Evaporation Heat Transfer in Sintered Porous Media

A two-dimensional model is presented to predict the overall heat transfer capability for a sintered wick structure. The model considers the absence of bulk fluid at the top surface of the wick, heat conduction resistance through the wick, capillary limitation, and the onset of nucleate boiling. The numerical results show that thin film evaporation occurring only at the top surface of a wick plays an important role in the enhancement of evaporating heat transfer and depends on the thin film evaporation, the particle size, the porosity, and the wick structure thickness. By decreasing the average particle radius, the evaporation heat transfer coefficient can be enhanced. Additionally, there exists an optimum characteristic thickness for maximum heat removal. The maximum superheat allowable for thin film evaporation at the top surface of a wick is presented to be a function of the particle radius, wick porosity, wick structure thickness, and effective thermal conductivity. In order to verify the theoretical analysis, an experimental system was established, and a comparison with the theoretical prediction conducted. Results of the investigation will assist in optimizing the heat transfer performance of sintered porous media in heat pipes and better understanding of thin film evaporation. [DOI: 10.1115/1.1560145]

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improvement possible. The work of Ma and Peterson [5] showed that most of the evaporating heat transfer passes through a small region, less than 1 μm. Unfortunately, current fabrication processes preclude the further increase of the groove density and the large-scale manufacture of these types of grooved surfaces. As a result, it is necessary to develop new kinds of evaporative surfaces, with heat transport capabilities comparable to the thin film evaporating heat transfer coefficient required. After examining traditional evaporative surfaces/wick structures used in the design of heat pipes, it was found that sintered metal powders and/or particles could provide a higher thermal conductivity while still facilitating the thin film evaporating heat transfer.

If heat is added to the sintered porous medium fabricated from the round particles, as shown in Fig. 1, the heat is transferred through the sintered particles filled with the working fluid, reaching the top surface where the liquid-vapor-solid interface exists. There, by utilizing the thin film evaporation, the heat is removed. If the radius of a single particle is known, the evaporating heat transfer for that particle may be approximated as

\[ Q_p = 2 \pi r_p k_s (T_s - T_v) \int_0^{10^{-6} \frac{s}{\delta}} ds \]

where all of the evaporating heat transfer is assumed to pass through the micro-region, i.e., the region less than 1 μm [5]. The film thickness profile can be described by

\[ \frac{d \xi}{ds} \frac{dp_d}{ds} = - \int \nabla \cdot \mathbf{u} = \frac{g(s)}{h_l g} ds \frac{2 \delta'(s) \rho_l}{\delta} \]

for the steady-state evaporating process of a thin film, where the meniscus curvature, \( \xi \), and the disjoining pressure, \( p_d \), can be found by

\[ \xi = \frac{d^2 \delta}{ds^2} \left[ 1 + \frac{d \delta}{ds} \right]^2 \]

\[ p_d = \rho R T_w \ln \left[ a \delta^b \right] \]

respectively. The boundary conditions corresponding to Eq. (2) are

\[ \delta = \delta_0; \quad \xi = 0; \quad \frac{d \delta}{ds} = 0; \quad \text{at} \ s = 0 \]

where the nonevaporating film thickness is determined by [5]

\[ \delta_0 = e \left[ \left( \frac{T_w}{T_v} - 1 \right) \frac{h_f}{RT_w} - \ln a \right] \frac{b}{\delta} \]

As the evaporating thin film profile is obtained, the temperature drop across the evaporating thin film can be determined.

If the total number of particles at the top surface of wick is \( N_b \), the total evaporating heat transfer may be determined by

\[ Q = \sum Q_p \]

where the total number of particles is approximately found by

\[ N_p = \frac{A_{tp}}{4 \pi \rho_p} \]

and \( A_{tp} \) is the total area of top surface of sintered porous medium.

