THERMAL RESISTANCE OF ROTATING CLOSED-LOOP PULSATING HEAT PIPES
Effects of Working Fluids and Internal Diameters

by

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The objective of this study was to experimentally investigate the effects of working fluids and internal diameters on the thermal resistance of rotating closed-loop pulsating heat pipes (RCLPHP). The RCLPHP were made of a copper tube with internal diameters of 1.50 mm and 1.78 mm, bent into the shape of a flower petal, and arranged into a circle with 11 turns. The evaporator section was located at the outer end of the tube bundle. R123, ethanol, and water were filled as the working fluids. The RCLPHP was rotated at centrifugal accelerations 0.5, 1, 3, 5, 10, and 20 times of the gravitational acceleration considered at the connection between the evaporator and the condenser sections. The heat input was varied from 30 W to 50 W, and then to 100 W, 150 W, and 200 W. It can be concluded that when the latent heat of evaporation increases, the pressure difference between the evaporator and the condenser sections decreases, and the thermal resistance increases. Moreover, when the internal diameter increases, the driving force increases and the frictional force proportionally decreases, or the Karman number increases, and the thermal resistance decreases.

Key words*: rotating closed-loop pulsating heat pipe, centrifugal acceleration, working fluid, internal diameter, thermal resistance*

Introduction

The closed-loop pulsating heat pipe (CLPHP), which is a heat transfer device with very high thermal conductivity, was firstly proposed by Akachi *et al.* [1]. It is made from a long copper capillary tube whose internal diameter does not exceed the critical diameter according to Maezawa's criterion [2]. Then, it is bent into an undulating tube and the ends of the tube are connected to each other to form a closed loop. An internal wick structure is not required. The tube is evacuated and partially filled in with the working fluid. While one end of the CLPHP, an *evaporator section*, is subjected to heat or high temperatures, the working fluid, in liquid slug form, evaporates, expands, and moves through the second part, the *adiabatic section*. This part has no heat transfer between the CLPHP and the environment. Then, the working fluid continuously moves toward the cooler section, the *condenser section*. Then, the vapor plug condenses, collapses, and releases the heat into the environment. The vapor plug evaporating in the evaporator section flows consequently to replace the vapor plug collapsing in the condenser

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section. Due to this mechanism, the working fluid can circulate and continuously transfer heat in a cycle. Thus, the heat transfer can be maintained by the replacement mechanism [3], which is totally different from the heat transfer mechanism taking place in other types of heat pipes, such as thermosyphons [4] and heat pipes [5]. The structure of an ordinary CLPHP is shown in fig. 1(a).

