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Heat transfer characteristics of a stainless steel/ethanol two-phase closed thermosyphon

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Abstract : An experimental investigation has been carried out to study the heat transfer characteristics of a stainless steel two-phase closed thermosyphon. Ethanol was used as working fluid. Since the filling ratio is expected to have a predominant effect on the steady-state thermal performance of a thermosyphon, three cases were considered: the underfilled, overfilled and optimally-filled sets. The effects of the input heat flux at the evaporator and the cooling fluid temperature at the condenser were also investigated. The steady state wall temperature was then monitored at various locations along the thermosyphon. The obtained results showed that the dry-out occurrence at the evaporator bottom resulted in an obvious temperature rise, especially for the underfilled case. The experimental heat transfer coefficient at the evaporator was estimated and compared with pool boiling correlation data. A good agreement between the experimental results and the prediction values was obtained. A reasonable agreement was found for water and acetone.

Keywords : thermosyphon ; working fluid ; filling ratio ; heat transfer coefficient.

1. Introduction

A two-phase thermosyphon, also known as gravity-assisted heat pipe, is a passive, wickless and very efficient heat transfer device. It consists of a metallic evacuated tube where a certain amount of working fluid is filled. Its operation is based on two-phase closed cycles where latent heat of evaporation (in the bottom extremity) and condensation (in the other extremity) is used to transfer heat, even for small temperature gradients. Considerable interest has been paid to wickless two-phase closed thermosiphon heat pipes due to their simple construction and low cost [1–3]. They have the advantages of low thermal resistance, compact and use small amount of working fluid. Thermosyphons are used in a wide range of applications such as electronics cooling, heat exchangers and solar collectors [1]. Many experimental studies have been performed to examine the impact of working fluid fill ratio and inclination angle on the performance of closed thermosyphons. Noie [4] studied the effect of filling ratio and the evaporator aspect ratio (evaporator length to evaporator diameter) on the heat transfer performance of the Thermosyphon for a range of heat input. It was found that changing the fill ratio can reduce the evaporator wall temperature depending on the aspect ratio. Jiao et al. [5] developed an analytical model to investigate the effect of filling ratio on the steady state heat transfer characteristics of a vertical wickless heat pipe and compared the results with their experimental work. They reported that the fill ratio depends on geometrical parameters and heat input. Jouhara and Robinson [6] investigated experimentally the effect of using different working fluids namely, water, FC-84 and FC-3283 and two filling ratios (100% and 50%) on the performance of thermosiphon heat pipe. A small size thermosiphon of 10W with different working fluids (water, methanol and acetone) and liquid fill at various input energy has been investigated by Mozumder et al. [7]. The study showed that the effect of charging liquid can be indicated by temperature difference, thermal resistance and overall heat transfer coefficient. The influence of the charged liquid and adiabatic length on the thermal performance of a long heat pipe charged with R-134a has been examined by Sukchana and Jaiboonma [8] who concluded that the optimum liquid charge and heat flux suitable for shorter adiabatic section were 15% and 5.92 kW/m^2, respectively. Chehade et al. [9] tested effects of fill ratio, inlet cooling water temperature and mass flow rate in condenser jacket on the performance of the two-phase closed loop Thermosyphon. They concluded that the best fill charge ratio is between 7% and 10% and the fastest start up occurs by using the optimal fill ratio. An experimental study has been performed by Manimaran et al. [10] to examine the effect of heat input, charge fill ratio, and angle of inclination on thermal characteristics of a heat pipe, who reported that the lower thermal resistance was obtained at fill ratio 75% and vertical orientation. Alizadhdakel et al. [11] have reported experimentally the effect of input energy and fill ratio on the performance of a wickless heat pipe. They have also carried out a numerical simulation to investigate the phase change phenomena with effect of non-condensable gases throughout the thermosyphon. An optimum value for fill ratio of 50% was obtained for the studied thermosyphon. A numerical CFD analysis and experimental work to investigate cooling water flow rate, input energy an orientation on the thermal performance of a thermosyphon have been carried out by Abdullahi [12]. Results show that the heat transfer characteristics of the heat pipe increase as inclination angle and input energy increase.

