CORRELATIONS TO PREDICT THERMAL PERFORMANCE AFFECTED BY WORKING FLUID'S PROPERTIES OF VERTICAL AND HORIZONTAL CLOSED-LOOP PULSATING HEAT PIPE

by

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Objectives of this paper are to investigate the effects of dimensionless numbers on the thermal performance, and to establish correlations to predict the thermal performance of the vertical and a horizontal closed-loop pulsating heat pipe. The heat pipes were made of long copper capillary tubes with 26 meandering turns and both the ends were connected together to form a loop. R123, R141b, acetone, ethanol, and water were chosen as variable working fluids with a constant filling ratio of 50% by total volume. The inlet temperature of the heating medium and the adiabatic section temperature were constantly controlled and maintained at 80 °C and 50 °C, respectively. The thermal performance was represented in terms of the Kutateladze number. It can be concluded that when the Prandtl number of the liquid working fluid, as well as the Karman number, increases, the thermal performance increases. On the other hand, when the Bond number, the Jacob number, and the aspect ratio increase, the thermal performance decreases. These effects of the dimensionless numbers on the thermal performance are valid for both the heat pipes, except in the case of Bond number which has no effect on the thermal performance as far as the horizontal heat pipe is concerned. Moreover, correlations to predict thermal performance have been successfully established.

Key words: *closed-loop pulsating heat pipe, correlation, working fluid, dimensionless parameter, thermal performance*

Introduction

The closed-loop pulsating heat pipe (CLPHP) is a heat exchanger with very high thermal conductivity. It was first proposed by Akachi *et al.* [1]. The CLPHP is made from a copper capillary tube, of which the internal diameter does not exceed the critical value following the Maezawa's criterion [2], is bent into an undulating tube, and is connected at both ends to form a closed loop. The tube is evacuated and, consequently, partially filled with the working fluid. Since the inner diameter of the tube is very small which then meets the capillary scale, the inside working fluid forms into liquid slugs alternating with vapor plugs along the entire length of the tube. This internal flow pattern is known widely as "slug-train" [3]. Heat can be transferred by means of the replacement mechanism [4]. When one end of the CLPHP, called "evaporator section," is subjected to heat or high temperature, the working fluid, which is in liquid slug

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form, will evaporate, expand, and move through the no-heat transferring zone, or the "adiabatic section," toward a cooler section, namely, the "condenser section." Then, the vapor plug will condense, collapse, and release the heat into the environment. 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.

It can be seen from the above-mentioned discussion that working fluid is a very important factor which significantly influences the thermal performance of the heat pipe. The thermal performance of the heat pipe, subsequently, depends on the thermodynamic properties of the working fluid inside the heat pipe. The thermodynamic properties involved in heat transfer and two-phase flow consist of the latent heat, specific heat capacity, viscosity, surface tension and density. However, different fluid types have different properties. Categorizing working fluids by only one thermodynamic property is not possible. Previous literature reviews on the effect of working fluids on the thermal performance of the CLPHP have frequently defined latent heat as the quantitative property used to identify the type of working fluid. Previous studies showed that when the latent heat of the working fluid increases, the thermal performance of the CLPHP shows a tendency to change in both ways, that is, increase or decrease [5-11].

The reason why the effect of latent heat on the thermal performance of the CLPHP is not clear is because the other thermodynamic properties also have a strong influence on the heat transfer mechanism and the circulation. In addition, the geometrical parameters of the CLPHP also have an influence on the thermal performance, depending on the working fluid type [12]. A number of past studies conducted [5, 7, 8, 13] on the effect of the working fluid on the thermal performance of the CLPHP could not determine the quantitative relationship between the working fluid type and the thermal performance. Only a relative thermal comparison could be directly investigated, for example, the CLPHP with the working fluid A has higher thermal performance than the CLPHP with the working fluid B. This causes a huge problem to heat pipe designers since thermal performance cannot be precisely predicted when working fluids other than the one investigated in each past study are used. In order to solve this issue, a group of dimensionless numbers should be considered and analyzed to be the quantitative property to identify the working fluid type and to predict the thermal performance of the CLPHP. Being dimensionless numbers, they are valid for any working fluid. To investigate the effect of the dimensionless numbers on the thermal performance and to establish a correlation to predict the thermal performance of the vertical and horizontal CLPHP – these are the primary and significant objectives of this study.

