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Correlation to Predict Thermal Performance According to Working Fluids of Vertical Closed-Loop Pulsating Heat Pipe

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Abstract**—**The objectives of this paper are to investigate effects of dimensionless numbers on thermal performance of the vertical closed-loop pulsating heat pipe (VCLPHP) and to establish a correlation to predict the thermal performance of the VCLPHP. The CLPHPs were made of long copper capillary tubes with inner diameters of 1.50, 1.78, and 2.16mm and bent into 26 turns. Then, both ends were connected together to form a loop. The evaporator, adiabatic, and condenser sections length were equal to 50 and 150 mm. R123, R141b, acetone, ethanol, and water were chosen as variable working fluids with constant filling ratio of 50% by total volume. Inlet temperature of heating medium and adiabatic section temperature was constantly controlled at 80 and 50°C, respectively. Thermal performance was represented in a term of Kutateladze number (Ku). It can be concluded that when Prandtl number of liquid working fluid (Pr_l), and Karman number (Ka) increases, thermal performance increases. On contrary, when Bond number (Bo), Jacob number (Ja), and Aspect ratio (L_e/D_i) increases, thermal performance decreases. Moreover, the correlation to predict more precise thermal performance has been successfully established by analyzing on all dimensionless numbers that have effect on the thermal performance of the VCLPHP.

Keywords**—**Vertical closed-loop pulsating heat pipe, working fluid, thermal performance, dimensionless parameter.

I. Introduction

THE closed-loop pulsating heat pipe (CLPHP) is a heat exchanger with a very high thermal conductivity. It was firstly invented by Akachi et al. [1]. The CLPHP is made from a copper capillary tube, of which internal diameter does not exceed the critical value following the Maezawa's criterion [2]. The tube is bent into an undulating tube, and connected at both ends of the tube to form a closed-loop. The tube is evacuated and consequently partially filled with working fluid. Since an inner diameter of the tube is very small and then meets a 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 well known 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 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. The structure of the CLPHP is shown in Fig. 1.

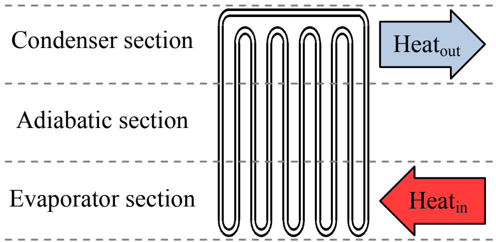

Fig. 1 Closed-loop pulsating heat pipe

It can be seen from the above mention that working fluid is a very important factor which significantly influences on thermal performance of the heat pipe since the working fluid acts as heat transferring medium between a heat source and a heat sink. Thermal performance of the heat pipe subsequently depends on thermodynamics properties of the working fluid inside the heat pipe. Thermodynamics properties involving in heat transfer and two-phase flow consist of latent heat, specific heat capacity, viscosity, surface tension, density, etc. However, individual working fluid type has different quantity in each property. Identification on working fluid type by only one thermodynamics property cannot be done successfully. It can be found from literature reviews on past studies that most of the studies on effect of working fluids on thermal performance of CLPHP frequently defined the latent heat as a quantitative property to identify a type of working fluid because heat transfer mechanism inside the CLPHP can be maintained due to evaporation and condensation of the working fluid, which directly relate to the latent heat. However, it was found that when latent heat of working fluid increased, thermal performance of the vertical CLPHP has possibility to change in both ways, i.e., increase and decrease. Results from the past studies that the thermal performance increased as an increase of the latent heat are concluded as follows: When working fluid changed from ethanol to water, or the latent heat increased from 970 to 2,400 kJ/kg (properties refers at 50°C), heat transfer rate increased from 560 to 940 W [5], and 55 to 58 W [6], respectively. When working fluid

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changed from FC-72 to ethanol and water, or the latent heat increased from 88 to 970 and 2,400 kJ/kg, thermal resistance decreased from 0.73 to 0.59 and 0.41°C/W, respectively (a decrease in thermal resistance implies to an increase in thermal performance) [7]. On contrary, Results from the past studies that the thermal performance decreased when the latent heat increased are concluded as follows: When working fluid changed from ethanol to water, or the latent heat increased from 970 to 2,400 kJ/kg, heat transfer rate decreased from 120 to 50 W, respectively [5]. When working fluid changed from R123 to ethanol and water, or the latent heat increased from 150 to 970 and 2,400 kJ/kg, heat flux decreased from 19 to 13 and 12 kW/m^2, respectively [8]. Moreover, when working fluid changed from R134a to MP39, or the latent heat increased from 181 to 190 kJ/kg, heat transfer rate decreased from 3,233 to 2,666 W, respectively [9]. Another study found that when the latent heat of evaporation increased from 161 to 1,000 and 2,382 kJ/kg, the thermal resistance increased from 3.3 to 5.4 and 9.0 m^2·K/kW, respectively [10].