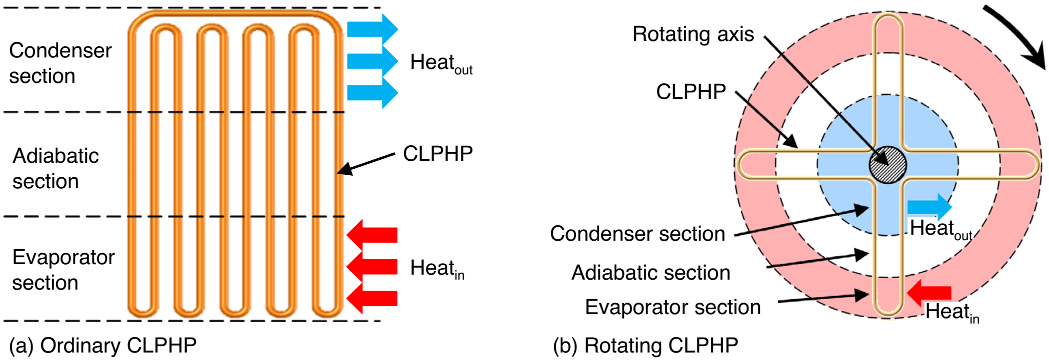


Figure 1. Structure of closed-loop pulsating heat pipes

The CLPHP can be easily employed as a heat exchanger installed on stationary devices, such as ice storage systems [6], condensers in vapor compression refrigeration systems [7], and race car engine radiators [8]. However, an ordinary CLPHP cannot be directly applied on rotating machines which operate with very high rotational speed, such as disk brakes, superconductor bearings, and steam turbine blades. In order to solve this problem, Aboutalebi *et al.* [9] firstly developed the shape of the CLPHP for it to suitably fit into the rotating machineries. The CLPHP was bent into the shape of a flower petal and arranged around the circumference of the radial part, which had accumulative heat, of a rotating machine. This modified CLPHP was named as a *rotating closed-loop pulsating heat pipe*, or RCLPHP, in short. In addition, it could be found to be named as a *closed-loop pulsating heat pipe under centrifugal acceleration*, or CLPHPuCA, in short. The basic structure of the RCLPHP is depicted in fig. 1(b). With this configuration, the RCLPHP is radially rotated together with the rotating machinery. In the meantime, a centrifugal acceleration occurs and exerts on the working fluid inside the RCLPHP. Since the direction of the acceleration is outward from the center of the rotating axis, the working fluid inside the RCLPHP is forced to flow from the inner to the outer part of the RCLPHP, especially for the liquid working fluid on account of its density being higher than that of vapor. Because of this reason, the evaporator section is generally located at the outer end of the CLPHP, and *vice versa* for the condenser section.

It was found from a past study [9] on water-filled RCLPHP with an internal diameter of 2.03 mm and four meandering turns that when the rotational speed increased from 50 rpm to 175 rpm, 300 rpm, 425 rpm, 550 rpm, 625 rpm, and 800 rpm, the thermal resistance decreased continuously. This is because the condensate flows back to receive the heat input in the evaporator section with higher velocity. In addition, they found that when the heat input increased from 25 W to 100 W with 15 W increments, the thermal resistance gradually decreased. This is because the internal flow pattern of the working fluid changed from slug flow to annular flow, as described in another past study [10]. Finally, a study on the effect of the working fluid's filling ratios revealed that the most suitable filling ratio for the RCLPHP was 50% by the total internal volume. Aside from these parameters which have been investigated, effects of other important parameters, such as, working fluids, internal diameters, numbers of meandering turns, evaporator section lengths, *etc*., on thermal performance of the RCLPHP have still not been discovered. Nevertheless, there has been a great number of studies in the case of ordinary CLPHP, for instance, a study on the effect of gravitational acceleration [11], qualitative studies on internal flow patterns of the working fluid inside ordinary CLPHP [3, 11-12], studies on the effect of internal diameters [13-15], studies on the effect of numbers of meandering turns [14-16], *etc*. The effects of the identical parameter on the thermal performance of an ordinary CLPHP and an RCLPHP are not absolutely the same since the geometrical shape of the tube and the acceleration exerting on the working fluid are totally different. From this viewpoint, the information obtained from studies on ordinary CLPHP can not be deduced in the case of the RCLPHP. For that reason, basic knowledge for correct design of the RCLPHP to be applied on the rotating machines is definitely required. The effect of each of the parameters must be carefully investigated quantitatively and qualitatively.

Because of the above-mentioned facts, these became the significant goals of this study: the objective was to experimentally investigate the effects of working fluids and internal diameters on the thermal resistance of the RCLPHP. The obtained knowledge will be very useful especially to any heat pipe designers and to people involved in heat pipe manufacturing industries since they can extensively utilize all the obtained information in order to design, produce, and utilize the RCLPHP in any future actual applications.

