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ABSTRACT

Micromembrane-enhanced evaporating surfaces were developed to enhance capillary evaporation heat transfer coefficient (*HTC*) and critical heat flux (*CHF*). Micromembranes made of sintered single-layer copper mesh screen were diffusion bonded on microchannels to effectively promote capillary pressure and reduce flow resistance. Compared with mono-porous evaporating surfaces such as microchannels and copper woven mesh laminates in the same thickness under the similar working conditions, *CHF* was substantially increased by 83% and 198%, respectively, because of the separation of the capillary pressure generation and fluid transport process that was enabled by the micromembrane. The major features such as “M”-shaped capillary evaporation heat transfer curves and the associated heat transfer regions were identified. Oscillating flows induced by the bubble growth and collapse as well as the capillary flows induced by the receding menisci were observed and believed to play imperative roles in enhancing the heat transfer by inducing advective flows and improving evaporation and nucleate boiling.

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1. Introduction

Capillary evaporation [1] is one of the most efficient heat transfer modes and has been widely used in heat exchangers [2] and heat pipes [3–5]. Evaporators with high heat transfer coefficient (*HTC*) and critical heat flux (*CHF*) are highly desirable for compact heat exchangers for high heat flux applications [6,7]. Most of porous coatings used in enhancing capillary evaporation are usually mono-porous structures [8]. For example, sintered particles and powders were developed to substantially enhance thin film evaporation *HTC* [8,9]. The effects of porosity, wick thickness and other factors on the optimal design of the wicking structures were also examined [8,9]. Copper woven mesh laminates [10–13] were extensively studied to enhance the capillary evaporation *HTC* due to the augmented surface areas and increased capillary forces. However, the flow resistances in these microscale mono-porous structures remain high, resulting in low *CHF* due to the liquid supply crisis. Micro-grooves [14,15] or channels [16] were superior for liquid supply because of the attributed low flow resistance, but the capillary forces induced by the disjoining pressure differences in grooves [15] were still too low to reach high *CHF*. This brief review shows that both the microscale mono-porous structures (such as sintered meshes or particles/powders) and micro-grooves or channels cannot meet the needs of high heat flux applications. To solve this dilemma, various types of bi-porous surfaces were proposed and developed [5,17–21]. Semenic et al. [18,20] found that bipor-

ous surfaces of sintered powders performed better than the mono-porous copper wicks because the working fluid can be supplied to hot spots through micropores inside the clusters even though the voids were filled with vapor. Cao et al. [22] reported that when a mono-dispersed wick was replaced by a bi-dispersed wick with the same small pore diameter, both *HTC* and *CHF* were increased significantly. Cai et al. [6] studied the heat transfer performance on the carbon nanotube (*CNT*) based bi-porous structures, which consisted of *CNT* array separated by microchannels. The nanoscale pores in the *CNT* bi-porous structure provided ultrahigh capillary pressure and augmented surface areas, which significantly reduced the menisci radii and increased thin-film evaporation area and evaporation efficiency. C̆oso et al. [23] examined a type of bi-porous media consisting of microscale pin fins periodically separated by microchannels to simultaneously increase the heat dissipation capacity as well as the *HTC* of the evaporator wick. Some of the bi-porous wicks have also been integrated in heat pipes [19,24–26] to decrease the thermal resistance and increase working heat fluxes. Heat pipe performance was found to be greatly enhanced by applying modulated wick because of enhanced axial capillary liquid flows and additional evaporation surface area resulting from the cross-sectional area [5]. In these reported bi-porous structures, the main fluid passages were still through the micro or nanoscale mono-porous structures (such as microscale powders or *CNTs*). As a result, the overall liquid flow resistances still remain high.