Experimental set-up and procedure

The CLPHP used in the experiment were made of long copper capillary tubes with inner diameters of 1.50 mm, 1.78 mm, and 2.16 mm, and bent into 26 turns (the number of meandering turns in this study was counted from the evaporator section). Then, both the ends were connected together to form a loop. The evaporator, adiabatic, and condenser section lengths were all equal and were 50 mm in the first case and 150 mm in the second case. Chromel-alumel thermocouples (Omega, Type K, accuracy ± 0.5 °C) were used for the temperature measurement. The thermocouples were installed on the outer surface in the middle of each part of the CLPHP, as follows: 10 points on the evaporator section, and 8 points each on adiabatic section and condenser section. Moreover, two thermocouples were installed on each inlet and outlet of the evaporator and condenser jackets. The ambient air temperature was also measured. R123, R141b, acetone, ethanol, and water were chosen as the working fluids in this study as these working fluids have clearly different thermodynamic properties. The filling ratio chosen was 50% of the total volume since the best thermal performance of the CLPHPs can be obtained with this ratio [2, 14]. A schematic diagram of the whole experimental setup is depicted in fig. 1. The evaporator section and the condenser section were fully enveloped in a zinc heating jacket and a cooling jacket. Heat was supplied to the evaporator section of the CLPHP by circulating hot distilled water whose inlet temperature was constantly controlled and maintained at 80 °C using a hot bath (Haake, N6, accuracy ±0.01 °C). The condenser section would then release the heat to the aqueous solution of ethylene glycol (50% by volume) whose inlet temperature and mass flow rate were adjusted to keep the adiabatic temperature constantly at 50 °C by using a cold bath (Bitzer, D7032, accuracy ±1 °C). The mass flow rate was obtained by measuring the time for a certain fluid quantity to flow. A digital scale (Ohaus, Adventure, accuracy ±0.01 g) and a high precision stopwatch (Casio, HS70W-1D, accuracy ±0.001 s) were used to measure the mass and time respectively. All the sections, including the cooling medium hoses, were well insulated by using a thermal-insulated sheet (Aeroflex, 3/8 in. thickness). At specified points, the temperature was monitored by using a data logger (Brainchild, VR18, accuracy ±0.1 °C). The procedure for the experiment was as follows: The CLPHP with R123 as the working fluid was first attached to a test rig in vertical orientation with bottom heat mode (with the evaporator beneath the condenser section). The CLPHP according to this setup is called a "vertical closed-loop pulsating heat pipe" (VCLPHP). The hot and cold baths were then started to flow the heating medium and the cooling medium. After the system was in a steady state, the evaporator, adiabatic, and condenser section temperatures, as well as the temperature differences across the condenser section, were simultaneously recorded in order to calculate the heat flux at specified times, as in eq. (1). The advantage of this method is that the actual throughput heat along the CLPHP can be obtained. Then, the CLPHP and the test rig were tilted into a horizontal orientation. The CLPHP in this configuration is called a "horizontal closed-loop pulsating heat pipe" (HCLPHP). The above-mentioned procedure was repeated until the heat flux for the HCLPHP was archived. Subsequently, the CLPHP was removed from the rig to be evacuated and filled with another working fluid, and then the procedure was repeated until all the working fluids in both the orientations were completely investigated.

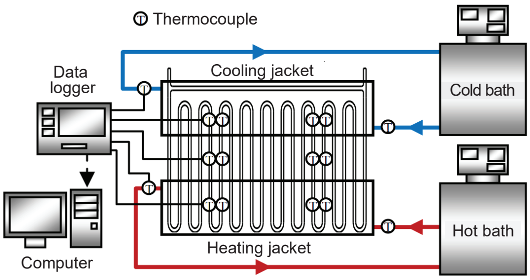

Figure 1. The experimental set-up

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

However, since each working fluid has a different "critical heat flux" – the highest thermal performance that a heat pipe can transfer before a dry-out of the liquid working fluid inside the evaporator section occurs – comparisons of the thermal performance of each working fluid through the heat flux would not be fair. The working fluid with a relatively low critical heat flux generally has a lower transferred heat flux. In order to normalize the experimental data, Kutateladze number (Ku) was chosen to be a representative of the thermal performance. The Ku is a well-known dimensionless number involved in heat transfer in a heat pipe. It is the ratio of the transferred heat flux to the calculated critical heat flux, which can be found from eq. (2). A higher

