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Heat transfer characteristics of a vertical flat thermosyphon (VFT) †

W. Srimuang, S. Rittidech* and B. Bubphachot

Heat-Pipe and Thermal Tools Design Research Lab. (HTDR), Faculty of Engineering, Mahasarakham University,
41/20 Khamriang Sub-District, Kantarawichai District,
Mahasarakham 44150, Thailand

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Abstract

This paper presents an experimental investigation on the heat transfer characteristics of a vertical flat thermosyphon (VFT). Several tests were performed to assess the effects of filling ratios, hydraulic radius, working fluid, and aspect ratio (L_e/4HR) at a vertical orientation on the heat transfer characteristics of the VFT. It was found that the filling ratios and hydraulic radius affect heat flux; while the aspect ratios of VFT increased, the heat flux decreased. In addition, the working fluid changed from water and ethanol to R123 as the heat flux increases.

Keywords: Thermosyphon; Pressed thickness; Flat shape; Heat pipe

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1. Introduction

There are numerous reasons to support the continued study of the use of a thermosyphon for heat recovery. The two-phase closed thermosyphon (TPCT) or conventional thermosyphon (CT) is thermodynamically similar to a wicked heat pipe, except that it relies on gravity to ensure liquid return from the condenser to the evaporator. One of the many reasons is an emerging type called the vertical flat thermosyphon (VFT), which can transfer heat flux better than the CT and uses limited energy to greater advantage. One possible way to save energy and reduce pollution is to increase heat recovery from exhaust gas. This can be done by improving the effectiveness of the heat pipe heat exchanger using different heat recovering processes. There are several interesting results from previous studies that have been carried out on the module design and thermal performance of heat pipes. Results from several studies on modified heat pipe cross sections showed that reducing these led to an increase in the performance of heat pipes. For instance, S.H. Moon et al. [1] studied miniature heat pipe performance for notebook PC cooling by modifying the cross section of the heat pipe. The heat pipes used were circular woven wire wick, composite wick, and central wick. These were pressed from circular cross sections which reduced 30 % of their cross-sectional area. The results showed that the composite wick heat pipe had the highest performance and can reduce heat generated by a notebook computer by up to 10% better than the ordinary one. This study was similar to that of [2], which was an experimental investigation of the micro heat pipe with the cross section of a polygon for cooling the CPU of a personal computer. The cross sections of the heat pipes used in this experiment were triangular with curved sides and rectangular with curved sides. The lengths of the evaporator, adiabatic, and condenser sections were 10, 15 and 25 mm, respectively. The results of the study showed that the thermal resistance of the triangular cross section heat pipes was higher than that of the rectangular one. Another study [3] was an experiment which investigated the effect of a wickless heat pipe's cross section geometry and filling ratio on the performance of flat plate solar collectors. This study

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*Corresponding author. Tel.: +66 43 754316, Fax.: +66 43 754316
E-mail address: s_rittidej@hotmail.com

involved heat pipes using three different geometric cross sections: circular, elliptical, and semi-circular. The results showed that deforming the circular cross section of a wickless heat pipe and reforming it into an elliptical one significantly improved the performance of the solar collector. However, there are no studies that could confirm the assumption that reducing thermosyphon cross sections would increase their performance in the heat pipes. Thus, this is an interesting point for further research. In [4-8], the parameters affecting heat transfer characteristics of CT were studied. These parameters included working fluid, filling ratio, input heat transfer rate, and aspect ratios. The results showed that these parameters could influence the heat transfer characteristics of CT. Nevertheless, it is not certain whether these parameters could likewise affect the heat transfer characteristics of the VFT.

Therefore, it is the purpose of the current study to investigate variables that can greatly impact the heat transfer characteristics of the VFT including the aforementioned filling ratio, hydraulic radius, working fluid, and aspect ratio.

2. Module design and experimental apparatus

The preparation began with a cross section drawing based on the calculated manner of VFT for creating a mold for pressed copper tubes.