On the other hand, the oscillating flow significantly increases *HTC* and *CHF* in closed mini/micro-channels as can be found in oscillating heat pipes [16,27]. However, the oscillating capillary

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Nomenclature

A	area of cross section, m^2
C_f	compress factor
d	diameter of copper mesh, m
D_h	hydrodynamic diameter, m
D_p	equivalent spherical diameter of porous media, m
f	friction factor
h	evaporation heat transfer coefficient, $W/(m^2 K)$
k	thermal conductivity of copper, $W/(m K)$
L	the length along the macroscopic, m
L_0	pressure gradient in porous media microchannel length, m
M	mesh number
Re	Reynolds number
S_v	surface area per unit volume of solid phase, m^2
T_5	temperature of the thermocouple 5, $^{\circ}C$
q''	input heat flux, W/cm^2
Q	flow rate, m^3/s

v_s	superficial velocity, m/s
Δp	pressure drop, kp_a
ΔT	temperature difference between two thermocouples, $^{\circ}C$
Δx	distance between two thermocouples, m
$\Delta x'$	distance from T_{C5} to the evaporating surface, m

Greek symbols

β, γ	constant number
μ	fluid viscosity, $kg/(s m)$
ε	porosity
ρ	fluid density, kg/m^3

Subscripts

w	wall
sat	saturated

evaporation in unconfined or open microchannels was not reported.

The objective of this study is to develop a new type of micromembrane-enhanced evaporating surfaces that are capable of both generating high capillary pressure and managing flow resistances. The effects on capillary evaporation were systematically examined. These effects include the separation of liquid supply and capillarity generation as well as the induced oscillating flows in unconfined micromembrane-enhanced evaporating surfaces.

2. Design of micromembrane-enhanced capillary evaporating surfaces

During the capillary evaporation, the counter interactions of flow resistance and capillary force determine the overall liquid supply and thus, the CHF. Fine copper woven meshes with microscale pores can generate high capillary pressure, but the associated flow resistance through the in-plane direction was significantly high. Microgrooves [14,15] or channels [16] were superior for liquid supply because of the low flow resistance, but with limited capillarity [15]. The combination of the advantages of single layer meshes and microchannels could lead to a new type of capillary evaporating surfaces with high capillary pressure and low flow resistance, which would consequently result in much higher CHF than each individual (Fig. 1(a)). The membrane with microscale pores was developed to generate high capillary pressure and augment heat transfer area. Smooth microchannels were designed as the primary fluid passages to reduce flow resistance. Moreover, the micromembranes would greatly enhance thin film evaporation because of the augmented area and high resistance to surface flooding using capillarity.

A theoretical model was developed to verify the design. Due to the unavailability of two-phase models for single layer mesh screens, the single-phase flow resistances through microchannels and single layer mesh (in-plane direction) were estimated. A modified Ergun equation for the porous media [28] was employed in this study to estimate the flow resistance through sintered woven meshes

$$\frac{\Delta p}{L} = \beta \frac{\mu(1-\varepsilon)^2 v_s}{D_p^2 \varepsilon^3} + \gamma \frac{1-\varepsilon}{\varepsilon^3} \frac{\rho v_s^2}{D_p} \quad (1)$$

where, Δp is pressure drop; L is the length along the macroscopic pressure gradient in porous media; v_s is the average velocity estimated from $v_s = Q/A$ (Q is flow rate through a cross-sectional area

A); μ is the absolute viscosity of fluids; ε is the volumetric porosity; D_p is the equivalent spherical diameter of porous media; and ρ is the fluid density. Here, the β and γ vary with different porous media [13] and were modified for the one layer woven screen meshes as $\beta = 15940$ and $\gamma = 12$ according to this experimental study using the method in Ref. [11]. The equivalent diameter of particles for a mesh is defined as, $D_p = 6/S_v$ [13], where, S_v is defined as the surface

Fig. 1. (a) Design of micromembrane-enhanced evaporating surfaces. (b) Comparisons of flow resistances between meshes, microchannels and the micromembrane-enhanced evaporating surfaces.

Fig. 2. The micromembrane-enhanced evaporating surfaces developed in this study. (a) Three dimensional (3D) Micro-XCT image of the micromembrane-enhanced evaporating surface. (b) Top view of the micromembrane-enhanced evaporating surface. (c) Sharp corners formed by individual wires. (d) Sintered four-layer copper woven mesh laminates. (e) Single-layer mesh. (f) 3D Micro-XCT image of microchannels.

area per unit volume of solid phase. The volumetric porosity of the mesh [29] is given by

$$\varepsilon = 1 - \frac{\pi d M \sqrt{1 + M^2 d^2}}{4 C_f} \quad (2)$$

where, d is the diameter of copper wires; M is mesh number; C_f is compression factor and equal to 0.9 for two-layer meshes.