From all mentioned experimental investigations, it can be concluded that the best fill ratio for any heat pipe depend on many factors such as geometry, heat input, type of liquid and operating conditions. Therefore, according to these parameters, investigations to identify the best fill ratio are needed whenever anyone of these parameters is changed. It is the first step in the design of more complex systems involving thermosyphon, like heat exchangers or solar panels. For that reason, an experimental analysis of the heat transfer characteristics of a stainless steel two-phase closed thermosyphon filled with ethanol is proposed. The effects of the input heat, the temperature of the cooling fluid at the condenser and the filling ratio are investigated. All tests were performed in the same experimental conditions for the same range of input heat loads.

2. Experimental setup and procedure

The schematic diagram of the experimental setup is shown in figure 1-a. It consists of the thermosyphon, a wire heater, a power supply module, a cooling water flow circuit, a vaccum pump, a graduated syringe for the thermosyphon filling and measuring instruments. The evaporator section is electrically heated by a wire heater connected to a power supply module. The electrical power input to the heater was controlled using a variable voltage transformer (variac). The power supplied to the evaporator section was monitored by measuring the applied voltage and current to the wire heater with two Metrix MX22 multimeters. The accuracy of the power input measurement was estimated to be around 5%. The evaporator and the adibatic sections were surrounded by a multi-layer insulation to minimise heat exchange with the ambient surroundings. The condenser section was inserted into a jacket exchanger to insure heat extraction from the thermosyphon. The temperature of the cooling water was monitored by a constant-temperature bath with an accuracy of ±1%. Two valves were included in the test rig to facilitate the charging procedure. In every test case, the thermosyphon was first cleaned with purified water. Then, the non-condensable gases were partially evacuated from the inside of the heat pipe using a mechanical vacuum pump (vacuum capacity of -50 kPa). A garduated syringe was used to charge the thermosyphon of the required volume of working fluid. The axial temperature distribution of the thermosyphon was measured by seven calibrated copper-constantan thermocouples set along the outer wall (Figure 1-b).

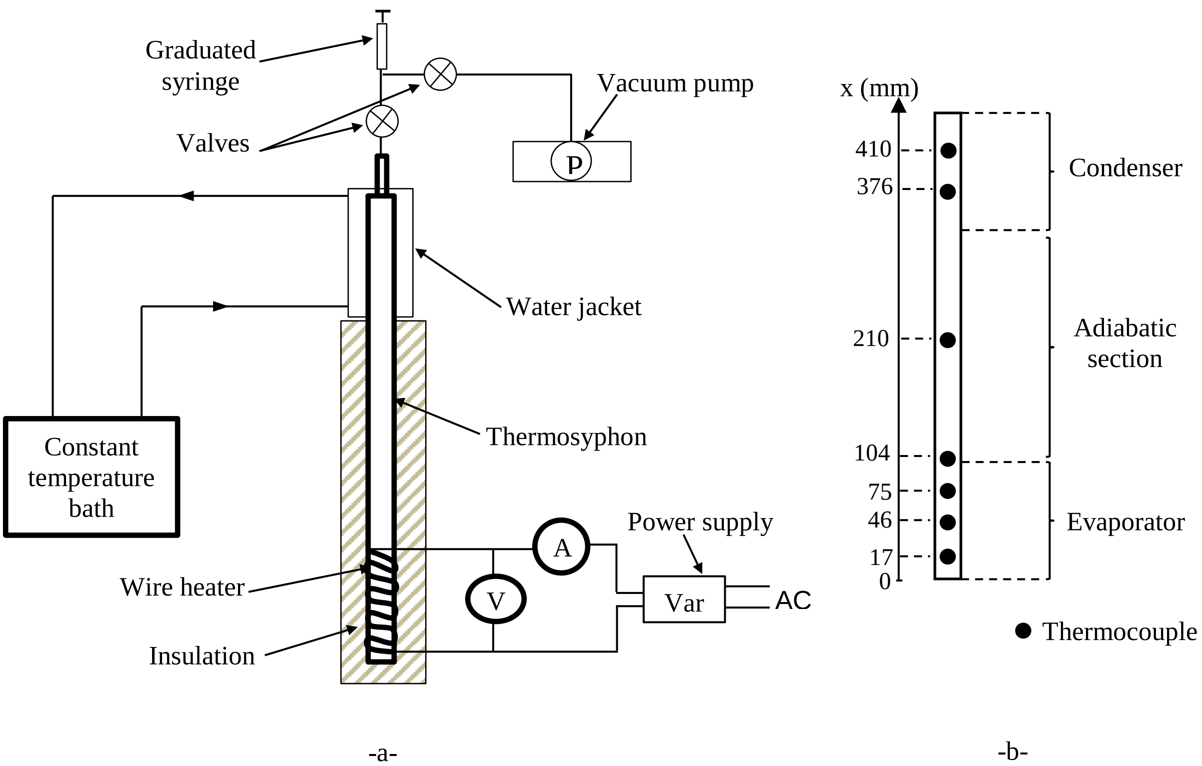

Figure 1 : -a- Schematic diagram of the thermosyphon test rig -b- axial locations of the thermocouples