Reason why effect of latent heats on thermal performance of the vertical CLPHP is not clear as mentioned above since other thermodynamics properties also has strong influence on heat transfer mechanism and circulation of each working fluid. In addition, geometrical parameters of the CLPHP also influence on thermal performance depending on working fluid type [11], [12]. Therefore, a group of dimensionless numbers or "correlation" should be considered and analyzed to be the quantitative property or parameter to identify the working fluid type and to predict the thermal performance of the vertical CLPHP. Since each dimensionless number in a correlation contains different thermodynamics properties of the working fluid and/or geometrical parameters of the heat pipe and the dimensionless numbers have no dimension or unit, the correlation is valid for any working fluids with wider condition. In order to investigate effect of dimensionless numbers on thermal performance of the vertical CLPHP and to establish a correlation to predict the thermal performance of vertical CLPHP according to working fluids from all corresponding dimensionless numbers, these become the significance and the objectives of this study. The correlation established in this study will be very useful for designers, industries, and people who are interesting in applications of the vertical CLPHP since the correlation can be use as a mathematical tool, which is a guideline for selecting the working fluid that is suitable for individual application. Moreover, obtained results and discussions also become a valuable basic knowledge for heat pipe researchers.

II. Experimental Setup and Procedure

The CLPHPs used in the experiment were made of long copper capillary tubes with inner diameters (D_i) of 1.50, 1.78, and 2.16mm and bent into 26 turns (N) (number of meandering turns in this study were counted from the evaporator section). Then, both ends were connected together to form a loop. The evaporator (L_e), adiabatic (L_a), and condenser (L_c) section length was equal to 50 and 150 mm. The CLPHP was arranged in the vertical plane and it is named shortly as "VCLPHP" throughout the article beyond this point. Chromel-Alumel thermocouples (Omega, Type K, accuracy ± 0.5°C) were used for temperature measurement. The thermocouples were installed on the outer surface in the middle of each part of the VCLPHP as follows: 10 points on evaporator section (T_e), 8 points on each adiabatic (T_a) and condenser section (T_c) as shown in Fig. 2. Moreover, two points of thermocouples were installed on each inlet and outlet of evaporator jacket ($T_{h,in}$ and $T_{h,out}$) and condenser jacket ($T_{cl,in}$ and $T_{cl,out}$) for measuring temperature difference across the evaporator and condenser section respectively, and also in the air for measuring the ambient temperature ($T_{ambient}$). R123, R141b, acetone, ethanol, and water were chosen as working fluids in this study because they had clearly different thermodynamics properties. The filling ratio was 50% of total volume since the best thermal performance of the VCLPHP can be obtained within this ratio [13].

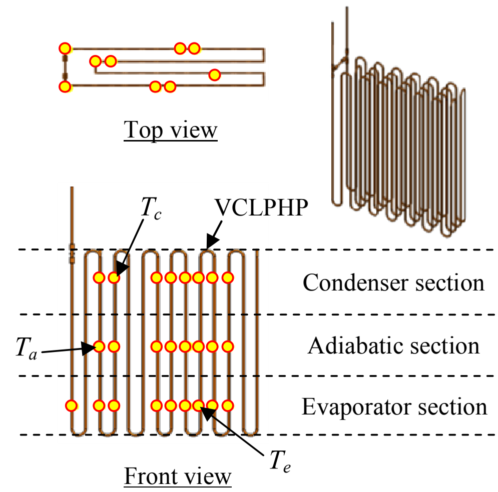

Fig. 2 Vertical CLPHP

A schematic diagram of the whole experimental setup is depicted in Fig. 3. The evaporator and condenser sections were fully enveloped in a zinc heating and cooling jacket respectively in order to circulate heating and cooling medium into each jacket. Heat was supplied to the evaporator section by circulating of hot distilled water of which inlet temperature was constantly controlled at 80°C by a hot bath (Haake, N6, accuracy ± 0.01°C). The condenser section then released the heat to aqueous solution of ethylene glycol (50% by volume) of which inlet temperature and mass flow rate were adjusted to keep the adiabatic temperature constantly at 50°C entire the test by a cold bath (Bitzer, D7032, accuracy ± 1°C). The mass flow rate could be obtained after the cooling medium, which flowed in a specified time, was weighed by the digital scale (Ohaus, Adventure, accuracy ± 0.01g) and simultaneously counted for the time by the high precision stopwatch (Casio, HS70W-1D, accuracy ± 0.001s). All sections including cooling medium hoses were well insulated by using a thermal-insulated sheet (Aeroflex, 3/8 in. thickness). At specified points, the temperature was monitored and recorded by a data logger (Brainchild, VR18, accuracy $\pm 0.1^{o}$C).