The single-phase flow resistance through a smooth, rectangular channel in the laminar flow regime can be estimated by Darcy–Weisbach equation [30]:

$$\Delta p = \frac{1}{2} \rho v^2 \frac{L_0}{D_h} f, \text{ where } f \cdot Re = 57 \quad (3)$$

where, Δp is the pressure drop through the channel; v is fluid velocity; L_0 is channel length; D_h is hydrodynamic diameter; f is the Darcy friction factor and Re is Reynolds number.

As shown in Fig. 1(b), the magnitude of flow resistance in the single layer mesh was more than four orders higher than that through the microchannels. Flow resistance on the micromembrane-enhanced evaporating surfaces was estimated to be only a small fraction of that in the single layer meshes at the same Reynolds number and slightly higher than the in microchannels as shown in Fig. 1(b).

Equally important, capillary force generated from the micromembrane can be significantly increased and was estimated to be approximately two times higher than that generated on the microchannels by comparing the minimum meniscus radius [31] when

the working fluid and substrate materials are identical. This suggests that the membrane-enhanced evaporating surfaces developed in this study could achieve high CHF with significantly enhanced HTC by increasing the capillary forces, augmenting the surface areas and reducing the flow resistance.

3. Experimental apparatus and data reduction

Fine sintered copper woven meshes were employed as the primary evaporating membranes because of their superior thermal conductivity, high permeability and large surface areas [29,32,33]. Copper woven meshes with mesh number of 1509 m^{-1} (145 inch^{-1}) and wire diameter of $56 \mu\text{m}$ (made by Belleville Wire Cloth, as shown in Fig. 2(a)–(c)) were attached on the microchannels by a diffusion bonding technique [32,34] to minimize the contact thermal resistance [29]. Samples as shown in Fig. 2(a) and (b) were sintered in a high temperature furnace at 1000°C in a hydrogen (H_2) atmosphere. Four-layer mesh laminates were made by sintering four layers of meshes with a diffusion bonding technique (Fig. 2(d)). In addition, we also studied the performances of single-layer mesh (Fig. 2(e)) and microchannels (Fig. 2(f)) as baselines in this study.

3.1. Experimental apparatus

Experimental study was conducted in a closed system. The $10 \times 15 \times 15 \text{ cm}^3$ test chamber was made from aluminum as shown in Fig. 3. One side of the chamber was covered by a piece

Fig. 3. Experimental setup for the capillary evaporation in the vertical direction. Four cartridge heaters are shown as 1, 2, 3 and 4. Five thermocouples (TC) are shown as TC1 to TC5.

Fig. 4. Schematic of the test sample assembly and the real sample images. (a) 1. Cartridge heater, 2. Copper heating block, 3. 1 cm × 1 cm square copper block with holes for thermocouple (TC block) 4. K type thermocouples, 5. Screw, 6. G-7 fiberglass, 7. High temperature RTV silicone, 8. Micromembrane-enhanced evaporating surface, (b) 9. Photo of the micromembrane-enhanced evaporating surface, 10. Copper TC block, (c) SEM image of the micromembrane-enhanced evaporating surface. (d) Thermocouple arrangement and parameters for data reduction.

of quartz glass as a visualization window. The other side was designed for sample installation. External lighting was provided for visualization studies. Four cartridge heaters were mounted at the four corners of the aluminum chamber to assure saturated working conditions (Fig. 3). A proportional-integral-derivative (PID) temperature controller was used to accurately control the water temperature between 99.9 ± 0.2 °C and 100.1 ± 0.2 °C. Highly purified water was degassed through boiling for more than two hours at approximately 100.0 ± 0.2 °C before tests. A built-in compact condenser was designed to keep a constant water level by timely recycling the condensate. The vapor pressure and temperature inside the chamber were monitored by a pressure gauge and two T-type thermocouples, respectively (Fig. 3). All tests were conducted with samples positioned in the vertical direction and with an approximately 15 mm distance from the center of heating area to the water level (Fig. 3), which aimed to minimize the impacts of pool boiling.