From Eqs. (1) through (8), it may be seen that a decrease in particle size results in an increase in the evaporating heat transfer. Alternatively, the smaller the particle size, the better the evaporating heat transfer performance. However, when the particle size is less than the characteristic length of the evaporating thin film region, the total heat transport capability in a sintered porous medium depends not only on the evaporating heat transfer, but also on the thermal conductivity of the sintered porous medium and the capillary flow. Neglecting the effects of convection heat transfer within the medium, the temperature drop occurring in a sintered porous layer can be approximated by \( T_w - T_v = QH/k_{eff} \), where \( H \) is the thickness of the sintered porous medium, and \( k_{eff} \) is the effective thermal conductivity. It is clear that for a given heat flux, an increase in the thermal conductivity would reduce the temperature drop across the sintered porous medium. The effective thermal conductivity of the porous medium of metallic spheres is a function of the solid conductivity, \( k_s \), the working fluid conductivity, \( k_f \), and the porosity, and it varies drastically between the limiting cases, i.e., \( \lim_{\delta \rightarrow 0} k_{eff} = k_s \) and \( \lim_{\delta \rightarrow \infty} k_{eff} = k_f \), depending upon the type of arrangement of the metallic particles. For the sintered particles investigated herein, the effective thermal conductivity can be determined by [14]

\[ k_{eff} = \frac{2k_s + k_f - 2e(k_s - k_f) \frac{2k_s + k_f + e(k_f - k_s)}}{2k_s + k_f + e(k_f - k_s)} \]

In order to sustain continuous capillary flow in the wick shown in Fig. 1, the capillary pumping pressure must meet or exceed the sum of all other pressure drops. The maximum capillary pumping pressure may typically be expressed as
\[ \Delta p_c = p_v - p_l = \frac{2\sigma}{r_c} \]  

where \( r_c \) is the meniscus radius at the evaporating surface. The effective capillary radius for a sintered evaporative surface is dependent upon the size of the sintered particles, the mode of packing, and sintering process. When the radius of the particles decreases, the capillary pumping increases, and it is expected that the size of the individual particles should be as small as possible. However, as the particle size becomes smaller, the permeability decreases.

When the fluid temperature reaches some pressure-dependant value above the saturation temperature, boiling occurs near the bottom of the evaporator resulting in two devastating consequences. First, thin film evaporation at the solid-liquid-vapor interface dramatically decreases as the boiling condition dominates the phase change behavior of the system. Secondly, the vapor generating at the base of the porous medium forms a blanket of vapor near the base of the medium, increasing the thermal resistance and preventing re-entry of the wetting fluid. Boiling heat transfer at the base of the wick should thus be avoided, as this condition could lead to an increased temperature drop across the wick structure and result in early wick dryout.

If the wick is constructed such that the temperature difference between the base temperature of the sintered porous medium and the saturation temperature, \( T_b - T_{sat} \), remains less than the superheat for the nucleation, bubble formation will not occur in the wick. Once the pressure distribution is determined, then by utilizing the Clausius-Clapyron relation and considering the surface tension, the minimum superheat for the onset of bubble formation can be determined by

\[ T_l - T_{sat}(p_l) = \frac{2\sigma T_{sat}(p_l)}{h_f p_{sat}} \left( \frac{1}{p_v} - \frac{1}{p_l} \right) \]  

Based on geometrical simplifications of the sintered perfect sphere particles, the maximum meniscus radius of the embryo bubble formed within the sintered sphere particles, i.e., the maximum cavity, may be approximated in terms of the particle radius as

\[ r_{b,p} = \frac{r_p}{4} \]  

Considering Eq. (13), Eq. (12) may then be expressed as

\[ T_l - T_{sat}(p_l) = \frac{8\sigma T_{sat}(p_l)}{h_f p_{sat}} \left( \frac{1}{p_v} - \frac{1}{p_l} \right) \]  

An embryo bubble will grow and a cavity will become an active nucleation site if the equilibrium superheat becomes equaled or exceeded one around the perimeter of the embryo. In order to avoid the bubble formation near the base of the porous medium, the temperature difference between the wall and the saturation temperature, \( T_b - T_{sat} \), must be less than the superheat indicated by Eq. (14).