Experimental set-up and procedure

The RCLPHP was made from a copper capillary tube with internal diameters of 1.50 mm and 1.78 mm. It was bent into the shape of a flower petal and arranged into a circle with 11 meandering turns. The evaporator section was located at the outer end of the bundle, while the condenser section was placed around the center of the RCLPHP. Both the sections had an identical length of 50 mm. The tested RCLPHP had no adiabatic section; this was so in order to shorten the distance between the evaporator and the condenser sections, as shown in fig. 2(a). The RCLPHP was evacuated, and R123, ethanol, and water were individually filled in as working fluid with a filling ratio of 50% of the total internal volume. An electrical annular-plate heater (220 VAC, maximum power 800 W) was attached to either side of the RCLPHP in order to generate the required heat input into the evaporator section. Then, ceramic papers (Alfiso, Isotek 1260 paper, 3 mm thickness), wooden plate, and insulation sheet (Aeroflex, 3/8 in. thickness) were consecutively attached on the outer side of the heater in order to prevent heat loss from the heater. Another insulation sheet was installed along the circumference of the RCLPHP set. The RCLPHP set was tightly secured by two steel crossbars, as shown in fig. 2(b). Finally, the RCLPHP set was installed on the test rig by securing both the crossbars onto a shaft on the test rig. The test rig and the experimental set-up are schematically shown in fig. 2(c). Electrical current (220 VAC, 50 Hz) was supplied to the heaters through slip rings. Power supply into each heater was controlled by a power controller (Shimax, MAC3D, accuracy ±0.25% full scale), and it was simultaneously monitored for validation of the heat input quantity by using watt meters (Axe, MMP, accuracy ±0.25% full scale). The quantities of heat input chosen for this study were 30 W, 50 W, 100 W, 150 W, and 200 W. The shaft was rotated by a DC motor (MY, model 1030, 24 VDC) with a maximum power of 750 W. The rotational speed of the RCLPHP set could be adjusted by a speed controller (MXA, model 066) powered by a DC power supply (ST, maximum power 720 W). At the same time, the rotational speed of the RCLPHP set was monitored by a digital speed sensor. Rotational speeds of 70 rpm, 100 rpm, 173 rpm, 223 rpm, 316 rpm, and 446 rpm were selected since they could be converted into centrifugal accelerations 0.5, 1, 3, 5, 10, and 20 times of the gravitational acceleration with the consideration of the position at the connection between the different sections. The temperature variation at each point was measured by 12 thermocouples (Omega, Type K, accuracy ±0.5 °C) which were connected to a handheld data logger (Lutron, BTM-4208SD, accuracy ±1.0 °C) secured on the shaft for temperature monitoring and recording. The locations of the thermocouples were as follows. Two points each at the middle of the evaporator and the condenser sections, two points for the air around the condenser section, one point for each heater, and one point each at the inner and the outer surfaces of the lateral and the radial insulation sheets, as shown in fig. 2(a) and fig. 2(b).