The visualization system consists of a high-speed camera (Phantom V 7.3), an infinity K2/SC long distance micro single port main body (VRI-INFINITY-K2SC), a CF-2 objective (VRI-MICRO-990214), and a NIKON T2 adapter (VRI-MICRO-770568). The fluid flow and bubble dynamics were captured by the high speed camera. The videos were taken at 3000 frames per second (fps) and replayed at 45 fps to show the whole oscillating cycles. The visualization setup was shown in Fig. 3.

As shown in Fig. 4(a), a test sample assembly consists of two component blocks: a heating block and a thermocouple (TC) block

with evaporating surfaces. The 25.4 mm (1 inch) squared copper heating block was made from copper Alloy 101 with four heater ports in one end. Heat was provided by four 50.8 mm (2 inches) long and 6.35 mm (1/4 inch) in the diameter cartridge heaters (250 W) through an alternative current (AC) power supply. The TC block was attached on the other end of the heating block. Thermal grease was applied to reduce the contact thermal resistance between the heating and TC blocks, which was then insulated by G-7 Fiberglass to ensure one dimensional (1D) heat flux input. Micromembrane-enhanced structure was sintered on the TC block (Fig. 4(b) and (c)) to minimize contact thermal resistance. Five K type thermocouples with 0.51 mm (0.02 inch) junctions in diameter were used to measure the axial temperature distribution, which was then used to estimate the linearity of temperature profile (Fig. 3(a)) and hence the approximately 1D input heat flux. High temperature RTV silicone[®] was used to achieve mechanical sealing. Only the evaporating surfaces were exposed to the saturated vapor. The heating elements were well insulated by high temperature Nelson Firestop Ceramic Fibers to applying high heat fluxes.

3.2. Data reduction

Data were reduced and categorized into the parameters from Eqs. (4)–(6)

$$q'' = k \frac{\Delta T}{\Delta x} \quad (4)$$

$$T_w = T_5 - q'' \frac{\Delta x'}{k} \quad (5)$$

$$h = \frac{q''}{T_w - T_{sat}} \quad (6)$$

where, q'' is the input heat flux, k is the thermal conductivity of copper; ΔT is the temperature difference between the two thermocouples; Δx is the distance between two thermocouples as shown in Fig. 4(d); T_w is the estimated temperature of the surface; T_5 is the temperature of the thermocouple TC #5; $\Delta x'$ is the distance from TC #5 to the evaporating surface; T_{sat} is the saturated water temperature; and h is the evaporation HTC.

Uncertainties of the temperature measurements and the length are ± 0.5 K for K type thermocouples (± 0.2 K for T type thermocouples) and 0.01 mm, respectively. Uncertainty propagations with 95% confidence level for the computed results in most of the ranges were estimated by Kline and McClintock method [35]. The uncertainties of heat flux, HTC, and superheat were ± 3.2 W/cm², ± 1.2 W/(cm² K), and ± 0.8 °C, respectively. Heat loss due to the fin effects was estimated to be less than 4% (Fig. 4(a)).

4. Results and discussion

In this study, four types of evaporating surfaces with identical heating areas (1×1 cm²) were experimentally investigated. These include copper microchannels (channel height and width: 250 μm, channel wall width: 250 μm), single layer copper mesh screen (thickness: 80 μm), four-layer sintered copper mesh screens (thickness: 320 μm) and the micromembrane-enhanced evaporating surfaces (total thickness of sintered microchannels and mesh: 320 μm). Dimensions of all evaporating surfaces are specified in Table 1.

4.1. Capillary evaporation curves

Fig. 5(a) shows the HTC-heat flux curves for all the tested samples. Four-layer mesh screens, which have an equal thickness with the micromembrane-enhanced evaporating surfaces, showed

Table 1
Sample specifics.

