EXPERIMENTAL STUDY ON HEAT TRANSFER CHARACTERISTICS OF A MODIFIED TWO-PHASE CLOSED THERMOSYPHON

by

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This study investigated the heat transfer characteristics of modified two-phase closed thermosyphon (TPCT) using water as the working fluid. In the modified TPCT, to reduce thermal resistance, a small TPCT was inserted inside the adiabatic section. For both the plain and modified thermosyphons the performances were determined at various heat inputs from 71-960 W. The results showed that the modified TPCT had less temperature difference between the evaporator and condenser sections than the plain one. According to the experimental data, in the modified TPCT, the thermal performance increased up to 20% over that of the unmodified one.

Key words: *two-phase closed thermosyphon, thermal resistance, thermal performance, heat transfer*

Introduction

Heat pipes are efficient heat transfer devices with extremely high thermal conductivity, which are used to transfer heat at a high rate by the evaporation and condensation of a working fluid. Many benefits of the heat pipe, such as low cost, simple design, low maintenance costs, and high rate of heat transfer, make this device extremely popular.

The TPCT is a simple form of heat pipe, in which its wick-like gravity helps to transport working fluid to the evaporator section. Based on a simple structure, they are used in many fields and many applications, such as chemical industry, spacecraft thermal control, and cooling of gas-turbine rotor blades [1].

Numerous studies have examined the design and application for various kinds of heat pipes, such as two-phase closed or open-cycle thermosyphons. Much of this research has focused on understanding the heat transfer characteristics and the effects of various parameters such as filling ratio (FR), the ratio of initial liquid volume per total volume of evaporation section), geometry, aspect ratio (AR), inclination angle, operational temperature and pressure on their performance [2-5]. Park *et al.* [6] reported the results of experimental studies on the effects of FR in the range of 10-70%, as well as the effect of heat-flow rate between 50-600 W on heat transfer characteristics in TPCT with both smooth and grooved surfaces. They concluded that the heat transfer coefficient of the evaporator increased with an increase in power, and that the effect of the fill-charge ratio was nearly negligible for both surfaces. In another study, Noie [7] investigated the effects of input heat transfer, FR, and AR of a two-phase closed thermosyphon

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under normal operating conditions, examining thermal performance, maximum heat transfer rates for each AR at different FR and the boiling heat transfer coefficients for various AR.

Various studies with mechanical and surface modification have also been carried out to improve the heat transfer characteristics of TPCT [8-10]. In recent research, Wang [11] studied the transient thermal performance of a bent heat pipe with a grooved surface at different inclination angles, finding a small difference in the established temperature of the condenser between straight and bent at the vertical orientation. Rahimi *et al.* [12] examined the effect of resurfacing the condenser and evaporator on overall performance of common closed two-phase thermosyphons. They modified the internal surface in the condenser and evaporator sections to transfer the heat in the evaporator and condenser sections. The result shows that the average thermal performance increased up to 15.27% over the unmodified one.

In another approach, some studies used more efficient working fluids to increase thermosyphon performance. This was done based on the thermal conductivity of working fluids, which plays an important role in the heat transfer rate for both evaporator and condenser sections [13-16]. The typical working fluids in TPCT are water, methanol, propanol, R123, ethanol and their mixtures [17, 18]. The effect of using an Al_2O_3-water nanofluid as a working fluid in a two-phase closed thermosyphon was analyzed by Noie *et al.* [19], who examined a TPCT at various input powers and found that the efficiency can be increased up to 14.7%. In addition, temperatures were less widely distributed within the TPCT when an Al_2O_3/water nanofluid was used instead of pure water.

In contrast to prior works, Khandekar *et al.* [20] used a nanofluid of Al_2O_3, CuO, and laponite clay as working fluid and investigated overall thermal resistance of a two-phase closed thermosyphon. The result showed that wettability of all nanofluids on a Cu surface was higher than that of pure water which led to poor thermal performance.

Considerable experimental and theoretical efforts have been expended on optimizing operating limits and design modifications for improving TPCT performance, such as reducing thermal resistance, increasing TPCT efficiency and augmenting the overall heat transfer coefficient [21-25].

