COMPARATIVE STUDY OF THE THERMAL PERFORMANCES OF TWO-PHASE CLOSED THERMOSYPHONS AND HEAT PIPES

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Abstract. The main goal of the present study was to investigate of the thermal performances of two-phase closed thermosyphons (TPCT) and heat pipes (HPs) using distilled water as working fluid. The TPCT and HP were fabricated from the copper tube with the outer diameter and length of 10 and 310 mm, respectively. The effect of operating temperature on the thermal performances of TPCTs and HPs were investigated. Current results showed that the heat transfer rates of the HP were much higher than those of the TPCT at temperature difference within the range of 35-55 K. Above of this temperature, the heat transfer rates of the TPCT become higher than those of the HP. The overall thermal resistance of the TPCT is reduced by 19% when compared to the HP in conditions of TPCT operation at maximum temperature difference. Also, the overall heat resistance of the HP is reduced by 22%, at low temperatures differences (~ 39 K).

Keywords: Two-phase closed thermosyphon, heat pipe, thermal performance.

1. INTRODUCTION

The investigation of two-phase closed thermosyphons (TPCT) and heat pipes (HPs) and their applications into thermal engineering are known for years, being used in various applications, such as heat exchangers (air preheaters or systems that use economizers for waste heat recovery), cooling of electronic components, communication devices, and solar heating systems [1]. The heat pipe differs from the thermosyphon by virtue of its ability to transport heat against gravity by an evaporation-condensation cycle. A typical two phase closed thermosyphon is shown in Fig 1. A small amount of liquid is placed in a tube from which the air is then evacuated and the tube sealed. The lower end (evaporator section) of the tube is heated causing the liquid to vaporise and the vapour to move to the cold end of the tube (condenser section) where it is condensed. The condensate is returned to the evaporator by gravity. Since the latent heat of evaporation is large, considerable quantities of heat can be transported with a very small temperature difference from end to end. Thus, the structure will also have a high effective thermal conductance [2]. The different heat and mass transfer processes occurring inside a TPCT are: convection, pool boiling, thin liquid film evaporation, countercurrent two- phase flow and film wise condensation [3].

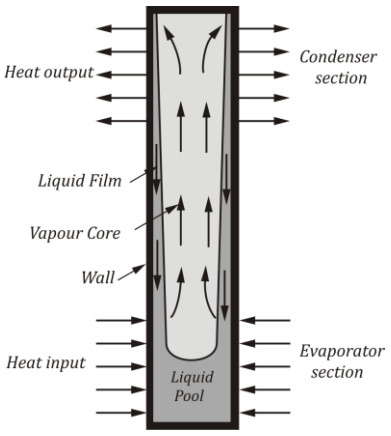


Fig. 1. Two-phase closed thermosyphon.

The heat pipes (Fig. 2) are thermodynamically similar to the thermosyphons and differs from the thermosyphon through the wick which is fixed on

the inside surface and the capillary forces return the condensate to the evaporator. HPs have been used for many years due to their high thermal conductivity, high efficiency without additional electric energy consumption, and suitable working temperature for electronic devices [4].

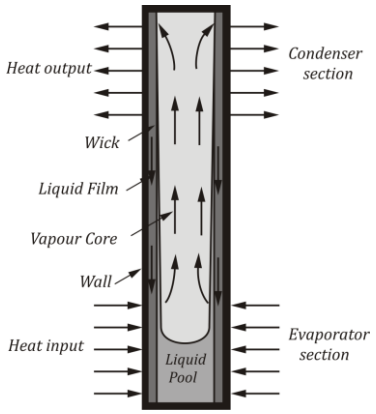


Fig. 2. Heat pipe.

The main factors which influence considerably of thermal performances of the TPCTs and HPs are geometry, thermo-physical properties of working fluid, filling ratio, inclination angle and operating conditions. There are many studies available in literature which investigated these parameters. Thus, Sukchana and Pratinthong [5] investigated the effects of bending position and tilt angle on the efficiency of a thermosyphon with a flexible hose adiabatic section that was bendable. The adiabatic section of the thermosyphon was made of a Teflon hose with the inner diameter of 12.7 mm, and the evaporator and condenser sections were made of straight copper tubing with the inner diameter of 17.4 mm.