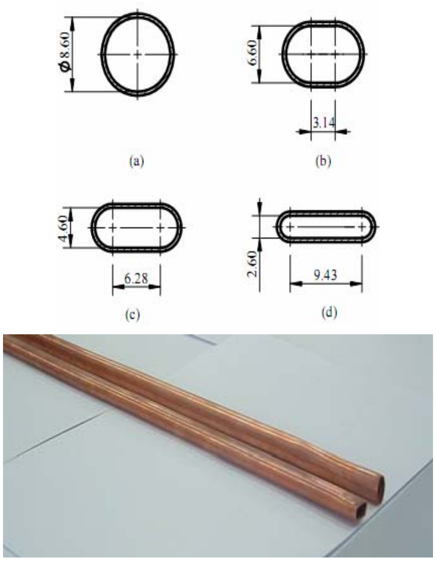
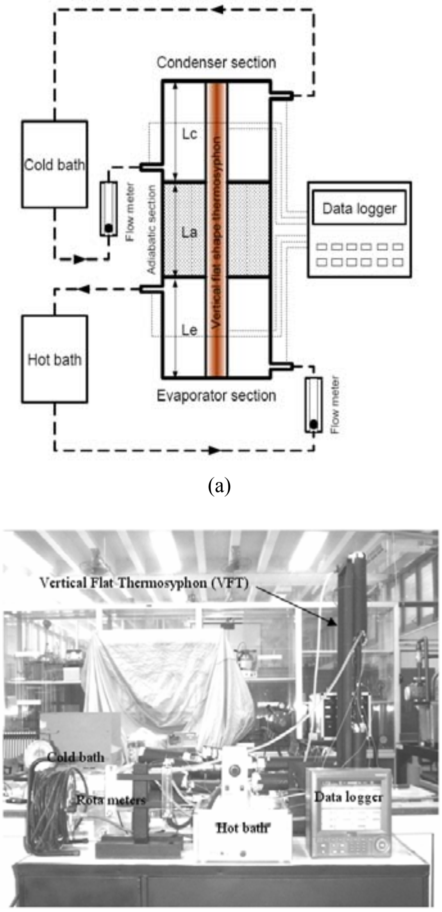


Fig. 1. Cross section area of the CT and VFT (a) circle (b), (c) and (d) flat shape [mm].

Fig. 1 shows the circular cross sections of a CT and the different cross sections of the VFTs, which were created by pressing copper tubes to create different VFTs. The outer and inner diameters of the copper tubes were 9.5 mm and 8.6 mm, respectively. The copper tubes were cut into different lengths: 390, 690, 990, and 1,290 mm. These were then pressed to form different cross sections. The pressed thicknesses were 6.6, 4.6, and 2.6 mm, respectively. The lengths of the evaporator, adiabatic, and condenser sections were equal. The evaporator, adiabatic, and condenser sections of the CT and VFT were carefully insulated to ensure that no heat losses occurred from the CT and VFT.

Fig. 2(a) and Fig. 2(b) show the schematic diagram and photograph of the experimental apparatus, respectively. It consists of a hot bath (a HAAKE W13



(a)

(b)

Fig. 2. (a) Schematic diagram and (b) photograph of the experimental apparatus.

with an inner-pump and a capacity of six liters of water) as an evaporator section. The other component, a cold bath, is an EYELA CA-1111. This cold bath contained 16 liters of water with a built-in pump and a control temperature that ranged from 20 to 100 ^oC with ± 2 ^oC accuracy. It was used to cool the heat out from the condenser section. Two floating rotameters (Omaga FL-2050 with 0.1-1.5 litre/min) were used to measure the water flow rate. A data logger (Yogogawa DX 200 with ± 0.1 ^oC accuracy, 20 channel input, and -200 to 1100 ^oC measurement temperature range) with type K thermocouples (Omega with ± 1^oC accuracy) was attached to the inlet and outlet water in the jacket. The thermocouples were then placed on the surface at three places: the evaporator, adiabatic, and condenser sections. These were connected to the data logger.

