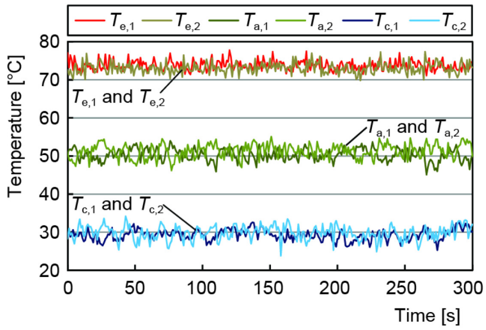

Figure 4. The temperature variation
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$$\dot{q}_c = \frac{\dot{m}_{cl} c_{pcl} (T_{out} - T_{in})_{cl}}{A_c} \tag{1}$$

In this study, the data in which the error was lower than 30% of the calculated heat flux were defined as passing the criterion for moving ahead to be analyzed in the next step. The experiment on each configuration was repeatedly conducted for five tests in order to verify the obtained results. The whole experimental procedure was repeated until all the independent parameters were observed. The average heat flux obtained from each configuration was brought into analyses in order to determine the quantitative relationship between the meandering turn number and arbitrary parameters, and, finally, to establish the correlation to predict the optimum meandering turn number.

Results and discussion

Effect of turn number on heat flux of vertical CLPHP

The effect of the meandering turn number on the heat flux of the vertical CLPHP with L_e = 50 mm is shown in fig. 5(a). It can be seen that when the meandering turn number is low, that is, 5 and 7, the heat flux is relatively low. When the meandering turn number was increased to 10, maximum heat flux was archived with magnitudes of 38,544 W/m^2 and 22,127 W/m^2 for the cases of R123 and water, respectively. After the meandering turn number increased to 16 and 30, the heat flux was observed to gradually decrease. It can be predicted that if the meandering turn number is further increased, the heat flux will continuously decrease. This result follows the same trend of the result obtained in a past study, that of Charoensawan *et al.* [10]. The CLPHP with water as working fluid and with L_e = 50 mm was investigated. It was found that when the meandering turn number increased from 13 to 28 and then to 44, the heat flux decreased. However, the level of the heat flux quantity was relatively higher than the one obtained in this study, in accordance with a few differences in the experimental conditions between the two studies. The evaporator section and the condenser section temperatures were constantly controlled at 80 °C and 20 °C, respectively, for the previous study. Nevertheless, the evaporator section and the adiabatic section temperatures were constantly controlled at 80 °C and 50 °C, respectively, for this study. This is the main cause of the difference in the thermal performance.

In the case of the vertical CLPHP with L_e = 150 mm, the effect of the meandering turn numbers is presented in fig. 5(b). It can be observed that when R123 was used as the working fluid, the optimum meandering turn number was 5, and this caused maximum heat flux at 27,783 W/m^2. However, when the working fluid was changed to water, the optimum meandering turn number was found to be 10 with the maximum heat flux of 18,075 W/m^2. The trend beyond this point which can be predicted is that the heat flux will gradually decrease and that its magnitude will fluctuate within a narrow range. This result follows the same trend as obtained in a past study, that of Charoensawan *et al.* [10]. The CLPHP with R123 as the working fluid and L_e = 150 mm was investigated. It was found that when the meandering turn number increased from 5 to 11 and then to 16, the heat flux decreased.

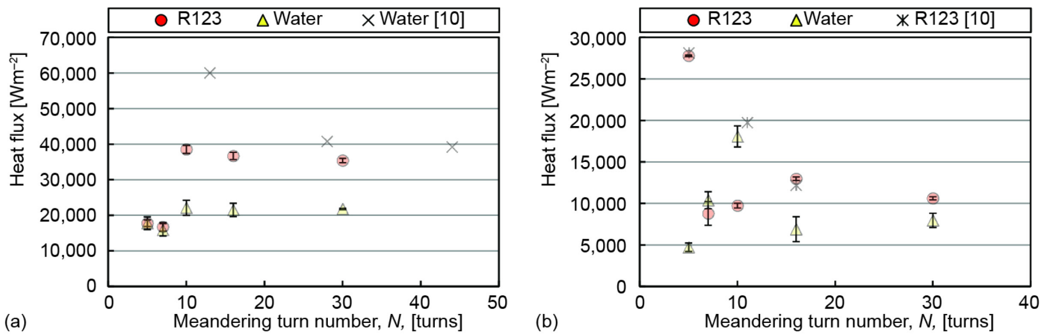

Figure 5. The effect of the turn number on the heat flux of vertical CLPHP; (a) L_e = 50 mm, (b) L_e = 150 mm