R-134a refrigerant was used as working fluid. The results showed that the bending positions and tilt angles of flexible hose had a significant effect on heat transfer characteristics, two-phase flow pattern and pressure drop in the pipe. Also, they founded that at the same tilt angle, bending at the upper end of the flexible hose would decrease thermosyphon performance more than bending at the lower end. Bending at both ends would result in the lowest thermosyphon performance.

The effect of the presence of the neutron absorber in only the evaporator section on the thermal performance was investigated experimentally by Kim and Bang [6]. Experiments were conducted to determine the effects of the working fluid fill ratio and the cross sectional area of the vapor path on the heat removal capacity and thermal performance of an annular thermosyphon that simulates a hybrid control rod. The results showed that the annular thermosyphon showed a lower evaporator thermal resistance compared to a concentric thermosyphon because the increased water level enhanced the convection between the condenser and the adiabatic section. Also, the flooding limits of annular thermosyphons were lower than those of concentric thermosyphon because of the reduction of the cross-sectional area for vapor flow and the resulting increase of the shear at the vapor–liquid interface.

Singh et al. [7] investigated the thermal performance of a flat thermosyphon with and without anodized inner surface. Anodization was performed to prepare a uniform coating on the inner side of the thermosyphon. Acetone was used as working fluid. Thermosyphon was tested at heat input range of 50–300 W, various inclination angles (0°, 45°, 90°) and fill ratios (40%, 60% and 100%). The results indicated that the fill ratio and inclination angle has a significant effect on the performance of the thermosyphon and the total thermal resistance of the anodized thermosyphon was reduced by 20% when compared to the non-anodized thermosyphon. Also, the authors compared the performances of flat and cylindrical anodized thermosyphons and founded that the flat thermosyphon performed better than the cylindrical one. The heat transfer coefficient in evaporator and condenser of flat thermosyphon was enhanced to 69% and 56% respectively when compared to the cylindrical one at a heat flux of 50 kW/m^2.

Jiang et al. [8] studied the heat transfer performances of the three-phase closed thermosyphon (THPCT). The thermally conductive polyamide 6 (PA6) particles and pure water was used as working fluid. The results were compared with two-phase closed thermosyphon (TPCT) using pure water. The experiments were performed for various filling ratios (30%–60%), heat fluxes (3.96–31.70 $kW \cdot m^{-2}$), and solid holdups (0%–50%). The results showed that by adding thermally conductive PA6 particles in a TPCT, increases evaporator convective heat transfer coefficient and decreases the overall thermal resistance. At a filling ratio of 30%, the maximum improvement was 25.6%.

An innovative design of a counter-current two-phase thermosyphon for the in-plane cooling of flat product structures was investigated experimentally and numerically by Schreiber et al. [9]. The goal

this two-phase thermosyphon is to minimize the temperature gradient and temperature fluctuation along the entire length of the evaporator. Experiments was carried out and compared with the numerical model to prove the working principle of this new type of thermosyphon. The results demonstrated the promising potential to implement such design features in thermosyphons. The thermal performances can be enhanced by optimizing the cascading pool design and the changing the glass plate for a metal cover.

Nair and Balaji [10] studied numerically the possibility of heat transfer enhancement by addition of extended surfaces inside the condenser section of the thermosyphon. Through incorporation of internal fins is expected to reduce the filling ratio and the effective thermal resistance by way of enhanced condensation. The placement of fins on internal surface of the thermosyphon aims to reduce the filling ratio and the effective thermal resistance by way of enhanced condensation. Water was used as working fluid. The results showed that the addition of fins increases the overall thermal conductivity of the thermosyphon. For the eight fin case around 43% increase in the thermal conductivity was observed.

The effects of bend angle and fill ratio on the performance of a naturally aspirated thermosyphon were investigated experimentally by Smith et al. [11]. The results showed that the naturally aspirated thermosyphon was capable of adequately cooling the heat source application under investigation and the installation of bend angles in the thermosyphon geometry did not pose any limitations to the heat transfer capabilities of the device. Relatively, the thermal resistance of the thermosyphon is a small percentage (~4%) compared to the natural convection heat sink and thermal interface of the heaters at high powers.

A novel modeling strategy, based upon an existing, multiphase 2D heat pipe (HP) model that is coupled with a new 3D single-phase (external flow) model was developed by Stark et al. [12]. The model has revealed temperature depressions in the HP wall that can affect the overall thermal resistance of the HP-fin array system.