The temperature measurements were monitored by a data acquisition system (Benchlink Data Logger) with the sampling period of 1.0 min during the test. The uncertainty of temperature measurement was estimated as ±0.2°C. At the beginning of each test, cooling water was pumped in the condenser jacket before power supplied. Then, input power was regulated to a desired level. The system needed enough time to reach steady state when each thermocouple temperature fluctuation was less than 0.5 °C within 5 min. At the steady state, the temperature at various locations were recorded. The same procedure was repeated at each power input level with various volume sets.
The main characteristics of the thermosyphon are summarized in table 1. It is made of stainless steel (1Cr18Ni9Ti) and charged with a compatible working fluid: ethanol.

Table 1 : Main characteristics of the thermosyphon

Envelope material	Stainless steel (1Cr18Ni9Ti)
Working fluid	Ethanol
Inner diameter (*mm*)	10
Outer diameter (*mm*)	12
Total length (*mm*)	450
The evaporator length (*mm*)	100
The condenser length (*mm*)	100
Input power (*W*)	9-18-25-40
Cooling Fluid temperature (*°C*)	5-10-15-20
Filling ratio (fluid volume)	10% (3.5 ml) – 20% (7 ml) – 35% (12 ml)

The expected optimal fluid filling volume was determined as the minimum of the effective overall thermal resistance Re was reached at a refrence temperature of 20°C (Figure 2). It was defined as [13]:

$$Re = \frac{\overline{T}_{evap} - \overline{T}_{cond}}{Q} \qquad (1)$$

A volume of 7 ml of ethanol was obtained. It was considered as the optimal case since minimum values of Re are reached (11.2 °C/W). In non-operational mode, this volume is sufficient to overfill the evaporator. In operational mode, it provides enough liquid to wet the evaporator section and to provide adequate thin liquid film along the condenser and the adibatic sections.

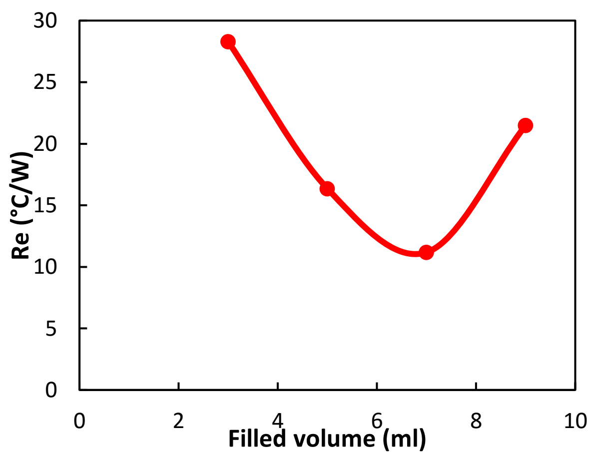

Figure 2 : The overall thermal resistance as a function of the ethanol filled volume

The filling ratio (FR) is defined as the volume ratio of the charged liquid to the whole thermosyphon. Three filling cases were then set according to the filling ratio of the thermosyphon: the optimal case (FR=20%), the underfilled case (FR=10%) and the overfilled case (FR=35%). To investigate the effect of the heat load on the thermosyphon operation four heat input level were considered : 9-18-25 and 40 W. The effect of the temperature of the cooloing fluid was also investigated. Four temperature levels were considered in the constant-temperature bath: 5, 10, 15 and 20°C, respectively (Table 1).

3. Results and discussion

To characterize the thermosyphon thermal behavior by worsening the operating conditions, a serie of tests have been performed with a simultaneous increase of the input heat load and the cooling fluid temperature. The three filling ratios have been considered: FR=10% (the underfilled case), 20% (the optimal case) and 35% (the overfilled case). The obtained results are shown in Figures 3, 4 and 5, respectively for the optimally filled, the underfilled and the overfillled cases.