Thin film evaporation can provide significantly higher overall heat transfer coefficients, but it is limited by the capillary force and by the onset of bubble nucleation. Clearly, there must exist a set of properties that will optimize the overall heat transfer capabilities in wick structures investigated here. In order to find the capillary limitation and the boiling limitation, the detailed distributions of temperature and pressure in wicks have to be first determined. Analogous to the actual heat pipe wick, fluid enters the wick from the side, and is pumped toward the mid-plane of the wick due to the capillary pumping force, as shown in Fig. 2. One of objectives in designing a wick structure for the evaporating region of a high heat flux heat pipe is to determine a set of geometric parameters that encourages thin film evaporation at the top surface of the wick. In defining the top surface boundary condition, it is thus assumed that phase-change heat transfer occurs only at the top surface of the sintered porous medium. It is then to determine what set of parameters could lead to such a condition for the largest anticipated heat flux input. Owing to the small size of the wicks investigated here and to the relatively small flow rates of the working fluids through the wick, the Darcy Modified
Rayleigh number, \( Ra = K_g \beta H \Delta T / \alpha \nu_f \) is less than 40 for the range of configurations considered herein. Since the Nusselt number due to the convection is less than 1 for these small Rayleigh values, it is reasonable to assume that the contribution of convection to the overall thermal transport may be neglected [16]. Because, by assumption, evaporation occurs only at the liquid-vapor-solid interface on the top surface, the flow in the wick is all single-phase liquid, and use of the area-averaged velocity, i.e.,

\[
u = \frac{1}{\Delta y \Delta z} \int_0^{\Delta z} \int_0^{\Delta y} u \, dy \, dz
\]  
(15)

in lieu of the actual local velocity of the fluid is appropriate. Using the area-averaged velocity technique, invoking Darcy’s Law, and introducing a buoyancy effect term to account for thermally induced flow in the vertical direction, the momentum equations for the system are

\[
u = -\frac{K_g}{\mu} \frac{\partial P}{\partial x}
\]  
(16)

\[
u = -\frac{K_g}{\mu} \left[ \frac{\partial P}{\partial y} - \rho g \beta (T - T_{sat}) \right]
\]  
(17)

Additional assumptions include constant property values except the wick effective conductivity, since it depends on the porosity. With these assumptions and the continuity relation within the wick, the governing equations describing the temperature distribution and pressure profile in wicks can be written as

\[
\nabla^2 T = 0
\]  
(18)

\[
\frac{\partial}{\partial x} \left( \rho \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \frac{\partial P}{\partial y} \right) = \rho g \beta \frac{\partial (T - T_{sat})}{\partial y}
\]  
(19)

It should be noted that in the development of the Darcy flow model, it is assumed that viscous effects are negligible and a slip condition must accordingly exist between each solid particles and the fluid. For high velocities, the slip condition no longer holds true. In these cases an additional friction term can be added as an extension to Darcy’s law, i.e.,

\[
p_f \left[ (V \cdot \nabla) V \right] = -\nabla P - \frac{\mu}{K} V
\]  
(20)

With the appropriate boundary conditions, Eqs. (18) and (19) can be readily solved for the temperature distribution and pressure distribution.

**Experimental System and Procedure**

In order to determine if a set of wick parameters exists for which the thin film evaporation condition may be established at the top surface, thus providing a maximum evaporating heat transfer coefficient as the forgoing discussion, the experimental system shown in Fig. 3(a) was established. The setup consisted of a machined copper housing, to which a sintered porous wick sample was attached shown in Fig. 3(b), a reservoir for the liquid supply, a data acquisition system for the temperature measurement, and a personal computer. Attached to the copper housing were barbed hose fittings that enabled plastic tubing to be used to connect the housing with a spigot on a large rectangular tank, which served as the reservoir for the working fluid. By adjusting the height of the tank, the surface height of the working fluid could be manipulated to cause the working fluid to flood the copper housing to the point where the fluid surface was leveled with the top surface of the wick sample. The liquid level maintained during the experiment because the liquid-vapor surface area of the tank was many times greater than the liquid-vapor surface area of the wicks tested.

