Research Article

Effect of Centrifugal Acceleration on Heat Transfer Characteristics of Rotating Closed-Loop Pulsating Heat Pipes

Kritsada On-ai*, Niti Kammuang-lue, Phrut Sakulchangsatjatai and Pradit Terdtoon

Department of Mechanical Engineering, Faculty of Engineering, Chiang Mai University, Chiang Mai, Thailand
** Corresponding author. Email address: gooddevil555@hotmail.com*

Abstract

The objective of this paper is to qualitatively study the effect of centrifugal acceleration on heat transfer characteristic of a rotating closed-loop pulsating heat pipe (RCLPHP). RCLPHP can be applied to cool rotating devices, such as disc brake or steam turbine. It improves the lifetime and reduces the wear of the device. In recent studies, effects of several parameters on thermal performance of CLPHP have been investigated e.g. inner diameter, length of evaporator section, meandering of turn and working fluid. Some researchers derived these parameters in dimensionless form, and then established correlation to predict thermal performance of CLPHP at specified inclination angles. Another parameter that affects rotating or moving heat pipe is centrifugal acceleration. The induced internal centrifugal acceleration of the RCLPHP affects the circulation of working fluid. When flow direction of condensate is in the same direction as the acceleration, the centrifugal acceleration is defined to be positive. In turn, the condensate is in the counter-flow direction to the acceleration, if centrifugal acceleration is negative. This acceleration affects the circulation of liquid phase. Because of heavier mass of the liquid, most of it accumulated at the end part of evaporator section. When the RCLPHP is heated, working fluid in this section changes from liquid to vapor phase. Then it circulates to condenser section, or another end, and condensed. When centrifugal acceleration is increased, thermal resistance decreases. The condensate can quickly circulate to evaporation section because working fluid velocity is higher than those at lower acceleration. In the future, researches to quantitatively study on the effects of centrifugal and parameters which related to the thermal performance of RCLPHP will be further conducted.

***Key Words*:** Rotating closed-loop pulsating heat pipe; Centrifugal acceleration; Thermal performance

Introduction

Pulsating heat pipes (PHPs) were a small capillary tube which was bent into meandering turn, inside the tube was vacuumed and contained working fluid. Flow patterns of working fluid inside the PHPs were vapor plug and liquid slug along an axial length of the tube (Akachi et al., 1995). The PHPs consisted of 3 sections, i.e., evaporator, adiabatic, and condenser sections. Heat transfer mechanisms corresponded to working fluid's circulation from evaporator section to condenser section. Heat could be transferred by means of the latent heat and the sensible heat. The PHPs could be classified into 3 types (Maezawa et al., 1995): First, a closed-end pulsating heat pipe (CEPHP), which both ends was disconnected. Therefore, the flow of the working fluid is restricted from both sides which affect the lower heat transfer of the PHPs. Second, a closed-loop pulsating heat pipe (CLPHP) that both end of the tube was connected, then, the working fluid could circulate freely along entire axis length. However, a flow direction of the working fluid was not always in the same direction. The last is a closed-loop pulsating heat pipe with check valve (CLPHP/CV), which is the same as the CLPHP, but it has one or more check valve for restrict the circulating direction. The latter has the best thermal performance; in turn, it had the most

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complication for construct, because dimensions of the check valve were much smaller than a diameter of the capillary tube.