Ku implies that the heat pipe has higher thermal performance and operates closer to the critical state; *vice versa* for a lower Ku. The relationship between the thermal performance and the working fluids through each dimensionless number, as depicted in the next section, will present the Ku on the vertical axis for comparing the thermal performance of the CLPHP on the horizontal axis.

$$\mathrm{Ku} = \frac{\dot{q}_c}{\rho_v h_{fg} \sqrt[4]{\sigma g \left(\dfrac{\rho_l - \rho_v}{\rho_v^2} \right)}} \tag{2}$$

Results and discussions

Effect of Prandtl numbers on thermal performance

Prandtl number (Pr) is a dimensionless number associated with working fluid's properties. It is the ratio between dynamic viscosity and thermal diffusion of the working fluid. The Prandtl number can be in two phases, that is, Prandtl number of liquid working fluid ($\mathrm{Pr_l}$) and Prandtl number of vapor working fluid ($\mathrm{Pr_v}$). However, as evident from the experiment, $\mathrm{Pr_v}$ rarely had an effect on the thermal performance and can be neglected from this analysis. It was discovered from the experiments in the cases of both the VCLPHP and the HCLPHP that when $\mathrm{Pr_l}$ increased the thermal performance increased. This effect of $\mathrm{Pr_l}$ on the thermal performance agrees very well with the results from a past study [5]. The effect of $\mathrm{Pr_l}$ on the thermal performance of the VCLPHP and the HCLPHP are demonstrated in figs. 2(a) and (b).

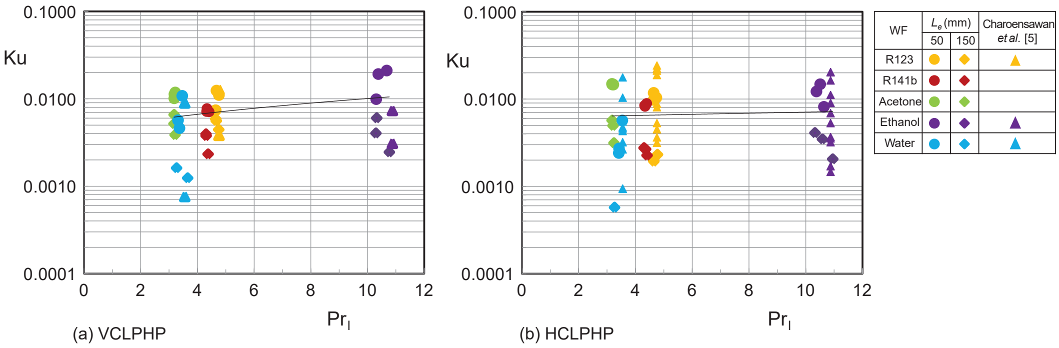

Figure 2. The effect of Prandtl numbers on thermal performance

Working fluids with greater liquid specific heat capacity can transfer larger heat quantities. This transferring of large quantities of heat causes the thermal performance to increase. Moreover, the increase in liquid viscosity leads the flow velocity of the working fluid to be retarded; leading to a longer time for receiving and releasing the heat in the evaporator and condenser sections and, consequently, the thermal performance increases. However, when a working fluid with higher liquid thermal conductivity is used, the heat will freely diffuse in the liquid slug in higher portions, compared with the remaining portion of the heat that causes evaporation; thus, the thermal performance decreases. These are the physical reasons to support the inference that when $\mathrm{Pr_l}$ increases, Ku, or the thermal performance of the VCLPHP and the HCLPHP, increases.

Effect of Bond numbers on thermal performance

Bond number (Bo) is a dimensionless number associated with working fluid's properties and geometry of the heat pipe. It expresses the ratio between the buoyancy force and the surface tension of the working fluid. It was found out from the study on the VCLPHP that when Bo increased, the thermal performance decreased in the case of a 50 mm evaporator section length. On the other hand, when Bo increased, the thermal performance increased in the case of a 150 mm evaporator section length, as illustrated in fig. 3(a). This happened irrespective of the working fluid type, the geometry of the heat pipe and the experimental conditions. Nevertheless, the physical reasons can be theoretically explained to support both the tendencies, as is given in the following explanation.