As aforementioned, most of previous studies focused on the processes of evaporation and condensation and reducing the thermal resistance of these sections. As the thermal resistance in the adiabatic section is quite important in reducing the thermal efficiency of a thermosyphon, the present study focused on reducing this section's thermal resistance.

For this purpose, an internal TPCT was placed in the adiabatic section of the studied thermosyphon, and the performance of this system was compared with that of an unmodified thermosyphon.

Experimental apparatus and procedure

Figure 1 shows the schematic diagrams of the unmodified and modified TPCT system. The main part of this system was a closed Cu tube with a length of 720 mm and an inner diameter of 14 mm, which formed the evaporator, adiabatic, and condenser sections. To increase the thermosyphon heat transfer performance, a small TPCT was inserted into the adiabatic section. The internal TPCT was made from Cu tube with an inner diameter of 4.5 mm, a thickness of 0.75 mm, and a length of 240 mm. It was filled with water to 75% full and an AR of 20. This modification was done to reduce the thermal resistance in the adiabatic section and increase the heat transfer rate from the evaporation section to the condenser section.

The condenser section was constructed from a cylindrical shell with a height of 300 mm that surrounded the upper region of the thermosyphon. The cooling water entered the condenser at a temperature of 15 °C, and its flow was controlled and measured by a flow

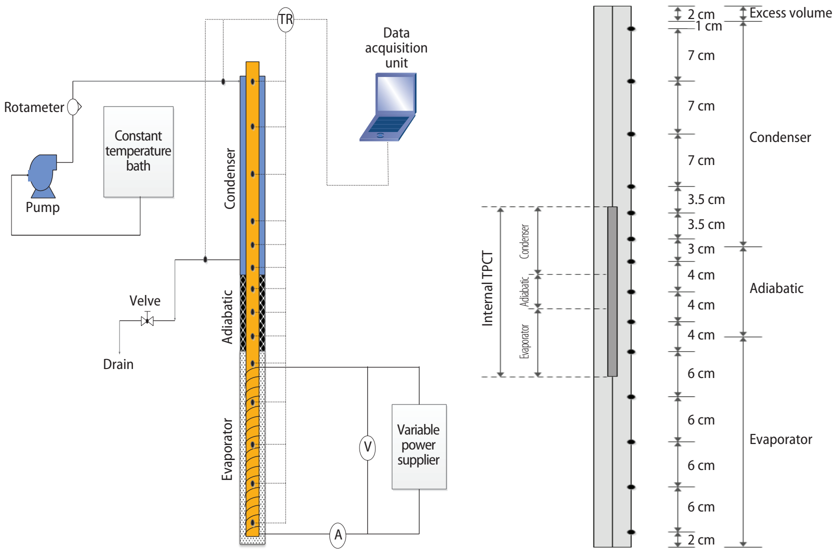

Figure 1. (a) A schematic of the experimental rig and (b) position of internal TPCT and thermocouples

meter (PLATON GRV14, 0.2-2.5 l/min). During the experiments, the cooling water flow rate was fixed at 0.8 l/min. Two thermocouples were mounted on the inlet and outlet to measure the temperature of cooling water. The evaporator section had a length of 280 mm, and the required energy for evaporation was supplied by an electric heater.

At the start of the experiment all the electrical (such as thermocouples and thermometers) and mechanical (such as flow meter) equipment were calibrated.

The input power to the evaporator section was adjusted using an electrical energy regulator (Variac) with a maximum power of 2 kW.

Uncertainties in the measurement may have resulted from measuring errors of parameters such as input power and flow rate, which were ±2% and ±0.4%, respectively.

To remove the non-condensable gases from the thermosyphon before each test, the distilled water was charged into the tube under a vacuum pressure of –86 kPa.

To prevent heat loss, all three sections were entirely insulated by rock wool with a thickness of 75 mm.

The local temperatures were measured with 14 k-type thermocouples. Six, three, and five thermocouples were installed at the condenser, adiabatic and evaporator sections, respectively. Figure 1 shows a schematic view of the configuration and their locations. A data logger (Lutron, BTM-4208SD) monitored all measured data. The uncertainty of the measurement for thermocouples, which included the data logger used in the experiment, was ±0.5 °C.

