Sweating-boosted air cooling using nanoscale CuO wick structures

Pengtao Wang, Raikan Dawas, Mohammad Alwazzan, Wei Chang, Jamil Khan, Chen Li*

Department of Mechanical Engineering, University of South Carolina, Columbia, SC 29208

Abstract

Low heat transfer coefficient (HTC) in air/fin-side is the bottleneck of dry cooling strategies for thermal power plants. Inspired by the phase change heat transfer during the perspiration of mammals, a sweating-boosted air cooling strategy with on-demand water dripping is proposed. The testing samples are featured with macroscale grooves for global liquid delivery, and with nanoscale hydrophilic copper oxide (CuO) wick structures for local liquid spreading. The experiments of sweating-boosted air cooling are conducted in a wind tunnel system. There are three wetting conditions with increasing dripping rates: dry, partially wetted, and flooded conditions. In the partially wetted conditions, the surface temperatures reduce and HTCs increase with increasing dripping rates. For a given dripping rate of water, HTCs are enhanced and surface temperatures are reduced with increasing air velocities. High air velocity and low surface temperature have a trade-off effect on the evaporation process. This effect results in almost constant saturated dripping rates for a given thermal load. A linear relationship between the saturated dripping rates and the thermal loads confirms that the evaporation dominates the heat transfer process of sweating-boosted air cooling. Complete surface wetting is obtained on the designed surfaces, but no obvious effect of groove width on HTCs is observed. Sweating-boosted air cooling can significantly increase air-fin side HTC in air cooled condenser (ACC), and dramatically reduce the water consumption compared to current water evaporative

* Corresponding author. Tel.: +1 803 777 7155. E-mail address: li01@cec.sc.edu (C. Li)

condenser (WEC). This research provides a fundamental understanding on the sweating-boosted effects on the air cooling.

Keywords: sweating cooling; air cooling; thin liquid film evaporation; convective heat and mass transfer; dry cooling condensers.

Nomenclature	
A	surface area of testing samples, cm^2
a, b	parameters for the empirical correlation of mass transfer
c_{pl}	thermal capacity of water
D	depth of grooves, mm
d	thickness of liquid film, μm
H	height of testing samples, mm
h	heat or mass transfer coefficient, $\text{W}/\text{m}^2\cdot\text{K}$
h_{fg}	latent heat of vaporization, kJ/kg
I	applied current on the heater, A
k	thermal conductivity, $\text{W}/\text{m}\cdot\text{K}$
L	length of testing samples, mm
m''	evaporation rate of water, ml/h
Nu	Nusselt number
p	partial vapor pressure, Pa
Pr	Prandtl number
q''	effective heat flux, W/cm^2
R	thermal resistance, $(\text{m}\cdot\text{K})/\text{W}$
Re	Reynolds number
T	temperature, $^{\circ}\text{C}$
t	thickness of CuO nanostructures, μm
V	applied voltage on the heat, V
U	air flow velocity, m/s
W	groove width, mm
Δx	distance from thermal couple and the bottom of grooves, mm
<i>Greek symbols</i>	

1. Introduction

The optimal performance of thermal-electric power plants requires effectively cooling the low-pressure

δ	boundary layer thickness
η	water used efficiency
ϕ	heat loss, %
<i>Subscripts</i>	
<i>Cu</i>	super-conductive copper
<i>CuO</i>	copper oxide
<i>conv</i>	convective heat transfer
<i>eq</i>	equivalent parameters
<i>f</i>	fluid, air or water
<i>i</i>	dripping water
<i>l</i>	liquid
<i>m</i>	convective mass transfer coefficient
<i>v</i>	vapor
<i>w</i>	wall
<i>s</i>	surface
<i>sweat</i>	sweating-boosted air cooling
<i>t</i>	convective heat transfer coefficient
<i>total</i>	total input power of heater
<i>x</i>	coordinate in x-direction
∞	averaged properties of air flow