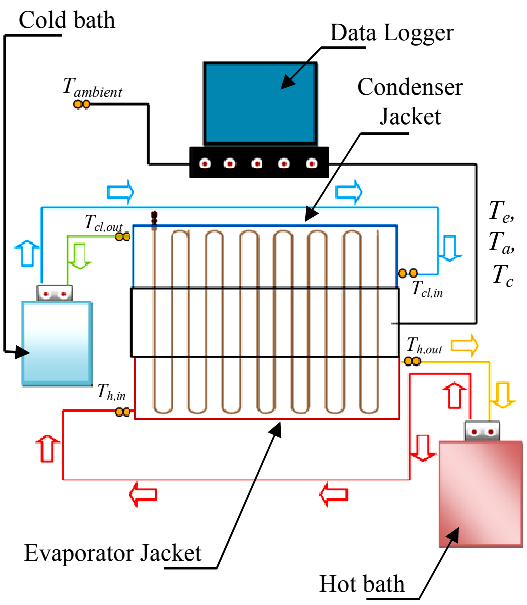

Fig. 3 Experimental setup

The procedure for the experiment is as follows: The VCLPHP successfully filled with R123 as working fluid was firstly attached to a test rig. The hot and cold bath were then started to flow the heating and cooling medium, respectively. All controlled parameters were adjusted and maintained constantly. After the system was in steady state, the evaporator, adiabatic, and condenser section temperatures, and also the temperature difference across the condenser section were simultaneously recorded in order to calculate the heat flux ($\dot{q}_c$) at specified times by means of the calorific as (1). The advantage of this way of measuring is that the actual throughput heat along the VCLPHP can be obtained. Subsequently, the VCLPHP was removed from the rig to be drained, evacuated, and filled in with another working fluid, and then the above mentioned procedure was repeatedly conducted until all working fluids were completely investigated.

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

where $\dot{q}_c$ is heat flux (kW/m^2), $\dot{m}_c$ is mass flow rate of the cooling medium (kg/s), c_{pc} is the specific heat of the cooling medium (kJ/kg·K), $(T_{out} - T_{in})_{cl}$ is the difference in temperature of the cooling medium (K), and A_c is the inner surface area of the tube in the condenser section (m^2).

In the calculation to determine heat flux from (1), it was important to know the mass flow rate of the cooling medium. This was accomplished by using the digital scale and both of the inlet and outlet temperature by using the data logger and thermocouples. Since these values were obtained from instruments, it was important to determine the error due to this measurement in order to correctly analyze the results. The error of the obtained heat flux can be calculated from (2).

$$d\dot{q} = \sqrt{\left(\frac{\partial \dot{q}}{\partial \dot{m}_c} d\dot{m}_c\right)^2 + \left(\frac{\partial \dot{q}}{\partial T_{cl,out}} dT_{cl,out}\right)^2 + \left(\frac{\partial \dot{q}}{\partial T_{cl,in}} dT_{cl,in}\right)^2} \tag{2}$$

where $d\dot{q}$ is the error of the heat flux, $d\dot{m}_c$ is the accuracy from measuring the mass flow rate of the cooling water, $dT_{cl,in}$ and $dT_{cl,out}$ is the accuracy from measuring the inlet and outlet temperatures respectively.

In this study, the heat fluxes where the error was lower than 30% of the calculated heat flux were defined to pass the criterion to be analyzed in the next step. Moreover, since each working fluid has different "critical heat flux" – the highest thermal performance that heat pipe can transfer before dry-out of liquid working fluid inside the evaporator section will occur, comparison in thermal performance of each working fluid through the heat flux is not reasonable. The working fluid with relatively low critical heat flux generally has lower transferred heat flux than the higher one. In order to normalize the experimental data, Kutateladze number (*Ku*) was chosen to be a representative of the thermal performance. *Ku* is a well known dimensionless number involving in heat transfer in the heat pipe. It is a ratio of transferred heat flux to calculated critical heat flux as found from (3). The higher *Ku* implies that the heat pipe has higher thermal performance and it operates closer to the critical state, and vice versa for the lower *Ku*. Relations between thermal performances and working fluids through each dimensionless number depicted in the next section will show the *Ku* on the vertical axis for comparing the thermal performance of the VCLPHP depending on dimensionless number on the horizontal axis.

$$Ku = \frac{\dot{q}_c}{\rho_v h_{fg} \left[\sigma g \left(\frac{\rho_l - \rho_v}{\rho_v^2} \right) \right]^{1/4}} \tag{3}$$

when g is gravitational acceleration (m/s^2), h_{fg} is latent heat (kJ/kg), ρ_l is liquid density (kg/m^3), ρ_v is vapor density (kg/m^3), and σ is surface tension (N/m).

