Numerical study on thermohydraulic behavior in evaporator section of wicked copper-water heat pipe at low superheat

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Abstract. Previous numerical studies have focused on the temperature profiles, flow pattern, heat transfer characteristics and transient behavior of wickless heat pipes. With the increase of heat dissipated power, the heat transfer capacity of the wickless heat pipe is no longer satisfied. Therefore, the research on flow and heat transfer of wicked heat pipe is more meaningful. In this paper, the numerical model of the heat pipe evaporator with a simplified rectangular wick structure was simulated. The present work to be aimed at the numerical model investigating on possible heat transfer performance. The effects of wick thickness, wick porosity, contact angle, single and double layer on heat transfer and flow characteristics were studied. For the wick thickness of 0.25, 0.5 and 1 mm, the 0.5 mm wick exhibits good versatility in heat transfer and flow. The porosity of 0.3 exhibits better heat transfer than that of 0.5, 0.7, 0.9. The value of the Nusselt number of the porosity of 0.3 reaches 142. For the contact Angle of 0, 30, 60, 90 degrees, 30 degrees has better heat transfer performance. Compared with single-layer wick, the double-layer wick can better promote thermal-hydraulic behavior.

1. Introduction
The demand to high heat flux for applications of the energy storage and transportation has identified heat pipe installations as systems suitable for this purpose \cite{1}. Heat pipes can transfer heat over a distance of a small temperature gradient using the principle of evaporative latent heat \cite{2}. Its stable operation depends on capillary forces and liquid returns in the wick. The capillary action is exerted by proper wicks or by proper axial or circumferential grooves \cite{3}. With the heat dissipation of electronic equipment increasingly precise, there is no doubt that the study of wicked evaporator in heat pipe is more significant than the macroscopic study of heat pipe.

In order to observe heat transfer promotion effects of the wick on evaporator, the visualization experiment \cite{4, 5} and the thermal hydraulic experiment \cite{6, 7} on the heat pipe are emerging in an endless stream. However, the selection of the wick relies too much on engineering experience and lacks scientific repeatability. Due to the high equipment price and complex screen specifications, the numerical simulation gradually improved the experiment and analyzed the flow pattern and heat transfer of the complex screen.

In recent years, the numerical simulation of heat pipe is divided into theoretical analysis, flow characteristics and heat transfer simulation. For the theoretical analysis of heat pipe, theoretical investigations, transient models and coupled equations were proposed and studied to predict the heat transfer performance of heat pipe \cite{8, 9, 10}. For the flow characteristics and heat transfer simulation of heat
pipe, the two-phase flow phenomena of flowing and boiling of thermosyphon was investigated using the Volume Of Fluid (VOF) model or other Eulerian multiphase domain [11-13]. Moreover, flow profiles and interfacial phenomena about the boiling regime and flow pattern in two phase flow in pipes were studied [14, 15].

Simulation of flow heat transfer in heat pipes has been achieved, but researchers hope to do more than that. In order to enhance the heat transfer in heat pipe, some simulation studies on enhanced surface and working medium have also emerged. Addition extended surfaces and different structures of the thermosyphon were numerically investigated to achieve the possibility of promoting heat transfer [16-18]. Fadhl et al. [19] used two refrigerants, R134a and R404a, as the working liquid of thermosyphon. On the other hand, heat pipe characteristics are also helpful to enhance heat transfer, such as inclination angle, liquid filling ratio, heat pipe structure parameters, etc. The effect of various filling ratios and inclination angles on the thermohydraulic behavior of thermosyphon were simulated and studied [20-24]. Ma et al. [25] introduced an evaporative model to study the effects of the fluid properties, the gravity level and the screen structure parameters on the vertical wicking.

The previous numerical study of heat transfer and flow pattern in wickless heat pipe has been sufficient, but the numerical study of evaporator in wicked heat pipe is still worth exploring. It can be seen that this aspect is still relatively deficient. Even the 2-d simulation is a good supplement to the traditional experimental study on the evaporator of the wick. In this way, regardless of the screen type, sintered type, groove type of the wick, their porosity, permeability will have a better reference. Rather than relying heavily on empirical formulas and experimental research. In combination with simulation research, the perfect combined semi-empirical formula is very convenient and necessary for the selection of the wick in engineering applications. In this paper, a 2-d transient VOF model is used for the wicked evaporator of heat pipe to account for the effects of screen thickness, porosity, contact angle and wick layers on thermal-hydraulic behavior. The continuum surface (CSF) model and evaporation model are coupled with simplified wick structure based on capillary limit and heat transfer.