steam in the condenser. A huge amount of freshwater is withdrawn and consumed to cool the low-pressure steam, accounting for approximately 143,000 million gallons per day in 2005 [1]. The insistent pursuit of annual reduction in freshwater usage motivates to replace water cooled condenser (WCC) with air cooled condenser (ACC). ACC can dramatically reduce water consumption, but its large footprint and high capital costs (about 3-5 times that of water cooled condenser [2]) are two major limitations. ACC is also susceptible to the ambient temperature and humidity, which may result in 10 % power production

penalty in hot weather [3]. These drawbacks of ACC are due to low heat transfer coefficient in air/fin-side (typically ~20 - 50 W/m²K). This challenge has been partially addressed by injecting water mist into air to enable an evaporation process, e.g., water evaporative condenser (WEC), where the heat dissipation performance is significantly improved through using the latent heat of vaporization. WEC is more energy efficient and environmentally friendly [4]. However, complex peripheral components are required in

WEC systems to generate the mist, and to eliminate the unevaporated mist to reduce the drift loss [5].

Inspired by the effective thermoregulation by a sweating/perspiration process of mammals, herein a sweating-boosted air cooling strategy is proposed to dramatically increase the air/fin-side heat transfer coefficient within ACC. The mechanism of this approach is the convective water evaporation enhanced by a forced air flow. Schematic of sweating-boosted air cooling strategy is illustrated in Figure 1. A water droplet is dripped on the wick structures on a copper block with wick structures, then it spreads literally and forms a thin liquid film. Synchronously, the water evaporates into vapor on this heated surface, then the vapor is carried away by the air flow. Therefore, the convective heat transfer is enhanced owing to the coupled phase change process. The water consumption is minimized owing to a near-surface on-demand dripping system.

Figure 1. Schematic of sweating-boosted air cooling strategy.

It has been well-established that the latent heat transport accompanied with convective mass transfer in liquid film evaporation can tremendously enhance the convective heat transfer. Yan [6] studied the transport of latent heat associated with a film evaporation in a

vertical channel, and reported that the magnitude of evaporative latent heat flux was five times greater than that of sensible heat flux. Jiang et al. [7] investigated the mixed convective heat transfer enhanced through a liquid film evaporation in inclined square ducts, and found that the heat transfer rate could be enhanced to be ten times of that without mass transfer.

The convective mass transfer coefficient, i.e., the evaporation rate, is the key parameter for evaporative cooling processes. With an assumption of a semi-permeable liquid-air interface, the convective mass transfer could be driven by the difference in vapor density [8], mass fraction [9], vapor pressure [10], or mole fraction [11]. The conjugation between convective heat and mass transfer makes the evaporation process more complex, which depends on the hydrodynamic, thermal, and concentration boundary layers near the liquid-air interface. Experimental and numerical studies have been performed to investigate the factors related to the evaporation rates. These include (a) flow conditions: natural [12], laminar [13], and turbulent convection [14]; (b) geometries, such as vertical channel [15], inclined planes [7], and tube bundles [16]; and (c) coolants, such as water, R134a [17], and ammonia-water mixtures [18].

Despite of extensive experimental, theoretical, and numerical studies on the liquid film evaporation, major discrepancies were observed between these results due to the challenge in obtaining a homogeneous liquid film [19]. The liquid film was experimentally obtained by flowing the liquid over vertical or inclined plates, whereas it was theoretically and/or numerically treated as a boundary condition with an assumption of zero thickness [20]. Few researchers utilized rough surfaces, such as rib, rod, and porous layers [21], to improve the convective mass transfer by controlling a small thickness of liquid film. Although there

were limited studies on the modified surfaces for the liquid film evaporation, various surface modification techniques have been successfully employed in enhancing the evaporation/boiling heat transfer. One method is to change surface morphology with micro- and/or nano- sized features, such as nanowires [22], thin capillary wicks [23, 24], micropillar arrays [25, 26], and sintered porous media [27]. The other method is modifying surface chemistry by coating a thin film of high surface energy material, such as silica [28, 29], and TiO_2 [30]. The mechanisms for these enhancements in the heat and mass transfer are as follows: the high capillary pressure enables liquid delivery to form a thin liquid film over the heat transfer surface, while the large extended menisci of liquid-vapor interface within microstructures provides an augmented evaporation area. We hypothesize that the surface modification can play a significant role in enhancing the convective mass transfer in the sweating-boost air cooling.