Consider the case when Bo increases, Ku, or thermal performance, decreases. This is, primarily, from a decrease in surface tension which appears as the denominator of Bo. When the surface tension decreases the vapor tends to form small bubbles instead of long vapor plugs. Since smaller bubbles have lower vapor mass than longer bubbles, this situation can be understood to imply that the heat in the evaporator section is transferred out from the tube's surface by means of evaporation in lower quantities. This causes the working fluid to transfer heat discontinuously and the thermal performance subsequently decreases. At the same time, another tendency was observed in the case of the longer evaporator section length, which is also found in a past study [5], that when Bo increases, Ku, increases. This is due to the buoyancy force. When the difference between the liquid and the vapor densities increases the buoyancy force is higher causing the vapor plugs to flow from the evaporator section to the condenser at the top of the VCLPHP in shorter time. Thus, the working fluid transfers the heat more actively and consequently, the thermal performance increases.

It was found out from the experiment of the HCLPHP that Bo does not have an effect on the Ku. Since the direction of the buoyancy force of the vapor bubbles is perpendicular to the direction of the flow passage of the working fluid between the evaporator and condenser sections, no matter how Bo increases or decreases the flow velocity of the working fluid is not affected. This causes the thermal performance of the HCLPHP to be independent of Bo. However, it needs to be mentioned that this conclusion is different to the relation found in a past study [5] which found that when Bo increased, the thermal performance increased. This is due to the differences in the number of variable parameters, such as the working fluid type, the geometry of the heat pipe and the experimental conditions. The effect of Bo on the thermal performance of the HCLPHP is illustrated in fig. 3(b).

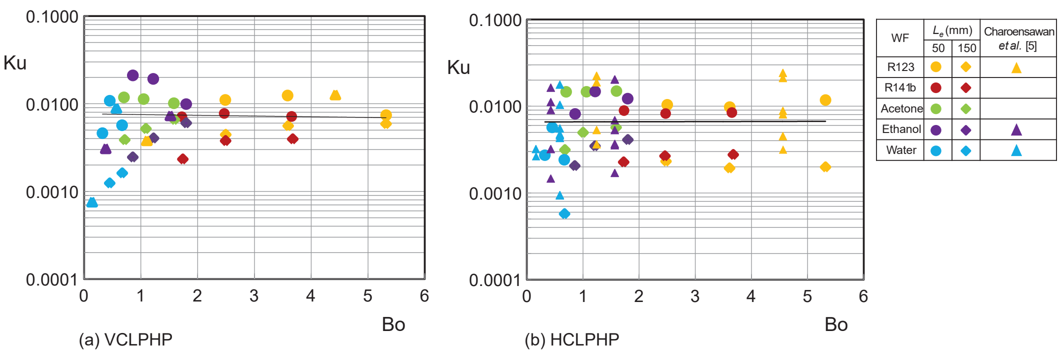

Figure 3. The effect of Bond numbers on thermal performance

Effect of Karman numbers on thermal performance

Karman number (Ka) is similar to Bo and represents the ratio between the driving force and the frictional force of the working fluid. It was found from the experiments in the cases of both the VCLPHP and the HCLPHP that when Ka increased, the thermal performance increased, exceptionally so in the case of ethanol since the thermal performance according to ethanol stands isolated from the overall data. Nevertheless, the relation between Ka and Ku agrees very well with the results from a past study [5]. The effect of Ka on the thermal performance of the VCLPHP and the HCLPHP is illustrated in figs. 4(a) and 4(b), respectively.