Majority of researches on CLPHP dealing with experimental works of stationary devices have been completely conducted by, e.g., Charoensawan and Terdtoon (2008), Khandekar et al. (2003), On-ai et al. (2013) and Sriwiset et al. (2013). These researchers investigated on parameters that affect the heat transfer of the CLPHPs, such as, inclination angles, working fluids (WF), inner diameters (Di), evaporator lengths (Le), filling ratios (FR) and heat inputs, as concluded in Table 1. In addition, they considered these parameters to establish correlations to predict thermal performance of the stationary CLPHPs. In recent day, the CLPHP, which was applied on the rotating devices, was firstly presented by Aboutalebi et al., 2013. It was called as a "rotating closed-loop pulsating heat pipe" or "RCLPHP". The RCLPHP in previous study was bent to form a 4-pedal flower shape and was studied on the effect of gravitation acceleration on thermal resistance of the RCLPHP. Heating electrical wire was twisted around U-shape in evaporator section as a heat input method and the condenser section was cooled by the air. The working fluid was water. An inner diameter was 2.03 mm. The experimental data showed that thermal resistanceof the RCLPHP decreased as an increase in a rotating speed. This research was similar to the one of Waowaew et al. (2003) who studied on a thermosyphon applied to cool the rotating devices. Waowaew et al. (2003) created a correlation consisting of centrifugal acceleration, working fluid, inner diameter, and evaporator length. Conclusion data from both researchers are shown in Table 1. As mentioned above, however, it was found that the experimental data of the RCLPHPs were not enough to establish the correlation. In the light of this reason, the authors are interested to investigate parameter affecting on the RCLPHPs. However, before a quantitative experiment on the parameters will be done, a qualitative analysis and also a physiology on thermal performance of the RCLPHP must be firstly considered. Moreover, the main parameter discussed in this analysis is a "centrifugal acceleration", which can be lower than, higher than, or equal to the gravitational acceleration related on thermal performance of the stationary CLPHPs.

The purpose of this research is to qualitatively study of effect of centrifugal acceleration on heat transfer characteristic of a rotating closed-loop pulsating heat pipe. A guideline for establishing the correlation to predict the thermal performance of the RCLPHP in which the effect of centrifugal acceleration is included should be investigated.

Various Forms of Correlation to Predict Thermal Performance of CLPHPs

Correlation is an equation with various independent parameters used for predicting the thermal performance of the CLPHPs and used to design heat pipes. From the past researches, parameters which affect to heat transfer characteristic of the CLPHPs have been studied. The Correlation for the CLPHPs at any inclines was established. Khandekar et al. (2003) presented the correlation to predict heat fluxes of the CLPHPs. Accuracy of the prediction was acceptable and it also showed the influence of a meandering turn on thermal performance, as shown in Equation (1). The effect of centrifugal acceleration is, however, not included in this correlation.

$$q_{pre} = 0.54\,\mathrm{Ka}^{0.47}\,Ja^{1.43}\,\mathrm{Pr}_{liq}^{0.27}\,N^{-0.27}\,(\exp(\beta))^{0.48} \quad (1)$$

Charoensawan and Terdtoon (2008) investigated and created the correlation for a horizontal CLPHP. It can predict heat transfer at any filling ratios by using the modified Jacob number. In condenser section, the air was used asa coolant. Moreover, one of dimensionless numbers in this equation was the ratio of conductivities between coolant and ambient air to increased accuracy of a prediction, as shown in Equation 2. However, this correction was established from the experiments in which the working fluid's flow inside the CLPHP was not in the same direction as the gravity.

$$Ku_{pre} = 21.3 \times 10^{-9}\,\mathrm{Pr}_l^{\,0.75}\,Ja^{*\,-0.38}\,Bo^{-0.84}\,Ka^{0.58}\,(k_c/k_a)^{1.21} \quad (2)$$

On-ai et al. (2013) presented two new correlations established from their experimental results and those of other researches to predict the thermal performance of the vertical and

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horizontal CLPHP saffected by the working fluid types as shown in Equation (3) and (4), respectively. These correlation scan extensively predict the thermal performance in only a case of the bottom heat mode. Since these correlations were established from the static condition, they were not suitable to predict the thermal performance when CLPHPs were rotated.

$$Ku_{model} = 5.27 \times 10^{-2}\, Ka^{\,0.057}\, Pr_l^{\,0.522}\, Ja^{-0.507}\, Bo^{-0.164} (L_e/D_i)^{-0.727} \quad (3)$$

$$Ku_{model} = 9.62 \times 10^{-3}\, Ka^{\,0.152}\, Pr_l^{\,0.905}\, Ja^{-0.110} (L_e/D_i)^{-1.212} \quad (4)$$

Sriwiset et al. (2013) showed correlationto determine the optimum number of turn for the vertical CLPHPs. It was developed from Equation (1) with their additional experimental data, as shown in Equation (5).