To ensure the accuracy of the test, the system ran for 60 min to reach the steady-state condition.

For both the unmodified and modified TPCT, the FR was 75%, and AR (the ratio of evaporator section length to inside diameter, Le/ID) was 20. These values were chosen due to some process limitations such as dry-out limitation, counter current flow limitation (CCFL) or flooding, boiling limitation (BL), *etc*. Dry-out limitation becomes important as soon as FR becomes less than 40%. However, if the FR is above 75%, the CCFL/flooding or BL may occur [26], which can limit the heat that is input to the evaporator section.

To check the repeatability of experimental results and the consistency of our observations, each test was repeated three times after reaching a steady-state. The difference between the obtained temperatures along the thermosyphon for all repetitions was less than 5%.

Data reduction

To investigate the effect of the internal TPCT on two-phase closed thermosyphon performance, the heat performance, as one of the most important characteristics in thermosyphon performance, has been used by [12]:

$$\eta = \frac{Q_{\text{out}}}{Q_{\text{in}}} \times 100 \tag{1}$$

where η is defined as the ratio of heat absorbed by the condenser section, Q_{out}, to heat input to the evaporator section via an electric power regulator, Q_{in}.

The amount of heat absorbed by the cooling water in the condenser section can be calculated from inlet and outlet water temperature of the cooling water, T_{in} and T_{out}, in the following relation:

$$Q_{\text{out}} = \dot{m} c_p (T_{\text{out}} - T_{\text{in}}) \tag{2}$$

where $\dot{m}$ and c_p are the mass flow rate and the heat capacity of the cooling water, respectively.

In addition, the heat input to the evaporator via an electric power regular may be approximated by:

$$Q_{in} = V I - Q_{\text{loss}} \tag{3}$$

where V and I are the input electrical voltage and current measured, respectively. The total heat loss, Q_{loss}, by radiation and free convection from the evaporator and condenser sections can be calculated from eqs. (4) and (5), respectively:

$$Q_{\text{loss}} = Q_{\text{rad}} + Q_{\text{conv}} \tag{4}$$

$$Q_{\text{loss}} = \varepsilon \sigma A (T_{\text{ins}} - T_{\text{surr}}) + h_{\text{conv}} A (T_{\text{ins}} - T_{\text{surr}}) \tag{5}$$

Here the free convection heat transfer coefficient is determined according to [27]:

$$h_{conv} = \frac{\text{Nu}\, k_{\text{surr}}}{L} = \frac{k_{\text{surr}}}{L} \left\{ 0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 \tag{6}$$

From eq. (6), the total heat loss was about 3.87% of the input power to the evaporator section.

The average temperature of the TPCT on each section or real function was determined:

$$T_{\text{ave}} = \frac{\int T \,\mathrm{d}l}{L} \tag{7}$$

The overall thermal resistance was defined:

$$R_{\text{th}} = \frac{1}{U_{\text{ave}}} = \frac{T_{\text{ave.}e} - T_{\text{ave.}c}}{Q_{\text{in}}} \tag{8}$$

where $T_{\text{ave.}e}$ and $T_{\text{ave.}c}$ are the average temperatures of the evaporator and condenser sections, respectively, and Q is the heat input to the evaporator section. The overall thermal difference was calculated:

$$\Delta T_{\text{overall}} = T_{\text{ave},e} - T_{\text{ave},c} \tag{9}$$

Results and discussion

In the current work, the experiments were carried out based on various heat inputs to the evaporator section in the range of $71 < Q_{\text{in}} < 960$ W. The wall temperatures achieved from the thermocouple were simultaneously processed to monitor the temperature distribution along the entire length of the TPCT, as illustrated in figs. 2(a)-2(d). In general, the results indicate that temperatures within the TPCT sections increased when the heat input increased from 71-960 W.

