International Journal of Heat and Mass Transfer 116 (2018) 675–684

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

The influences of the inclination angle and evaporator wettability on the heat performance of a thermosyphon by simulation and experiment

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ARTICLE INFO

Article history:
Received 23 June 2017
Received in revised form 7 September 2017
Accepted 9 September 2017

Keywords:
Thermosyphon
Inclination angle
Evaporator wettability
Bubble emission frequency
Contact angle
Fluent CFD modelling

ABSTRACT

A model considering evaporator wettability in terms of the contact angle is developed in this study. In addition, experiments are conducted using a copper thermosyphon with a length of 240 mm and internal and external diameters of 22.2 mm and 25.4 mm. The influences of the inclination angle and evaporator wettability on the heat performance of a thermosyphon charged with water are investigated. It is observed that simulated temperatures agree well with experimental data with a relative error of 0.12% for a modified thermosyphon. The results show that bubbles attaching to the wall of the evaporator decrease as the inclination angle increases from 15° to 90°, which decreases thermal resistance by 59.5%. As the heating power increases from 10 W to 14 W, thermal resistance reduces more significantly (44.1% decrease) for an evaporator with a hydrophilic surface than for an evaporator with a hydrophobic surface (20.6% decrease) because the bubble emission frequency of a hydrophilic surface rises more sharply (265% increase) than that of a hydrophobic surface (100% increase) at an inclination angle of 90°.

1. Introduction

The heating power per unit area of light emitting diode (LED) light chips has been increased to obtain higher heat performance, which results in the occurrence of high local temperatures in LED light chips. If heat is not effectively released, the reliability and the lifetime of the LED light will decrease. Therefore, heat dissipation is an important factor in the design of LED lights [1,2]. The thermosyphon is considered an efficient heat dissipation device because of its low thermal resistance [3]. Thermosyphons are used to cool LED light bulbs and are also applied in electronic equipment cooling [4], nuclear reactor spent fuel storage pools [5], solar water heater systems [6], air preheating [7], heat recovery systems in ventilation and air conditioning [8], railway transportation systems [9], defrosting applications in heat pumps [10] and the Rankine cycle [11].

Several researchers have investigated the heat performance of thermosyphons experimentally. Gedik [12] and Huminic et al. [13] reported the thermal performance of inclined thermosyphons and found that heat performance was optimum at an inclination angle of 90°. Huminic [14] reported that the heat performance was optimum at an inclination angle of 90° for a thermosyphon charged with nanoparticle concentration levels of 5.3%. Paramethanuwat et al. [15] reported that the heat performance was optimum at an inclination angle of 90° for a thermosyphon with a diameter of 7.5 mm charged with nanofluids. Jouhara et al. [16] reported that thermal performance of an inclined-condenser wickless heat pipe charged with an ethanol-water azeotropic mixture was optimal at an inclination angle of 0°. Grooten and van der Geld [17] reported that operation limiting heat flux of a long thermosyphon charged with R-134a increased with increasing inclination angles in the range of 0–90°. Negishi and Sawada [18] reported that high heat transfer rates of a thermosyphon charged with water and ethanol were obtained at inclination angles of 20–40° and more than 5°. Zhang et al. [19] reported that thermal resistance of a thermosyphon charged with ammonia was the minimum at an inclination angle of 20° and low temperature. Ong et al. [20] reported that thermal resistance of a thermosyphon charged with R410A was the minimum at an inclination angle of 60° with a filling ratio (the ratio of liquid volume to evaporator volume) of 100%. Dangeton et al. [21] proposed a correlation to predict the heat flux of a miniature loop thermosyphon at an inclination angle of 90°. Noie et al. [22] reported that the output heat transfer rate of a thermosyphon charged with water was the maximum at inclination angles of 15–60°. Emami et al. [23] reported that the heat transfer rate of a thermosyphon was the maximum at an inclination angle of 30° for a filling ratio of 15%.

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https://doi.org/10.1016/j.ijheatmasstransfer.2017.09.028
0017-9310/

Nomenclature

C_p	specific heat (J/kg K)
E	energy (J/kg)
I	current (A)
g	gravitational acceleration (m/s^2)
k	thermal conductivity (W/m K)
$\hat{n}$	the surface normal next to the wall
p	pressure (Pa)
Q	heating power (W)
R_T	thermal resistance (°C/W)
S_m	mass source term (kg/m^3 s)
S_E	energy source term (J/m^3 s)
$\hat{s}_w$	the unit vector normal to the wall
T	temperature (K)
t	time (s)
$\hat{t}_w$	the unit vector parallel to the wall
U	voltage (V)
$\vec{u}$	velocity vector (m/s)
$\vec{u}_{contline}$	the contact line velocity (m/s)