A special mounting stand was manufactured from Lexan polycarbonate sheet to enable the heater to be firmly applied to the base of the copper housing. A thermally conductive paste was used at the interface between the copper heater and the copper housing to reduce the effects of contact resistance between the two components. Thermocouples were implemented to monitor the temperatures of the top and bottom surfaces of the sintered wick structure as well as the ambient temperature and outer heater surface temperatures, which were connected in differential mode to a cold junction compensated IO/Technical Personal DAQ56. In order to measure the interfacial temperature, a small groove was

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**Fig. 3** (a) Schematic of the experimental system; (b) test section; and (c) solid model of the heater fabricated for experimentation (cylinder diameter is 2.54 cm, top surface is 1x2 cm², and length of rectangular neck is 2 cm)
machined in the base of the sintered porous medium, where the thermocouple tip was mounted using Omegabond® 100 epoxy.

The sintered porous wick sample was fabricated from 100 Mesh, 99.99% pure copper particles prepared based on the recommended sintering temperature and time ranges of 840–900°C and 12–45 minutes [17]. Since the thermal and fluid transport properties of a sintered porous medium can vary significantly between identically prepared samples, it was difficult to isolate the thickness effect using multiple samples. Thus, an alternative method was proposed wherein a single, somewhat thick sample, was prepared for the testing process. Once testing had been completed on the initial sample, the thickness was decreased by removing layers of the sintered copper particles to allow for subsequent testing and the isolation of the thickness effect.

In order to achieve a heat flux as high as 1.0 MW/m², a special pure copper block wrapped by a Minco BB010047 bar heater was designed and fabricated. The heater was well insulated to reduce the effects of heat loss from the heating elements. To obtain a uniform heat flux added on the wick structure, the copper block was analyzed by means of the ALGOR heat transfer analysis software. In establishing the desired level of uniformity of the heat flux at the contacting surface, the copper block with a top contact surface of 1×2 cm² and a rectangular neck length of 2 cm, as shown in Fig. 3(c), was fabricated.

A voltage transformer was used to alter the electrical power into the resistive heating element. The voltage and current into the heater was monitored using a digital multi-meter. The temperatures of the base and top surface were monitored using the strip chart and digital display features in the DAQView software. When the steady-state temperature was reached for a given power level, the temperature readings of all thermocouples were recorded. The steady-state temperature was defined as the temperature for which less than one degree of change was observed for 5 minutes of continuously applied heat. Once the steady-state temperature readings for a given heat rate were recorded, the power was increased in steps of 5 to 10 W and the process was repeated.

Multiple tests were run for each thickness value tested. After the range of heat input values of interest had been tested multiple times, several layers of copper particles were scraped from the top surface of the wick. Once the particles were removed from the test container, the new thickness of the wick was measured and recorded, and the steady-state test process was repeated. While the test housing and heater were well insulated, some heat losses due to the combined effects of radiation and natural convection were inevitable. Calculations for the amount of nonevaporative heat loss based on outer insulation temperatures indicated that the loss was equal to 2–5% of the total heat added on the heater.

**Numerical Calculation Procedure**

Based the experimental investigation specially designed for the two-dimensional physical model shown in Fig. 1, the two-dimensional model, shown in Fig. 2, was employed. Owing to the symmetry, only half of the model was used to calculate the pressure and temperature distributions. The boundary conditions except for the top surface were expressed as follows

\[ q'' = 0 \quad \text{and} \quad \frac{\partial P}{\partial x} = 0 \quad \text{at} \quad x = 0 \quad \text{and} \quad y \in [0, H] \quad (21) \]

\[ q'' = 0 \quad \text{and} \quad P = P_v \quad \text{at} \quad x = L/2 \quad \text{and} \quad y \in [0, H] \quad (22) \]

\[ q'' = 0, \quad \frac{\partial P}{\partial y} = 0 \quad \text{at} \quad y = 0 \quad \text{and} \quad x \in [L_d/2, L/2] \quad (23) \]

\[ q'' = q_{sat}'' = -k \frac{\partial T}{\partial y} \quad \text{at} \quad y = 0 \quad \text{and} \quad \frac{\partial P}{\partial y} = 0 \quad \text{at} \quad y = 0 \quad \text{and} \quad x \in [0, L_d/2] \quad (24) \]

At the top surface, thin film evaporation occurs and vapor departs from the wick. Considering Eq. (1) due to the thermal resistance across the thin liquid film, the temperature on the top surface can be determined as

\[ T |_{y = H} = T_{ad}(P_v) + \frac{Q_p}{2\pi r_p k_{fg} l_{fg}^{1/4} \sqrt{rac{1}{\delta}}} \quad (24) \]

