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Effect of Nanoparticle Deposition on the Thermal Performance of Evaporator in Thermosyphons

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Abstract. Rapid advancements in technology have led to the miniaturization of electronic devices which typically dissipate heat fluxes in the order of 100 W/cm^2. This has brought about an unprecedented challenge to develop efficient and reliable thermal management systems. Novel cooling technologies such as Two-Phase Thermosyphons that make use of nanofluids provide a promising alternative to the use of conventional systems. This article analytically estimates the effects caused by nanoparticles that deposit on the evaporator surface and their effect on the heat transfer process.

1. Introduction

It is estimated that air-based cooling infrastructures consume up to 45% of the total power [1] in data centers. In comparison, Two-Phase Thermosyphons (TPT's) have proven to be more efficient, sustainable, and economically viable cooling systems under high heat fluxes [2-4]. In these systems, the working fluid in its liquid phase absorbs heat from the source at the *evaporator* section converting into a two-phase fluid that travels upward via a *riser* to the *condenser* section where it exchanges heat with a secondary coolant resulting in a dense liquid that travels back to the evaporator through a *downcomer* thus forming a closed loop.

To enhance heat transfer in TPT, conventional working fluids can be replaced by nanofluids which are suspensions of solid (metal, metal oxide, carbon) particles with average crystal sizes below 100 nm (nanoparticles) [5-7]. Few studies have also reported a deteriorated performance [8-10] due to the changes induced in the two-phase heat transfer mechanism of its evaporator. The present article focuses on gaining insights into the heat transfer process in the evaporator by developing an analytical model of TPT and analyse effects induced by nanoparticles on the heat transfer process.

2. Modelling of TPT

The analytical model to evaluate performance uses the dimensions of TPT and heat flux experienced at the evaporator as input parameters. A mini TPT with following dimensions (evaporator: 4 parallel channels each 15 mm, 500 μm; riser: 30 cm, 4 mm; condenser: 25 cm, 4 mm; downcomer: 30 cm, 4 mm; dimensions in terms of length, radius) is modelled. The evaporator is assumed to consist of multiple parallel channels so that sufficient surface area is available to dissipate high heat fluxes. It is assumed that evaporator and condenser sections are horizontal with flow entering and exiting them at saturated conditions respectively while the riser and downcomer are adiabatic. Based on these dimensions and heat flux experienced by the evaporator from the source, hydrodynamic parameters are evaluated in each section of TPT [11-15] following a numerical methodology detailed in [16]. The outputs of this model namely, equilibrium mass flow rate and evaporator exit flow quality can then be used to evaluate the thermal performance of the evaporator. This is quantified in terms of its two-phase heat transfer coefficient (h_{tp}) which consists of nucleate (h_{nb}) and convective boiling (h_{cb}) mechanisms and is expressed as [17]

$$h_{tp} = \left(h_{nb}^2 + h_{cb}^2\right)^{0.5}. \tag{1}$$

The developed model was validated against data from the literature as shown in Fig. 1. The resulting h_{tp} for various heat flux inputs for R134a as working fluid at 16.8 bar operating pressure is shown in Fig. 2.