Greek symbols

α	volume fraction
α_1	volume fraction of liquid phase
α_v	volume fraction of gas phase
κ	interface curvature (1/m)
μ	dynamic viscosity (kg/m s)
ρ	density (kg/m^3)
σ	surface tension coefficient (N/m)
σ_V	uncertainty in voltage
σ_I	uncertainty in current
ℓ	the direction of contact line motion
θ_w	contact angle (°)
$\nabla\cdot$	divergence
∇	gradient

Subscripts

l	liquid
v	vapor

Khazaee et al. [24] proposed an empirical equation based on various parameters, including inclination angle, aspect ratio, filling ratio, and heating power. Rahimi et al. [25] reported that the average thermal resistance of a thermosyphon with hydrophilic evaporator and hydrophobic condenser decreased 2.35 times. Solomon et al. [26] reported that the heat transfer coefficient of a thermosyphon with hydrophilic evaporator improved by 15% compared to an unmodified thermosyphon. Singh et al. [27] reported that the total thermal resistance of an anodized thermosyphon decreased by 20% compared to a non-anodized thermosyphon at a heating power of 50 W. Solomon et al. [28] reported that the average evaporator temperature of a thermosyphon with a coating decreased by 22.7 °C compared to that of an uncoated thermosyphon with an inclination angle of 60° and heating power of 50 W.

It is difficult to describe the phase change process of thermosyphons and to explore their mechanism in detail using experiments. Numerical simulations are an effective approach for predicting heat performance before experiments are performed to reduce the time and cost of investigations; some researchers have utilized different approaches to simulate the heat and mass transfer process of thermosyphons.

Alizadehdakhel et al. [29] simulated the phase change process in a thermosyphon by using a VOF approach. Fadhl et al. [30,31] developed a volume of fluid (VOF) model considering the time relaxation parameter as a constant for simulating the heat and mass transfer process in a thermosyphon. Xu et al. [32] developed a model for simulating the heat and mass transfer process for a thermosyphon by considering transient time relaxation parameter. Alammar et al. [33] carried out computational fluid dynamics (CFD) simulations of a two-phase flow to investigate the influences of the inclination angle and filling ratio on heat performance using FLUENT software; they found that the optimum of filling ratio and inclination angle were 65% and 90°. Zhang et al. [34] investigated the effect of inclination angle on the efficiency of the collector and found high efficiency of the collector at inclination angles of 30–45°.

The above studies show that the inclination angle and surface modifications changing wettability have significant effects on the heat performance of thermosyphons. However, the heat transfer mechanisms of thermosyphons at various inclination angles and different evaporator wettabilities are not well described. The use of both numerical studies that consider the contact angle and experimental studies of the heat performance of a thermosyphon at different inclination angles and evaporator wettabilities for different heating powers has never been explored. Therefore, the objectives of this study are as follows: (a) to develop a model to investigate the heat performance of a thermosyphon by considering evaporator wettability in terms of the contact angle, (b) to validate the model by determining the relative errors of the absolute temperature between the simulation and experiment, and (c) to explore the mechanisms by which the inclination angle and evaporator wettability influence the heat performances of a thermosyphon.

2. Experiment

2.1. Setup

A copper surface was rubbed with sand paper. Then, the rubbed copper surface was washed with de-ionized water four times and washed with ethanol. The contact angle measurements were conducted using a contact angle meter (Drop Shape Analyzer-DSA100) before the experiment was performed. The contact angle of the modified copper surface was approximately 70°, thereby making the surface hydrophilic. Before the copper surface of the evaporator was airbrushed using the nanoparticle coating method, the nanosilica particles were prepared using the sol-gel method. Solution C consisted of solution A and solution B at a volume ratio of 1.5/100. Solution A included tetraethoxysilane and de-ionized water at a molar ratio of 1/4. Solution B included ethanol and de-ionized water at a molar ratio of 1/3. Solution C was mixed with 2-grams nanoparticles and stirred magnetically for 9 h at a constant temperature. To prepare the hydrophobic surface, in the first step, a copper surface was airbrushed repeatedly using the nanoparticle coating method. In the second step, the modified surface was coated with the mixture of trichlorosilane and methyl alcohol at a volume ratio of 1/100. Finally, the contact angle of the modified surface was 128°, thereby making the copper surface hydrophobic. The contact angle of the unmodified surface was 98°.

Fig. 1 illustrates the schematic diagram of the experimental setup. The setup includes a thermosyphon with internal and external diameters of 22.2 mm and 25.4 mm, a tilt platform, Bakelite, a