In the present study, the heat dissipation performance of sweating-boosted air cooling is experimentally evaluated. CuO nanostructures are fabricated on a grooved copper substrate, serving as a wick structure. The typical performance of sweating-boosted air cooling under various dripping rates are studied. The effects of air velocities, thermal loads, and width of grooves are also systematically examined.

2. Fabrication and characterization of the CuO nanostructures

To enhance the sweating cooling, the dripping water needs to be delivered uniformly and forms a thin liquid film for evaporation over the entire heat transfer surface. The liquid transportation is determined by the trade-off between capillary pressure (i.e., the driving force) and flow resistance. Nanoengineered wick structures can generate high capillary pressure, but with large flow resistance. Whereas macroscale grooves with a relatively

smaller flow resistance can be combined with micro-scale wick structures (e.g., copper woven mesh) and microchannels [28, 29] to achieve high capillary pressure and small flow resistance. In this study, the evaporation surface is designed by coating nanoscale wick structures on a grooved copper substrate. The macroscale grooves provide the global liquid transportation with a reduced flow resistance, while nanoscale wick structures act a pump to rapidly spread liquid. Moreover, the nanoscale wick structures provide an augmented evaporation area, resulting in an enhanced mass and heat transfer.

3 mm-thick super-conductive copper blocks (50 mm × 50 mm) are used to prepare the test samples. 0.5 mm-deep rectangular grooves are milled with a spacing of 5 mm, and with three widths of 0.5 mm, 0.75 mm, and 1.0 mm, respectively. To prepare the wick structure, first, the samples are ultrasonic cleaned in acetone for 10 mins and then rinsed with ethanol, isopropyl alcohol, and deionized (DI) water. Second, the samples are dipped into 1.0 M hydrochloric acid solution for 10 mins to remove the native oxide film and rinsed with DI water, then dried by nitrogen gas. Third, the samples are immersed into an alkaline solution composed of NaClO₂ (3.75 g), NaOH (5 g), Na₃PO₄·12H₂O (10 g), and distilled water (100 ml) at 95 °C for 30 minutes [31, 32]. Finally, the samples are rinsed with DI water and dried by nitrogen gas. During this quasi-self-limiting chemically oxidation process, a thin film of brown Cu₂O is initially formed, and subsequently oxidized into a uniform film of black CuO, as shown in Figure 2(a) and (b). CuO nanostructures have sharp blade-like morphology with a height about 1 μm and a tip dimension less than 10 nm, as shown in Figure 2(c). Atomic force microscope (AFM) measurement shows that the surface roughness of CuO nanostructures is Ra = 373 nm, Rq = 454 nm, as illustrated in Figure 2(d). The static contact angle of water on these CuO nanostructures is ~ 9.6°, compared

with the contact angle of 46° on a smooth CuO surface [33]. It indicates that the wettability is significantly enhanced by these nanoscale features. CuO nanostructures also show superior mechanical properties with reduced modulus of 41.53 GPa and surface hardness of 0.34 GPa as measured in a nanoindentation test.

Figure 2. Testing sample with grooves (a) before and (b) after hot alkaline treatment ($W = 0.75$ mm). CuO nanostructures characterized with FE-SEM (c) and (d) AFM.

The liquid transportation within single groove patterned with CuO nanostructures is characterized by a water rise experiment [34]. The sample ($25 \text{ mm} \times 50 \text{ mm}$) is vertically positioned on a sliding block, and slowly dripped into the liquid reservoir. Once the sample touches the liquid surface, the liquid raises within the groove and CuO nanostructures due to the capillary force. The water wetting behavior as a function of time is recorded by a high-speed camera (Micro-Ex4, Phantom) operating at 100 frames per second (fps).