2. Mathematical methodology
In this paper, the Volume of Fluid (VOF) method and the Euler-Euler approach have been used to simulate the wicked evaporator [26]. The VOF method can clearly track the interface and the motion of different phases through defining the volume fraction [27, 28].

2.1. Navier-Stokes equations

2.1.1. Continuity equation. For the vapor phase, this equation is expressed as follows:

\[
\frac{\partial}{\partial t}(\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \vec{u}_v) = S_M
\]

(1)

\[\alpha_l + \alpha_v = 1\]

(2)

where \( \alpha_v, \rho_v, \vec{u}_v \) represent the volume fraction, density and velocity vector of the vapor phase, respectively; \( S_M \) is the mass transfer from liquid to vapor during evaporation process.

2.1.2. Momentum equation. The momentum equation is dependent on the volume-averaged density \( \rho \) and volume-averaged dynamic viscosity \( \mu \), which are the following term:

\[
\frac{\partial}{\partial t}(\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla P + \nabla \cdot [\mu (\nabla \vec{u} + (\nabla \vec{u})^T)] + \rho \vec{g} + \vec{F}
\]

(3)

\[\mu = \alpha_l \mu_l + \alpha_v \mu_v\]

(4)

\[\rho = \alpha_l \rho_l + \alpha_v \rho_v\]

(5)

where \( \vec{g} \) is the acceleration of gravity, \( P \) is the pressure, \( \vec{F} \) is the external force acting in the fluid.

In order to include the effects of surface tension along the interface between liquid and vapor, the continuum surface (CSF) model proposed by Blackbill [29] is applied to the momentum equation:

\[
F_{CS} = \sigma_{lv} \frac{\rho_l c_v \alpha_l}{\frac{\rho_l + \rho_v}{2}}
\]

(6)

where \( \sigma_{lv} \) is the surface tension along the interface between the liquid and vapor phases.
2.1.3. Energy equation. The energy equation has the following term:

\[
\frac{\partial}{\partial t} (\rho E) + \nabla \cdot \left[ \bar{u} \left( \rho E + \rho \right) \right] = \nabla \cdot \left( k \nabla T \right) + S_E
\]

where \( S_E \) is the energy source term contained contribution from evaporation in this study, and volume-averaged thermal conductivity \( k \) is given by:

\[
k = \alpha_l k_l + \alpha_v k_v
\]

Both temperature \( T \) and energy \( E \) are treated as mass-averaged variables:

\[
E = \frac{\rho_l E_l + \rho_v E_v}{\rho_l + \rho_v}
\]

where \( E_l, E_v \) is the energy for the liquid and vapor phase, respectively. Both are based on the specific heat of the phase and the shared temperature.

2.2. Heat and mass transfer

In order to simulate the evaporation process precisely, a user-defined function (UDF) is developed to calculate the heat and mass transfer between liquid and vapor. The mass source term \( S_{M-iv} \), which represent the mass transfer from evaporation, are based on the previous research by Schepper et al.[11] and Fadhl et al.[30]. In addition, the energy source term of evaporation can be calculated by multiplying the mass transfer rate by the latent heat of the phase.

\[
S_{M-iv} = \beta_e \alpha_l \rho_l \frac{T_{sat} - T}{T_{sat}}
\]

\[
S_E = -\beta_e \alpha_l \rho_l \frac{T_{sat} - T}{T_{sat}} \Delta H
\]

where \( T_{sat} \) is the saturation temperature and \( T \) is the temperature of mixture phase of liquid and vapor. \( \beta_e \) is the so-called mass transfer time relaxation parameters for evaporation, which need to be tuned to match experimental data:

\[
\beta_e = \frac{6}{D_{sm} \sqrt{2\pi RT_{sat}}} \frac{\rho_l \Delta H}{\rho_v}
\]

where \( D_{sm} \) is the mean Sauter diameter, \( M \) is the molecular weight, \( R \) is universal gas constant and \( \Delta H \) is the latent.

