Evaporation/boiling on Thin Capillary Wick (II): Effects of Volumetric Porosity and Mesh Size

Chen Li and G. P. Peterson
Evaporation/Boiling in Thin Capillary Wicks (II)—Effects of Volumetric Porosity and Mesh Size

Chen Li
Graduate Research Assistant
Rensselaer Polytechnic Institute,
Department of Mechanical, Aerospace and Nuclear Engineering
Troy, NY 12180
e-mail: lic4@rpi.edu

G. P. Peterson
Professor and Chancellor
University of Colorado,
Boulder, CO 80309
e-mail: Bud.Peterson@colorado.edu

1 Introduction

Evaporation/boiling in capillary wicking structures has been shown to be an effective heat transfer mechanism in a wide variety of applications, such as heat pipes, loop heat pipes (LHP), capillary pumped loops (CPL), etc. As a result, the parameters that govern these phenomena are of considerable interest. A summary of the recent investigations has been presented in Part I of this two-part investigation [1]. In that review, the range of capillary wick structures previously investigated was summarized, along with the fabrication procedures used, the size of the heat source employed, and the contact condition between the heated wall and the capillary wick [2–12]. The resulting conclusions were that of all the various parameters investigated to date, one of the most critical parameters, the contact condition, was also the one most often neglected. In Part I, the contact condition between the capillary wicking structure and the heated surface was found to be a key factor that affects both the heat transfer efficiency and the CHF. In addition, the effects of capillary wick thickness on the evaporation/boiling in capillary wicking structures made from uniform layers of sintered isotropic copper mesh were also presented. The experimental results indicated that the sintering process developed in Part I could achieve nearly perfect contact at the heated surface/capillary wick interface. It was also demonstrated that the evaporation/boiling heat transfer is nearly independent of the capillary thickness, while the CHF increases with increasing thickness, if all other geometric properties, i.e. volumetric porosity and mesh size, were held constant. In Part I, extremely high heat transfer performance and CHF values were successfully achieved from thin capillary wicks fabricated from sintered isotropic copper mesh. These structures achieved some of the highest heat transfer enhancement and CHF values reported in the literature. In addition, this structure was easy to fabricate and allowed precise control of both the thickness and volumetric porosity.

In addition to the effects of the capillary wick thickness, two fundamental questions must be addressed before the evaporation/boiling mechanism from these structures is fully understood. These are: First, what is and where does the critical meniscus radius, which is controlled by the wire diameter and the mesh number, occur in these types of structures? And second, how does the volumetric porosity affect the evaporation/boiling performance, characteristics and CHF? For thin capillary wicking structures fabricated from sintered isotropic copper mesh, the critical meniscus radius and the effective pore size in the liquid flow direction are controlled by the mesh size (including wire diameter and mesh number) and the compression factor. In [14] the value of \((W+d)/2\) was recommended as the effective pore radius to estimate the capillary pressure for multiple wire-mesh screens, for both sintered and simple contact situations. Because the evaporation/boiling phenomenon is a complicated and dynamic process, the determination of the critical meniscus radius is quite challenging. In the evaporation/boiling process from a capillary wicking structure, the menisci are initially formed at the horizontal mesh openings. As the heat flux increases, the evaporation at the liquid-vapor interface is intensified. In these situations, the liquid meniscus recedes into the wick, reducing the meniscus radius, which results in an increase in the capillary pressure. However, in some cases, the capillary pressure generated through the
meniscus curvature may not be sufficient to drive or pump the required amount of working fluid to the heating area. Visual observations indicate that these menisci are not only formed at the horizontal mesh openings, but may also be formed at other locations in the capillary wick structure, i.e., the vertical mesh openings or the corners formed at the junction of the wire and the heating wall.

In the current investigation, the effects of the critical meniscus radius/the effective pore size and volumetric porosity are investigated systematically by varying the mesh size including the wire diameter and mesh number, and the distance between wire layers, respectively. This approach provides new physical insights into the evaporation/boiling phenomena from the capillary wicking structures. To accomplish this, the optimum sintering process developed in Part I was employed to minimize the contact thermal resistances between the individual layers of copper mesh, as well as between the copper mesh and the heated wall. All of the experimental results presented herein utilized an identical sintering process.