Figure 3 shows the selected video frames from the liquid transportation within a single groove ($W = 0.75$ mm) coated with CuO nanostructures. Immediately after immersing the samples into DI water, a macroscopic meniscus forms due to the competition between the capillary force and the gravity. The measured height of this meniscus is about 4 mm, which agrees well with both the theoretical and experimental data reported in [33]. The water front moves fast vertically along the groove, then propagates horizontally within the wick structures owing to a substantially smaller flow resistance resulting from the integration of a microscale groove. After 5 seconds, the rising height within a groove ($W = 0.75$ mm) is approximately 19.53 mm, whereas the horizontal wetting width is 1.51 mm.

Figure 4 shows the capillary rise heights as a function of time for different groove widths. On a flat surface with wick structures, the water front reaches a relative steady state within 1 second, corresponding to a capillary height of 4.73 mm. While for the surface with a groove, the water wets the surface very fast in the first few seconds, after that the water propagation speed slows down continuously. The groove with a smaller width generates higher capillary pressure to drive the water propagation, therefore, a higher capillary rise height is observed.

Figure 3. Selected video frames from the liquid transportation within single groove patterned with CuO nanostructures. ($W = 0.75$ mm)

Figure 4. Capillary rise heights as a function of time.

3. Experimental system and data processing

3.1 Experimental apparatus

Sweating-boosted air cooling is conducted in a wind tunnel (ScanTEK 200, Aerostream), as shown in Figure 5. The air velocity over the testing sample is measured with an air speed

transmitter (FMA904R-V1, Omega). The temperature and humidity of air flow are measured with a duct style relative humidity/temperature transmitter (HX-94C, Omega). In this research, the temperature and relative humidity of inlet air are 21 ± 1 °C and 55 ± 3 %, respectively. All relevant data are channeled into a data acquisition unit (34972A, Agilent) and recorded with a LabVIEW program built-in PC. Distilled water is delivered by a syringe pump (ProSpense, Cole-parmer) and directly dripped on the center of testing samples.

Three T-type thermal couples (TMQSS-020G-12, Omega) are mounted on the backside of testing samples, along the central line with a spacing of 1.5 cm and a depth of 2 mm. A flexible silicone rubber heater (SRFG-202/10-P, Omega), connecting with a DC power supply, is used to provide a constant heat flux. The heater is bonded with the back side of testing sample using thermal conductive epoxy (OB-101, Omega), and supported with a plastic block for thermal insulation. This testing unit is inserted into an acrylic block, where the surface of testing sample is adjusted horizontally to ensure a parallel air stream over the testing sample.

Figure 5. Experimental setup for the sweating-boosted air cooling.

3.2 Data processing and uncertainty analysis

Thermal load is the effective heat flux provided by the heater, given as

$$q'' = \frac{VI}{A}(1 - \phi) \quad (1)$$

where

q'' is the effective thermal loads dissipated from the heated surface;

V and I are the voltage and current applied to the heater, respectively;

A is the projected surface area of testing sample;

ϕ is the heat loss of experimental system.

Figure 6. Thermal resistance analysis for the grooved testing sample with CuO nanostructures. (a) Schematic of a unit cell, (b) Thermal resistance network.

As shown in Figure 6(b), the thermal resistance network for the grooved substrate patterned with CuO nanostructures is

$$\bar{T}_w - T_\infty = q''(R_{Cu} + R_w + R_l + R_{sweat}) \quad (2)$$

where

$\bar{T}_w$ is the average wall temperature measured with three thermal couples;

T_∞ is the temperature of air flow;

R_{Cu} is the thermal resistance of copper block from the bottom of grooves to the location of thermal couple;

R_w is the thermal resistance from the bottom of grooves to the top of wick structures;

R_l is the thermal resistance of the liquid film within wick structures;

R_{sweat} is the thermal resistance of the convective heat and mass transfer due to the sweating-boosted air cooling.

The thermal resistance of the copper block between the location of thermal couples and the bottom of grooves is

$$R_{Cu} = \frac{\Delta x}{k_{Cu}} \quad (3)$$

where

Δx is the distance between the location of thermal couples and the bottom of grooves, and $\Delta x = 0.5 \text{ mm}$;

k_{Cu} is the thermal resistance of copper.