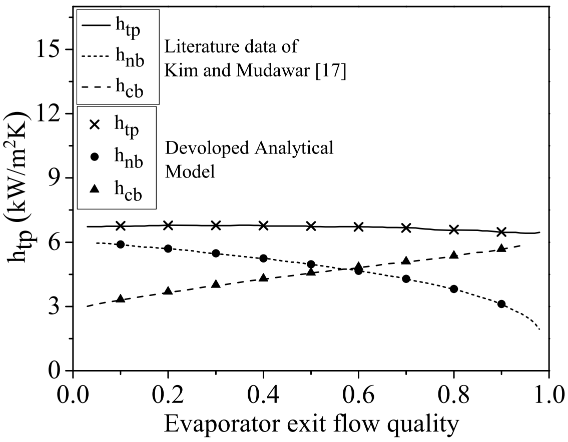

Figure 1. Validation of developed model

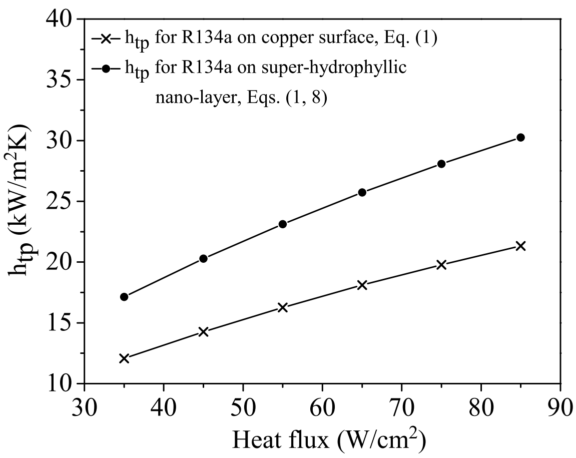

Figure 2. Evaporator thermal performance

In the given range of heat fluxes in Fig. 2, it is found that the heat transfer process is dominated by nucleate boiling mechanism (average $h_{nb}/h_{cb} = 6.7$). This implies that analysing the influence of nanoparticles on this mechanism is of primary importance. From Eq. (1) it can be derived that the relative change in h_{tp} with respect to h_{nb} can be expressed as

$$\Delta h_{tp}/h_{tp} = \left[h_{nb}^2/h_{tp}^2\right](\Delta h_{nb}/h_{nb}). \tag{2}$$

Equation (2) will be used in the next section to evaluate the effects of nanoparticles on the nucleate boiling mechanism which in turn affects the overall heat transfer rate.

3. Effect of nanoparticles on nucleate boiling

In general, a heating surface (evaporator in case of TPT) has certain micro-level cavities where the working fluid in its liquid state upon experiencing heat flux converts into vapor thus creating a bubble. This is the mechanism underlying nucleation boiling and can be expressed as [17]

$$h_{nb} \propto \left(Bo\frac{P_H}{P_F}\right)^{0.70} p_r^{0.38}(1-x)^{-0.51}. \tag{3}$$

The ratio of heated to the wetted perimeter (P_H/P_F), reduced pressure (p_r), and diameter of the channel (D) are constants and thus remain the same for both conventional fluids and nanofluids. The liquid mass fraction (x) depends on the heat transfer rate. Boiling number (Bo) is directly proportional to the heat flux ($q^{''}$) experienced by the working fluid which can be quantified as $q^{''} \propto \Delta T_s^{1.4} N_a^{0.4}$ [18] where ΔT_s is the wall super-heat (difference between surface temperature T_s and saturation temperature of the working fluid) and N_a is the active nucleation site density (i.e. micro-level cavities). Although this has been derived for pool boiling mode (stationary fluid), it is used in the present study for a TPT in which flow boiling takes place (working fluid is driven by pressure gradients); as the underlying nucleation mechanism is quite similar in these modes for a mini/micro channel [19]. Substituting this into Eq. (3) yields

$$h_{nb} \propto Bo^{0.7} \propto (q^{''})^{0.7} \propto [\Delta T_s^{1.4} N_a^{0.4}]^{0.7}. \tag{4}$$

Here N_a depends on the non-dimensional surface roughness R_{nd} and wall super-heat ΔT_s [20]