3. Procedure for experiments

In order to determine the effect of filling ratios, thermal resistance, hydraulic radius (HR), aspect ratios, and working fluid on the heat transfer characteristics of a CT and the VFTs, a series of tests were carried out. First, we investigated the effects of filling ratios of working fluid on thermal performance with the constant length of the evaporator of a CT and the VFTs. For the variable parameters, the filling ratio of the working fluid was 20, 40, 60, and 80%, determined by the total volume of the thermosyphon and the pressed thickness.

In addition, several experiments comparing thermal resistance on heat flux of the CT and VFT were also conducted. In this section, the control parameter was 4.6 mm for the pressed thickness of the VFT with varied filling ratios.

Next, the effects of hydraulic radius on the heat flux of the VFTs were investigated. The control filling ratio of the working fluid was 20% of the total volume with varied hydraulic radius and evaporator length (Le). The hydraulic radius of VFT was calculated from the following equation:.

$$HR = \frac{A_{cr}}{WP}, \tag{1}$$

where A_{cr} is the cross-sectional area of the CT or VFT; and WP is the wetted perimeter, which is defined as the sum of the length of the boundaries of the section actually in contact with the working fluid.

Finally, experiments to study the effect of aspect ratios and working fluid on the heat flux of the VFTs were carried out. The aspect ratio of CT, which was the circular cross-sectional area, was L_e/D_i. However, the cross-sectional area of the VFT was noncircular. Thus, 4 (HR) was used as the characteristic dimension, which was 4 (HR) equivalent D_i. In this study, the use of 4 (HR) as the characteristic dimension for noncircular cross sections is deemed appropriate. This approach will provide reasonable results as long as the cross section has an aspect ratio that was not too different from that of the circular cross section. The aspect ratio of the VFT was $L_e/4HR$.

The experiment began by turning the switch of a cold bath for supplying cold water inlet through the condenser section. The controlling temperature of the cold bath was set at a constant of 20 ^oC. When the cooling water reached 20 ^oC, the switch for the hot bath was turned on, effectively controlling the temperature of the hot water at a constant of 80 ^oC. During the experiment, the adjusted flow rate of the cold water for temperature at the adiabatic section was maintained at 50 ^oC (working temperature). At this point, temperatures at all five positions (evaporator section, condenser section, adiabatic section, cold water inlet to cooling jacket, and water outlet from cooling jacket) were recorded. The flow rate of the cooling water was also recorded. The rate of heat removed from the condenser section was obtained from the following equation:

$$Q = \dot{m}C_p(T_{out} - T_{in}), \tag{2}$$

Where $\dot{m}$ is the mass flow rate of the cooling water; C_p is the specific heat of water at average temperature; and T_{out} and T_{in} represent the outlet and inlet water temperatures, respectively. The heat losses from the evaporator and condenser section were negligible.

It is necessary to change the heat transfer (Q) of the CT and VFTs into heat flux (q) by using the following equation:

$$q = \frac{Q}{A_s}, \tag{3}$$

where A_s is the surface area of the condenser section. The thermal resistance (TR) of a CT or VFT can be calculated by using the following:

$$\mathrm{TR} = \frac{\mathrm{T_e} - \mathrm{T_c}}{\mathrm{Q}}, \tag{4}$$

where T_e and T_c are wall temperatures of the evaporator and condenser of the CT or VFT, respectively; and q is the heat flux going on the condenser section.