Li et al. [13] investigated experimentally the effects of vacuuming process parameters on the thermal performance of a composite heat pipe by conducting transient and steady-state tests. Under the conditions of the first vacuuming process, the effective working length showed a more remarkable effect on the start-up performance of the heat pipes than the first vacuuming time and filling ratio. Under the conditions of the second vacuuming process, the second vacuuming temperature showed a remarkable effect on the isothermal performance. The thermal resistances were less than 0.02 K/W at the evaporator and less than 0.09 K/W at the condenser with respect to those less than 0.16 K/W after the first vacuuming process.

A high temperature special-shaped heat pipe (HTSSHP) coupling the flat plate heat pipe (FHP) and cylindrical heat pipes (CHPs), respectively as the thermal receiver and heat transfer unit, was proposed as the key component in a solar thermo-chemical reactor by Wang et al. [14]. Its startup characteristics, isothermal performance, and the effect of heat inputs and cooling rates on thermal resistance were experimentally investigated. The results showed that the overall thermal resistance in HTSSHP reduces with increasing operating temperature, ranging from 0.12 to 0.19 °C/W that is in the same order of magnitude in a typical heat pipe.

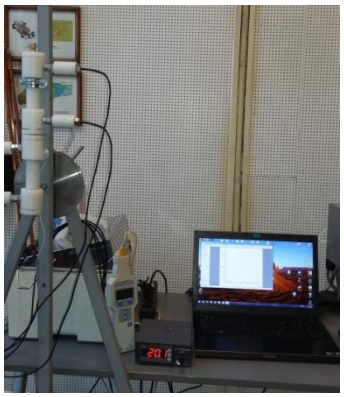
The influence factors of heat pipe structure on heat transfer coefficient were analyzed by Tan and Zhang [15] to enhance the heat transfer performance of the WIHP, such as the working temperature, the ratio of the evaporating section length (RESL) and the diameter of the heat pipe. They founded that the average equivalent heat transfer coefficient (EHTC) of the WIHP reaches the maximum of 1.24 W/(m2 °C) at a RESL of 75%, and also, the RESL should be optimized based on the working-hours-weighted mean temperature.

This work presents a comparative study between thermal performances of the two-phase closed thermosyphons (TPCTs) and heat pipes (HPs) using distilled water as working fluid. The experiments were conducted at different operating temperatures (45 -90 °C) of the two-phase closed thermosyphons and heat pipes. The effect of the operating temperature on the heat transfer characteristics of TPCT and HP were studied.

2. EXPERIMENTAL PROCEDURE

The system used for thermal performance measurement of TPCT is shown in Fig. 3. The TPCT made of copper tube with outer diameter of 10 mm, 1 mm in thickness and 310 mm in length. The evaporator and condenser sections had 126 and 130 mm length, respectively. Local temperatures on the TPCT were measured by five

thermocouples (type-PT-100-thermo-resistance). The uncertainty in temperature measurements was ±0.1°C. Two thermo-resistances were mounted on the evaporator section, one on the adiabatic section and two on the condenser section. The temperature of the heating water from the evaporator section was kept constant by a thermostatic bath (GD 120-S26 with an operating range of -15 °C to 120 °C and ±0.1°C accuracy) and the operating temperature was varied between 45 °C and 90 °C.



a)



b)

Fig. 3. The experimental set-up for measuring the thermal performance of the TPCT and HP (a), and detail (b).

The temperature of the cooling water from the condenser were kept constant through a thermostatic bath (Haake C10 – P5/U with an operating range of 25–100 °C) with an accuracy of ±0.04 °C. The mass flow rate of the cooling water was measured by flow meter with accuracy of 0.001 kg/s. The inlet and outlet temperatures from evaporator and condensers sections and the mass flow rates of the cooling water were also measured. All the tests so far conducted have been performed at an inclination angle of 90° (vertical) and a filling ratio of 25%.

After placing the amount of the working fluid in heat pipe, these were vacuumed using a vacuum pump Rothenberger. The vacuum pressure inside of the heat pipe was measured by digital vacuumeter Testo 552.

3. RESULTS AND DISCUSSIONS

Thermal performances of the TPCT and HP were characterized by the heat transfer rate, the evaporator and condenser heat transfer coefficients and the overall thermal resistance.