Table 1 Parameters in the past research

Inclination angle	Researchers	Working fluid	Centrifugal accelearion (m/s^2)	Di (mm)	Le (mm)	Turn	Filling ration (%)	Heat input (W)	Te (ºC)	Ta (ºC)	Tc (ºC)	Coolant
Rotation	Adoutalebi et al. (2013)	water	1g 11g 33g 67g 112g 169g 238g	2.00	100	4	25 50 75	25 40 55 70 85 100				air
	Waowaew et al.(2003)*	R123 ethanol water	0.2g 1g 3g 5g 10g	11.00 26.00 50.40	5 10 20 40		60% of evaporator			90	20	air
0 - 90º	Khandekar et al.(2003)	R123 ethanol water		2.00 1.00	100 150	5 7 11 16 23	50		80		20	aqueous solution of ethylene glycol
0º	Charoensawan and Terdtoon (2007)	ethanol water		1.00 1.50 2.00	50 150	5 11 16 26	30 50 80		40-90	50	25	air
0º, 90º	On-ai et al. (2013)	R123 R141b ethanol water		1.50 1.78 2.16	50 150	26	50		80 80	50		aqueous solution of ethylene glycol
	Sriwiset et al. (2013)	R123 water		2.03	50 150	5 7 10 16 30	50		80			aqueous solution of ethylene glycol

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$$N = [(Ja^{1.43}\ \mathrm{Pr}^{0.27}\ Ka^{0.47}) / 50000]^{3.7040} \tag{5}$$

In addition, the correlations to predict the maximum heat flux at the critical state of the vertical and horizontal CLPHPs were established by Kammuang-Lue et al. (2010) and Kammuang-Lue et al. (2009), as shown in Equations (6) and (7), respectively. However, the effect of centrifugal acceleration did not involve in these correlations.

$$Ku_{max} = 6.25\ (\mathrm{CGV})^{0.34}\ (D_i / L_e)^{0.91}\ (L_t / L_e)^{-0.26} \tag{6}$$

$$Ku_{max,0} = 0.004849\ \mathrm{Bo}^{0.5696}\ Ja^{-0.1396}\ (L_e / D_i)^{-1.5341} (L_t / L_e)^{1.3733} \tag{7}$$

All correlations mentioned, had acceleration in term of Bo which was gravity (g = 9.81 m/s^2) only. For RCLPHP, centrifugal accelerations were lower, higher and equal to gravity. Therefore, when centrifugal acceleration is changed as in a case of the RCLPHPs, theses equation cannot be used to predict thermal performance. The correlations to predict the thermal performance of RCLPHP should be improved by adding dimensionless numbers consisting of the centrifugal acceleration.

Correlation Related to Centrifugal Acceleration of Thermosyphon

In the past studies, there were investigations on the thermal performance of the rotating heat pipes used for applying on the rotating devices. Aboutalebi et al. (2013) studied effect of the centrifugal accelerations, heat inputs, and filling ratios on thermal performance of the RCLPHPs, but the correlation was not established since the experimental data were not sufficient. In a case of the thermosyphon, Waowaew et al. (2003) investigated radially rotating thermosyphons and derived correlation to predict the heat flux of the thermosyphons, as shown in Equation (8)

$$q_{90}^{*} = 6\times10^{7}\left(\left(\frac{L_e}{D_i}\right)^{4.6} Fr^{0.5}\ Bo^{-0.85}\ Ek^{0.1}\ Ja^{-0.15}\ \mathrm{Pr}_v^{-0.35}\left(\frac{C_{pv}}{C_{pl}}\right)\left(1-\left(\frac{\rho_v}{\rho_l}\right)\right)^{0.19}\right)^{0.57} \tag{8}$$

When q_{90}^{*} is dimensionless heat flux at vertical position. Equation (8) showed the dimensionless numbers which related to the centrifugal acceleration and some terms can be applied with the RCLPHPs, e.g., Bond number (Bo), Ekman number (Ek), and Froude number (Fr). However, since Ek has the smallest effect on the heat transfer among the others. Then Bo and Fr were considered in this research to apply in the correlation to predict thermal performance of RCLPHP. The characteristic of RCLPHP is shown in Figure 1.