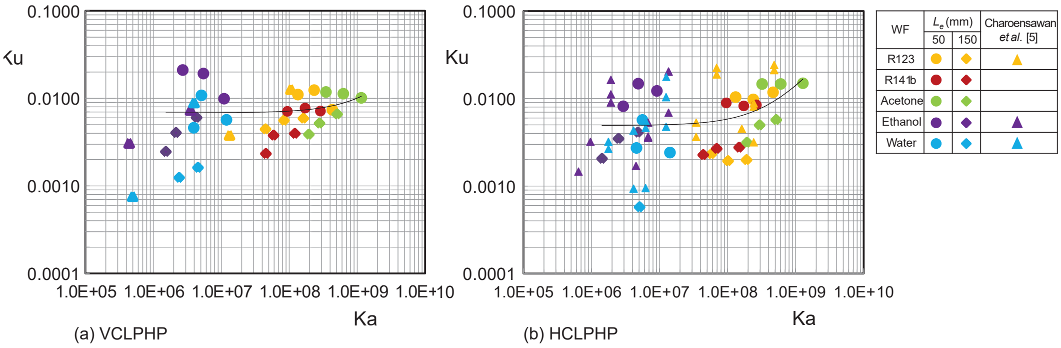

Figure 4. The effect of Karman numbers on thermal performance

The driving force is the main mechanism of the working fluid's circulation in the CLPHP [1]. The magnitude of the driving force depends on the saturated pressure differences of the working fluid between the evaporator and the condenser sections. Therefore, when the pressure difference increases, or Ka increases, the driving force increases. This causes the working fluid's flow velocity and the transferred heat quantity to increase. In addition, when the internal diameter increases, the pressure loss along the flow passage decreases, and the Ku, increases. On the contrary, an increase in the liquid viscosity and the effective length of the heat pipe causes a decrease in Ka and in the thermal performance, since the working fluid's flow velocity decreases and the flow distance increases. These phenomena, originally, were the major cause of thermal performance degradation.

Effect of Jacob numbers on thermal performance

Jacob number (Ja) is a dimensionless number given by the ratio between the fluids latent heat and sensible heat. The latent heat corresponds to the working fluid's phase change, and the sensible heat relates to the working fluid's temperature change. It was observed that when Ja increased, the thermal performance increased in the case of the VCLPHP with the evaporator section length 50 mm. In contrast, the opposite tendency was found in the case of the VCLPHP with the longer evaporator section length, that is, 150 mm. This finding agrees very well with the result obtained in a past study [5], as shown in fig. 5(a).

The working fluid with a higher Ja, causes the CLPHP to have better thermal performance since the working fluid transfers a greater quantity of the heat by means of a phase change in the evaporator section relative to the remaining heat that is transferred by means of temperature change. This situation corresponds to the heat transfer mechanism of the CLPHP [4]; thus, the heat pipe operates with near-theoretical optimized ability. However, increasing the evaporator section length increases the frictional force and the pressure loss. For this reason, heat will be transferred discontinuously. This causes the heat transferred through the CLPHP with higher Ja not to increase as usual. This is a piece of evidence to support the viewpoint that evaporator section length has a significant effect on the thermal performance. Therefore, another dimensionless number, associated with the evaporator section length has to be investigated. In addition, it was discovered from some past studies that when a portion of the sensible heat decreased, or when

Ja increased, the thermal performance decreased. This led to the realization that the CLPHP primarily transfers heat by means of temperature change mechanism rather than by phase change [15-18]. At present, information regarding the actual portion of heat transferred in the CLPHP according to latent heat and sensible heat is still not confirmed and established. Therefore, when Ja increases, it is possible that thermal performance can change in either direction.

It was observed from a case of the HCLPHP that when Ja increased, the thermal performance decreased, with the evaporator section lengths of 50 mm and 150 mm, as presented in fig. 5(b). This can be taken to imply that when the ratio of the sensible heat increases the thermal performance increases. This is because the direction of the vapor's flow passage inside the HCLPHP is perpendicular to the buoyancy force, as described in the section discussing the effect of Bo. Although a working fluid with high latent heat can evaporate and transfer the heat with high quantity, the buoyancy force does not actuate the vapor plugs to flow toward the condenser section. Therefore, an increase in the ratio of latent heat does not change the thermal performance. It can be concluded from this observation that the thermal performance of the HCLPHP primarily depends on the ratio of the sensible heat.