Overviewed consideration on the trend of every tested CLPHP can be given the physical reason of the optimum meandering turn number of the vertical CLPHP: When the meandering turn number is less than the optimum, the driving force normally generated in the evaporator section is relatively low. This causes inactive circulation of the working fluid inside the CLPHP, and the thermal performance is consequently unappreciated. When the CLPHP has the optimum meandering turn number, for example, 5 or 10, the driving force is sufficient to activate the active circulation inside the CLPHP. This subsequently causes a rapid increase in the thermal performance. However, when the meandering turn number is higher than the optimum, for example, 30, although the sufficient driving force continuously increases, the higher meandering turn number reflects the higher total length, L_t, of the CLPHP. This strongly affects the working fluid to have longer circulating distance. This consequently causes a higher frictional force between the working fluid and the tube's wall. The circulation, therefore, is not more active as the increase in the meandering turn number, and the thermal performance beyond the optimum meandering turn number will finally decrease.

Effect of turn number on heat flux of horizontal CLPHP

The effect of the meandering turn number on the heat flux of horizontal CLPHP with an evaporator section length of 50 mm is shown in fig. 6(a). It can be seen that when the meandering turn number is low, that is, 5 and 7, the heat flux is relatively low. When the meandering turn number was increased to 10, maximum heat flux was archived with a magnitude of 36,279 W/m^2. After the meandering turn number increased to 16 and 30, the heat flux was observed to slightly decrease in the case of R123. It can be predicted that when the meandering turn number is increased further, the heat flux will tend to decrease gradually. In the case of the horizontal CLPHP with water, it was found that when the minimum meandering turn number of 5 was used, the CLPHP was not able to transfer the heat. The heat started to be transferred after the meandering turn number was increased to 7. When the meandering turn number was increased to 10, maximum heat flux was archived with a magnitude of 21,480 W/m^2. After the meandering turn number increased to 16 and 30, the heat flux obviously fluctuated to be decreasing and increasing, respectively. If the meandering turn number increases beyond this point, the heat flux is predicted to increase continuously. This result has the same trend as the result obtained in a past study, that of Charoensawan and Terdtoon [9]. The horizontal CLPHP with water as the working fluid and an evaporator section length of 50 mm was investigated. It was found that the CLPHP with the meandering turn number of 5 could not start the heat transfer operation since this number is less than the critical meandering turn number of the horizontal CLPHP, which is 16, as firstly defined in [10]. Moreover, when the meandering turns number increased from 11 to 16 and then to 26, the thermal resistance continuously decreased, and, in turn, the thermal performance increased.

In the case of the horizontal CLPHP with L_e = 150 mm, the effect of the meandering turn numbers is as illustrated in fig. 6(b). It can be seen in the case of the R123 that the values of heat flux in the cases of all the meandering turn numbers were very low compared to the