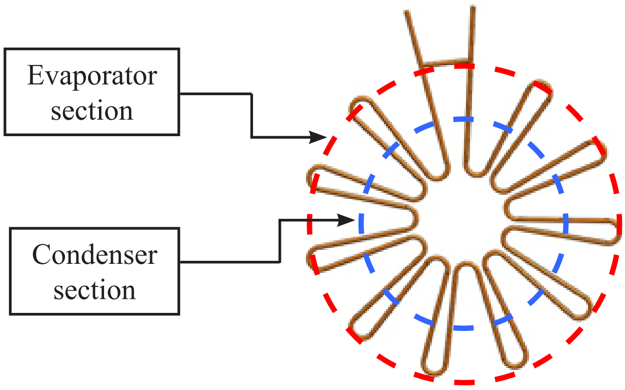

Figure 1 Characteristic of RCLPHP

Bo is a ratio of buoyancy force to surface tension. Direction of buoyancy force is opposite to gravitational acceleration for the CLPHP in bottom heat mode. In a case of the RCLPHP, the directions of buoyancy force and centrifugal acceleration are the same, which is leaving the center of a rotation. Buoyancy force affects the circulating velocity of the working fluid, which condensed in condenser section. It could drive liquid slug to evaporator section and received heat from heat source. When centrifugal acceleration increased, circulation velocity is increased and resulting in adecreasing in the thermal resistance. This result is shown in Figure 2. On the other hand, when Bo is less, vapor slug motion was obstructed from surface tension. Then, the working fluid slowly circulates from condenser section to evaporator section, thermal resistance consequently increased.

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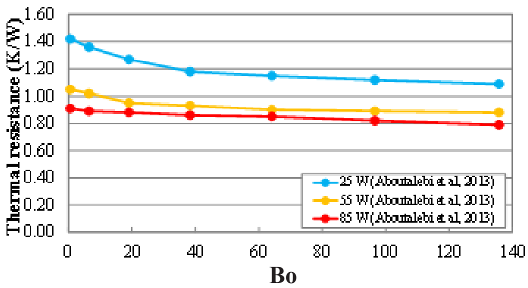
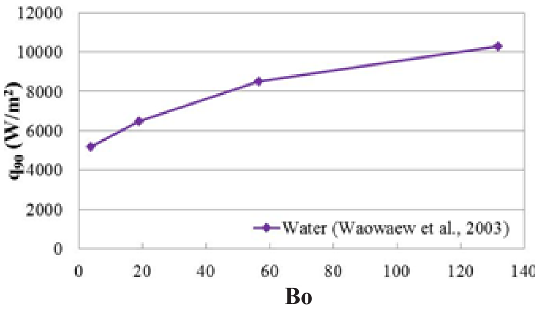


(a) Effect on thermal resistance



(b) Effect on heat flux

Figure 2 Effect of Bo on thermal performance

Fr is a ratio of inertia force to gravity force of liquid film that condenses and circulates to evaporator section. When centrifugal acceleration increases, the accelerating force exerting on the working fluid increases. Since liquid has higher weight than vapor, the centrifugal acceleration primarily affects the circulation of the liquid to flow to the evaporator section faster. This causes the thermal resistance to decrease. Effect of Fr on thermal resistance is shown in Figure 3.