3. Model description and simulation strategies

3.1. Geometry and mesh

A schematic diagram of the evaporative wicking process is reported in Figure 1. The model was
performed on a conventional copper heat pipe \((d_e=8 \, \text{mm})\), internally lined by simplified copper rectangular structure, with water as working fluid and operating in vertical. The evaporator zone at the pipe section and the condenser at the bottom pool. In addition, a schematic diagram of the evaporator with physical parameters is shown in Figure 2. The geometry and dimensions of the model are presented in Figure 2 and Table 1. In this study, the complex wick type was replaced by rectangular copper solid structure. For wick structure, the permeability is neglected because it is a kind of simplified structure model and the porosity is assumed to be the ratio of \(L_1\) to \(L_1+L_h\). The variation along the 3-d dimension is consequently neglected. According to Canti et al. [4], the temperature difference between the packages of center, left and right is less than 1 °C. This result leads to suppose that the 2-d assumption is valid. Moreover, all the cases are shown in Table 2.

In Figure 3, triangle mesh was applied to mesh the model because of its good connectivity. Capture curvature and proximity options were adopted to subdivide the mesh through which the liquid film flows. The grid near the boundary was refined because much more details about wall adhesion were needed, especially the details in the boundary layer. Different mesh sizes were used to test grid independence as shown in Figure 4. The average temperature of the inner heating wall downside and upside for different sizes for the case of these were monitored. As shown in Table 3, the numerical results of inner heating wall temperature were almost unchangeable when grid number reached 46482.

### Table 1 Specifications of the model

| Parameter | Value | Parameter | Value |
|-----------|-------|-----------|-------|
| \(L_e\) (mm) | 50 | \(L_{ph}\) (mm) | 10 |
| \(d_e\) (mm) | 8 | \(L_{pw}\) (mm) | 60 |
| \(L_w\) (mm) | 0.2 | \(L_{pt}\) (mm) | 5 |
| \(L_h\) (mm) | 0.25 0.5 1 |
| \(L_1\) (mm) | change with porosity |
| \(L_2\) (mm) | 0.3 |

### Table 2 Range of parameters under different cases

| Case No. | Porosity | Layer | Contact angle | \(L_h\) | \(L_w\) |
|----------|----------|-------|---------------|-------|-------|
| 1        | 0.3      | 1     | 150           | 0.5   | 0.2   |
| 2        | 0.5      | 1     | 150           | 0.5   | 0.2   |
| 3        | 0.5      | 2     | 150           | 0.5   | 0.2   |
| 4        | 0.7      | 1     | 100           | 0.5   | 0.2   |
| 5        | 0.7      | 1     | 120           | 0.5   | 0.2   |
| 6        | 0.7      | 1     | 150           | 0.5   | 0.2   |
| 7        | 0.7      | 1     | 180           | 0.5   | 0.2   |
| 8        | 0.7      | 1     | 150           | 0.25  | 0.2   |
| 9        | 0.7      | 1     | 150           | 1     | 0.2   |
| 10       | 0.7      | 2     | 150           | 0.5   | 0.2   |
| 11       | 0.9      | 1     | 150           | 0.5   | 0.2   |

### Table 3 Grid independence results

| Mesh size (cells) | 27614 | 46482 | 89631 |
|------------------|-------|-------|-------|
| \(T_{\text{heating wall downside}}\) (K) | 375.1692 | 375.7393 | 376.0685 |
| \(T_{\text{heating wall upside}}\) (K) | 374.1534 | 374.8427 | 375.1386 |
3.2. Boundary and initial conditions
In order to simulate the heating and evaporation, the first boundary condition was applied on the heating wall. A non-slip boundary condition was imposed at the walls of the model. Surface roughness was ignored, while wall adhesion and surface tension were considered. The working fluid is water and the surface tension is proportional to the volume-average density where initial value is 0.059 N/m. Water as the primary phase and vapor as the second phase. The gravity is considered along the y axis.

The total heat contains radial and axial heat. Referring to reference [4], the radial heat flux for the pipe at different degrees of the wall superheat ($T_w - T_s$) is then calculated by:

$$ Q = M_t (c_p \Delta T_{sup} + \Delta H) $$  

(13)

As a criterion for heat transfer, the $Nu$ coefficient was defined as boundary condition on the condenser's wall. The corresponding heat transfer $Nu$ coefficients have been calculated using the formula:

$$ h = \frac{q}{\Delta T_m} $$  

(14)

$$ \Delta T_m = \frac{\Delta T_{max} - \Delta T_{min}}{\ln \left( \frac{\Delta T_{max}}{\Delta T_{min}} \right)} $$  

(15)

$$ Nu = \frac{h d_e}{\lambda} $$  

(16)

3.3. Solution methods and techniques
For simulating surface tension and buoyancy effect, the explicit formulation and the body force formulation are adopted. Geo-reconstruct and PRESTO! were adopted. Double precision and transient condition are started. The Coupled algorithm was used for pressure-velocity coupling. The SST $k$-omega model was adopted. The momentum was discretized by Second order upwind. In addition, the turbulent kinetic energy and the specific dissipation rate were discretized by first order upwind. The time step size varies from $10^{-4}$ to $10^{-6}$ based on the reasons of considering accuracy and time cost.