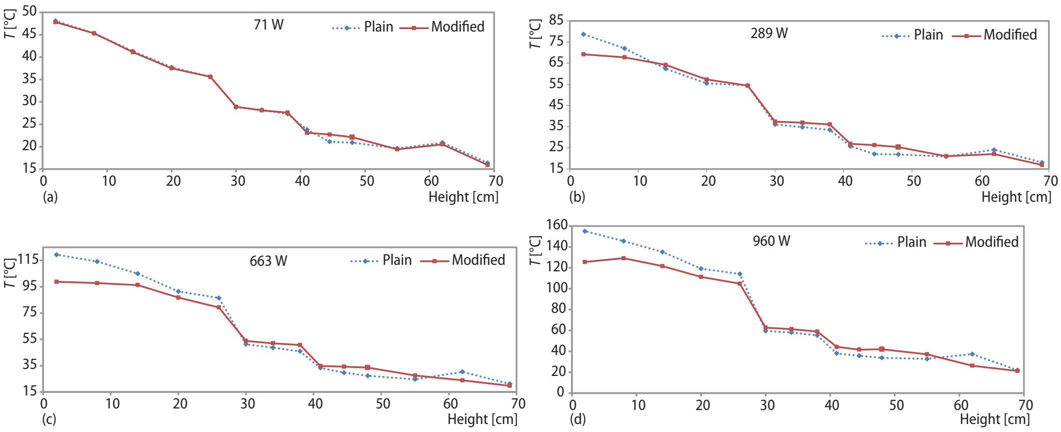

Figure 2. The TPCT temperature profiles at various heat inputs

To accurately review what happened in the thermosyphon, observations were conducted separately on the evaporation, adiabatic, and condensation sections. According to the results, there were significant differences in the temperature distribution along the evaporator section between both TPCT. However, as the figure indicates, at low heat inputs, the temperature distributions along the TCPT wall in the whole section, particularly for 71 W, were almost the same for both layouts. These differences were greater at higher heat inputs, and the temperature of the wall along the evaporator was lower in the modified TPCT. This might indicate that the internal TPCT decreased the heat transfer resistance from the wall to the working fluid in the thermosyphon. In addition, in the modified TPCT the temperature distribution along the evaporator section was more uniform.

Moreover, in both TPCT the maximum difference in temperature occurred within the adiabatic section. This suggests that within the adiabatic section thermal resistance prevented the heat-flux transfer from the evaporator to the condenser section.

This modification had a direct effect on the condenser section, which is the main feature in the thermosyphon operation. In other words, by using a modified TPCT it is possible to enhance the evaporation-condensation phenomenon, and consequently the condensation

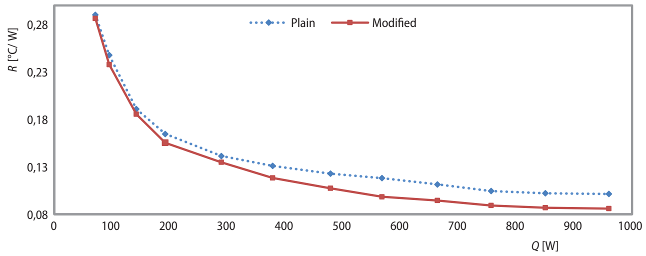

Figure 3. A comparison between the TPCT overall thermal resistance at various heat inputs

section has a lower thermal resistance at higher heat inputs.

The overall heat resistances at various heat inputs were calculated in fig. 3. Increasing the evaporator power input increased the difference in overall heat resistance between the TPCT, and remarkable decreases in the overall heat resistances were observed at higher heat inputs.

At a heat input of 71 W, the difference in overall heat resistance between the two TPCT was about 1%. However, at a heat input of 378 W the difference approached 10%. Moreover, an increase in heat input from 378 to 567 W caused a 20% decrease in the overall heat resistance for the modified TPCT. However, an additional increase in heat input from 567-663 W reduced the overall heat resistance difference to 17.9%. Finally, between 663-960 W, the difference in thermal resistance remained almost constant at 17%. As aforementioned, the experimental results show that an increase in power to more than 960 W causes instability in the unmodified TPCT. In contrast, the modified TPCT was stable at around 1140 W. Therefore, the ability to transfer heat at higher input power is another advantage of using a modified TPCT.