The literature review indicates that heat transport models and regimes for evaporation/boiling from capillary wicking structures have been studied by a number of investigators. Hanlon and Ma [9] studied evaporation on sintered copper particle beds, both analytically and experimentally. This study indicated that there exist two heat transfer models: one for thin liquid film evaporation and one for nucleate boiling; that the thin film evaporation heat transfer on the top surface of the wick is the dominant factor in the enhancement of the evaporating heat transfer, and that nucleate boiling causes a decrease in heat transfer performance. The heat transfer models for heat pipes shown later as Fig. 11 were presented and discussed by Faghri [14]. Four models were presented and discussed: conduction-convection, receding liquid, nucleate boiling, and film boiling. In addition, the dynamics of the liquid-vapor interface and its effects on the evaporation/boiling in capillary wicks were mentioned indirectly. In the following, visual observations and analytical analyses, along with the heat transfer regimes, local bubble dynamics, and liquid vapor interface dynamics are all systematically presented and discussed.

2 Results and Discussion

All of the experiments were conducted using the experimental test facility and test procedures described in Part I [1]. Specifications of the test samples used in the current investigation are listed in Table 1. Typical test results are summarized in this section and are presented in terms of the heat flux including the CHF, wall superheat, and heat transfer coefficient.

2.1 Effects of Mesh Size. To investigate the effects of variations in the mesh size on the evaporation/boiling heat transfer performance and CHF at steady-state condition, three test articles, E145-4, E100-2 and E60-1, with approximately identical thicknesses and volumetric porosities, were evaluated in an atmospheric environment. For comparisons, the pool boiling curve on a plain surface is added in Figs. 1(a) and 1(b).

![Diagram](image-url)

**Fig. 1** (a) Heat flux as a function of superheat \([T_{\text{wall}} - T_{\text{sat}}]\) as a function of mesh size; (b) heat transfer coefficient as a function of heat flux as a function of mesh size.

2.1.1 Effects of Mesh Size on Evaporation/Boiling Heat Transfer Performance. Figure 1(a) presents the heat flux as a function of the superheat, while Fig. 1(b) illustrates the effective heat transfer coefficient. As shown, the evaporation/boiling heat transfer coefficient increases with incremental increases in the input power until a maximum value has been reached. At this point the evaporation/boiling heat transfer coefficient begins to decrease due to partial dry out of the wick structure and the heated surface. This implies that the capillary force is still effective in helping to provide fluid to the heated area even at partial wick dry out, which is consistent with the findings of the experimental investigation in Part I [1]. Furthermore, the heat transfer performance on the 2362 m\(^{-3}\) copper mesh (60 in.\(^{-3}\)), a relatively coarse mesh, reached values as high as 117.3 kW/m\(^2\) K, which is still superior to pool boiling on a plain surface.

With the exception of the initial region and the region following partial dry out, the input heat flux, \(q''\), and the wall super heat, \(T_{\text{w}} - T_{\text{sat}}\), exhibit a strong linear relationship. Figure 1(a) also indicates that the evaporation/boiling heat transfer performance of the capillary wick increases with increasing mesh number or decreases in the mean pore size. While it is clear that both the total surface area and the exposed surface area would increase proportionally with an increase in the mesh number, an increase in the total surface area or an increase in the exposed surface could account for the increase in the heat transfer coefficient with wire diameter. However, in Part I [1] it was illustrated that only the

<table>
<thead>
<tr>
<th>Sample #</th>
<th>Wire diameter (µm)</th>
<th>Porosity</th>
<th>Pore size (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E145-4</td>
<td>56</td>
<td>0.692</td>
<td>119.3</td>
</tr>
<tr>
<td>E145-6c</td>
<td>56</td>
<td>0.56</td>
<td>119.3</td>
</tr>
<tr>
<td>E145-7c</td>
<td>56</td>
<td>0.409</td>
<td>119.3</td>
</tr>
<tr>
<td>E100-2</td>
<td>114</td>
<td>0.632</td>
<td>139.7</td>
</tr>
<tr>
<td>E60-1</td>
<td>191</td>
<td>0.67</td>
<td>232.8</td>
</tr>
<tr>
<td>E145-8c</td>
<td>56</td>
<td>0.698</td>
<td>119.3</td>
</tr>
</tbody>
</table>

*This sample is 0.74 mm thick.*
exposed surface area plays a role in the evaporation/boiling process in these types of capillary wick structures in gravitational fields. When the volumetric porosity and thickness are held constant, increases in the mesh number may result in an enhanced exposed surface area which, in turn, would result in substantially improved performance.