For the optimally filled thermosyphon, a temperature increase is obtained with the heat load and cooling fluid temperature increase in the three sections (Figure 3). An isothermal state is almost observed in the evaporator for all the operating conditions. The vapor temperature T_v, which is generally defined as the adiabatic section wall temperature, increases to adapt the thermosyphon to the operating conditions worsening. At the condenser section, the heat load increment results in wall temperature increase as a consequence of the heat transfer rate decrease.

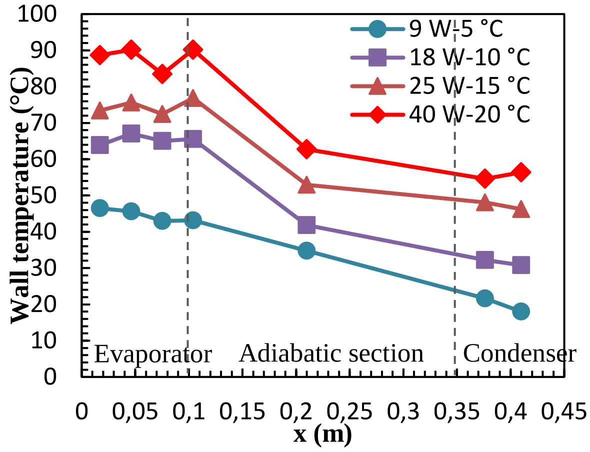

Figure 3 : Wall temperature distribution along the thermosyphon (FR=20%) for various operating conditions

If the thermosyphon is underfilled (FR=10%) then a temperature gradient from the evaporator and the condenser is established since the first heat load level (9 W). It regresses when the input heat flux increases due an important increase of the condenser wall temperature (Figure 4). For high heat loads (Q= 25-40 W), the wall temperatures in the condenser and the adiabatic section are already uniform. For an underfilled thermosyphon, Shabgard et al. [14] showed that the liquid pool vaporizes and forms a thin liquid film on the inner surface of the condenser section which starts to move downwords. In this case, the liquid pool is completely vaporized before the liquid film reaches the pool. As a result, the dry-out occurs at the bottom part of the evaporator and wall temperature increases dramatically.

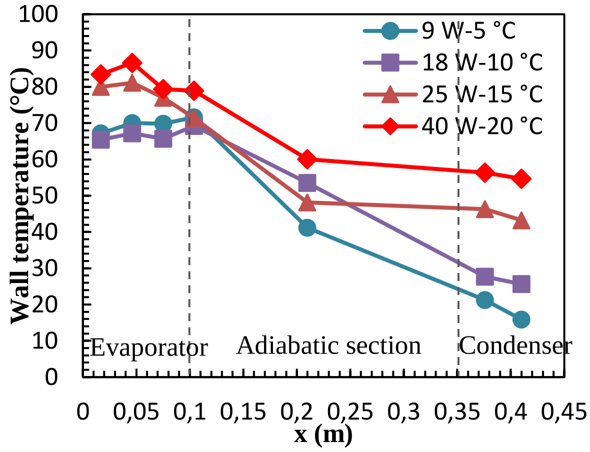

Figure 4 : Wall temperature distribution along the thermosyphon (FR=10%) for various operating conditions

Figure 5 shows the variation of the wall temperature along the thermosyphon according to the heat load and the cooling fluid temperature for the overfilled case. It can be seen that it is similar to the results of the optimal case (Figure 3). The main differences are that the isothermal characteristic of the evaporator is more obvious and that the temperature gradient along the thermosyphon is more important in the ovefilled case. For this last case, pool boiling is the predominant heat tansfer mechanism in the evaporator which is totally liquid submerged (FR=35%). In the optimally filled thermosyphon (FR=20%), a mixed pool boiling – thin film heat transfer mode appears in the heated section [5].