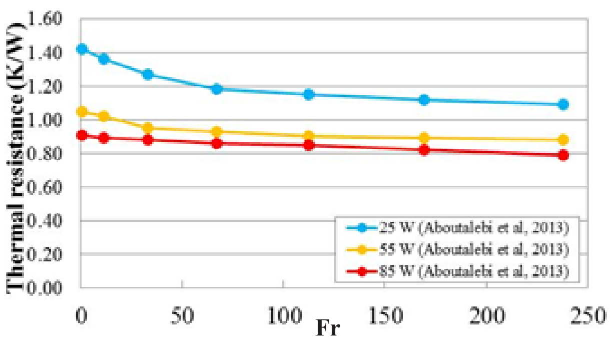


Figure 3 Effect of Fr on thermal resistance

The Figure 2 (a) and Figure 3 were rewritten from the data of Aboutalebi et al. (2013) to compare the effect of Bo and Fr on the thermal resistance. It is found that Bo and Fr had effect on the thermal performance of the RCLPHPs as a case of the rotating thermosyphon. From this point, the correlation to predict thermal performance of the RCLPHPs reasonably involved with the Bo and Fr as main parameters and the correlation must be essentially established.

Results and Discussions

From above mention, Equation (3) was established from a number of researches, therefore, it could be predicted the thermal performance of the CLPHPs in various conditions. Then, Fr was combined with a term of $Ku = a(Ka^{0.057}\ Pr_l^{0.522}\ Ja^{-0.507}\ Bo^{-0.164}(L_e / D_i)^{-0.727})^b\ Fr^c$ to establish the correlation to predict the thermal performance of the RCLPHPs. A dependent parameter was changed to be Ku, since Ku was a dimensionless and it could be used to compare experimental data from different conditions. A new correlation is shown in Equation (9), when ‘a’ was a coefficient of the correlation, ‘b’ and ‘c’ were the power of the dimensionless number.

$$Ku = a(Ka^{0.057}\ Pr_l^{0.522}\ Ja^{-0.507}\ Bo^{-0.164}(L_e / D_i)^{-0.727})^b\ Fr^c \qquad (9)$$

In addition, least square method was conducted to find the best value of ‘b’ and ‘c’ together with data obtained from the RCLPHP with 2.03-mm inner diameter, 100-mm evaporator length, 4-turns and water was used as working fluid (Aboutalebi et al., 2013). Comparison between dimensionless group and Ku model found that the coefficient of the correlationwas $1.6x10^{-6}$, as shown in Figure 4 and a new correlation is shown in Equation (10).

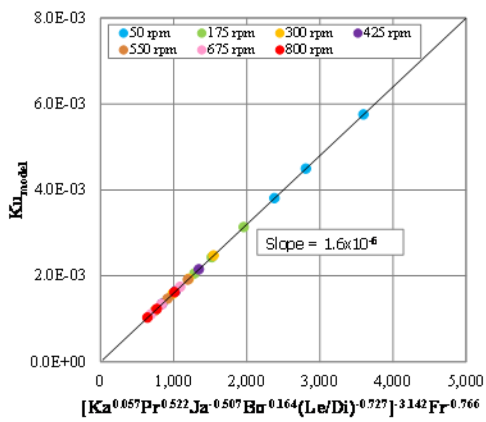


Figure 4 Coefficient of the new correlation of RCLPHP

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$$Ku^{*}_{model}=1.6\times10^{-6}[Ka^{0.057}Pr_l^{0.522}Ja^{-0.507}Bo^{-0.164}(L_e/D_i)^{-0.727}]^{-3.142}Fr^{-0.766} \quad (10)$$

It was found from the proposed correlation that, a power of Fr was negative. This means that when Fr increases, Ku decreases. It was opposite to the effect of Fr on physiology inside the RCLPHPs, since the centrifugal acceleration has stronger effecton theoretical critical heat flux than that of the experimental heat flux. The refore when centrifugal acceleration increased, theoretical heat flux increased. Comparison between Ku^{*}_{model} and Ku_{exp}, showed that, a standard deviation (STD) of error was 1%, as shown in Figure 5. However, if correlation was established from wider conditions or consisted with higher number of independent parameters, such as, inner diameters, evaporator lengths, working fluid types, numbers of turn, and filling ratios, the STD of the correlation would be increased, since the boundary condition was widened. However, in this research, the correlation could be predicted only the trend of the thermal performance of the RCLPHPs. The correlation needed an improvement as mentioned above before it can be used in the design of the RCLPHPs.