Once the temperature distribution in the wick is obtained, one can determine the heat flux into the top surface boundary. For steady-state heat transfer, the heat entering the boundary elements through heat conduction must be rejected via phase-change heat transfer at the top surface, i.e.,

\[ -k_{eff} \frac{\partial T}{\partial y} (1 \cdot dx) = \dot{m}_j h_{fg} \quad \text{at} \quad y = H \quad \text{and} \quad x \in [0, L/2] \quad (25) \]

where a segment of the top surface of unit depth and width, \( dx \), is considered. With a mass balance relation for the segment, \( \dot{m}_j = \rho_{v} v (dx \cdot 1) \), and the Darcy flow equation, \( v = -K/\mu \partial P/\partial y \), the top surface boundary condition for Eq. (19) can be found as

\[ \frac{\partial P}{\partial y} = \frac{\mu}{\rho_v h_{fg} \sqrt{2g}} \frac{\partial T}{\partial y} \quad \text{at} \quad y = H \quad \text{and} \quad x \in [0, L/2] \quad (26) \]

Once the energy equation with the required boundary conditions was solved, the temperature distribution was used to determine the temperature gradient and the heat flux across the liquid-vapor interface at the top surface of the wick. With the top surface boundary condition specified, the pressure distribution was then calculated. The top surface pressure on the vapor side of the meniscus was then determined using the Laplace-Young relation. The saturation temperature being a function of pressure was next found for the vapor side of the meniscus for each grid point in the wick. Using Eq. (14) and starting with an artificially large heat input value, the critical superheat was calculated for each grid point and compared to the local superheat for the onset of bubble nucleation. If the superheat at any point in the wick exceeded the critical superheat, the heat flux was reduced and the process re-started. After several repetitions, the applied heat flux was finally reduced to the one that every point in the grid remained below the local superheat for bubble nucleation. At that point, the maximum allowable heat flux without bubble nucleation was bounded. Once the heat flux for the nucleate boiling limit was determined for a given wick parameter set, one parameter value, the wick thickness for example, was increased or decreased and the entire process was repeated.

As the thickness, for example, is reduced, the cross-sectional area for fluid flow is subsequently reduced. For a given heat flux input value, assuming all other properties held constant, a decrease in the wick thickness results in an increase in both the liquid flow velocity and the pressure drop through the wick. The capillary pumping pressure must be large enough to compensate for the liquid pressure drop in the wick as well as those occurring along the rest of the working fluid flow path. If the total liquid pressure drop exceeded the capillary limit defined by Eq. (10), the heat flux was reduced and the process re-started. Once the capillary limit was obtained for a given condition, the entire process was repeated. Once both the nucleate boiling and capillary pumping heat transport limits had been obtained for a given thickness, the lesser of the two calculated transfer heat was accepted as the active dryout threshold.

**Results and Discussion**

Unless stated otherwise, the parameter values for each of the studies below were as follows: working fluid—water; \( r_p = 0.635 \) mm; \( L = 50.8 \) mm; \( L_d = 10 \) mm; \( \epsilon = 43\% \). Because of the difficulty in determining the onset of nucleate boiling in the wick and thin film evaporation on the top surface of the wick, the
Experimental data were obtained by the temperature difference between the base (bottom surface) and the top surface as a function of heat flux for a given wick including a known wick thickness, and continuously from the single-phase flow with the thin film evaporating on the top surface to the two-phase flow with the nucleate boiling in the wick.

Experimental results describing the relationship between wick thickness and temperature drop across the sintered wick are presented in Fig. 4 by plotting the dimensionless temperature defined by

\[ \theta = \frac{T_b - T_{sat}}{T_{sat}} \]