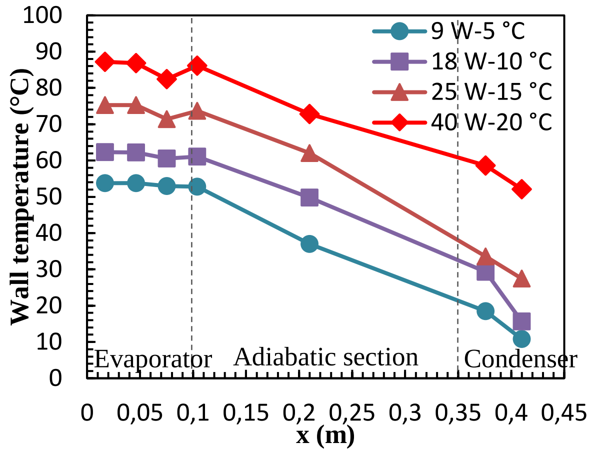

Figure 5: Wall temperature distribution along the thermosyphon (FR=35%) for various operating conditions

The heat transfer capacity of the evaporator is characterized by the heat transfer coefficient h_{evap}. From the measured data, it can be evaluated using the following equation [13]:

$$h_{evap} = \frac{Q}{\pi d_i L_{evap}(\overline{T}_{evap} - T_v)} \quad (2)$$

The vapor temperature T_v is commonly equivalent to the wall temperature of the adiabatic section [12]. Figure 6 shows the experimental values of the heat transfer coefficient h_{evap} as a function of the axial heat flux ($=Q/(\pi d_i L_{evap})$) for various filling levels. The highest heat transfer coefficient is obtained for the overfilled thermosyphon. It increases with the increment of heat flux for the three filling rates (10%, 20% and 35%). However, it is more significant for the overfilled case. Shabgard et al. (2014) [14] recommended a small amount of additional working fluid over the optimal filling ratio to ensure optimal and stable operation of a thermosyphon at steady state.

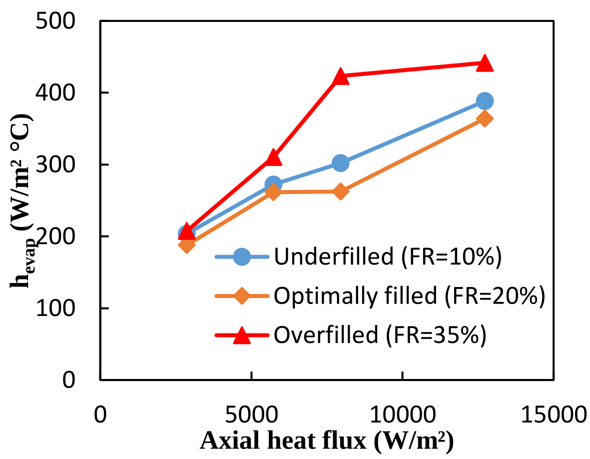

Figure 6 : Heat transfer coefficient in the evaporator vs the axial heat flux for various filling cases

Complex fluid flow and heat transfer behaviors such as natural convection, evaporation, and nucleate boiling, which result from circulation of the working fluid inside the thermosyphon, occur in the evaporator [13]. There is no general expression to simulate the heat transfer process of the evaporator. However, in a previous study, it has been found that nucleate boiling is the dominant mechanism in the evaporator when the filling ratio FR is higher than 30% [16]. In the literature, there have been many expressions of the heat transfer coefficient in the evaporator with some agreement with the experimental values [15]. Imura's correlation based on nucleate boiling in thermosyphons [17] is chosen to compare with the experimental data obtained by equation (3):

$$h_{evap,th} = 0.32 \frac{\rho_l^{0.65} k_l^{0.3} C_{pl}^{0.7} g^{0.2} Q^{0.4}}{\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1} (\pi d_i L_{evap})^{0.4}} \left(\frac{P_{sat}}{P_a}\right)^{0.3} \quad (3)$$

It showed indeed good agreement with the experimental values of h_{evap} for various working fluids and operating temperature ranges [4, 13, 14, 15]. As shown in figure 7, Imura's correlation (equation 3) over-predicts the heat transfer coefficient in the evaporator h_{evap} in comparison with the experimental data. However, it can be seen that the best agreement between the analytical and the experimental results is obtained for the overfilled case and low input heat flux. It can be explained by the fact that the considered correlation was carried out for nucleate boiling in thermosyphons, which is expected to be the dominant heat transfer mechanism in the liquid pool exceeding the evaporator section. This last configuration is used to compare the numerical and the experimental values. The obtained results are compared in figure 8. Almost all the data points are within the error range of ±20%. The experimental results of h_{evap} are in good agreement with Imura's correlation for ethanol and the overfilled case.