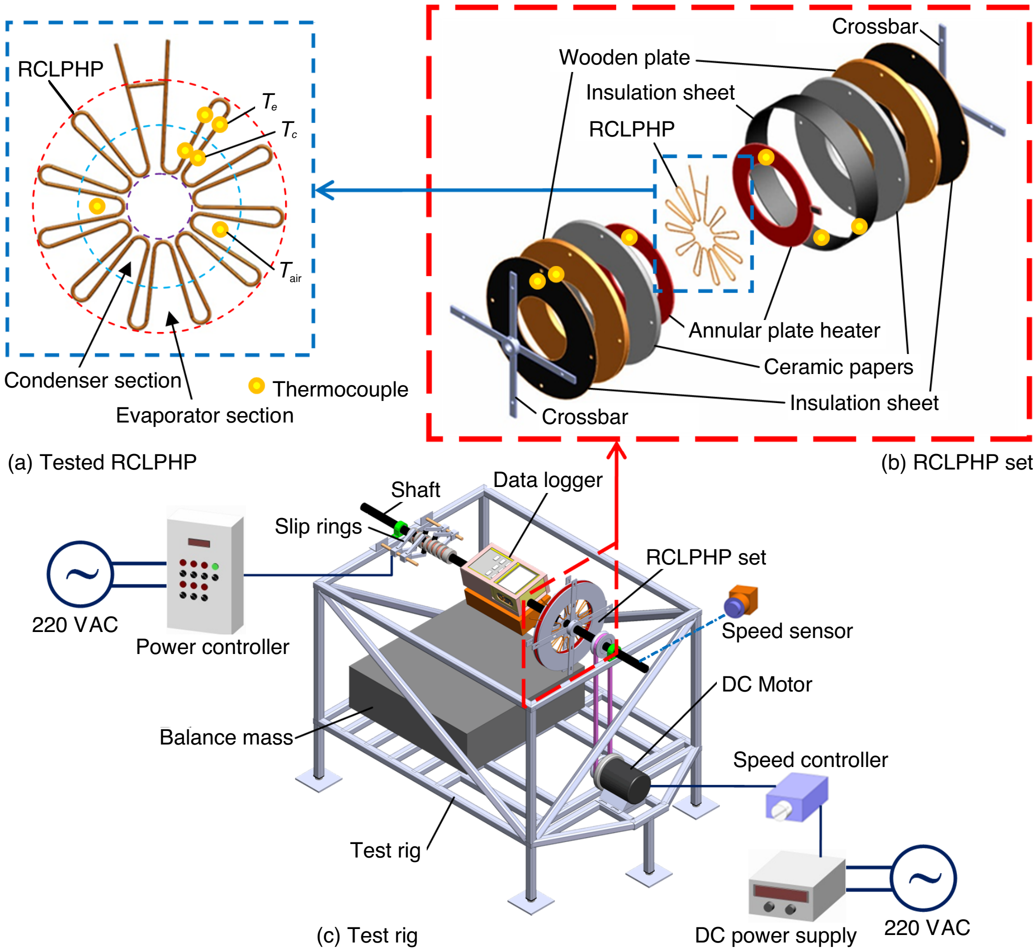


Figure 2. The RCLPHP and the experimental set-up

The experimental procedure was as follows. The RCLPHP set was rotated at a constant speed corresponding to the slowest one. The power of the heaters was then adjusted to the lowest one. After the RCLPHP had reached a steady state, the temperatures at all the points were recorded for a certain duration. The procedure was repeatedly conducted for higher reliability. The experiment was consequently conducted until all the variable parameters were completely covered. Thermal resistance per unit area between the evaporator and the condenser sections (z), or *thermal resistance*, in short, beyond this point represented the thermal performance of the RCLPHP in this study since it can be directly compared to other RCLPHP with different geometrical sizes and heat inputs. The thermal resistance could be found from eq. (1):

$$z_{\text{e-c}} = \frac{T_e - T_c}{\dot{q}} \tag{1}$$

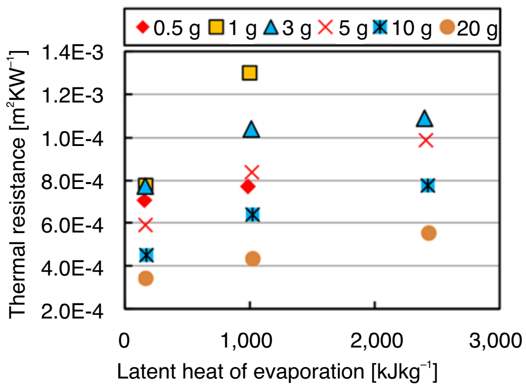
where $\dot{q}$ is the input heat flux entering the evaporator section, and it could be found from dividing the power of the heaters, $\dot{Q}$, by the internal surface area in the evaporator section, as derived in eq. (2). In this study, the data in which the error was lower than 30% of the calculated 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. After that, the relationships between the thermal resistance and each of the variable parameters were plotted and discussed.