III. Results and Discussions

A. Effect of Prandtl Numbers on Thermal Performance

Prandtl number (*Pr*) is a dimensionless number involved with working fluid's properties. It is implied to a ratio between dynamic viscosity and thermal diffusion of the working fluid. Pr can be in two phases, i.e., Prandtl number of liquid working fluid (Pr_l) and Prandtl number of vapor working fluid (Pr_v). However, from the experiment, since Pr_v rarely had an effect on thermal performance, Pr_v could be neglected from this analysis. Equation for Pr_l is expressed in (4) when $c_{p,l}$ is liquid specific heat capacity (kJ/kg·K), μ_l is liquid viscosity (kg/m·s), and k_l is liquid thermal conductivity (kW/m·K).

$$Pr_l = \left(\frac{c_{p,l}\mu_l}{k_l} \right) \tag{4}$$

It was found from the experiment that when Pr_l increased or working fluid respectively changed from acetone to water, R141b, R123, and ethanol –which has the highest Pr_l, the thermal performance increased. This effect of Pr_l on thermal performance very well agreed with the results from the past study [5] as shown in Fig. 4.

Working fluid with higher liquid specific heat capacity can transfer higher heat quantity since the working fluid carries the higher quantity of the heat from evaporator to release at condenser section, comparing within the same working fluid's mass. This causes the thermal performance to increase. Moreover, Increase in liquid viscosity leads flow velocity of the working fluid to be retarded; therefore, time duration that the working fluid receives and releases the heat in the evaporator and condenser section is longer, respectively and the thermal performance consequently increases. However, when the working fluid with higher liquid thermal conductivity is used, the heat will freely diffuse in liquid slug with higher portion compared with remaining portion of the heat that causes evaporation, then, thermal performance decreases. These are the physical reasons to support that when Pr_l increases, *Ku* or thermal performance of the VCLPHP increases. As in this study, ethanol has the highest Pr_l compared to other working fluids, e.g., R123, R141b, acetone, and water. This tendency causes the VCLPHP with ethanol to have the highest thermal performance within a viewpoint of Pr_l only.

B. Effect of Bond Numbers on Thermal Performance

Bond number (*Bo*) is a dimensionless number involved with working fluid's properties and geometry of the heat pipe. It is implied to a ratio between buoyancy force and surface tension of the working fluid. *Bo* can be calculated from (5) when D_i is internal diameter of the heat pipe (m).

$$Bo = \frac{g\left(\rho_l - \rho_v\right) D_i^2}{\sigma} \tag{5}$$

It was found from the study that when *Bo* increased, the thermal performance decreased. However, the obtained relation is in an opposite direction to that of the relation found in the past study [5] as shown in Fig. 5. This argument occurred according to differences in number of variable parameters, such as working fluid types, geometry of the heat pipe, and also experimental condition. Nevertheless, physical reasons can be theoretically explained to support both tendencies as follows.

In a case of the relation that when *Bo* increases, *Ku* or thermal performance decreases. This is primarily affected from a decrease in surface tension appearing in a denominator of *Bo*. When surface tension decreases, vapor tends to form into small bubbles instead of long vapor plugs. Since smaller bubbles have lower vapor's mass than longer bubbles, this situation can be implied that the heat in evaporator section transfers out from the tube's surface by means of evaporation with lower quantity in a case of smaller bubbles. This causes the working fluid to transfer the heat less continuously and the thermal performance is subsequently lower. On the other hand, another tendency was found from the past study that when *Bo* increases, *Ku* or thermal performance increases. This is a major effect due to buoyancy force. When a difference between liquid and vapor densities increases, it can be implied that vapor plug is obviously lighter than liquid slug compared in the same volume. This causes the buoyancy force to be higher and vapor plugs can flow from evaporator to condenser section which locates at the top of the VCLPHP with shorter time duration. Moreover, an increase in the internal diameter of the heat pipe promotes the working fluid's circulation throughout the heat pipe since a cross-sectional area of a flow passage is wider and the frictional force at contact surface between the working fluid and inside tube's wall decreases. The working fluid transfers the heat more actively and thermal performance consequently increases.

It could be seen from both experimental results and physical reasons as mentioned above that the effect of *Bo* on thermal performance is still not clear. Thus, one or more dimensionless number that has stronger effect on the thermal performance than that of *Bo* is possibly existed, and also both thermodynamics properties of the working fluid and geometry of the heat pipe must involve in this dimensionless number as same as a case of *Bo*.