2.1.2 Effects of Mesh Size on Evaporation/Boiling Characteristics. Figure 1(a) indicates that the evaporation/boiling inception superheat is generally dependent on the mesh size and can be reduced through the use of capillary wicking structures when compared with pool boiling on a plain surface. For a given wick thickness and volumetric porosity, the evaporation/boiling inception superheat increases with mesh size increases. One possible reason is due to the pore size reduction. For example, sample E60-1 is made from relatively coarse copper mesh and has evaporation/boiling inception point close to that of pool boiling on a plain surface, i.e., when pore size is big enough, the effect of capillary wicking is deteriorated and the reduction of boiling inception superheat would diminish. The superheat at CHF also demonstrates an increasing relationship with increasing mesh size.

2.1.3 Effects of Mesh Size on the CHF. As discussed in Part I [1], the CHF for any evaporation/boiling system, is determined by the mechanism of liquid supply to, and vapor escape from, the phase change interface. For the evaporation/boiling from thin sintered isotropic mesh surfaces, the bubbles break up or collapse at the free liquid-vapor interface and the liquid-vapor counterflow resistance is reduced without bubble flow through the capillary wick. In addition, the contact points connecting the wick and wall, serve to interrupt the formation of the vapor film and/or reduce the critical hydrodynamic wavelength. Thus the CHF for evaporation/boiling from these porous surfaces is greatly enhanced.

Figure 2 demonstrates the dependence of the CHF on the wire diameter and indicates that increases in the wire diameter would result in an increase in the CHF. This demonstrates that besides the menisci formed between the wires in the horizontal direction [14], menisci are also formed between the wire and the wall, as shown in Fig. 3 [17]. These factors also play a role in the resupply of the liquid. In the evaporation/boiling process in these capillary wicking structures, the capillary pressure is the only pressure source available to pump water to the heated area and the contact angles are assumed to be constant, since identical materials, sintering and cleaning processes (Duraclean™ 1075) were employed to assure identical wetting characteristics. In Fig. 3, α, r_m, and R_w denote the contact angle between the liquid and solid surface, the meniscus radius formed between the wire and heating wall, and the wire diameter, respectively. From Fig. 3, it is clear that wires with smaller diameters also can have a small meniscus radius when the interface is close to the bottom of the wire, but from a fluid mechanics perspective, when a meniscus with a small radius is formed between wires with smaller diameters and the heating wall, the cross-sectional flow area of the liquid is much smaller than that formed between wires with larger diameters. Thus, the flow resistance for smaller wires is much higher than that for larger wires when the meniscus radius is the same, i.e. the capillary pressure generated is the same, but the flow resistance is greater. Generally larger wires can generate higher capillary pressures with a relatively lower flow resistance; therefore the CHF would be higher for wires with larger diameters when the thickness and volumetric porosity of the capillary wicking structures are held constant.

2.2 Effects of Wick Volumetric Porosity. In order to determine how the volumetric porosity of a thin capillary wick structure affects the evaporation/boiling heat transfer performance and the CHF on thin capillary wicking structures, three samples, E145-4, E145-6c, and E145-7c, which, as shown in Table 1, are approximately the same thickness, were fabricated using an identical copper mesh. The required variations were achieved by compressing the layers, thereby changing the distance between layers. From Figs. 4(a) and 4(b), the thickness of the six-layer test article is reduced significantly when compared with the regular six-layer test article. This compression process does not change the horizontal pore size, but does impact the vertical pore size, which is greatly reduced. Results from these three samples are plotted and compared in Figs. 5(a) and 5(b). For comparison, the pool boiling curve on a plain surface has also been added to Figs. 5(a) and 5(b).

2.2.1 Effects of Volumetric Porosity on Evaporation/Boiling Heat Transfer Performance. Figure 5(a) presents the heat flux as a function of the superheat, while Fig. 5(b) illustrates the effective heat transfer coefficient as a function of the heat flux. These two figures present some of the same characteristics as previous cases studied and the resulting curves are consistent with results previously reported by Li et al. [1]. In addition, the lower the volumetric porosity of the wick, the higher the heat transfer performance, even though the improvement is relatively small. The reason for this is thought to be due to the increase in the effective thermal conductivity with decreases in the volumetric porosity, i.e., for a given heat flux, the capillary wick could be utilized efficiently and hence, more cavities could be activated in wicks with higher ef-
fective thermal conductivities. Generally, for porous media with the same structure, the effective thermal conductivity increases with decreases in the volumetric porosity.

It is also apparent that wicks with a lower volumetric porosity would result in higher heat flux values after partial dry out. This is because the pore size in the vertical direction becomes smaller after compression, resulting in a smaller capillary radius and hence, a higher capillary pressure. This explains the existence of an optimum porosity when the wick thickness and mesh size are both held constant.