Samples	Parameters		
1 layer mesh	Thickness: 0.08 mm	Wire diameter: 0.56 mm	Porosity: 0.737
4 layer meshes	Thickness: 0.32 mm	Wire diameter: 0.56 mm	Porosity: 0.693
Microchannel	Height: 0.25 mm	Width: 0.25 mm	Pillar width: 0.25 mm
Micromembrane-enhanced evaporating surfaces	Thickness: 0.32 mm	Width: 10 mm	Length: 60 mm

higher HTC in the operating heat flux range. HTC on four-layer mesh screens reached a peak value of $23.1 \text{ W}/(\text{cm}^2 \text{ K})$ at the heat flux of $17.0 \text{ W}/\text{cm}^2$ and then kept dropping. Such a high HTC should result from a combination of the enhanced thin film evaporation and nucleate boiling on the large surface areas. However, the high HTC on four-layer mesh screens could not sustain and decreased quickly with the increasing heat flux as shown in Fig. 5(a). This was primarily caused by the increasing drying areas due to the high flow resistance. For the same reason, HTC on single-layer copper mesh dropped even more sharply because the single-layer mesh stored less water initially and had a significantly higher flow

resistance than four-layer meshes. Microchannels reached a CHF of $83.1 \text{ W}/\text{cm}^2$, but the capillary forces generated in microchannels could not maintain the liquid supply for high heat flux conditions.

As shown in Fig. 5(a) and (b), two micromembrane-enhanced evaporating surfaces show a good repeatability on two individual samples. The capillary evaporation HTC increased with increasing input heat flux in the low heat flux region and decreased after reaching the first peak value. Then HTC started to increase after approaching the valley and reached the second peak value (Fig. 5(a)). Finally HTC decreased a little bit due to the partial dryout. The whole curve exhibits an “M” shape. It shall be noted that the first peak HTC , $14.1 \text{ W}/(\text{cm}^2 \text{ K})$, on the micromembrane-enhanced evaporating surfaces was observed at a heat flux of $11.2 \text{ W}/\text{cm}^2$. However, HTC decreased gradually to $5.9 \text{ W}/(\text{cm}^2 \text{ K})$ and then started to increase again with the increasing heat flux between the heat fluxes of 11.2 and $57.2 \text{ W}/\text{cm}^2$. In the high heat flux region, the peak HTC , $9.1 \text{ W}/(\text{cm}^2 \text{ K})$, was achieved at a heat flux of $140.4 \text{ W}/\text{cm}^2$ (Fig. 5(a)) with a superheat of 16.0°C (Fig. 5(b)). The overall HTC curves on the micromembrane-enhanced evaporating surfaces showed two peaks, which were significantly different from reported results and evaporating curves of the other three interfaces.

The HTC on the micromembrane-enhanced evaporating surfaces was superior to the microchannels and single-layer mesh when input heat fluxes were less than $37 \text{ W}/\text{cm}^2$ (Fig. 5(a)). The primary mechanism behind the enhanced HTC on the micromembrane-enhanced evaporating surfaces in the low heat flux regime is the enhanced nucleate boiling and advection induced by oscillating flow. However, microchannels showed better evaporating performance than the micromembrane-enhanced evaporating surfaces when the working heat flux exceeded $30.3 \text{ W}/\text{cm}^2$ due to the fully activated flow oscillations in microchannels.

CHF on the developed micromembrane-enhanced evaporating surfaces was substantially enhanced to 151.7 and $152.2 \text{ W}/\text{cm}^2$ on the two individual samples. Compared to microchannels, four-layer mesh screens, and single-layer mesh screen, CHF was enhanced approximately 83%, 198%, and 1000%, respectively. The dramatic enhancement in CHF primarily resulted from the separation of the evaporation and liquid transport process that was enabled by the micromembrane. As a result, capillary force was increased by fine meshes and flow resistance was reduced by microchannels.

4.2. Capillary evaporation phenomena on micromembrane-enhanced evaporating surfaces

According to the heat transfer characteristics and visualization study, as shown in Fig. 5(a), the whole heat transfer process can be divided into three regions. Region I (Figs. 5(a) and 6(a)) was dominant by the advective flows induced by oscillating capillary flows and by the nucleate boiling in the microchannels since the microchannels were nearly flooded [36]. It should be noted that in the micromembrane covered microchannels, the oscillating flows and the bubbles were not confined. Although the bubbles were not able to detach due to the absence of buoyancy force, these bubbles could keep growing until collapsed. The visible bubbles and menisci distribution in the region I are shown in Fig. 6(b) and (c). In region I, the micromembrane (i.e., mesh screen) was well wetted, but nucleate boiling was not fully developed on micromeshes due to the relative low superheat (less than 0.82°C on the bottom surface of microchannels as estimated). The thin film evaporation was also weak due to the flooded surfaces in region I (Fig. 6(a)).