C. Effect of Karman Numbers on Thermal Performance

Karman number (*Ka*) is a dimensionless number involved with working fluid's properties and geometry of the heat pipe, which is similar to *Bo*. It represents a ratio between driving force and frictional force of the working fluid. *Ka* can be written as in (6) when $(\Delta P)_{sat}^{e\text{-}c}$ is saturated pressure difference between evaporator and condenser section (Pa), L_{eff} is effective length of the heat pipe (m), which is calculated from $(0.5L_e+L_a+0.5L_c)$ [6].

$$Ka = \frac{\rho_l \left(\Delta P\right)_{sat}^{e\text{-}c} D_i^3}{\mu_l^2 L_{eff}} \tag{6}$$

It was found from the experiment that when *Ka* increased, the thermal performance increased, exceptionally to ethanol since the thermal performance according to ethanol isolates from overall data. Nevertheless, relation between *Ka* and *Ku* or thermal performance very well agreed with the results from the past study [5] as shown in Fig. 6.

Since driving force is the main mechanism of working fluid's circulation in the CLPHP, the working fluid flows forth and back between evaporator and condenser section due to this force [1]. Magnitude of the driving force depends on saturated pressure difference of the working fluid between evaporator and condenser section. Therefore, when the pressure difference increases or *Ka* increases, the driving force increases. This causes the working fluid's flow velocity and heat transferred quantity to increase. In addition, when internal diameter increases, pressure loss along flow passage between evaporator and condenser section decreases. Net pressure difference of CLPHP with bigger internal diameter is higher than that of the smaller one. Since an increase in internal diameter strongly diminishes the effect of pressure difference, when internal diameter increases, *Ka* increases, and *Ku* or thermal performance obviously increases. On contrary, increases in liquid viscosity and effective length of the heat pipe causes a decrease in *Ka* and then the thermal performance, since the working fluid's flow velocity decreases and flow distance, which directly affects to an increases in the pressure loss, increases, respectively. These phenomena originally are the major cause of degradation in the thermal performance.

D. Effect of Jacob Numbers on Thermal Performance

Jacob number (*Ja*) is a dimensionless number involved with working fluid's properties. It implies to be a ratio of heat quantity that the heat pipe can transfer between two different mechanisms, i.e., latent heat and sensible heat. The former mechanism corresponds to the working fluid's phase change, the latter relates to the working fluid's temperature change. *Ja* can be expressed as in (7) when ΔT_{sat}^{e-c} is saturated temperature difference between evaporator and condenser section (K).

$$J_a = \frac{h_{fg}}{C_{p,l}\Delta T_{sat}^{e-c}} \tag{7}$$

It could be seen that when *Ja* increased, the thermal performance increased in a case of the VCLPHP with the evaporator section length of 50mm. On the other hand, opposite tendency was found in a case of the VCLPHP with longer evaporator section length, i.e., 150mm. This relation very well agreed with the relation obtained from the past study [5] as shown in Fig. 7.

Working fluid with higher ratio between latent to sensible heat or higher *Ja* causes the CLPHP to have higher thermal performance, since the working fluid transfers greater quantity of the heat by means of the phase change from liquid to vapor in evaporator section relatively to remaining heat quantity 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 near the theoretically optimized ability.

However, when evaporator section length increases, distance that the working fluid has to flow towards the condenser section increases. The frictional force and also pressure loss increase. From this reason, heat will be transferred with un-continuity. This causes the heat transferred through the CLPHP with higher *Ja* not to increase as usual. This is an obvious evidence to support that the evaporator section length has significant effect on the thermal performance. Therefore, another dimensionless number, which has influence over *Ja* and involved with the evaporator section length, has to be further investigated. In addition, it was found from some past studies that when a portion of the sensible heat decreased or *Ja* increased, the thermal performance decreased. Reason was discussed and realized that the CLPHP primarily transfers the heat by means of temperature change mechanism rather than the phase change [13]-[16]. On the recent day, actual portion of heat transferred in the CLPHP according to latent heat and sensible heat is still in a black box. Therefore, when *Ja* increases, the thermal performance can possibly change in either direction.

E. Effect of Aspect Ratios on Thermal Performance

Aspect ratio (L_e/D_i) is a dimensionless number involved with geometry of the heat pipe. It represents a ratio between evaporator section length (L_e) and internal diameter (D_i). It could be seen that when Aspect ratio increased, thermal performance decreased. This result very well agreed with the results from the past study [5] as shown in Fig. 8.