The average temperatures along the TPCT for both layouts at various heat inputs are depicted in figs. 4(a)-4(c).

The difference in the average temperatures along the entire lengths of the unmodified and modified TPCT increased with increasing heat input and the modified TPCT had a higher average temperature in the adiabatic and condenser sections than did the unmodified TPCT. In contrast, the evaporator section of the modified TPCT had a relatively lower average temperature profile than that of the unmodified one, although at higher heat inputs this difference was comparable with that at lower heat inputs. This can be explained by the fact that at low heat input, the rate of water evaporation – and thus the rate of condensation – is low along the entire length. This means that the evaporation-condensation cycle is not completely running, which directly affects thermosyphon operation.

At higher input heat the modified TPCT can transfer heat from the evaporator section more easily than can the unmodified one. Therefore, the evaporator section has a lower average

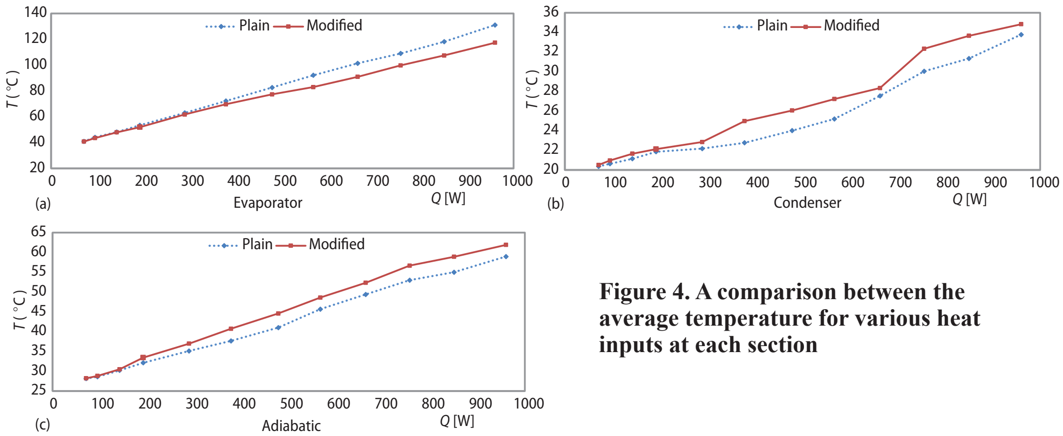

Figure 4. A comparison between the average temperature for various heat inputs at each section

temperature and the heat bulk is simply from this section to other sections, such as the adiabatic and condenser sections. Therefore, it can be concluded that the modified TPCT works more efficiently.

Figure 5 shows the overall temperature difference across the unmodified and modified TPCT, according to eq. (9), *vs.* heat input. With an increase in heat input the difference in $\Delta T_{\mathrm{overall}}$ between the modified and unmodified TPCT increased noticeably.

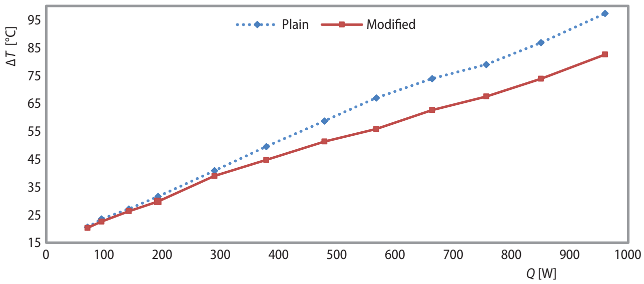

Figure 5. The overall temperature difference for both layouts at various heat inputs

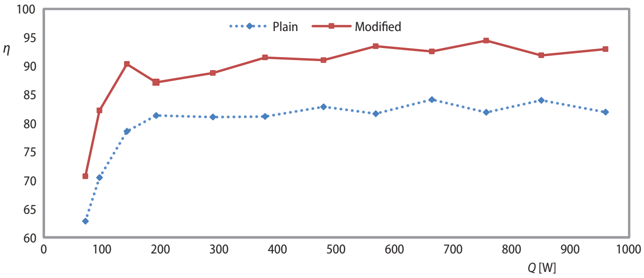

Figure 6. The thermal efficiency of both layouts at various heat inputs

The results show that $\Delta T_{\mathrm{overall}}$ values in the modified TPCT are lower than in the unmodified one. Thus the modified TPCT has a lower thermal resistance than the unmodified one, and transfers heat more efficiently from the evaporator to the condenser.