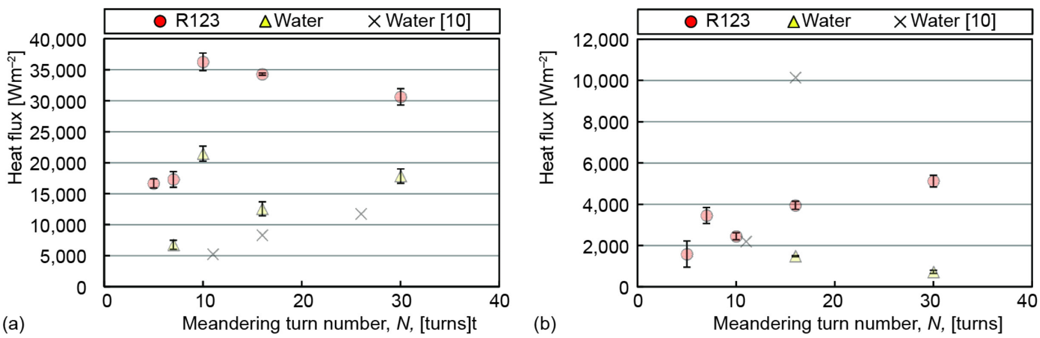

Figure 6. The effect of the turn number on the heat flux of horizontal CLPHP; (a) L_e = 50 mm, (b) L_e = 150 mm

horizontal CLPHP with the evaporator section length of 50 mm. At the minimum meandering turn number of 5, the heat flux was just 1,586 W/m^2. After the meandering turn number was increased to 30, the heat flux gradually increased to 5,126 W/m^2, which was the maximum heat flux in this test. It can be predicted from the overview that the heat flux will continuously increase beyond this particular meandering turn number. In the case of the CLPHP with water as the working fluid, it was found that the CLPHP was not able to operate when the meandering turn number was relatively low, at 5, 7, and 10. The CLPHP started transferring the heat at the meandering turn number of 16 with a heat flux of 1,486 W/m^2. This finding corresponds to the results obtained in a past study, that of Charoensawan *et al.* [10], which revealed that the critical meandering turn number of the horizontal CLPHP with an internal diameter of 2 mm was 16. As an addition to the finding of this previous study, when the meandering turn number increased from 11 to 16, the heat flux was observed to dramatically increase from 2,204 W/m^2 to 10,147 W/m^2.

Although it can be seen that the optimum meandering turn number of the horizontal CLPHP with R123 and an evaporator section length of 50 mm is 10, from an overview of other cases, it is evident that the optimum meandering turn number cannot be found since the heat flux continuously increases with increase in the meandering turn number. The physical reason to explain this relationship as regards the horizontal CLPHP is: When the CLPHP is tiled to be in the horizontal plane, the longitudinal axis of the capillary tube becomes perpendicular to the direction of gravity, and, as a result, the working fluid's circulation from the condenser to the evaporation section is not activated by gravitational acceleration. At the same time, the buoyancy force which generally drives the vapor plugs to move toward the condenser section is not in the direction along the longitudinal axis of the tube as in the case of the vertical CLPHP. By conjugating these phenomena, it can be concluded that the horizontal CLPHP itself needs an extra driving force to start and maintain the working fluid's circulation in order to transfer the heat continuously. Since an increase in the turn number is also equivalent to an increase in the number of nucleation sites for the working fluid's boiling system inside the heat pipe, the nucleation site can be compared to the working fluid's driving force generating site [4]. For this reason, when the turn number of the horizontal CLPHP increases, the working fluid has more driving force and tends to circulate more actively, and this causes the heat flux to directly vary with the turn number. In the light of these reasons, it can be stated that the correlation to predict the optimum meandering turn number of horizontal CLPHP is not necessary to be established since higher turn numbers cause higher thermal performance of horizontal CLPHP.

Correlation to predict optimum turn number of vertical CLPHP

It can be seen from the experiment in the previous topic that although each of the vertical CLPHP has an optimum meandering turn number, the exact optimum meandering turn number for every configuration of the CLPHP individually still can not be defined. This is because other parameters involved in the configuration of CLPHP also have an effect on the thermal performance. In order to evaluate the optimum meandering turn number, the effect of those parameters, such as the internal diameter, the evaporator section length, the meandering turn number, the temperature at each section, and the thermodynamic properties of the working fluid, must be included in the evaluation. Tracing back to the recent study [15], the most well-known correlation to predict the heat flux of the CLPHP, which is widely accepted, is as expressed in eq. (2):

$$\dot{q}_c = 0.54\left[\exp(\beta)\right]^{0.48} \mathrm{Ka}^{0.47} \mathrm{Pr}_\mathrm{l}^{0.27} \mathrm{Ja}^{1.43} N^{-0.27} \tag{2}$$