Figure 5(a) clearly indicates that the evaporation/boiling inception point is greatly reduced when compared with pool boiling on a plain surface and can be shown to be independent of the volumetric porosity for a given mesh size and capillary wick thickness.

2.2.2 Effects of Volumetric Porosity on the CHF. Mathematically there exist three distinct cases for volumetric porosity: a regular capillary wick covered surface, a solid coated surface, and an empty coated surface, each distinguished by the relative porosity of the coating material or wick. This experimental study investigated the effects of porosity, which varied from 0.409 to 0.692 by changing the distance between the wires in the vertical direction. For \( \epsilon = 0 \), the heating area is filled with solid metal and thus no phase change can occur on top of the heated wall without the liquid supply. For this case, the CHF is assumed to be 0 W/cm². However, when the heating area becomes empty and a pool of liquid is formed above the top of the heater, the CHF is similar to the value obtained for pool boiling in [1]. This experimentally measured value in Part I is presented here in Fig. 6.

Figure 6 illustrates that for a given thickness and mesh size, there exists an optimum volumetric porosity that governs the CHF and evaporation/boiling from thin capillary wicks made of sintered isotropic copper mesh. From a fluid mechanics perspective, increases in the volumetric porosity results in a decrease in the flow head loss. If the maximum capillary pressure generated in the wick is dependent only on the wire diameter and horizontal pore size [14], the CHF would increase with increases in the wick volumetric porosity, because of a decrease in the flow resistance.

![Fig. 4](image_url) (a) SEM image of the compact six layer sample; SEM images of sintered isotropic copper mesh with 1509 m⁻¹ (145 in⁻¹), 56 μm (0.0022 in.) wire diameter, and fabricated at sintering temperature of 1030 °C with gas mixture protection (75% N₂ and 25% H₂) for two hours

![Fig. 5](image_url) (a) Heat flux as a function of superheat \( [T_{\text{wall}} - T_{\text{sat}}] \) as a function of volumetric porosity; (b) heat transfer coefficient as a function of heat flux as a function of volumetric porosity

![Fig. 6](image_url) Test data of CHF as a function of volumetric porosity of the sintered isotropic copper mesh
However, as shown in Fig. 6, the CHF decreases after a volumetric porosity of approximately 0.50, which implies an optimum volumetric porosity in the present capillary wick structure. Furthermore, the existence of an optimum volumetric porosity implies that the menisci formed between the wires in the vertical direction also play an important role in the capillary pressure generation. From this analysis and the discussion in Sec. 2.2.3, it is apparent that the minimum meniscus radius, which can occur at the pores between the wires in the horizontal direction or the vertical direction, as well as between the wire and the wall, determines the capillary limit in evaporation/boiling from capillary wick surfaces.

3 Evaporation/Boiling on Thin Capillary Wicks

3.1 Heat Transfer Regime on Thin Capillary Wicks. The results presented in Part I, coupled with the previous discussions, illustrate the characteristics of evaporation/boiling from uniformly sintered copper mesh coated surfaces. It is important to understand what happens in this complex process, in order to optimize the use of these structures in actual applications. Evaporation/boiling phenomenon on uniformly sintered copper mesh surfaces is complicated by the existence of several different regimes and irregular geometric characteristics of the liquid-vapor interface. Based on visual observations of the phase change phenomenon and comparisons of the experimental test data, characteristic $q^*$-\(T_{\text{super}}\) and \(h-a^*q^*\) curves have been proposed, as shown in Figs. 7(a) and 7(b), respectively. These two curves clearly demonstrate the three regimes that exist: convection, nucleate boiling and thin film evaporation. The transition from one regime to another is accompanied by marked changes in the hydrodynamic and thermal states of the system. When the temperature or heat flux is below a certain value (i.e., the onset of strong nucleate boiling), heat is transferred by convection (AB), which is driven by the temperature difference or a small number of relatively large bubbles. This, then transitions to a region where the nucleate boiling (CD) is the dominant heat transfer mechanism when the temperature or heat flux exceeds the saturation temperature by a value that varies with the capillary wick thickness, pore size and surface characteristics. In Figs. 7(a) and 7(b), the transition region, BC, does not actually exist. The wall temperature effectively jumps from B to C when it exceeds the evaporation/boiling inception point. In the nucleate boiling regime, numerous bubbles are generated and grow from the nucleation sites on the heated surfaces, i.e. heater wall and wire surfaces, and finally break up or collapse at the free liquid-vapor interface. Increases in the wall temperature or heat flux are accompanied by large increases in the bubble population. These processes, as well as the mutual interaction among bubbles and nucleation sites, result in significant enhancement in the heat transfer performance. It is worth noting that the capillary evaporation also plays an important role in the improvement of performance in the nucleate boiling regime. If the temperature or heat flux is further increased, the meniscus recedes further into the porous material and thin liquid film evaporation (DE) begins. In this region, the liquid film thickness becomes very thin due to a combination of the surface tension and disjoining pressure, making it very hard for bubbles to form and grow from the heated surfaces, and hence, the liquid just evaporates directly from the heated surfaces until they dry out completely, which results in the best performance among these three regimes.