where \( T_b \) is the bottom surface (base) temperature, and \( T_{sat} \) is the saturation temperature corresponding to the local atmospheric pressure, equal to 743 mm Hg. As the heat flux increased, as shown in Figs. 4 and 5, the temperature drop across the wick increased. When the heat flux was increased to some value between 50 and 200 kW/m\(^2\) for the range of wick sizes tested, the heat flux necessary for bubble nucleation was reached, which could be visually observed. Due to the bubble formation, the region of the wick directly above the heat application zone became saturated with vapor thus blocking the re-entrance of liquid to the center of the wick, and thin film evaporation on the top surface of wick no longer existed. As a result, for the heat flux level higher than approximately 200 kW/m\(^2\), the slope of the dimensionless temperature, \( \theta \), versus the applied heat flux, \( q^\prime \), was highly linear resulting in a significant decrease of the evaporating heat transfer coefficient shown in Fig. 6. This phenomenon was particularly pronounced for the thinner wick thickness values. It can be concluded that thin film evaporation plays an important role for the maximum evaporating heat transfer coefficient. The evaporation at the liquid-vapor-solid interface, i.e., at the top surface, of the sintered wicks can optimize the thin film evaporation.

Achieving thin film evaporation on the top surface of wick means the vaporization occurs only on the top surface of wick, and a single-phase flow exists in the wick, which is also the basis for the numerical model developed herein. Clearly, the model presented in the paper cannot predict the results shown in Figs. 4, 5, and 6, where the two-phase flow existed already in wicks when the heat flux level was higher than approximate 200 kW/m\(^2\). Utilizing the highest evaporating heat transfer coefficient obtained for each wick, which occurred when the heat flux was equal to some value between 50 and 200 kW/m\(^2\) for the range of wick sizes tested, the relationship between the maximum evaporating heat transfer coefficient and wick thickness was presented in Fig. 7 and compared with the numerical prediction. While the theoretical prediction was higher than the experimental data, the comparison provides insight regarding the importance of thin film evaporation in maximizing evaporating heat transfer coefficient, and it illustrates how failing to establish a thin film evaporation condition results in a significant decrease in the evaporating heat transport capability.
As discussed previously, achieving thin film evaporation on the top surface of wicks is limited by the liquid pressure drop and superheat for the onset of bubble nucleation. The numerical simulation results, as shown in Fig. 8, indicate that the boiling limit is always less than the limit determined by the capillary pumping failure criterion for the wick configurations investigated here, which has an agreement with the experimental observation. As the heat flux was increased to some value between 50 and 200 kW/m² for the range of wick sizes tested, the bubble nucleation was visually observed. For the range of test wicks, the capillary limit was never reached, even with the maximum heat flux of 1.0 MW/m² available from the experimental system. Clearly, the bubble nucleation defined by Eq. (14) is the primary limit affecting the evaporating heat transfer coefficient for the sintered porous media investigated herein.

As shown in Fig. 9, when the wick thickness was decreased from 6 mm to 0.25 mm, the dimensionless temperature difference between the base and the top surface decreased considerably, indicating a dependency on the wick thickness for a given porosity and particle size. Numerical results, shown in Figs. 10, 11, and 12, all indicate that there exists an optimum wick thickness for maximum heat flux, which depends on the particle radius, the porosity, and the entry length, $L - L_b/2$. Results shown in Fig. 10 indicate that when the particle radius decreases, the optimum thickness increases. The effect of porosity on the optimum wick thickness is shown in Fig. 11. Based on the results of Figs. 10 and 11, it is concluded that, if it is possible to decrease the bead radius while maintaining a constant porosity, the bead radius should be as small as possible. The impact of these results is that thicker wicks, which are more readily manufactured and assembled into heat pipes, can provide heat removal capabilities equivalent to the more delicate, thin wicks. For wick thicknesses small enough that the capillary limit becomes dominant, as shown in Fig. 12, the model predicts that shorter entry lengths will yield larger dry-out heat fluxes. Results presented here can effectively direct the new design of high heat flux cooling device. For example, if an evaporating surface for a high heat flux heat pipe is fabricated using the optimum wick thickness shown in Fig. 10, the simple calculation from Figs. 7 and 10 indicates that the temperature drop across the wick is only several degrees for a heat flux of 800 kW/m².