4. Results and discussion

4.1 Effect of filling ratio on heat flux

Water was used as the working fluid with the constant length of the evaporator set at 430 mm. The results, illustrated by Fig. 3, confirm that filling ratios have a significant influence on the heat flux for both CT and VFT. These results clearly demonstrate the effects of the heat flux. The Fig. also compares the current findings with those of the literature (K. S. Ong et al. [7]). It can be observed that the present and the literature data follow similar trends. The maximum heat flux occurred at a 32.5 kW/m^2. It occurred when applied to 20% filling ratio and 4.6 mm pressed thickness. On the other hand, a 20% filling ratio for the CT may cause a dry out phenomena, given that the quantity of working fluid was too low to operate. Nevertheless, the 20% filling ratio was sufficient for the VFT with 4.6 mm pressed thickness. This could likely be due to the boiling phenomena and the two-phase liquid being moved better from the evaporator to the condenser section than in the CT and other pressed thicknesses. When the filling ratio was more than 60%, heat flux decreased significantly. In addition, when the pressed thickness was decreased to 2.6 mm, the friction losses were affected by the two-phase movement, after which the blocking phenomena occurred in the VFT.

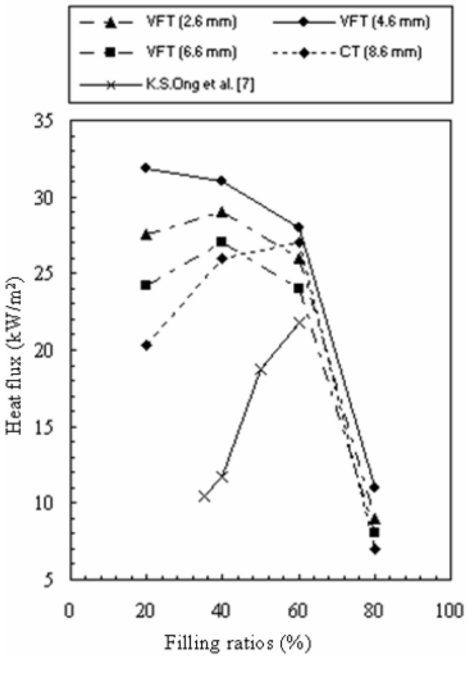


Fig. 3. Relationship between the heat flux and filling ratios of the CT and VFT.

4.2 Effect of the thermal resistance on heat flux

The next experimental result presented thermal resistance on the heat flux. In this experiment, heat load was varied by changing the volumetric flow rate of the hot water flow though evaporator section. Figure 4 compares thermal resistance on a CT and the VFT with 4.6 mm pressed thickness. It can be found that the thermal resistance of the VFT is lower than that of the CT. However, when heat flux increased, thermal resistance decreased. Comparisons of heat flux in this experiment with the results of K. S. Ong et al. [7] were also made. The present results were found to be

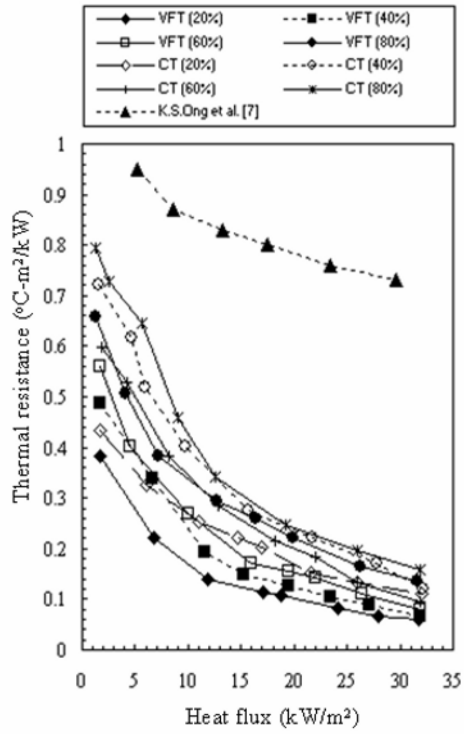


Fig. 4. Comparative thermal resistance of the CT and VFT with varying heat flux.

in reasonable agreement with those of K. S. Ong et al. [7]. The observed trend of both data was similar. However, the TR in this research was lower than that of K. S. Ong et al. [7]. The thermal resistance reached the minimum value of 0.06 m^2 - $^oC/kW$ at a filling ratio of 20%.