3.4. Model Validation
According the experimental investigations performed by Canti et al. [4], the numerical model in this paper, can be referred and belonging to his vertical evaporator test. In Figure 5, the validity of the numerical simulation was verified by comparing its heat transfer performance with the experimental results.

4. Simulation results

4.1. Effect of screen thickness ($L_h$)
Prior to investigating the thermal-hydraulic behavior of the model, the one set of data thermal performance is benchmarked. A given constant 5 K superheat is allowed to reach steady state. All the following working conditions are carried out under 5 K superheat. Steady state is defined as when the
standard deviation in the time-averaged temperature profiles of inner wall temperature is less than 5%.

Under the same porosity, the thickness of the wick structure is changed to compare the effect of different thickness on heat transfer and flow. It can be seen from Figure 6 that while keeping the porosity constant, the thickness of the wick structure is changed to obtain the axial change of the $Nu$ number. If the thickness is maintained at 1 mm, the heat transfer is obviously inefficient. The wick structure plays the role of capillary adhesion. However, if it is microscopically speaking, the capillary height of the wick in this section cannot be supported to the next section. This explains why the heat transfer efficiency is not high in the case of a thicker screen. Because the liquid can be carried, it cannot flow through the wall and maintain a stable liquid film.

In Figure 7, the pressure drop curves of the three cases are compared. The pressure drop curve with a thickness of 1 mm shows that the capillary pressure is not able to flow through the wall due to the limitation of the wick. For the working conditions of 0.5 mm and 0.25 mm, the difference in the first half is due to the difference in the density of the wick and the imparity in the capillary pressure produced. In the second half, the pressure drop of 0.5 mm is higher than that of 0.25 mm. This is because the dense arrangement of the wick structure is not conducive to the separation and diffusion of bubbles. Then the liquid accumulates between the cracks, which causes additional thermal resistance.

4.2. Effect of porosity (the ratio of $L_1$ to $L_1 + L_h$)

In this work, observation is positioned at the heating wall or the center of the pipe. For the same wick structure, the heat transfer performance with four kinds of porosity is first examined. The heat transfer performance under different porosity is compared here. Figure 8 compares the Nusselt number ($Nu$) for a wick ($L_h=0.5$ mm, $L_w=0.2$ mm) with different porosity under the superheat of 5 K. Driven by capillary
action, the water gradually rises along the heating wall. It can be seen from the Figure 8 that $Nu$ shows a sharp downward trend in the first half of the y axis, while it gradually stabilizes in the second half. Since it is the inlet end of liquid reflux, the value of $Nu$ is larger. The end tends to be stable because it is restricted by capillary forces. $Nu$ with a porosity of 0.3 is stable at 142. However, there is no significant difference between the values of 0.5 and 0.7, which are 116 and 94 respectively. The minimum value with a porosity of 0.9 is 25. It can be concluded that the reduction of porosity is conducive to the enhancement of heat transfer. Figure 9 shows the variation curves of static temperature of different temperature. On the other hand, it shows that the temperature of the vapor-liquid mixture in the tube is related to the porosity, which affects the heat transfer in the evaporation section. However, only high porosity of 0.9 has a greater impact, while the other three have little difference in temperature. Then, in addition to the case where the porosity is 0.9, the factor affecting the heat transfer is the absorption capacity of the wick.

![Figure 8. Variation of $Nu$ at axial direction at different porosity](image1)

![Figure 9. Variation of averaged-temperature at axial direction](image2)

![Figure 10. Variation of flow characteristics at different porosity ($L_h=0.5$ mm)](image3)

Figure 10 shows the corresponding visualization at the evaporator area. The corresponding film climbing height and film thickness based on capillary force will also be compared. Some of the liquid absorbed by the wick will remain on the wall over time. In the gradual ascent, with the continuous evaporation, the liquid film gradually thinner, and the heat transfer becomes regionally stable. Four kinds of liquid film climbing heights with porosity from 0.3 to 0.9 can reach 28, 25, 22 and 15 mm, respectively. The study of porosity is meaningful if a stable and efficient heat transfer state can be obtained faster in the same time. Therefore, it is instructive for the arrangement of the suction core. However, in terms of bubble dynamics, lower porosity is not necessarily better. When the porosity is
0.3, the water vapor in the tube is not conducive to the transfer through the wick.