To understand the oscillation mechanism on the micromembrane-enhanced evaporating surfaces, the schematic liquid and vapor distribution in this region was proposed in Fig. 6. In the region I, the micromembrane-enhanced evaporating surfaces, which include microchannels and micromembranes, should be saturated

Fig. 5. Evaporation curves and the proposed heat transfer regions. (a) Capillary evaporation heat transfer curves on the microchannels, single layer mesh, four-layer meshes, and micromembrane-enhanced evaporating surfaces. Heat transfer region I (0 – $11.2 \text{ W}/\text{cm}^2$), region II (11.2 – $57.2 \text{ W}/\text{cm}^2$), and region III (57.2 – $152.2 \text{ W}/\text{cm}^2$). (b) q'' – ΔT curves.

Fig. 6. Schematic of fluid flow on the micromembrane-enhanced evaporating surfaces in region I. (a) Proposed fluid distribution in heat transfer region. ((b) and (c)) Bubble and meniscus distributions on the surfaces at the heat flux of 9.0 W/cm², respectively. ((d)–(h)) Hypothesized interactions between fluid and vapor inside microchannels. (i) The number of visible bubbles in a single channel, n_b , and total number of visible bubbles in active channels, $N_c n_b$, in the low heat flux region. (j) Oscillating frequency in a single channel, f , and the total frequency in the channel array, i.e., the product of active channel number and the frequency of a single channel, $N_c f$, in the low heat flux region.

(or flooded) with water initially (Fig. 6(d)). The liquid columns inside microchannels were separated by the growing bubbles periodically, which resulted in the periodic change of the meniscus curvature primarily in microchannels (Fig. 6(e)). As a result, the upper liquid columns tended to move downwards due to the net capillary force ($F_2 - F_1$) generated on the caps of the upper liquid columns. Simultaneously, the lower liquid columns were pumped up by capillary force (Fig. 6(f)). The liquid columns with motions in the opposite directions ended up with a merging in a microchannel. Then, the merged liquid columns were pumped up by the capillary force generated on the meniscus located on the top ends of microchannels. The incompletely wetted microchannels were then fully rewetted by the working fluid at a velocity of more than 10 m/s as measured (Fig. 6(h)). This rewetting process is similar to the multilayer liquid spreading as mentioned in Xiao's work [37]. This periodic process resulted in oscillating capillary flows and hence enhanced *HTC* during capillary evaporation in the vertical direction in region I, which is radically different from the oscillating flows observed in the confined channels and closed chambers [27,38,39]. The liquid droplets were observed to flash out of the microchannels from the top openings at the low heat flux region, which further indicated oscillating fluid motions on the microm-

embrane-enhanced evaporating surfaces. Nucleate boiling occurred primarily on the microchannel walls because the surface temperature on the microchannel wall was higher than that on the membrane. A higher superheat is required to initiate nucleate boiling on membrane according to the nucleate boiling theory [40,41]. As illustrated in Fig. 6(i) and (j), in a single microchannel, the oscillating frequency increased because the nucleate boiling was intensified and the menisci receded more rapidly due to the increased evaporating rate. However, it was also observed that not all the channels were activated because of the variations in the roughness and wettability. Hence, the capillary flows were governed by the boiling and menisci inside these active microchannels in region I.

Region II (Fig. 7(a)) is a transition heat transfer region, where the heat transfer should be dominated by thin film evaporation. When the heat flux exceeded 11.2 W/cm², the phase change rate was getting stronger and the liquid supply through capillary pressure generated in microchannels was not sufficient in some of the microchannels. As a result, some microchannels started to be partially dry and the oscillating liquid flows were gradually suspended in these microchannels. The primary reason that results in this transition region could be the transformation of the dominant capillary

Fig. 7. Liquid distributions in region II. (a) Hypothesized liquid distribution inside the structure. ((b) and (c)) Bubble and meniscus distributions on the surfaces at the heat flux of 40.4 W/cm^2 , respectively.

force from microchannels to micromembranes. This speculation could be evidenced in Fig. 7(b) and (c) [36], where visible bubbles were not observed. When the heat flux approached to 57.2 W/cm^2 , only one microchannel was observed to be active with fluid oscillations. Accordingly, HTC dropped with the increasing number of non-active channels in this transition regime, as thin film evaporation started to prevail. As shown in Fig. 5(a), the shape of the capillary evaporation heat transfer curves in regions I and II was

well predicted by the trend of total frequency, which was defined by the product of the oscillatory frequency, f , and the active channel numbers N_c , as a function of input heat flux (Fig. 6(i) and (j)).