The evaporator section length and the internal diameter simultaneously affect to the thermal performance of the heat pipe. Working fluid's flow pattern inside the tube depends on the evaporator section length. The working fluid with "slug flow" pattern can be observed when the evaporator section length is short, e.g., 50 mm as in this study. Vapor plugs in the slug flow form in a core of the tube surrounding with liquid film. Since liquid film contacts around the tube's wall in the evaporator section, heat input can transfer to the liquid film directly and evaporation rate consequently increases. This causes the thermal performance to increase. On contrary, the working fluid's flow pattern in the CLPHP with longer evaporator section length will change into "churn flow" in which vapor's shape is not stable. Liquid working fluid can be entrained as droplets into vapor core. Heat input cannot conduct through vapor core to liquid droplets easily. The thermal performance according to this flow pattern will be low. Moreover, it was found that when the distance between evaporator and condenser section increases, the working fluid's circulation velocity decreases. This leads the thermal performance to decrease. In addition, it was found that an increase in the internal diameter not only causes the heat transferring area between the heat pipe and working fluid to increase but also causes the cross-sectional area of the working fluid's flow inside the CLPHP to increase. When the CLPHP has larger internal diameter or wider cross-sectional area of the flow passage, vapor plugs evaporating in the evaporator section consequently flow toward the condenser section more continuously with higher working fluid's quantity. The pressure loss of working fluid's flow decreases. From this physical reason, the CLPHP can transfer more heat and the thermal performance increases. It can be seen from the experimental results that geometric of the heat pipe strongly influences on the thermal performance. It could be concluded that the CLPHP with low Aspect ratio (approx. 20 to 40) has higher thermal performance comparing with the CLPHP with high Aspect ratio (approx. 70 to 1000).

F. Correlation to Predict Thermal Performance

Correlation is a mathematical product from a combination among 5 dimensionless numbers that affect to the VCLPHP as discussed above. More precise thermal performance according to certain operating condition of the VCLPHP can be predicted by this established correlation. The correlation was started from a function as of in (8).

$$Ku = f\left(Pr_l, Ja, L_e / D_i, Ka, Bo\right) \quad (8)$$

All dimensionless numbers were arranged into the correlation by means of the least-square curve fitting. Experimental data involving in the curve fitting consisted of the results obtained from this study and from the past studies on VCLPHPs of [5] and [17], in order to expand the correlation's available condition and to increase prediction's precision of the correlation. Availability of combining these results from other studies into the correlation's establishment was acceptable, since scopes of experiment in both studies were nearly the same. Parameters, which were identical between [5] and this study, were shown as follows. Working fluids were R123, ethanol, and water. Evaporator section lengths were 50, and 150 mm. Inlet temperature of the heating medium was controlled at 80°C. Only inlet temperature of the cooling medium (20°C) and internal diameters (1.06 and 2.03 mm) were slightly different. Comparison between [17] and this study shows the identical parameters as follows. Working fluids were R123, and water. Evaporator section lengths were 50, and 150 mm. Inlet temperature of the heating medium was controlled at 80°C. Adiabatic section temperature was constantly adjusted at 50°C. Only number of meandering turns (5, 7, 10, 16, and 30 turns) and internal diameter (2.03 mm) were nearly the same. The correlation to predict the thermal performance of the VCLPHP was finally established as expressed in (9). Thermal performances calculated from the correlation (Ku_{model}) and obtained from the experiment (Ku_{exp}) were plotted against each other to verify the precision of the correlation as shown in Fig. 9. Percentage of data deviation between Ku_{model} and Ku_{exp} was ± 37%. It should be noted that quantity of each thermodynamics property involving in the correlation is defined to be the quantity which is corresponding to the adiabatic section temperature. In a design that the adiabatic section temperature is not exactly known, the adiabatic section temperature can be estimated from $T_a = (T_e+T_c)/2$, when T_e and T_c is evaporator and condenser section temperature, respectively.

$$Ku_{model} = 5.27x10^{-2}\, Pr_l^{0.522}\, Ja^{-0.507} \left(L_e / D_i\right)^{-0.727} Ka^{0.057}\, Bo^{-0.164} \quad (9)$$