Conclusions

A theoretical analysis and experimental investigation was conducted to determine that thin film evaporation occurring only at the top surface of a horizontally oriented wick with localized bottom heating plays an important role in the enhancement of evaporating heat transfer and further that there exists a wick that can make this occur. When evaporation occurs only at the top surface of the wick and the effect of bulk liquid on the vapor flow, one of factors limiting the evaporating heat transfer, is removed, the heat conduction resistance, capillary pumping pressure, frictional pressure drop, and the onset of bubble formation become primary...
factors determining the maximum evaporating heat transfer in the sintered wick structure. A two-dimensional model incorporating the heat conduction, capillary limitation, and the onset of nucleate boiling was developed to predict the overall heat transfer capability in the sintered wick structure. The numerical results show that it is possible to promote thin film evaporation from the top surface of a sintered wick by selecting the appropriate particle size, porosity, and thickness. By decreasing the average particle radius, the evaporation heat transfer coefficient can be enhanced. Furthermore, there exists an optimum characteristic thickness for maximum heat removal. In order to verify the theoretical analysis, an experimental investigation was conducted to determine the effect of the thin film evaporation at the top surface in the total evaporation heat transfer in the sintered wick structure. Experimental results show that thin film evaporation plays an important role in the enhancement of evaporating heat transfer, and evaporation occurring at the top surface, where the thin film regions were optimized, could reach the maximum evaporating heat transfer coefficient. Results also indicate that the maximum dry-out heat flux significantly depends on the wick thickness. While the theoretical prediction was higher than the experimental data, the comparison provides insights into how thin film evaporation occurring at the top surface of wicks plays an important role for the maximum evaporating heat transfer coefficient and how failure to establish a thin film evaporation condition results in significant decreases in the evaporating heat transport capability. Results of the investigation will assist in optimizing the heat transfer performance of sintered porous media in heat pipes and better understanding of thin film evaporation.

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Nomenclature

\[ a = \text{constant}, \ 1.5787 \]
\[ A = \text{area}, \ m^2 \]
\[ b = \text{constant}, \ 0.0243 \]
\[ f = \text{friction factor} \]
\[ h_f = \text{latent heat}, \ J/kg \]
\[ H = \text{thickness}, \ m \]
\[ k = \text{thermal conductivity}, \ W/m\cdot K \]
\[ K = \text{permeability}, \ m^2 \]
\[ L = \text{length}, \ m \]
\[ N = \text{quantity of units} \]
\[ p = \text{pressure}, \ N/m^2 \]
\[ p_d = \text{disjoining pressure}, \ N/m^2 \]
\[ q = \text{heat flux}, \ W/cm^2 \]
\[ Q = \text{Heat transfer, W} \]
\[ r = \text{radius}, \ m \]
\[ R = \text{gas constant}, \ J/kg\cdot K \]
\[ Ra = \text{Rayleigh number} \]
\[ Re = \text{Reynolds number} \]
\[ s = \text{coordinate}, \ m \]
\[ T = \text{temperature}, \ K \]
\[ u = \text{velocity}, \ m/s \]
\[ v = \text{velocity}, \ m/s \]
\[ x = \text{coordinate}, \ m \]
\[ y = \text{coordinate}, \ m \]

\[ \alpha = \text{thermal diffusivity}, \ m^2/s \]
\[ \beta = \text{volume thermal expansion coefficient}, \ K^{-1} \]
\[ \delta = \text{film thickness}, \ m \]
\[ \delta_0 = \text{novevaporating film thickness}, \ m \]
\[ \varepsilon = \text{porosity} \]
\[ \eta = \text{coordinate}, \ m \]
\[ \mu = \text{dynamic viscosity}, \ Ns/m^2 \]
\[ \nu = \text{kinematic viscosity}, \ m^2/s \]
\[ \theta = \text{dimensionless temperature} \]
\[ \rho = \text{density}, \ kg/m^3 \]
\[ \sigma = \text{surface tension}, \ N/m \]
\[ \xi = \text{curvature}, \ m^{-1} \]

Subscripts

\[ c = \text{capillary} \]
\[ \text{eff} = \text{effective} \]
\[ f = \text{fluid} \]
\[ H = \text{heated} \]
\[ l = \text{liquid} \]
\[ p = \text{particle} \]
\[ s = \text{solid} \]
\[ \text{sat} = \text{saturation} \]
\[ \text{sp} = \text{particle surface} \]
\[ v = \text{vapor} \]
\[ w = \text{wall} \]

References