$$N_a \approx R_{nd}^{-0.4} \Delta T_s^3. \tag{5}$$

When nanofluids are boiled, nanoparticles form an irregular porous deposition (nano-layer) on the heating surface [21]. Formation of such nano-layer alters the non-dimensional surface roughness ($R_{nd,nf}$) which can be expressed from [22] as $R_{nd,nf} = R_{nd} \cdot \lambda^3 \cdot \psi^{-0.5}$. Where λ is the wettability parameter (depends on contact angle between fluid and nano-layer) and ψ is the interaction parameter (ratios of average roughness of pure surface and diameter of nano-particle used). The nano-layer also affects ΔT_s, as the working fluid is now in contact with the top of the nano-layer. The temperature (T_n) of the nano-layer differs from the inner wall temperature due to the thermal resistance of the nano-layer. Let the new super-heat be ΔT_n after the nano-particles have deposited. Substituting these into Eq. (5) yields the new nucleation site density ($N_{a,nf}$) after the formation of nano-layer i.e. $N_{a,nf} \approx R_{nd}^{-0.4} \lambda^{-1.2} \psi^{0.2} \Delta T_n^3$. Using this along with Eq. (5), the relative change in nucleation site density can be expressed as

$$\Delta N_a/N_a = (N_{a,nf} - N_a)/N_a \approx [\lambda^{-1.2}\psi^{0.2}\Delta T_n^3 - \Delta T_s^3]/\Delta T_s^3. \tag{6}$$

Scaling analysis for dimensions as assumed before is performed to determine the difference between T_n and T_s for nano-layer thickness (δ) using 100 nm aluminium oxide nanoparticles. For simplicity, it is assumed that nanoparticles stack vertically to form a nano-layer. In the range of heat fluxes indicated in Fig. 2, at least 5% (240 layers) of the evaporator radius must be filled by nano-layer for T_n to differ from T_s by at least 1%. This is highly unlikely considering the dimension of evaporator channels, hence thermal resistance is neglected (i.e. $T_n = T_s$). Consequently, Eq. (6) is reduced to

$$\Delta N_a/N_a \approx \lambda^{-1.2}\psi^{0.2} - 1. \tag{7}$$

From Eq. (4), the dependence of h_{nb} on N_a can be expressed as $\Delta h_{nb}/h_{nb} \approx 0.28(\Delta N_a/N_a)$ where the value of $\Delta N_a/N_a$ can be obtained from Eq. (7). Substituting the resulting expression ($\Delta h_{nb}/h_{nb} \approx 0.28[\lambda^{-1.2}\psi^{0.2} - 1]$) into Eq. (2) yields

$$\frac{\Delta h_{tp}}{h_{tp}} \approx 0.28 \left[\frac{h_{nb}^2}{h_{tp}^2}\right] \left[\lambda^{-1.2}\psi^{0.2} - 1\right]. \tag{8}$$

From this relation, it can be concluded that h_{tp} is maximized when the working fluid has high wettability with the formed nano-layer (i.e. λ is low) and the same has been experimentally observed [23, 24] in which low contact angles lead to high rates of thin micro-layer evaporation.

h_{tp} is also enhanced when the nanoparticles of relatively smaller size (ψ is high) split up the existing nucleation sites and this is in agreement with observations [25]. A similar trend is observed in experiments in which h_{tp} is related to the surface roughness as $\sim R^{0.2}$ [26, 27].

Equation (8) is used in the earlier developed model of TPT, assuming nanoparticles of diameter 100 nm on copper evaporator with the average surface roughness of 1 μm ($\psi = 10$). R134a has a contact angle of 4.7° at 16.8 bar [28] on the copper surface. To compute the theoretical maximum heat transfer enhancement with nanoparticles, it is assumed that the nano-layer acts as a super-hydrophilic surface ($\lambda = 0.003$). The resulting h_{tp} is shown in Fig. 2, where an average increase of 44% is predicted for the given range of heat fluxes.

4. Conclusions

On the basis of a model developed for a TPT with a mini channeled evaporator section, it has been observed that the nucleate boiling mechanism dominates the heat transfer process. The effect of nano-layer formed by nanofluids in a TPT is analytically analyzed in terms of changes in surface wettability and nucleation site density. An average increase of 44% is predicted in the heat transfer coefficient. The present model is based on steady-state conditions. However, due to the inertial forces of a working fluid, nanoparticles might lift off the nano-layer, making it a transient process. To determine this, a more detailed understanding of the deposition mechanism is needed. It would also help to quantitatively analyse increased surface area due to such a layer.

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