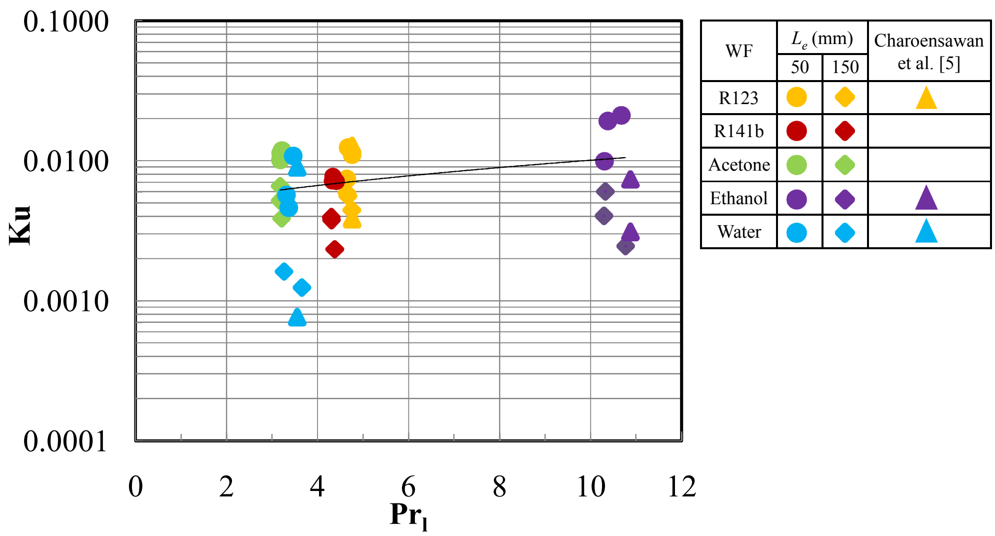

Fig. 4 Effect of Prandtl numbers on thermal performance

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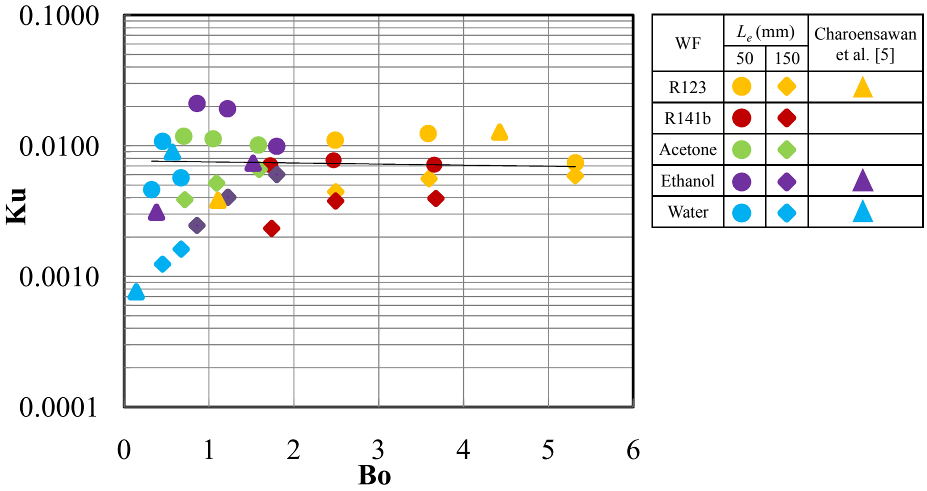