3.2 Onset Point of Nucleate Boiling From a Horizontal Heated Wall. As was the case for pool boiling, in order to reach nucleate boiling, the surface temperature must exceed the saturation temperature by several degrees. Based on the previous discussions, this onset of nucleate boiling from sintered copper mesh surfaces varies with the thickness and mesh size, and is nearly independent of the volumetric porosity. Figure 8 presents the superheat of the heated wall as a function of the heat flux applied, as in the cases of E145-4, 6, and 8. When the wall temperature is lower than the onset of nucleate boiling, it increases with heat flux; however, once it exceeds that critical value, nucleate boiling begins and the wall temperature drops sharply due to the high heat transfer capability of nucleate boiling. Figure 8 demonstrates the transition from convection heat transfer to nucleate boiling heat transfer that occurs on sintered copper mesh surfaces, which can be easily identified on either a typical $q^*$-\(T_{\text{super}}\) or \(h-a^*q^*\) curve, i.e., point B to C. This figure also illustrates that nucleate boiling
occurs in the capillary wicks and significantly improves the heat transfer performance, which is different from the findings of Hanlon and Ma [9].

For some sections, surface AB can be easily overlooked because the value of the onset of nucleate boiling is small and is not easily predicted. In addition, in some cases the step increase of the heat flux used may be larger than the value at which the onset of nucleate boiling occurs, making it impossible to distinguish one region from another.

The reason that the superheat necessary to activate nucleate boiling varies with the capillary wick thickness and mesh size, remains somewhat unclear. The current experimental investigation only illustrates that these phenomena do, in fact, occur for evaporation/boiling on the capillary wick. The available data indicate that the onset point of nucleate boiling increases with decreases in the mesh size and thickness, but this appears to have no direct relationship with the capillary wick volumetric porosity.

3.3 Local Bubble Generation from a Horizontal Heated Wall. From classical nucleate boiling theory, it is clear that the characteristics of the actual surface are very complicated and that the many nucleation cavities with various shapes and sizes all affect the boiling characteristics. Bubbles are typically generated from the largest cavities first, since this process requires less energy, with the smaller cavities subsequently activated as the heat flux increases. Although the mechanisms of bubble generation from sintered copper mesh surfaces agree with the classical theory, there appears to be some subtle influences due to the presence of the capillary wicking structure that should be noted.

Inside the sintered copper mesh, there are a variety of different two-dimensional menisci formed at the various corners, due to the surface tension. The liquid-phase pressure that exists prior to thin liquid film evaporation in this kind of porous media is determined by the capillary forces generated at the different menisci, and the thin liquid film thickness is determined by the disjoining pressure. Since copper, when treated as described herein, wets well with water, the pressure in the cavity is given by Eqs. (1) and (2):

\[ P_v = P_l + 2\sigma r \]
\[ P_l = P_{am} - 2\sigma r m \]

where \( r \) and \( r_m \) are the radius of the curvature of the bubble interface and meniscus, respectively.

Equations (1) and (2) indicate that both the liquid pressure and the liquid thickness would decrease as the size of the meniscus radius decreases. The local evaporation/boiling process on the uniformly sintered copper mesh coated surface has five typical configurations, as illustrated in Fig. 9: the initial state (a), convection (b), nucleate boiling (c), thin liquid film evaporation (d), and dry out (e). In this process, water is supplied only via the capillary forces, which can be deduced from the surface tension and the geometric parameters of the meniscus. At a lower heat flux, smaller amounts of larger bubbles are generated and grow within the heated liquid. In addition, a thin liquid film is formed on the upper wire surfaces. In most cases, boiling and evaporation occur simultaneously and the bubble movement causes liquid convection. If the heat flux increases, the meniscus radius must decrease in order to provide the capillary force needed to pump water into the heated area. Thus, the bubble population and bubble generation frequency, as well as the thin liquid film area all increase. As the heat flux continues to increase, the meniscus radius recedes towards the corner along the solid heated surfaces, forming a thin liquid film, which, due to the disjoining pressure makes it difficult for a bubble to form and grow. At this point, liquid film evaporation becomes the dominant mode. If the heat flux continues to increase, the central portion of the mesh will dry out, due to a loss of liquid supply.