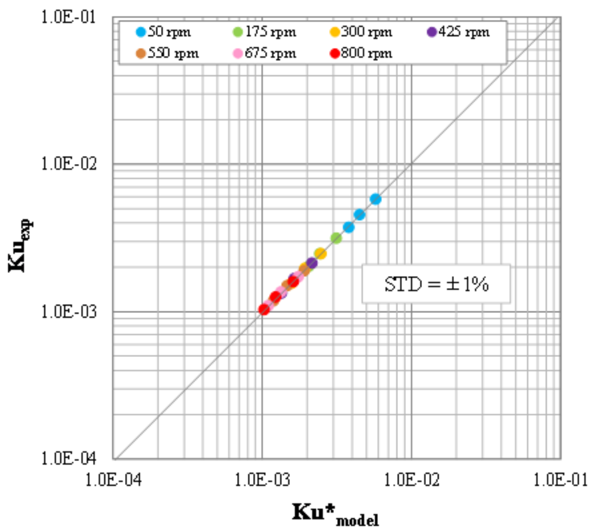

Figure 5 Comparison between Ku^{*}_{model} and Ku_{exp}

Finally, the qualitative analysis found that the centrifugal acceleration is a main parameter which affectsto a circulation of the working fluid contained in the RCLPHP, the centrifugal force will improve a restoring force to increase and liquid smoothly circulated to evaporator section. When the centrifugal acceleration increased, restoring force increased. Most of liquid inside RCLPHP circulated to the end of tube and continuously received the heat input. Therefore, the RCLPHP can continuously transfer the heat from evaporator to condenser section, and thermal resistance finally increased.

Conclusions

The qualitative analysis on effect of the centrifugal acceleration on thermal performance of the RCLPHPs showed that, when the centrifugal acceleration increased, thermal resistance of the RCLPHPs increased. Since the circulation's velocity increased, working fluid condensed and smoothly circulated. In addition, the heat input in the evaporator section is transferred to the condenser section more continuously. In the future, the quantitative study on the effect of the centrifugal acceleration and other parameters which related to the thermal performance of the RCLPHPs will be further considered. However, Bo and Fr were found to be an important dimensionless number considered in the establishment of the new correlation of the RCLPHPs.

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Nomenclature

Bo Bond number, $Bo = g(\rho_l-\rho_v)D_i^2/\sigma$

CGV dimensionless group of critical gas velocity

$$CGV = \frac{j_v\mu_v}{\sigma}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}$$

c_p specific heat(kJ/kg-K)

D diameter (m)

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Ek Ekman number, $EK = \nu/D_i^2\omega$

Fr Froude number, $Fr = \omega^2 R/g$

g gravitational acceleration (m/s^2)

h_{lv} latent heat (kJ/kg)

j volumetric flux (m/s)

Ja Jacob number, $Ja = C_{pl}(T_e\text{-}T_c)/h_{lv}$

Ja* modified Jacob number,

$$Ja^* = \frac{(FR)c_{pl}(T_e\text{-}T_c)}{(1\text{-}FR)h_{lv}}$$

Ka Kaman number,

$$Ku = \frac{q''}{h_{lv}\rho_v\left(\frac{\sigma(\rho_l\text{-}\rho_v)}{\rho_v^2}\right)^{1/4}}$$

k thermal conductivity (W/m-K)

L length (m)

N turn

Pr Prandlt number, $Pr = c_p\mu/k$

q_{90} heat flux at vertical position (W/m^2)

q" heat flux in evaporator section (W/m^2)

q^* dimensionless heat flux, $q^* = q/\mu_l g$

T temperature (K)

z thermal resistance per unit area ($K\text{-}m^2/W$)

β inclination angle (degree)

$\Delta P_{sat}^{e\text{-}c}$ pressure difference (Pa)

μ viscosity (Pa.s)

ρ density (kg/m^3)

σ surface tension (N/m)

Subscripts

a adiabatic section, ambient

c condenser section

e evaporator section

eff effective

exp experiment

i inner

l liquid

pre predict

t total

v vapor

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