Finally, to show the overall advantage, both TPCT efficiencies, according to eq. (1) at various heat inputs were evaluated in fig. 6.

The experimental results show that higher efficiencies were obtained as the internal TPCT was placed in the adiabatic section (modified TPCT). This can be explained by the fact that by using an internal TPCT and decreasing the heat resistance inside the thermosyphon it is possible to transfer heat from the evaporator to the condenser more efficiently.

Conclusions

This study proposed a new two-phase closed thermosyphon to reduce the thermal resistance of conventional thermosyphons. For this purpose, a small TPCT with specific dimensions was placed in the adiabatic section. This internal thermosyphon's effects on heat transfer enhancement and characteristics were investigated. The results showed that the average thermal resistance of the studied thermosyphon decreased 20% with these modifications, and that the overall thermal differences at various heat inputs were as much as 16.67% less than those in the unmodified configuration. The average wall temperature on the evaporator section was lower than that in the unmodified TCPT, which represents better heat transfer. The results reveal that the modified TPCT can transfer heat more efficiently at higher power than the unmodified one. From this study it can be concluded that using a small internal thermosyphon inside the adiabatic section of a conventional thermosyphon can enhance its efficiency.

Nomenclature

A – area of cross-section, [m^2]
c_p – specific heat at constant pressure, [Jkg^{-1}K^{-1}]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length, [m]
l – lenght of experimental section, [m]

R_{th} – deat resistance, [°CW^{-1}]
T – temperature, [°C]