3.4 Local Bubble Dynamics From the Heated Surface. The most noticeable difference between classical nucleate boiling in pool boiling and the evaporation/boiling processes from a thin, i.e., less than 1 mm, sintered copper mesh, is that in the latter, the bubbles never grow large enough to release, hence, there is no bubble departure from the horizontal heated wall. Generally, the bubble departure diameter is estimated by

\[ D_d = \frac{2\sigma}{g(p_l - p_v)} \]

The Bond number depends on the bubble contact angle and stochastic processes; i.e., in actuality the observed bubble departure diameter has a statistical distribution around some mean. Numerous investigations over the past 70 years have been carried out to estimate the Bond number. This departure diameter model was first proposed by Fritz [13] and is shown below as Eq. (4),

\[ D_d = \frac{2\sigma}{g(p_l - p_v)} \]

Two more models are selected to compare with the model of Fritz. One model was developed by Cole and Rohsenow [13] and is shown as Eq. (5) below

\[ \left( \frac{g(p_l - p_v)D^2}{\sigma} \right)^{1/2} = C\frac{\rho C_p T_{sat}}{p_l h_b} \]

where, \( C_f = 0.0148 \) for hydrogen bubbles and water vapor, in water, and the contact angle, \( \phi \), is in degrees. Here, \( C_f = 1.5 \times 10^{-4} \) for water. The second model presented here was proposed by Jensen and Memmel [15].

\[ \frac{\sigma}{g(p_l - p_v)D^2} = 0.19(1.8 + 10^3 K)^{1/2} \]

where, in Eq. (6) \( K = (Ja/P_r)\left[\left(\frac{g\rho(p_l - p_v)/p_l}{\mu_l^2}\right)\sigma/g(p_l - p_v)\right]^{3/2} \) and the Jacob number, \( Ja \), is defined by \( T_w - T_{sat} = C_p p_l/(p_l h_b) \). For copper-water combinations, the contact angle \( \sigma \) is 55°, which was experimentally determined by Wu and Peterson [16]. The departure diameters estimated using Eqs. (4)–(6), for this situation are all greater than 1 mm. These estimations are listed in Table 2, where the minimum value given is even greater than the thickest capillary wick presented herein, 0.82 mm.

The typical localized bubble generation and growth from the heated surfaces, i.e., the top wall of the heater and wire surfaces, are illustrated in Fig. 10. When a relatively low heat flux is applied, only large cavities are activated, so large bubbles are generated and grow from the heated surface. These large bubbles grow, but never depart from the heated wall. Instead, they collapse at the free liquid-vapor interface, due to local condensation and pressure differences before they can grow large enough to gain

![Diagram of local bubble dynamics](https://example.com/diagram.png)
sufficient buoyancy energy to detach from the wall surface. A higher heat flux activates smaller cavities, and causes an increase in the number of bubbles and a decrease in the average bubble size. The mutual interaction among bubbles results in higher heat transfer performance. With a further increase in the heat flux, the mean bubble size continues to decrease and the bubble generation frequency increases dramatically. In addition, for thin liquid films with high pressures (induced by the disjoining forces), it is hard for the bubbles to grow, and as a result, the liquid evaporates directly from the thin liquid surface. In other words, when bubbles cannot depart from the heated wall, there is a lower flow resistance between the bubble and the liquid, as well as a higher bubble generation frequency and the resulting heat transfer performance will be significantly enhanced.

### 3.5 Liquid Vapor Interface Dynamic and Drying-out Process on Thin Capillary Wicks

The behavior of the liquid-vapor interface is one of the key characteristics that govern the evaporation/boiling on capillary wicking structures. In [14], the heat transfer from capillary wicking structure surfaces was discussed and four models were presented as shown in Fig. 11. In addition, the dynamics of the liquid-vapor interface and its effects on the evaporation/boiling in capillary wicks were discussed indirectly. The principal points in [14] can be summarized as follows:

At low heat fluxes, combined conduction and convection is the main heat transfer mode and, if the wick is horizontal, the liquid-vapor interface is parallel to the heated surface. As the heat flux increases, the evaporation at the liquid surface is intensified, and the liquid-vapor interface recedes uniformly, generating a greater capillary force. In the receding liquid model, there is no boiling and conduction still governs the heat transfer across the liquid surface. The capillary limitation may be encountered in this phase. When the heat flux is further increased, nucleate boiling may take place within the wick. Bubbles grow at the heated wall and escape to the liquid surface, where they burst rapidly. Nucleate boiling may represent a heat transfer limit in the nucleate boiling model, since, as the heat flux increases to a specific value, large quantities of bubbles are generated at the heated wall. These bubbles coalesce and form a vapor layer adjacent to the heated wall. This vapor layer is the principal cause of the heat transfer limit often encountered. In [14], the heat transfer limit for evaporation in capillary wicks was described as being similar to that occurring in pool boiling heat transfer. The thickness factor was not considered in [14] when evaporation/boiling behavior was discussed.

In the present work, however, the wick thickness is confined to thicknesses of less than 1 mm, which is smaller than the bubble departure diameter, \( D_b \). As discussed previously, at this thickness, a much higher heat transfer performance and higher CHF were achieved. The control volume shown in Fig. 12 is utilized to analyze the liquid vapor interface behavior in evaporation/boiling in capillary wicks. At saturation conditions, the mass and energy conservation equations can be defined as:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0
\]

\[
\rho \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p + \nabla \cdot \mathbf{S}
\]

\[
\rho \mathbf{u} \cdot \nabla \phi = -\nabla \cdot \mathbf{j}
\]

where \( \mathbf{u} \) is the velocity vector, \( p \) is the pressure, \( \mathbf{S} \) is the stress tensor, \( \mathbf{j} \) is the current density, \( \phi \) is the scalar potential, and \( \rho \) is the density.

Solving Eqs. (7) and (8), the liquid-vapor interface configuration at steady state can be determined by

\[
y = y_0 - \frac{q^*}{G_f h_f}
\]

where \( G_f \) is the liquid flow rate and \( y_0 \) is the initial position of the liquid, i.e., the wick thickness. If the heat flux, \( q^* \), is ideally uniform, the liquid-vapor interface would be a straight line with a slope of \( -q^*/G_f h_f \). However, because the thermal insulation is not perfect, the actual heat flux profile along the heated wall is parabolic, as shown in Fig. 13. The interface would be irregular due to the meniscus, but the overall shape would be curved, thinner in the center, and thicker at the edges. The overall liquid-vapor interface dynamics has four typical modes, which are presented in Fig. 14. Mode A is the initial state, in which the liquid-vapor interface is parallel to the heated wall. As the heat flux is increased, vaporization begins and the meniscus begins receding, which generates a greater capillary force, increasing the amount of liquid supplied to the heated area. The receding of the menisci is

**Table 2** A comparison of the bubble departure diameter

<table>
<thead>
<tr>
<th>Model Description</th>
<th>( D_b) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fritz's model [13]</td>
<td>2.884</td>
</tr>
<tr>
<td>Cole and Rohsensow’s model [13]</td>
<td>2.426</td>
</tr>
</tbody>
</table>

**Fig. 11** Models of heat transfer and vapor formation in wicks [14]

**Fig. 12** Control volume of the liquid-vapor interface
not uniform because of the constant and continuous liquid mass reduction along the heated wall, which results from more vaporization of the liquid as the center of the heated area is approached. During this process, menisci are formed between the wires in the horizontal and vertical directions, and even between the wire and the heated wall after the initial dry out occurs.

This analysis is consistent with the experimental results presented in Sec. 2.2.2. As the heat flux is further increased, the curvature of the liquid-vapor interface becomes smaller and the initial dry out point first appears at the center of the heated section of the wall. For evaporation/boiling on a capillary wick structure, the appearance of this initial dry out does not represent the CHF. The capillary wick will still function under higher heat fluxes until sufficient liquid can no longer be provided. As shown in Fig. 15, the dry out regions are clearly evident for each of the different heat flux levels tested. Here, the white rings represent the dry out region for each heat flux level tested. At each heat flux, steady-state operation is achieved and a specified region of the wick remains dry. These rings represent the capillary limit for the evaporation/boiling in the capillary wick for a specified heat flux.