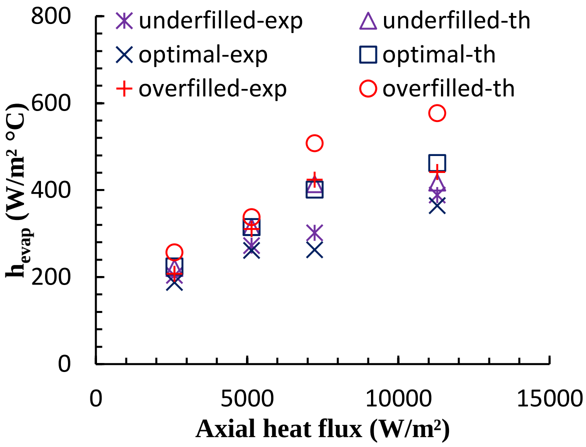

Figure 7: Experimental results and Imura's predictions of the heat transfer coefficient in the evaporator vs the axial heat flux for various filling cases

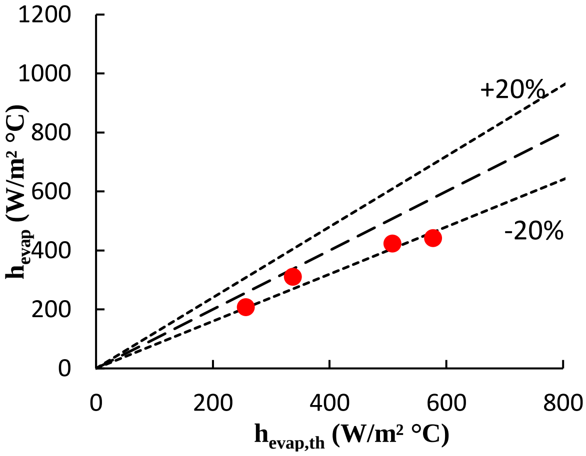

Figure 8: Predicted and experimentally determined heat transfer coefficient values in the evaporator for overfilled thermosyphon (FR=35%)

At the condenser section, the heat transfer capacity is much less affected by the increment of heat flux [18]. Moreover, the variations of the vapor temperature and the heat transfer coefficient with boundary conditions at the condenser are expected to be much smaller than those associated with the change in the input power [14].

4. Conclusions

A compact stainless steel thermosyphon filled with ethanol was experimentally tested. The effects of the input heat load, the temperature of the cooling fluid at the condenser and the filling ratio on the heat transfer characteristics were investigated at steady state. The obtained results showed that not only the increment of the input heat flux in the evaporator but also the increase of the cooling fluid temperature at the condenser had a meaningful effect on the temperature distribution along the thermosyphon. In particular, increasing the input heat load and the cooling fluid temperature resulted in a progressive increase in the wall temperature and then in the temperature gradient between the two thermosyphon extremities. For the underfilled thermosyphon (FR=10%), a significant temperature rise was obtained as soon as the first input heat flux was applied. It could be explained by the occurrence of the dry-out at the evaporator bottom. For the optimal (FR=20%) and overfilled (FR=35%) cases, the thermosyphon thermal behaviour was almost the same. Minor differences, such as the wall temperature at the coldest extremity and the temperature gradient between the evaporator and the condenser, could be noticed. The overfilled thermosyphon had a more specific isothermal behaviour in comparison with the two other filling cases.

The heat transfer capacity of the evaporator was characterized by the heat transfer coefficient h_{evap}. It was shown that the overfilled case involved the best h_e. It increased obviously with the heat flux increment for the overfilled thermosyphon. We also compared the experimental h_e values with the Imura's correlation predictions specifically developed for the pool boiling heat transfer in thermosyphons. It was then better convenient with the overfilled device. The results were indeed within the error range of ±20%.

Nomenclature

C_{pl}	specific heat, J/kg°C
cond	condenser
d_i	innner diameter, *m*
evap	evaporator
FR	filling ratio, *%*
g	gravitational accelation, *m/s^2*
h	heat transfer coefficient, *W/m^2 °C*
h_{fg}	latent heat of vaporization, *kJ/kg*
k_l	thermal conductivity of liquid,*W/m °C*
L	length, *m*
P_{sat}	vapor saturation pressure, *Pa*
P_a	atmospheric pressure, *Pa*
Q	heat load, *W*
Re	overall thermal resistance, *°C/W*
T	temperature, *°C*
$\overline{T}$	average temperature, *°C*
th	theoritical
v	vapor
x	axial location along the thermosyphon, m
ρ_l	density of liquid, *kg/m^3*
ρ_v	density of vapor, *kg/m^3*
μ_l	viscosity of liquid, *Ns/m^2*

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