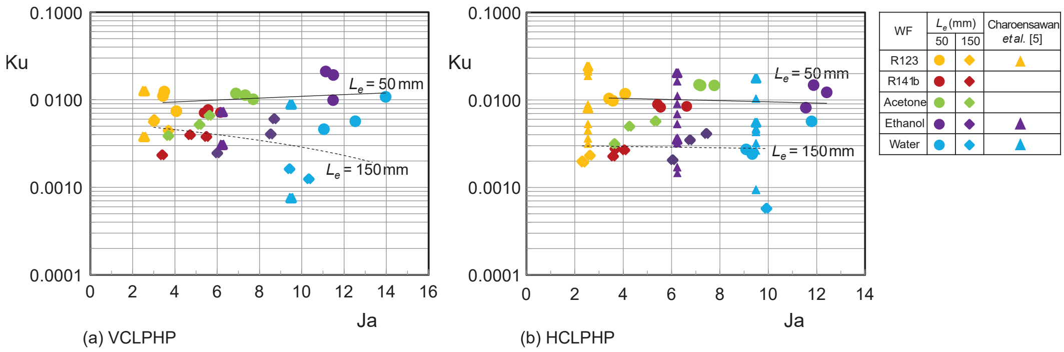

Figure 5. The effect of Jacob numbers on thermal performance

Effect of aspect ratios on thermal performance

Aspect ratio (L_e/D_i) is a dimensionless number associated with the geometry of the heat pipe. It represents the ratio between the evaporator section length and the internal diameter. It could be seen from the investigation regarding both the VCLPHP and the HCLPHP that when the aspect ratio increased, the thermal performance decreased. This result agrees very well with the results from a past study [5]. The effect of the aspect ratio on the thermal performance of the VCLPHP and the HCLPHP can be observed in figs. 6(a) and (b). The evaporator section length and the internal diameter affect the thermal performance of the heat pipe. It was found that when the distance between the evaporator section and the condenser section increases the working fluid's circulation friction increases. This leads to a decrease in the thermal performance. In addition, it was found that an increase in the internal diameter causes not only the heat transferring area between the heat pipe and the working fluid to increase but also the cross-sectional area of the working fluid's flow inside the CLPHP. The pressure loss in the working fluid's flow decreases. This is the physical reason why the CLPHP can transfer more heat and thus, the thermal performance increases. It is understood from the experimental results that the geometrical parameters of the heat pipe have a strong influence on the thermal performance. Therefore, it can be concluded that CLPHP with lower aspect ratios (approx 20 to 40) have higher thermal performance in comparison with CLPHP with higher aspect ratios (approx 70 to 1,000).

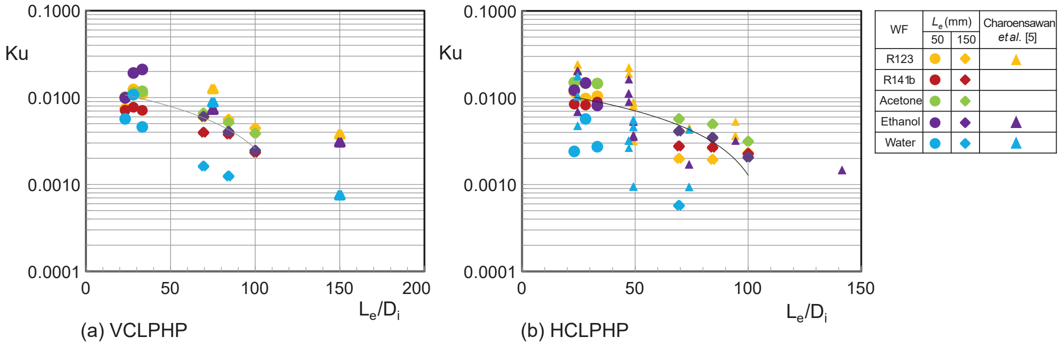

Figure 6. The effect of aspect ratios on thermal performance

Correlation to predict thermal performance

A correlation is desired between the five dimensionless numbers that affect the VCLPHP and the HCLPHP, as previously discussed. This would allow for more precise prediction of thermal performance of VCLPHP and HCLPHP. The correlations originated from the functions; $\mathrm{Ku_v} = \mathrm{f(Pr_l, Ja}, L_e/D_i, \mathrm{Ka, Bo)}$ and $\mathrm{Ku_h} = \mathrm{f(Pr_l, Ja}, L_e/D_i, \mathrm{Ka)}$ for the VCLPHP and the HCLPHP, respectively. It can be seen that there is no Bo in the function of the HCLPHP since Bo does not have any effect on the thermal performance of the HCLPHP, as described in the section *Effect of Bond numbers on thermal performance*.

All dimensionless numbers were arranged in the correlation by means of least-square curve fitting. The experimental data involved in the curve fitting consisted of the results obtained in this study and in past studies on VCLPHP [5, 19]. The data was used to increase the predictive precision of the correlation. Combining these results with those from other studies was accepted, since the scope of the experiment in both studies was nearly the same. The resulting correlation is expressed in eq. (3) and eq. (4), respectively:

$$\mathrm{Ku_{v,model}} = 5.27\cdot 10^{-2}\,\mathrm{Ka}^{0.057}\,\mathrm{Pr_l}^{0.522}\,\mathrm{Ja}^{-0.507}\mathrm{Bo}^{-0.164}\left(\frac{L_e}{D_i}\right)^{-0.727} \tag{3}$$

$$\mathrm{Ku_{h,model}} = 9.62\cdot 10^{-3}\,\mathrm{Ka}^{0.152}\,\mathrm{Pr_l}^{0.905}\,\mathrm{Ja}^{-0.110}\left(\frac{L_e}{D_i}\right)^{-1.212} \tag{4}$$