The quantity of heat transferred to the coolant water can be calculated from inlet and outlet water temperature difference, taking into account the water mass flow rate and specific heat:

$$\dot{Q} = \dot{m} \cdot c_p \cdot (T_{C2} - T_{C1}) \, [W] \qquad (1)$$

where $\dot{Q}$ is heat transfer rate, in W, $\dot{m}$ is mass flow rate of the cooling water, in kg/m-3, c_p is specific heat, in J/kg K and T_{C1}, T_{C2} are inlet and outlet temperatures from condenser section, in K.

The heat transfer rate of the TPCT and HP is shown in Fig. 4.

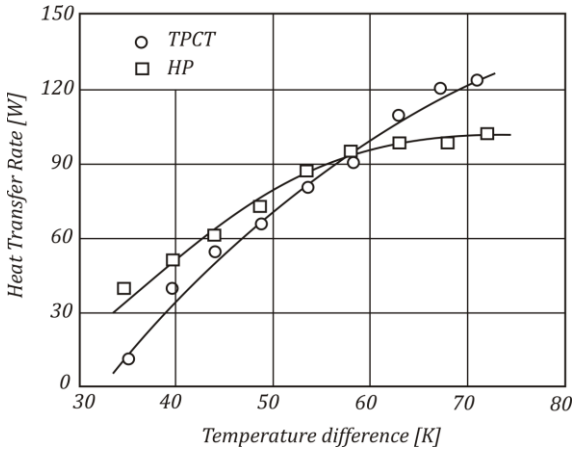


Fig. 4. Heat transfer rate of the TPCT and HP.

As shown in Fig. 4, the heat transfer rates of the HP were much higher than those of the TPCT at temperatures differences within the range of 35-55K. Above of temperature of 55 K, the heat transfer rates of the TPCT become higher than those of the HP. The heat transfer rate increases up to 21.43% for the TPCT, in conditions of TPCT operation at maximum temperature difference. At low temperatures differences (~ 39 K), the increase of the heat transfer rate of the HP was 27.27%. Based on the analysis of the experimental data it can be seen that the better thermal performances of the HP using water were obtained at the low

temperatures differences, while for TPCT improved thermal performances were obtained to high temperatures differences (above 55 K).

Figs. 5 and 6 illustrate the evaporation and condenser heat transfer coefficients of the TCPT and HP at various operating temperatures. The evaporator heat transfer coefficient is calculated by:

$$h_E = \frac{\dot{Q}}{A_E (T_{Em} - T_S)} \left[\frac{W}{m^2 K} \right] \quad (2)$$

where A_E is evaporator surface area, in m^2, T_{Em} is mean temperature on evaporator section, in K, T_S is vapor temperature, in K.

The heat transfer coefficient at the condenser section is defined as:

$$h_C = \frac{\dot{Q}}{A_C (T_S - T_{Cm})} \left[\frac{W}{m^2 K} \right] \quad (3)$$

where A_C is condenser surface area, in m^2, and T_{Cm} is mean temperature on condenser section, in K.

The vapor temperature T_S was measured on adiabatic section.

As can see in Fig. 5 the evaporation heat transfer coefficients of the HP were much higher than those of the TPCT.

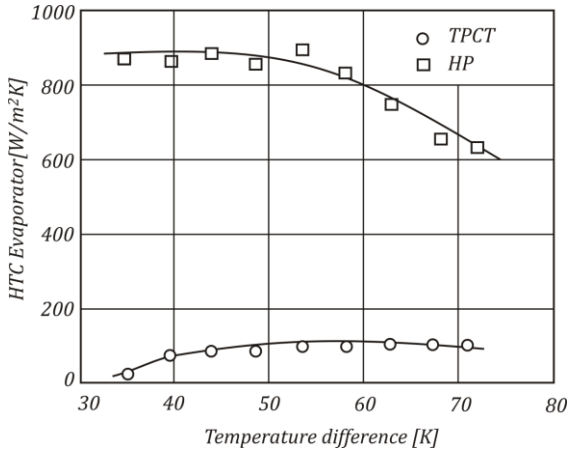


Fig. 5. HTC evaporator of the TPCT and HP.

The evaporation heat transfer coefficient of the HP increases slowly with the increase of the operating temperature up to temperatures differences of 55 K, and then, the decreases significantly with increase of the operating temperature (above of the temperature difference of 55 K). Concerning the evaporation heat transfer coefficient of the TPCT this increases with the increase of the operating temperature.