It can be noted that there is a meandering turn number in this correlation. Therefore, if the experimental data obtained in this study are taken into account with this correlation, the modified correlation to predict the optimum meandering turn number of the CLPHP will be consequently archived. The procedure to establish the correlation is: The parameter about the working orientation, β, was firstly omitted since only the vertical CLPHP was considered in the correlation. The constant 0.54 was also neglected. The heat flux in the unit of [W/m^2], which was the dependent parameter, was substituted with the dimensionless group of the Kutateladze number. The Kutateladze number indicates the ratio between the actual heat flux and the critical heat flux. From this procedure, a non-dimensional function used for establishing the correlation was obtained, as given in eq. (3):

$$\mathrm{Ku} = f(\mathrm{Ka}^{0.47} \mathrm{Pr}_\mathrm{l}^{0.27} \mathrm{Ja}^{1.43} N^{-0.27}) \tag{3}$$

After that, the experimental data obtained from this study were plotted by following the previous function in which the Kutateladze number was on the vertical axis and the dimensionless groups were on the horizontal axis. Moreover, in order to increase the reliability of the correlation, the experimental data obtained from the past studies of Charoensawan *et al.* [10] (in which the working fluids were R123, ethanol, and water, with various meandering turn numbers) and On-ai *et al.* [22] (in which the working fluids were R123, R141b, acetone, ethanol, and water; N = 26 turns) were also plotted on the graph, as demonstrated in fig. 7(a). It should be noted that the thermodynamic properties of the working fluids involved in the function were considered at the working temperature, T_w, of the CLPHP by estimating the working temperature from the equation $T_\mathrm{w} = (T_\mathrm{e} + T_\mathrm{c})/2$. It can be seen in fig. 7(a) that when the horizontal axis has a value of about 50,000, the maximum Kutateladze number is obtained. If the value of the dimensionless groups is higher than 50,000, the Kutateladze number will dramatically decrease. It can be verified that this is the optimum point for establishing the correlation. In the next step, when the Kutateladze number in eq. (3) was substituted with the value 50.000, the correlation was subsequently rearranged, therefore, the correlation to predict the optimum meandering turn of the vertical CLPHP was finally obtained:

$$N_\mathrm{opt} = \left(\frac{\mathrm{Ka}^{0.47} \mathrm{Pr}_\mathrm{l}^{0.27} \mathrm{Ja}^{1.43}}{50,000}\right)^{3.704} \tag{4}$$

From this correlation, the relationship between the optimum meandering turn number, N_opt, and the dimensionless group of Karman, Prandtl, and Jacob numbers was obtained, as indicated by the solid line in fig. 7(b). It can be concluded that the meandering turn numbers on this line are the optimum meandering turn numbers which bring about the optimum thermal performance for each configuration of the CLPHP along the horizontal axis. When all the experimental data are plotted in the graph, it can be observed that all the points above the solid line are the data whose horizontal axis value in fig. 7(a) is lower than 50,000. Since the Kutateladze number increases with increase in the horizontal axis value within this range, the meandering turn number in the zone above the solid line is called as the *suitable meandering turn number*, or the meandering turn number that gives rise to acceptable thermal performance. In contrast, the experimental data located beneath the solid line are the data whose horizontal axis value in fig. 7(a) is higher than 50,000. Within this range, the Kutateladze number dramatically decreases. In the light of this reason, the meandering turn number in the zone beneath the solid line is called as the *unsuitable meandering turn number*, or the meandering turn number that causes obvious degradation of the thermal performance.

The solid line calculated from the correlation, as shown in fig. 7(b), can be used in the design of the CLPHP by the following procedure. The working fluid type and the evaporator section and the condenser section temperatures, T_e and T_c, are initially defined. The thermodynamic properties of the working fluid (μ, ρ, c_p, h_{fg}, k, and P) corresponding to the working temperature, T_w, are determined in order to calculate the value of the right-hand side of the correlation in eq. (4). This becomes the value in the horizontal axis in fig. 7(b). After that, a vertical line is projected to the solid line and perpendicularly projected to cross the vertical axis where it is the optimum meandering turn number, N_{opt}, of the designed CLPHP. It should be noted that the limitations of using the correlation are: it is available for vertical CLPHP with the working fluid's filling ratio of 50% by total volume inside the CLPHP, the value of the dimensionless group must be within the range of $20,000 \leq [\mathrm{Ka}^{0.47}\mathrm{Pr}_l^{0.27}\mathrm{Ja}^{1.43}N^{-0.27}] \leq 90,000$, and the percentage of standard deviation of error between predicted and actual N_{opt} was ±25.64%.