4 Conclusions

The minimum meniscus radius is ultimately the determining factor in CHF for the evaporation/boiling from capillary wicking structures, and the results presented here illustrate that the menisci are formed not just between the wires in the horizontal direction, but also between the wires in the vertical direction, and between the wire and the wall. An optimal volumetric porosity has been shown to exist for CHF in the capillary wicks, depending upon the thickness and mesh size. When the wire diameter and wick thickness are held constant, the experimental data shows that the heat transfer performance increases with decreases in the volumetric porosity of the wick, but the improvement is not significant. The reason for this small improvement is thought to be due to the increase in the effective thermal conductivity with decreases in the porosity, i.e., for a given heat flux, more cavities are activated in wicks with higher effective thermal conductivities.

Some key characteristics of the evaporation/boiling from capillary wicks are detailed in the discussion. When the capillary wick thickness is less than 1 mm, the bubbles break up or collapse at
the free liquid-vapor interface, due to local condensation and the pressure difference, instead of departing from the heated wall. This feature could enhance the heat transfer capability and CHF dramatically, due to reductions in the counterflow liquid and vapor resistance and the enhancement in the bubble generation frequency. Analysis of the liquid-vapor interface dynamics shows that the liquid thickness in the wick, decreases nonlinearly along the heated wall, and the first dry out point occurs in the center of the heated area. This analysis also demonstrates that menisci are formed not just between wires in the horizontal direction, but also between wires in the vertical direction and between wires and the heated wall. These new theoretical findings are consistent with both the current and previously presented experimental results.

Both the experimental test data and the visual observations support the concept that the heat transfer limit for evaporation/boiling in capillary wicks is the result of a capillary limit, and that the characteristic receding of the meniscus into the wick structure is not uniform, but rather decreases as the center of the heated area is approached. The different heat transfer regimes for evaporation/boiling in these different capillary wick structures have been proposed and discussed based on the current and previous experimental investigations as well as the visual observations of the phase change phenomena and the heat flux—superheat relationship.

The test data obtained herein, illustrate that the evaporation/boiling inception is strongly dependent on the pore size in the horizontal direction and the wick thickness, however, it is weakly dependent on volumetric porosity. Hysteresis was also observed from evaporation/boiling on the capillary wick in this experimental investigation.

Acknowledgment

The authors would like to acknowledge the support of the National Science Foundation under award CTS-0312848. Fruitful discussions offered by Dr. Ji Li and Mr. Hong Li are greatly appreciated.

Nomenclature

\[ A-E = \text{points number in Fig. 3} \]
\[ B_0 = \text{Bond number} \]
\[ C_d = \text{Constant in Eq. (3)} \]
\[ C_p = \text{Specific heat at constant pressure [J/kg.K]} \]
\[ D_p = \text{Bubble departure diameter (mm)} \]
\[ G = \text{Flow rate (kg/m²s)} \]
\[ g = \text{Gravitational acceleration (m/s²)} \]
\[ h_{tg} = \text{Latent heat (kJ/kg)} \]
\[ J_a = \text{Jacob number} \]
\[ K = \text{Thermal conductivity (W/MK)} \]
\[ K_i = \text{Parameter in Eq. (6)} \]
\[ m = \text{Mass flow rate (kg/s)} \]
\[ M = \text{Mesh number (m⁻¹)} \]
\[ O = \text{Center of circle} \]
\[ P = \text{Pressure (Pa)} \]
\[ Pr = \text{Prandtl number} \]
\[ q'' = \text{Heat flux (W/cm²)} \]
\[ R = \text{Bubble radius (mm)} \]
\[ r_w = \text{Meniscus radius (mm)} \]
\[ R = \text{Wire radius (mm)} \]
\[ t = \text{Distance or thickness (mm)} \]
\[ t_w = \text{Time required for bubble growth (s)} \]
\[ t_d = \text{Time required for bubble departure (s)} \]
\[ T = \text{Temperature (K)} \]
\[ W = \text{Width of mesh opening} \]
\[ X = \text{X coordinate} \]
\[ Y = \text{X coordinate} \]

Greek Symbols

\[ \alpha = \text{Contact angle (degree)} \]
\[ \beta = \text{Angle (degree)} \]
\[ \varepsilon = \text{Volumetric porosity} \]
\[ \sigma = \text{Surface tension (N/m)} \]
\[ \rho = \text{Density (kg/m³)} \]

Subscripts

\[ \text{in} = \text{Flow in direction} \]
\[ l = \text{Parameter related to the liquid phase} \]
\[ \text{out} = \text{Flow out direction} \]
\[ v = \text{Parameter related to the vapor phase} \]
\[ w = \text{Parameter related with the wall} \]
\[ \text{sat} = \text{Parameter related with the saturation condition} \]
\[ \text{eff} = \text{Effective parameter} \]

References