As can be seen in Fig. 6 the condenser heat transfer coefficients of both TPCT and HP increases with increase of the operating temperature, the condenser heat transfer coefficients of the HP being higher than those of TPCT, for all the operating temperatures. The condenser heat transfer coefficient increases up to 8.0% for the HP compared with that of the TPCT at maximum temperature difference. At low temperatures differences (~ 39 K) the increase of the heat transfer coefficient of the HP was 34.46%.

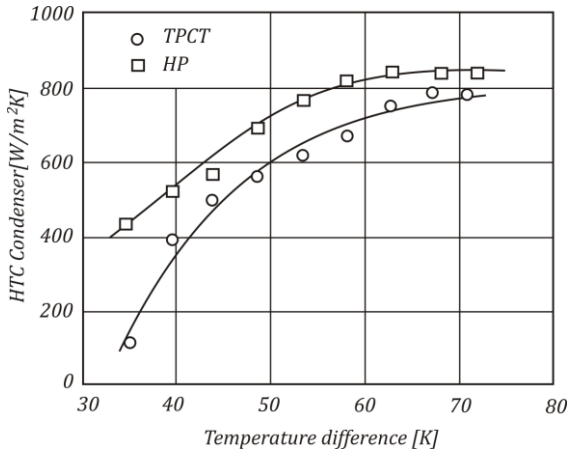


Fig. 6. HTC condenser of the TPCT and HP.

The overall thermal resistance of the TPCT and HP is defines as ratio between the temperature difference between the evaporator and condenser sections and the quantity of heat transferred to the coolant water:

$$R = \frac{\Delta T}{\dot{Q}} = \frac{T_{Em} - T_{Cm}}{\dot{Q}} \left[\frac{K}{W} \right] \quad (4)$$

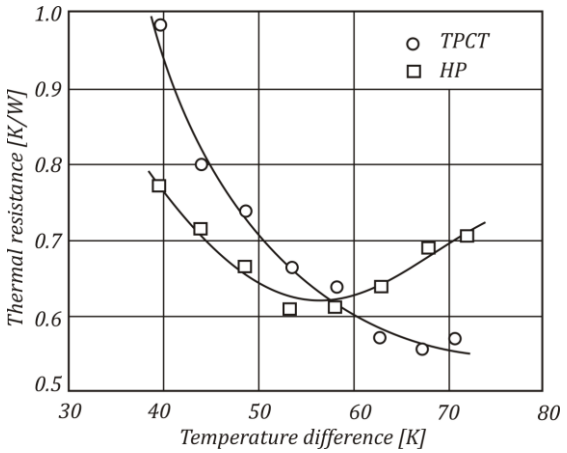


Fig. 7. The overall thermal resistance of the TPCT and HP.

The overall thermal resistance of the TPCT and HP at various operating temperatures is shown in Fig. 7. Experimental results revealed that the overall thermal resistance of the HP was much lower than those of the TPCT at the temperature difference within the range of 35-55 K. Above of temperature of 55 K, the overall thermal resistance of the HP becomes higher than those of the TPCT. The overall heat resistance decreases up to 18.86% for the TPCT, in conditions of TPCT operation at maximum temperature difference. At low temperatures differences (~ 39 K), the decrease of the overall heat resistance of the HP was 21.6%.

4. CONCLUSION

In this study was performed an experimental study concerning thermal performances of the two-phase closed thermosyphons (TPCTs) and heat pipes (HPs). Thermal performances of the TPCTs and HPs were characterized by the heat transfer rate, the evaporator and condenser heat transfer coefficients and the overall thermal resistance. The experimental results showed that the heat transfer rates of the HP were much higher than those of the TPCT at temperature difference within the range of 35-55K. Above 55 K, the heat transfer rates of the TPCT become higher than those of the HP. The evaporation heat transfer coefficient of the HP increases slowly with the increase of the operating temperature up to temperatures differences of 55 K, and then, the decreases significantly with increase of the operating temperature, while the evaporation heat transfer coefficient of the TPCT this increases with the increase of the operating temperature. Current results indicate that the condenser heat transfer coefficients of both TPCT and HP increases with increase of the operating temperature, the condenser heat transfer coefficients of the HP being higher than those of TPCT, for all the operating temperatures. Also, experimental results revealed that the overall thermal resistance of the HP was much lower than those of the TPCT at temperature difference within the range of 35-55 K. Above of temperature of 55 K, the overall thermal resistance of the HP becomes higher than those of the TPCT.

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