4.3. Effect of surface wettability

Figure 11. Variation of $Nu$ at axial direction at different contact angle

Figure 12. Variation of liquid film at axial direction at different contact angle

In this case, non-wetting conditions are not considered. The purpose of introducing the wick is to promote liquid reflux, not to further inhibit it. Figure 11 presents the Nusselt number of different contact angle of the porosity of 0.7. It can be concluded that, at different wall contact angles, the change of $Nu$ number in the second half of the evaporation segment is not very obvious. When the contact Angle is 30 degrees, the number of $Nu$ is the highest and the value is about 60; when the contact Angle is 0 degrees, the number of $Nu$ is the lowest and the value is about 108. According to the principle of capillary action, for the capillary effect of liquid column, it is undoubtedly the larger the contact Angle, the better. For the heat transfer in the evaporation section of the heat pipe, it is also important to maintain a certain thickness of the liquid film and not to be too thick to hinder heat transfer.

In Figure 12, the length and thickness of the liquid film under the fully wetted condition are better than other three conditions under different contact angles. However, if the liquid in the film does not evaporate quickly at this porosity, it will increase the thermal resistance of the evaporation section of the heat pipe. By comparing the number of $Nu$ and the thickness of the liquid film, a contact angle of 30 degrees has the best effect on heat transfer. In Figure 13, we can see that the wettability restricts the temperature field. The temperature field with high contact angle fluctuates little, and it cannot achieve a good heat transfer effect. At high contact angles of 0 and 30 degrees, the temperature field reacts violently. However, there are too many vortices on the wall, which will cause obstacles to the flow. Therefore, this also explains why the heat transfer becomes worse when the contact angle is 0 degrees.

Figure 13. Contours of temperature field and streamlines of the evaporator at different contact angle
4.4. Effect of wick layer

The porosity and contact angle of the single layer are discussed above. Then some composite double-layer wick structures will be analyzed. For the study of the number of layers in the wick, a comparison is made here between single and double layers with porosity of 0.5 and 0.7. As you can see from Figure 14, the number of $Nu$ of the two-layer is generally better than that of the single-layer. Monolayer $Nu$ with porosity of 0.5 and 0.7 is around 115 and 93. Then the $Nu$ number in the modified two-layer structure can reach around 166 and 145. In addition, from the distribution of the average temperature inside the tube, it can also be seen in Figure 15 that the double-layer structure has a more obvious improvement in temperature. The temperature difference between the two-layer structure and the wall surface is higher than that of the single-layer structure and is maintained in a stable range.

In Figure 16, the reinforcement of the double layer can also be explained by a visualization of the liquid volume contour. With the addition of a layer of structure, more walls can be attached to by water. Under the action of surface tension, the liquid film can be stable, rather than like a single layer wick, easy to fall off or evaporate. Moreover, the composite structure can have more space for bubble movement based on the inherent porosity. This is undoubtedly conducive to the evaporation and separation of the bubble. Compared with the heat transfer from the wall to the water, increasing the role of the wick also increases the heat transfer inside the evaporation section of the heat pipe.

5. Conclusion

In the present study, the 2-d transient VOF model of wicked evaporator of heat pipe was investigated.
The effects of screen thickness, porosity, contact angle and wick layers on thermal-hydraulic behavior were analyzed. The conclusions are as follows:

In the wick structure of 0.25, 0.5, and 1mm, based on the capillary pressure and the limitation of bubble detachment, the 0.5mm wick exhibits good versatility in heat transfer and flow. Among the porosities of 0.3, 0.5, 0.7, and 0.9, the porosity of 0.3 exhibits most heat transfer efficiency, and the value of $Nu$ reaches 142. From the point of view of evaporation heat transfer, the porosity of 0.5 and 0.7 has important effects on bubble dynamics than that of 0.3 and 0.9.

For wettability, the contact angle varies from 0, 30, 60, and 80 degrees. With a contact angle of 0 degrees, complete wetting is not conducive to heat transfer, and too thick liquid film will increase additional contact thermal resistance. The contact angle of 30 degrees will reach the working condition of better thermohydraulic performance and the $Nu$ will be around 108. Finally, the double-layer wick has better adhesion compared to the single-layer. The wick structure promotes heat transfer and facilitates the flow of vapor and liquid. A double-layer wick with a porosity of 0.5 can achieve a $Nu$ of about 166.

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