The thermal performances calculated from the correlation and obtained from the experiment ($\mathrm{Ku_{v,exp}}$ or $\mathrm{Ku_{h,exp}}$) were plotted against each other to verify the precision of the correlation for the VCLPHP and the HCLPHP, as illustrated in fig. 7(a) and fig. 7(b), respectively. The data deviation between $\mathrm{Ku_{model}}$ and $\mathrm{Ku_{exp}}$ was ± 37% for both the CLPHPs. It should be noted that the quantity of each thermodynamic property involved in the correlation is defined at the adiabatic section temperature. In a design in which the adiabatic section temperature is not exactly known, the adiabatic section temperature can be estimated from the equation $T_a = (T_e+T_c)/2$. In addition, it needs to be mentioned that both the correlations can be used subject to the following three conditions:
(1) $0.001 < \mathrm{Ku} < 0.030$,
(2) the working fluid's filling ratio is 50% of the total volume as the optimum ratio [2, 14], and
(3) the evaporator section, the adiabatic section, and the condenser section have the same length.

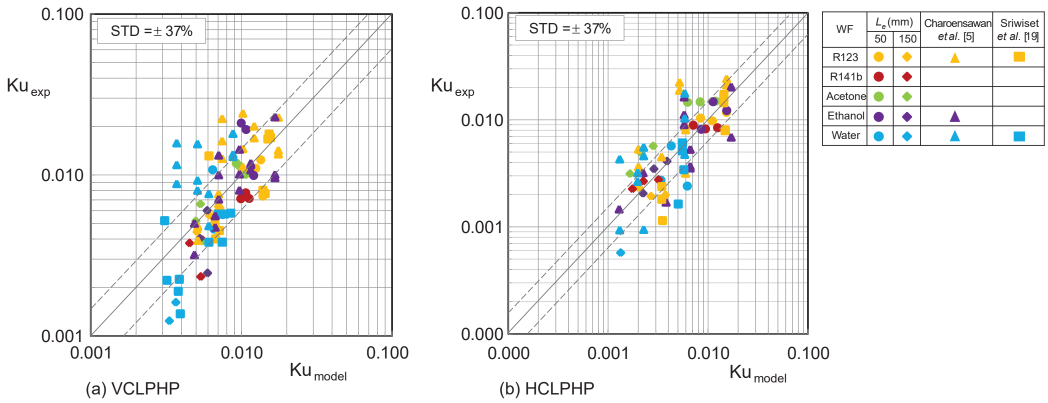

Figure 7. The comparison between Ku_{model} and Ku_{exp}

Conclusions

The effects of dimensionless numbers on the thermal performance of the VCLPHP and the HCLPHP have been thoroughly investigated, and the correlations to predict the thermal performance of the VCLPHP and the HCLPHP have been successfully established in this study. Thermal performance was represented in terms of Kutateladze number, which is associated with heat transfer in heat pipes. It can be concluded that when the Prandtl number of liquid working fluid and the Karman number increase, the thermal performance increases. On the other hand, when the Bond number, the Jacob number, and the aspect ratio increase, the thermal performance decreases. These effects of the dimensionless numbers on the thermal performance are valid for both the VCLPHP and the HCLPHP except in the case of the Bond number which has no effect on the thermal performance as far as the HCLPHP is concerned. Additionally, all the dimensionless numbers that have an effect on the thermal performance of the VCLPHP and the HCLPHP have been analyzed. The correlations will be very useful for designers, industrialists and people who are interested in the applications of the CLPHP, in addition to being valuable basic knowledge for heat pipe researchers.

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Nomenclature

A – area, [m^2]
Bo – Bond number, ($=\Delta\rho_{l-v}gD^2/\sigma$), [–]
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, [$=h_{fg}/(c_{p,l}\Delta T_{e-c})$], [–]
Ka – Karman number, [$=\rho_l\Delta P_{e-c}D^3/(\mu_l^2 L_{eff})$], [–]
k – thermal conductivity, [kWm^{-1}K^{-1}]
L – length, [m]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
Pr – 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
out – outlet
v – vapor

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