Fig. 5 Effect of Bond numbers on thermal performance

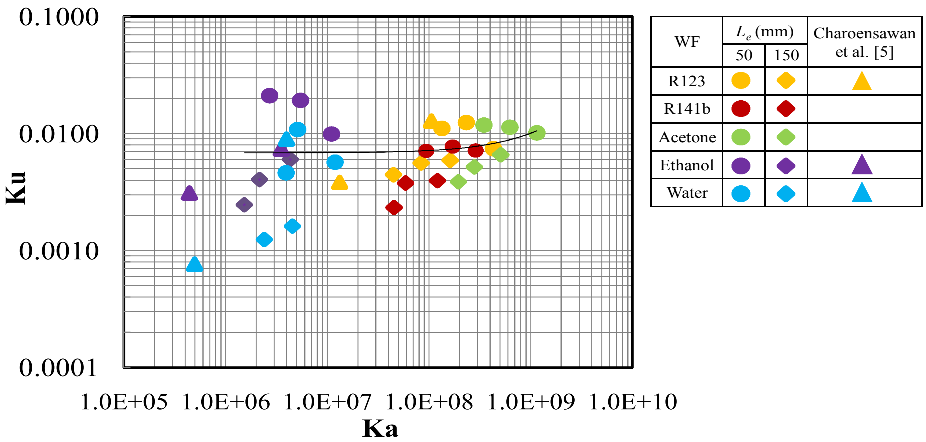

Fig. 6 Effect of Karman numbers on thermal performance

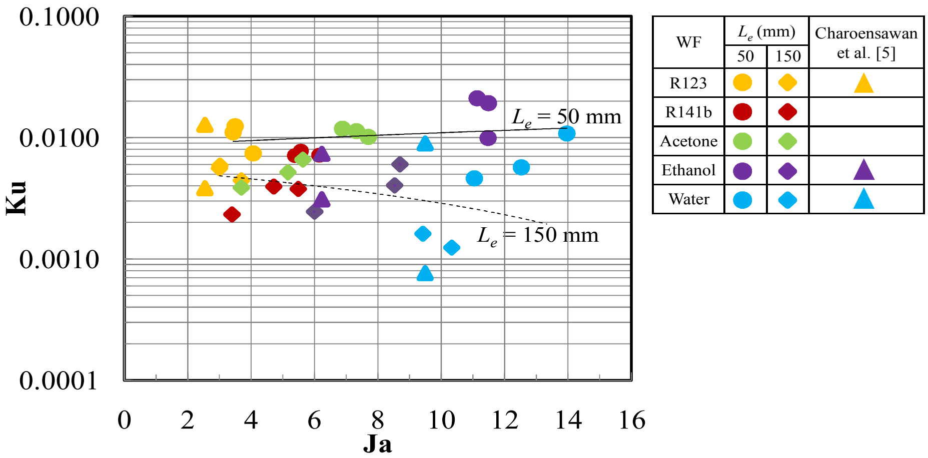

Fig. 7 Effect of Jacob numbers on thermal performance

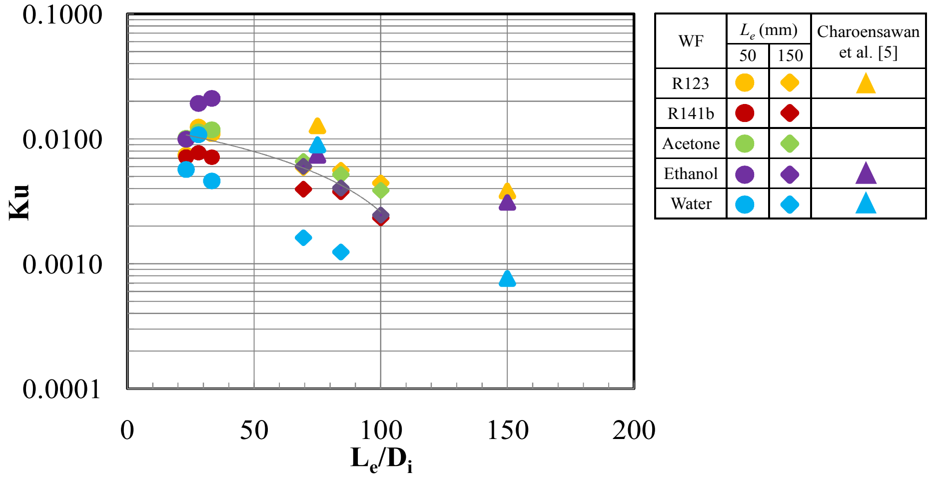

Fig. 8 Effect of Aspect ratios on thermal performance

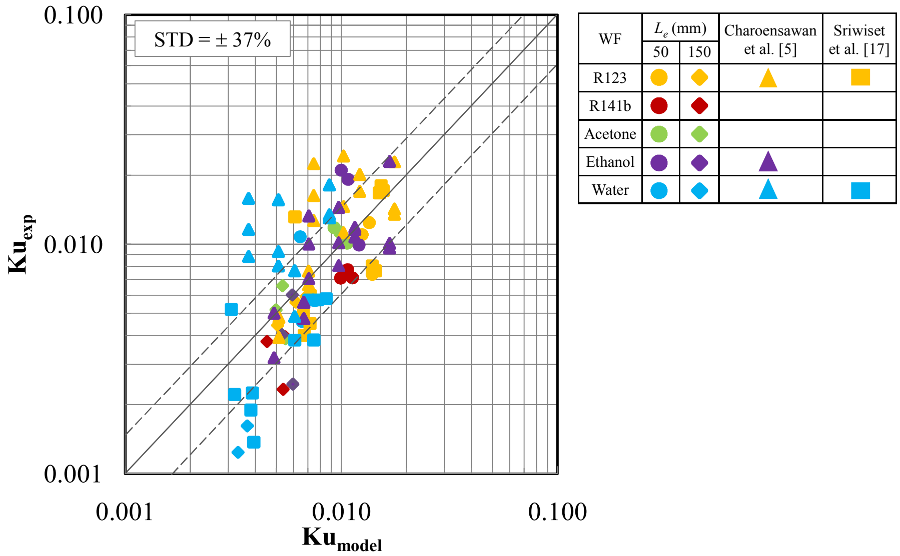

Fig. 9 Comparison between Ku_{model} and Ku_{exp}

IV. Conclusions

Effects of dimensionless numbers on thermal performance of the vertical closed-loop pulsating heat pipe (VCLPHP) have been thoroughly investigated and a correlation to predict the thermal performance of the VCLPHP has been successfully established in this study. Thermal performance was represented in a term of Kutateladze number (*Ku*), which is a dimensionless number involving in the heat transfer in heat pipes. It can be concluded that when Prandtl number of liquid working fluid (Pr_l), and Karman number (*Ka*) increases, thermal performance increases. On contrary, when Bond number (*Bo*), Jacob number (*Ja*), and Aspect ratio (L_e/D_i) increases, thermal performance decreases. All dimensionless numbers that have effect on the thermal performance were analyzed together to establish the correlation to predict more precise thermal performance. The correlation will be very useful for designers, industries, and people who are interesting in applications of the vertical CLPHP and also a valuable basic knowledge for heat pipe researchers.

Acknowledgment

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