$$\dot{q} = \frac{\dot{Q}}{\pi D_i L_e 2N} \tag{2}$$

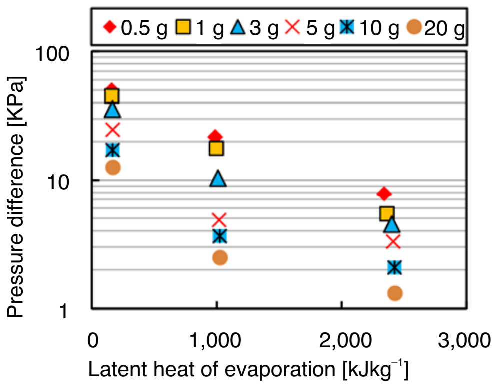
Results and discussion

Effect of working fluids on thermal resistance

In a study on the effect of working fluids, it is important to separate the different types of working fluids by their thermodynamic properties, for example, latent heat of evaporation, specific heat, surface tension, viscosity, thermal conductivity, *etc*. Since the RCLPHP transfers heat by phase change of the working fluid [3], the most suitable quantitative parameter to identify the type of working fluid in this study was the latent heat of evaporation. It can be seen from the experiment on the RCLPHP with an internal diameter of 1.5 mm and a heat input of 50 W that when the working fluid changed from R123 to ethanol and water, the latent heat of evaporation, in turn, increased from about 167 kJ/kg to 1,015 kJ/kg and 2,411 kJ/kg, the thermal resistance obviously increased in every case of centrifugal acceleration, as shown in fig. 3(a).



(a) Effect on thermal resistance



(b) Effect on pressure difference

Figure 3. The effect of the working fluids on thermal resistance and pressure difference

This is because working fluids with lower latent heat of evaporation, such as R123, generally require lower quantity of heat to be used in the complete evaporation of liquid working fluids of certain mass than working fluids with higher latent heat of evaporation, namely ethanol or water. Thus, the liquid working fluid with lower latent heat of evaporation will evaporate, carry the heat out from the evaporator section, and then flow to the condenser section more rapidly. The thermal resistance is consequently low. Moreover, consideration of the relationship between the pressure differences between the inside of the evaporator and the condenser sections, and the latent heat of evaporation found that the higher the latent heat, the lower the pressure difference. This variation is true in every case of centrifugal acceleration, as shown in fig. 3(b). This evidence supports the effect of latent heat of evaporation, as mentioned above. When a working fluid with lower latent heat of evaporation evaporates, the vapor plug expands rapidly. This causes the pressure inside the evaporator section to increase instantly and, subsequently, causes the pressure difference between the working fluid inside the evaporator and the condenser sections to increase. It is clearly known in the case of ordinary CLPHP that when pressure difference increases, the working fluid will have a higher driving force exerted on it. This causes the circulation's velocity to increase, and the thermal resistance consequently decreases. Although there is no past study on the effect of working fluids on the thermal performance of the RCLPHP, the trend of the effect of the latent heat of evaporation well agrees with the result obtained from a past study on an ordinary vertical CLPHP which was applied to release the heat from multiple heat sources [17].

It can be concluded that when the latent heat of evaporation increases, the pressure difference between the evaporator and the condenser sections decreases. This causes the working fluid to circulate with lower velocity. The thermal performance of the RCLPHP finally decreases in every case of centrifugal acceleration.

Effect of inner diameters on thermal resistance

It was found from the study on the RCLPHP with R123 as the working fluid and a heat input of 100 W that when the internal diameter increased from 1.50 mm to 1.78 mm, the thermal resistance decreased in every case of centrifugal acceleration, as shown in fig. 4(a).