The thermal resistance of a grooved sample patterned with copper oxidation nanostructures is given as [35]

$$R_w = \frac{(W+L)t}{(L+2t)k_{CuO} + (W-2t)k_f} + \frac{(W+L)(D-t)}{Lk_{Cu} + 2tk_{CuO} + (W-2t)k_f} + \frac{(W+L)t}{Lk_{Cu} + Wk_{CuO}} \quad (4)$$

where

W is the width of grooves, and W is 0.5 mm, 0.75 mm, and 1.0 mm, respectively;

L is the length of square region surrounded by grooves, and $L = 5.0 \text{ mm}$;

D is the depth of grooves, and $D = 0.5 \text{ mm}$;

t is the total thickness of CuO nanostructures, and $t = 1.5 \text{ }\mu\text{m}$;

k_{CuO} is the thermal resistance of CuO nanostructures, and $k_{CuO} = 21.43 \text{ W/m} \cdot \text{K}$; [31]

k_f is the thermal resistance of the fluid in grooves, i.e., air or water.

The thermal resistance of the liquid film within the nanostructures is

$$R_l = \frac{d}{k_l} \quad (5)$$

where d is the thickness of the liquid film within the wick structures. With the assumption that the water fully fills the cavities of CuO nanostructures, $d = 1 \mu\text{m}$.

The surface temperature for the evaporation (T_s) can be determined as a linear function of its thermal resistance under a 1-D assumption,

$$T_s = \bar{T}_w + q''(R_{cu} + R_w + R_l) \quad (6)$$

The equivalent heat transfer coefficient (HTC, h_{eq}) for the sweating-boosted air cooling is extracted as follows

$$h_{eq} = q''/(T_s - T_\infty) \quad (7)$$

The uncertainties of key parameters are listed in Table 1. For all the samples, uncertainties propagation in the calculated values is estimated using the Kline and McClintock method [36].

Table 1. Uncertainties of key parameters.

Name of parameters	Errors
Length, L	0.01 mm
Wall temperature, T_w	0.5 °C

Air temperature, T_{∞}	0.6 °C
Air velocity, U_{∞}	0.5 m/s
Water dripping rate	2.5 %
Input power of heater, q''_{total}	0.20 %
Heat Loss, ϕ	3.0 %
HTC, h_t	2.1 ~ 41.5 W/m ² K

3.3 Calibration of experimental system

The dominant error in evaluating the heat transfer coefficient arises from the estimation of conductive heat losses from the testing samples to surroundings. To estimate the heat loss, an identical flat copper block is used as the testing sample. No evaporation process is employed in these calibration experiments. The velocities of air are in the range of 2 - 10 m/s, corresponding to the Reynolds number in a range of 4,921 - 31,444. The total input powers of heater are 4.8 W, 7.5 W, and 10.7 W, respectively.

The convective heat flux over the flat surface is measured by thin film heat flux sensors (HSF-4, Omega). The experimental HTC is calculated by Equation (7). The measured heat flux is assumed to be the effective heat flux. The theoretical HTC is calculated from the empirical relation based on average Nusselt number ($\overline{Nu}$) given as [37]

$$\overline{Nu} = 0.906Re^{1/2}Pr^{1/3} \quad (8)$$

where Re and Pr is Reynolds number and Prandtl number of air, respectively. It is restricted to a flat plate with constant heat flux in a laminar flow ($Re < 3 \times 10^5$). A great

agreement between the experimental and theoretical HTC is achieved, as shown in Figure 7, indicating an accurate measurement of convective heat flux.

Figure 7. The experimental and theoretical HTCs for a flat copper plate.

The heat loss (ϕ) is estimated from measured convective heat flux, given as

$$\phi = \frac{q''_{total} - q''_{conv}}{q''_{total}} \times 100\% \quad (9)$$

where

q''_{total} is the input power of heater, and $q''_{total} = VI/A$,

q''_{conv} is the convective heat flux measured with the heat flux sensors.