4.3 Effect of hydraulic radius on heat flux

To study the effect of hydraulics radius on the heat flux of the VFT, water was used as the working fluid and hydraulic was varied from 1.1 to 2.0 mm. The results are plotted in Fig. 5. It can be seen that the heat flux were 32, 29, and 26 kW/m^2, while the evaporator lengths were 130, 230, and 330 mm, respectively. This experiment cannot be compared with [4, 7], because they were conducted with only 6.25 mm hydraulics radius. However, when the hydraulics radius was increased, the heat flux decreased. These heat rates could be efficiently transferred with the vapor bubble moving to the condenser at a low speed. The maximum heat flux was obtained when using 1.7 mm hydraulic radius. This is the cross-sectional area that had an appropriated size to allow the bubble vapor to move better from the evaporator section to the condenser section compared with other hydraulics radii. The movement of the bubble vapor depended on its density, as well as on the thickness of the liquid film that resulted from the condensation of the bubble vapor in the condenser section. If the density is low and the thickness of the liquid film is thin, the bubble vapor will move faster. These phenomena occurred when the overall heat transfer coefficient (U) was high.

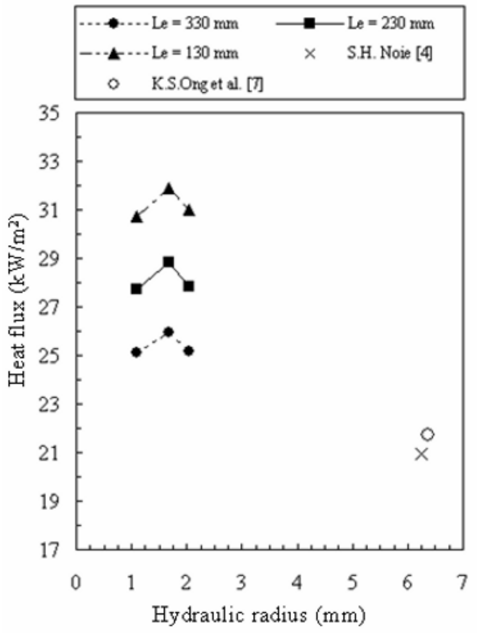


Fig. 5. Comparison of the VFT with different hydraulics radii on heat flux.

However, the U depends on the convective coefficient (h) and thermal conductivity of the thermosyphon wall. In this experiment, the thermal conductivity (k) was constant, because the VFT was made from a copper tube only. Consequently, it can be concluded that the 4.6 mm pressed thickness had the highest convective coefficient in relation to other pressed thicknesses studied.

4.4 Effect of aspect ratio and working fluid on heat flux

To investigate the effect of aspect ratio and working fluid on heat flux, the controlled parameters included pressed thickness of 4.6 mm and a working fluid filling ratio that was 20% of the total volume.

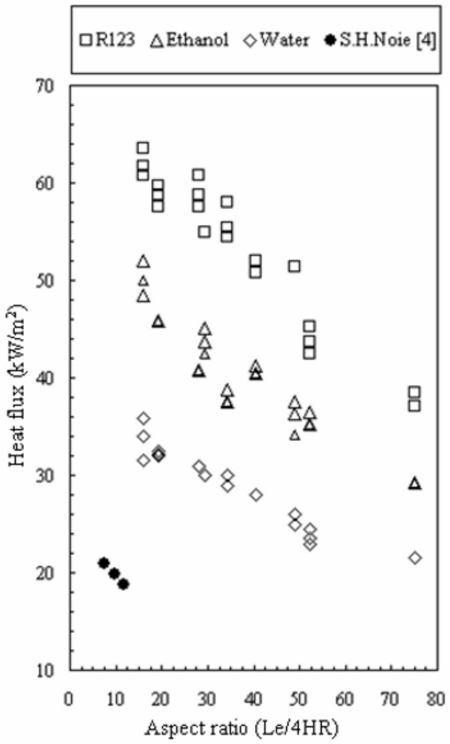


Fig. 6. Relationship aspect ratio to heat flux of the VFT.