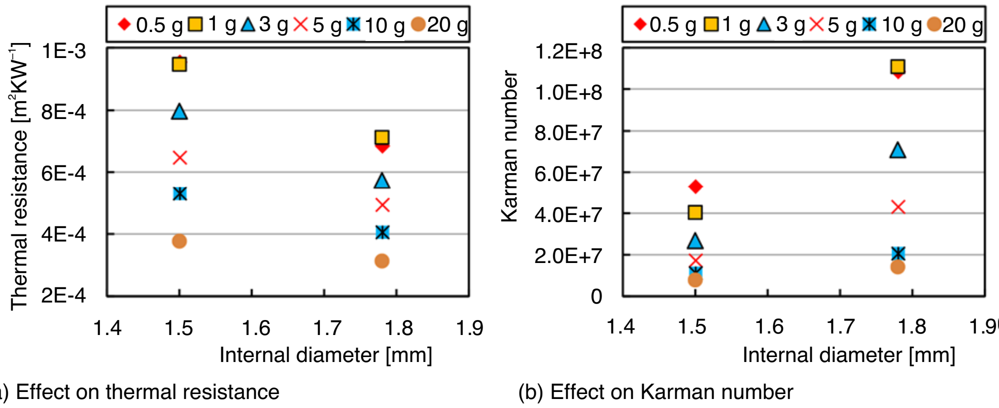


Figure 4. Effect of internal diameters on thermal resistance and Karman number

The physical phenomena can be described as follows. When the internal diameter increases, not only is the heat transferring area between the RCLPHP and the working fluid increased but also the cross-sectional area of the working fluid's flow inside the RCLPHP. When the heat transferring area increases, the heat input has a wider entrance area to reach the liquid working fluid. This causes more active evaporation to occur. The more active the evaporation, the higher the driving force exerting on the vapor plugs to flow from the evaporator section to the condenser section. Thus, the working fluid's flow velocity increases and the thermal resistance subsequently decreases. In the meantime, the increase in the cross-sectional area of the working fluid's flowing passage causes the frictional force of the fluid flow to decrease. These conjugated effects of the increase in the driving force and the decrease in the frictional force as the internal diameter of the RCLPHP increases can generally be explained via a dimensionless group of the Karman number (Ka). The Ka can be written:

$$\mathrm{Ka}=\frac{\rho_\mathrm{l}\left(\Delta P\right)_\mathrm{sat}^\text{e-c}D_\mathrm{i}^3}{\mu_\mathrm{l}^2 L_\mathrm{eff}}=\frac{\rho_\mathrm{l}\left(\Delta P\right)_\mathrm{sat}^\text{e-c}D_\mathrm{i}^3}{\mu_\mathrm{l}^2 0.5\left(L_\mathrm{e}+L_\mathrm{c}\right)} \tag{3}$$

The relationship between Ka and the internal diameter is shown in fig. 4(b). It shows that when the internal diameter increases, the Ka obviously increases in every case of centrifugal acceleration. This quantitative trend completely proves the physical reason that when the internal diameter increases, the driving force increases and the frictional force proportionally decreases. Because of this reason, the thermal resistance decreases. Although there is no past study on the effect of the internal diameter on the thermal performance of the RCLPHP, the trend of the effect of the internal diameter well agrees with the results obtained in a past study on ordinary vertical CLPHP [13] as well as ordinary horizontal CLPHP [14].

It can be concluded that when the internal diameter increases, the driving force increases and the frictional force proportionally decreases, which in turn, increases the Karman number. This causes the thermal performance of the RCLPHP to finally increase in every case of centrifugal acceleration.

Conclusion

Effects of working fluids and internal diameters have been thoroughly investigated in this study. It can be concluded that when the latent heat of evaporation increases, the pressure difference between the evaporator section and the condenser section decreases, and the thermal resistance increases. Moreover, when the internal diameter increases, the driving force increases and the frictional force proportionally decreases, or the Karman number increases, and the thermal resistance decreases. Effects of other factors, such as, number of turns, evaporator section length, *etc*., are suggested to be investigated in the future in order to enhance the basic knowledge regarding RCLPHP.

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Nomenclature

D – diameter, [m]
g – gravitational acceleration, [ms^{-2}]
L – length, [m]
N – number of turns, [turns]
ΔP – pressure difference, [Pa]
$\dot{q}$ – heat transfer rate, [W]
$\dot{Q}$ – heat flux, [Wm^{-2}]
T – temperature, [K]
z – thermal resistance per unit area, [m^{2}KW^{-1}]

Subscripts

c – condenser section
e – evaporator section
eff – effective
i – inner
l – liquid
sat – saturated

Greek symbols

μ – viscosity, [kgm^{-1}s^{-1}]
ρ – density, [kgm^{-3}]

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