Figure 8 shows the heat loss estimated from the measured heat flux. It illustrates that the heat loss is directly related to the Reynolds numbers, and extremely high heat loss is found at a low Reynolds number, e.g., $Re \approx 5,000$. The averaged heat losses at different Reynolds numbers are expressed using an exponential fitting, as

$$\phi = 23.01\% + 31.80\% \times \exp(-2.267 \times 10^{-4} Re) \quad (10)$$

Equation (10) is used in the data reduction processing for the experimental results. The uncertainty for the estimated heat loss for input powers of 4.8 W, 7.5 W, and 10.7 W was less than 3 %.

Figure 8. Heat loss estimated with the heat flux sensor.

4. Results and discussion

During the sweating-boosted air cooling process, thermal loads are dissipated in forms of sensible heat and latent heat, given as

$$q = h_t(T_s - T_\infty) + m''c_{p,l}(T_s - T_i) + m''h_{fg} = h_{eq}(T_s - T_\infty) \quad (11)$$

where

h_t is the convective heat transfer coefficient;

T_s is the surface temperature of wetted wall;

m'' is the evaporation rate of water, i.e., the convective mass flux;

$c_{p,l}$ is the thermal capacity of water;

T_i is the temperature of dripping water;

h_{fg} is the latent heat of water.

h_{eq} is the equivalent heat transfer coefficient of sweating-boost air cooling.

In Equation (11), the first term presents the convective air cooling, the second term for heating up of dripping water, and the third term for the latent heat of water, respectively. The second term is much less than the other two terms, and hence neglected in this study.

The mass transfer process is assumed to be under the conditions that the Fick's law of diffusion is valid and the interface is semi-permeable. By taking the difference in the vapor pressure as the driving force, the evaporation rates can be expressed as [9]

$$m'' = h_m A (p_{v,s}(T_s) - p_{v,\infty}(T_{v,\infty}, RH)) \quad (12)$$

Where

h_m is the convective mass transfer coefficient;

$p_{v,s}$ is the partial vapor pressures at the surface of liquid film;

$p_{v,\infty}$ is the partial vapor pressures in the air flow, which is determined from the temperature and relative humidity.

Pauken [38] proposed an empirical correlation between the convective mass transfer coefficients and the mean air velocity (U_∞) as

$$h_m = a + bU_\infty \quad (a > 0, b > 0) \quad (13)$$

With an assumption of saturated vapor condition at the liquid-air interface, $p_{v,s}$ can be determined from the surface temperature of liquid film. $p_{v,\infty}$ is taken as a constant in this research, as the temperature and relative humidity of the inlet air are 21 ± 1 °C and 55 ± 3 %, respectively. As a summary, high air flow velocity and high surface temperature can enhance the convective heat and mass transfer in the sweating-boosted air cooling.

4.1 Typical heat dissipation performance of sweating-boosted air cooling

During the sweating-boosted air cooling process, the dripped water spreads and evaporates synchronously. The wetting conditions with various dripping rates are captured with an IR camera (A325sc, FLIR), as shown in Figure 9. The infrared images are used only for qualitatively distinguishing the wetting conditions, rather than quantitatively measuring the surface temperature. The grooves filled with water are dark-colored, while the dry grooves are light purple-colored, because of different radiation emissivity of CuO and water. However, no obvious difference between the wetted and unwetted CuO wick structures is observed, which may be due to an extremely thin liquid film formed over the CuO wick structures and/or high thermal conductivity of copper substrate.

The competition between the dripping rates and the evaporation rates results in various wetted conditions, namely, dry, partially wetted, and flooded conditions. At the lowest dripping rate of 2 ml/h conducted in this study, the water dripped on the testing samples generates a small wetted spot, but appears to evaporate completely before the next liquid drop is delivered. More water drops are generated with increasing of dripping rates, as the droplet volume is almost constant [39]. A stable wetted area is observed with a dripping

rates of 4 ml/h. The wetted area is continuously enlarged with the increasing of dripping rates from 4 ml/h to 10 ml/h, resulting in a partially wetted condition. The wick surface is completely wetted with a dripping rate of 11 ml/h, which is defined as the saturated wetting point. Beyond this point, water accumulates on the surface and hence forms a thick liquid film, resulting in a flooded condition. This flooded condition needs to be avoided to minimize the draft loss in the practical application.