Subscripts

ins – insulation
surr – surraunding

References

[1] Vafai, K., Wang W., Analysis of Flow and Heat Transfer Characteristics of an Asymmetrical Flat Plate Heat Pipe, *Int. J. Heat Mass Transfer, 35* (1992), 9, pp. 2087-2099
[2] Sakulchangsatjatai, P., *et al.*, Correlations to Predict Thermal Performance Affected by Working Fluids Properties of Vertical and Horizontal Closed-Loop Pulsating Heat Pipe, *Thermal Science, 20* (2016), 5, pp. 1555-1564
[3] Jiao, B., *et al*., Investigation on the Effect of Filling Ratio on the Steady-State Heat Transfer Performance of a Vertical Two-Phase Closed Thermosyphon, *Appl. Therm. Eng*, *28* (2008), 11-12, pp. 1417-1426
[4] Kammuang-Lue, N., *et al.*, Effect of Working Fluids and Internal Diameters on Thermal Performance of Vertical and Horizontal Closed-Loop Pulsating Heat Pipes with Multiple Heat Sources, *Thermal Science, 20* (2016), 1, pp. 77-87
[5] Hussein, H. M. S., *et al*., Performance of Wickless Heat Pipe Flat Plate Solar Collectors Having Different Pipes Cross Sections Geometries and Filling Ratios, *Energy Convers. Manag*, *47* (2006), 11-12, pp. 1539-1549
[6] Park, Y. J., *et al*., Heat Transfer Characteristics of a Two-Phase Closed Thermosyphon to the Fill Charge Ratio, *Int. J. Heat Mass Transfer*, *45* (2002), 23, pp. 4655-4661
[7] Noie, S. H., Heat Transfer Characteristics of a Two-Phase Closed Thermosyphon, *Appl. Therm. Eng*, *25* (2005), 4, pp. 495-506
[8] Tang, Y., *et al*., Experimental Investigation on Isothermal Performance of the Micro-Grooved Heat Pipe, *J. Exp.Therm. Fluids Sci*, *47* (2013), May, pp. 143-149
[9] Chang, S. W., Lin, C. Y., Thermal Performance Improvement with Scale Imprints over Boiling Surface of Two-Phase Loop Thermosyphon at Sub-Atmospheric Conditions, *Int. Commun. Heat Mass Transfer*, *56* (2013), 1-2, pp. 294-308
[10] Khodabandeh, R., Furberg, R., Heat Transfer, Flow Regime and Instability of a Nano- and Micro-Porous Structure Evaporator in a Two-Phase Thermosyphon Loop, *Int. J. Thermal Sci.*, *49* (2010), 7, pp. 1183-1192
[11] Wang, J., Experimental Investigation of the Transient Thermal Performance of a Bent Heat Pipe with Grooved Surface, *Applied Energy*, *86* (2009), 10, pp. 2030-2037
[12] Rahimi, M., *et al*., Thermal Characteristics of a Resurfaced Condenser and Evaporator Closed Two-Phase Thermosyphon, *Int. J. Heat Mass Transfer*, *37* (2010), 6, pp. 703-710
[13] Ming, Z., *et al*., The Experimental Study on Flat Plate Heat Pipe of Magnetic Working Fluid, *Int. J. Exp. Therm. Fluids Sci*, *33* (2009), 7, pp. 1100-1105
[14] Lin, Y. H., *et al.*, Effect of Silver Nano-Fluid on Pulsating Heat Pipe Thermal Performance, *Appl. Therm. Eng*, *28* (2008), 11-12, pp. 1312-1317
[15] Huminic, G., Huminic, A., Heat Transfer Characteristics of a Two-Phase Closed Thermosyphons Using Nanofluids, *Int. J. Exp.Therm. Fluids Sci*, *35* (2011), 3, pp. 550-557
[16] Ji, Y., *et al*., Particle Size Effect on Heat Transfer Performance in an Oscillating Heat Pipe, *Int. J. Exp. Therm. Fluids Sci*, *35* (2011) 4, pp. 724-727
[17] Wong, S. C., et al., Visualization and Evaporator Resistance Measurement in Heat Pipes Charged with Water, Methanol or Acetone, *Int. J. Therm. Sci*, *52* (2012), Feb., pp. 154-160
[18] Armijo, K. M., Carey, V. P., An Analytical and Experimental Study of Heat Pipe Performance with a Working Fluid Exhibiting Strong Concentration Marangoni Effects, *Int. J. Heat Mass Transfer*, *64* (2013), Sept., pp. 70-78
[19] Noie, S. H., *et al*., Heat Transfer Enhancement Using Al_2O_3/Water Nanofluid in a Two-Phase Closed Thermosyphon, *Int. J. Heat Fluid Flow*, *30* (2009), 4, pp. 700-705
[20] Khandekar, S., *et al*., Thermal Performance of Closed Two-Phase Thermosyphon Using Nanofluids, *Int. J. Therm. Sci*, *47* (2007), 6, pp. 659-667
[21] Hung, Y. H., *et al*., Evaluation of the Thermal Performance of a Heat Pipe Using Alumina Nanofluids, *Int. J. Exp. Therm. Fluids Sci*, 44 (2013), Jan., pp. 504-511
[22] Putra, N., *et al*., Thermal Performance of Screen Mesh Wick Heat Pipes with Nanofluids, *Int. J. Exp. Therm. Fluids Sci*, *40* (2012), July, pp. 10-17

[23] Kang, S. W., *et al*., Experimental Investigation of Silver Nanofluid on Heat Pipe Thermal Performance, *Appl. Therm. Eng*, *26* (2006), 17-18, pp. 2377-2382
[24] Tsaia, C. Y., *et al*., Effect of Structural Character of Gold Nanoparticles in Nanofluid on Heat Pipe Thermal Performance, *Mater. Lett*, *58* (2004), 9, pp. 1461-1465
[25] Lin, Y., *et al*., Effect of Silver Nanofluid on Pulsating Heat Pipe Thermal Performance, *Appl, Therm. Eng, 28* (2008), 11-12, pp. 1312-1317
[26] Faghri, A., *Heat Pipe Science and Technology*, Taylor and Francis, Oxford, UK, 1995
[27] Churchill, S. W., Chu, H. H. S., Correlating Equations for Laminar and Turbulent Free Convection from a Vertical Plate, *Int. J. Heat Mass Transfer, 18* (1975), 11, pp. 1323-1325

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