The variable parameters were working fluid (R123, ethanol, and water) and aspect ratios (16, 19, 28, 30, 34, 40, 49, 52, and 75).

These results which are plotted in Fig. 6 show the relationship between the aspect ratio and the heat flux of VFT. It can be seen that the maximum heat flux of each pressed thickness was obtained at an aspect ratio of 16. As the $L_e/4HR$ increased from 16 to 75, the heat flux decreased. At a high $L_e/4HR$, boiling ensued inside a confined channel and a low heat flux occurred. Generally, if the working fluid changes from water and ethanol to R123, the heat flux increases. This may be because R123 has a low latent heat of vaporization, as well as the fact that the boiling point of R123 is lower than that of water and ethanol. Boiling point is an important parameter for the thermosyphon's working temperature and performance. If the boiling point is low, the thermosyphon will work at a low temperature. However, the latent heat and boiling point of working fluid also have an effect on thermosyphon's performance. When comparing the results of this research with those of S. H. Noie [4], it was found that the heat flux found in this research was higher than that found by S. H. Noie [4] as shown in Fig. 6.

5. Conclusions

From all of the results of the effect of working fluid, filling ratios, and hydraulic radius on the heat transfer characteristics of the VFT, it can be concluded that:

- The VFT has a higher heat flux when compared with the CT.
- The filling ratios and hydraulic radius affect heat flux. That is, heat flux reached its maximum value at a filling ratio of 20% of the hydraulic radius of 1.7 mm (4.6 mm pressed thickness).
- While the aspect ratios decreased, the heat flux increased.
- Using R123 as the working fluid showed that heat flux transferred better than with water and ethanol.

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Nomenclature

A : Total heat transfer area (m^2)
D : Diameter (m)
C : Specific heat (kJ/kg °C)
L : Length of tube (mm)
$\dot{m}$: Mass flow rate (kg/s)
Q : Heat transfer rate (kW)
q : Heat flux (kW/m^2)
TR : Thermal resistance (m^2 °C/kW)
T : Temperature (°C)
HR : Hydraulic radius (mm)
WP : Wetted perimeter (mm)
U : Overall heat transfer coefficient (kW/ m^2 °C)
h : Convective coefficient (kW/ m^2 °C)
k : Thermal conductivity (kW/ m °C)

Subscripts

c : Condenser
e : Evaporator
a : Adiabatic
cold : Cold water
hot : Hot water
s : Surface area
cr : Cross section
in : Input
out : Output
p : Pressure constant

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Wasan Srimuang received his B.Eng. degree in Mechanical Engineering from Rajamangala Institute of Technology, Northeastern Campus, Thailand in 1997. He then received his M.Eng. degree in Mechanical Engineering from Srinakarinwirot University, Thailand, in 2004. He is currently a Ph.D. student in Mechanical Engineering at Mahasarakham University, Thailand.

Sampan Rittidech received his Ph.D. degree in Mechanical Engineering from Chiang Mai University, Thailand in 2002. He is currently an Associate Professor at the Faculty of Engineering at Mahasarakham University, Thailand. His research interests include heat transfer, heat pipe, and heat exchanger. Dr. Sampan is the head of the Heat-Pipe and Thermal Tools Design Research Laboratory (HTDR).

Bopit Bubphachot received his Ph.D. degree in Engineering Mechanics and Energy from University of Tsukuba, Japan in 2008. He is currently a Lecturer at the Faculty of Engineering at Mahasarakham University, Thailand. Dr. Bopit's research interests include Metal forming (Fine Blanking Process), FEM Simulation in Metal forming process, and Fatigue and Creep Fatigue Strength for Perforated Plate.