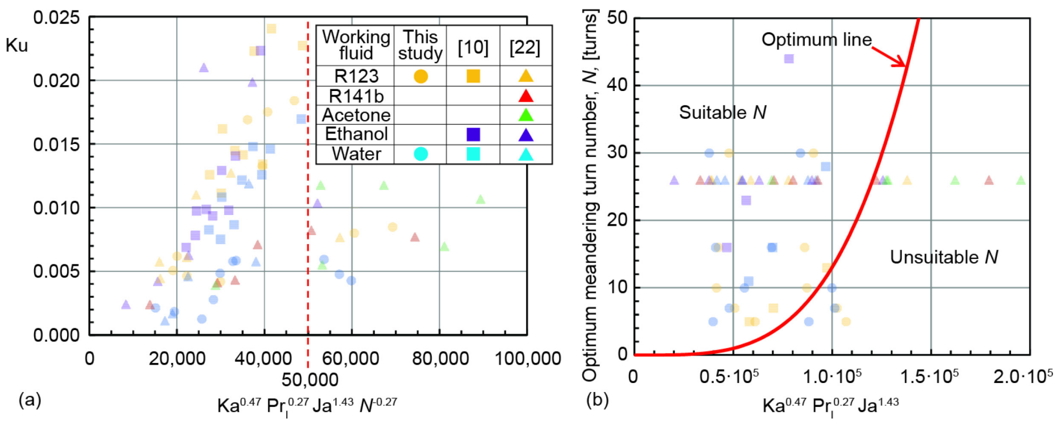

Figure 7. The relationship with the dimensionless groups; (a) Kutateladze number, (b) optimum turn number (*for color image see journal web site*)

Conclusion

It could be concluded that the optimum meandering turn number of vertical CLPHP with an evaporator section length of 50 mm is 10 for both R123 and water, and the optimum meandering turn numbers of vertical CLPHP with an evaporator section length of 150 mm is 5 and 10, respectively, for R123 and water. In addition, the correlation to predict the optimum meandering turn number of vertical CLPHP was successfully established. However, the optimum meandering turn number of horizontal CLPHP could not be found since higher turn numbers cause higher heat flux. For this reason, the correlation to predict the optimum meandering turn number of horizontal CLPHP is not necessary to be established. The experiments on the optimum meandering turn number of CLPHP with different working fluids and structural configurations are extensively conducted in the future.

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were authorized to be used in the correlation. The authors would like to express their sincere appreciation for all of the support provided.

Nomenclature

A – area, [m^2]
c_p – specific heat, [kJkg^{-1}K^{-1}]
D – diameter of heat pipe, [m]
h_{fg} – latent heat of evaporation, [kJkg^{-1}]
Ja – Jacob number, [$= h_{fg}/(c_{pcl}\Delta T_{e-c})$], [–]
k – thermal conductivity, [kWm^{-1}K^{-1}]
Ka – Karman number, [$= \rho_l \Delta P_{e-c} D^3/(\mu_l^2 L_{eff})$], [–]
Ku – Kutateladze number
L – length, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – pressure, [Pa]
Pr_l – Prandtl number, ($= c_{p,l}\mu_l/k_l$), [–]
$\dot{q}$ – heat flux, [kWm^{-2}]
T – temperature, [K]

Greek symbols

Δ – difference
μ – viscosity, [kgm^{-1}s^{-1}]
ρ – density, [kgm^{-3}]
σ – surface tension, [Nm^{-1}]

Subscripts

a – adiabatic
c – condenser
cl – cooling medium
e – evaporator
eff – effective
l – liquid
in – inlet
opt – optimum
out – outlet
w – working

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