Region III (Fig. 8) should be dominated by the thin film evaporation and nucleate boiling in microchannels as well as on the micromeshes. Fig. 8(a)–(c) show that the menisci were inherently interacting with bubble dynamics [36]. When the superheat on meshes was getting higher than the onset of the nucleate boiling with increasing heat fluxes, bubbles started to nucleate on mesh wires (Fig. 8(a)). Liquid oscillations were retarded during the bubble expansion, while strong liquid pulls were observed during the menisci receding into the sharp corners formed by individual wires when bubbles collapsed [36]. Thus, the liquid motions were significantly intensified by the induced wetting and dewetting [42] processes. Consequently, the fluid flows were resumed by the receding menisci in region III (Fig. 8(b) and (c)), where nucleate boiling occurred on the flooded microchannel walls, while thin film evaporation primarily occurred on the micromeshes. As shown in Fig. 8(d), the bubble number increased continuously with increasing heat fluxes. It was challenging to visualize individual bubbles when the heat flux exceeded 100 W/cm^2 . The microchannels could not be fully saturated with working fluid due to the high velocity of vapor flow in the high heat flux region. The frequency of bubble growth and collapse as well as the menisci receding on the mesh screens would determine the frequency of local liquid wetting and dewetting processes, which are different from the liquid motions in region I. The frequency of wetting and dewetting, which was found to be closely associated with the bubble growth and collapse, increased with increasing heat fluxes as shown in Fig. 8(d). The copper meshes provide larger surface areas and more nucleate

Fig. 8. Schematic of fluid flow on the micromembrane-enhanced evaporating surfaces in region III. (a) Hypothesized fluid flow and liquid distributions inside the structure. ((b) and (c)) Bubble and meniscus distributions on the surfaces at the heat flux of 61.4 W/cm^2 , respectively. (d) The number of visible bubbles in a single channel, n_b , and total number of visible bubbles in active channels, $N_c n_b$, in the high heat flux region. (e) Bubble growth and collapse frequency, f_{bc} , in the high heat flux region.

sites for heat transfer. The peak HTC in the region III was measured at $9.1 \text{ W}/(\text{cm}^2 \text{ K})$, which is approximately 50% lower than the peak value ($14.1 \text{ W}/(\text{cm}^2 \text{ K})$) in the region I. The reason could be that the additional thermal resistance resulting from the vapor core that started to form in microchannels in the region II and further developed in region III. The assumption of the existence of vapor core inside microchannels (Fig. 8(a)) can be validated by the fact that liquid droplets splash from the top opening of the microchannels in region I, but was not observed in regions II and III [36].

5. Conclusions

In summary, significantly high HTC and CHF were achieved using the micromembrane-enhanced evaporating surfaces that were developed in this study. The separation of capillary pressure generation and water transport processes enabled by the micromembrane should be favorable to supply liquid. CHF on the developed micromembrane-enhanced evaporating surfaces was substantially enhanced to $152.2 \text{ W}/\text{cm}^2$. Compared to microchannels, four-layer mesh screens, and single-layer mesh screen, CHF was enhanced approximately 83%, 198%, and 1000%, respectively. The increased capillary forces on micromeshes and the reduced flow resistance using microchannels should be the primary reasons for the dramatically increased CHF . The generation of capillary forces on the newly developed interfaces was governed by the liquid evaporative rate, bubble growth and collapse and receding menisci, which were not continuous or constant. Oscillating flows were observed in the low heat flux region and HTC was increased by the induced advection. Thin film evaporation and nucleate boiling were improved by the enhanced nucleation site density and augmented surface areas through the use of micromembrane that was made of sintered single-layer micromesh, resulting in the enhanced capillary evaporation HTC in the high heat flux region.

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Appendix A. Supplementary data

Supplementary data associated with this article can be found, in the online version, at <http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.05.030>.

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