Figure 9. Wetting conditions of testing samples with increasing water dripping rates. ($q'' = 0.29 \text{ W/cm}^2$, $U_\infty = 6 \text{ m/s}$, $W = 0.75 \text{ mm}$)

The performance of sweating-boosted air cooling is summarized as the surface temperature and HTC vs. water dripping rates, as shown in Figure 10. The surface temperatures reduce linearly but the HTCs increase with increasing water dripping rates, until they reach the saturated wetted point. Under a partially wetted condition, dripped water completely evaporates. The evaporative heat transfer becomes more dominative with more water evaporating at higher dripping rates, resulting in a higher HTC. For water dripping rates increasing from 2 ml/h to 11 ml/h, the surface temperature is reduced from 67.1 °C to 37.9 °C, while the HTC sharply increases from 65.6 W/m²K to 185.4 W/m²K. At the saturated

point, the minimum surface temperature and the optimized HTC are achieved. Under the flooded condition, the surface temperature and the HTC are almost constant, as the excess dripping water cannot be effectively evaporated. Moreover, high uncertainties of HTC are observed due to low surface temperatures. As a summary, the thermal performance is dramatically enhanced by 182.6 % with air flow velocity of 6 m/s for a thermal load of 0.29 W/cm^2 . It proves the concept of sweating-boosted air cooling.

Figure 10. Typical thermal performance of the sweating-boosted air cooling. The solid (■) and hollow (□) symbols indicate the surface temperature and the HTC, respectively. ($q'' = 0.29 \text{ W/cm}^2$, $U_\infty = 6 \text{ m/s}$, $W = 0.75 \text{ mm}$)

4.2 Effects of air velocities

Figure 11. The effects of air velocities on the surface temperatures. Symbol legend: solid square (■) for $U_{\infty} = 6 \text{ m/s}$; solid circle (●) for $U_{\infty} = 4 \text{ m/s}$; solid up-triangle (▲) for $U_{\infty} = 2 \text{ m/s}$; solid down-triangle (▼) for $U_{\infty} = 0 \text{ m/s}$. ($q'' = 0.43 \text{ W/cm}^2$, $W = 0.75 \text{ mm}$)

Figure 12. The effects of air velocities on the HTC. Symbol legend: hollow square (□) for $U_{\infty} = 6 \text{ m/s}$; hollow circle (○) for $U_{\infty} = 4 \text{ m/s}$; hollow up-triangle (△) for $U_{\infty} = 2 \text{ m/s}$; hollow down-triangle (▽) for $U_{\infty} = 0 \text{ m/s}$. ($q'' = 0.43 \text{ W/cm}^2$, $W = 0.75 \text{ mm}$)

The performance of sweating-boosted air cooling under a forced convection are performed at air velocities of 2, 4, and 6 m/s, corresponding to the average Reynolds numbers of 5,832, 11,530, and 17,538 on the flat surface, respectively. Cooling process under natural convection is also included for comparison, as shown in Figures 11 and 12. With a given dripping rate, the surface temperature drops as the air velocity increases. This follows the general concept that the convective heat transfer increases with higher Reynolds numbers. Under given flow conditions, lower surface temperatures are observed for higher dripping rates of water, which are similar as the typical thermal performance of sweating-boosted air cooling as discussed previously. According to the boundary layer theory, the thickness of thermal and mass boundary layers is directly related to the air velocity, as $\delta \sim 1/\sqrt{Re_x}$. These boundary layers become thinner and less developed with increasing of air velocities, resulting in a larger mass concentration gradient at the liquid-air interface. The grooves and CuO nanostructure may cause the velocity fluctuations, which further enhance the heat and mass transfer within the boundary layers. Thus, higher HTCs are achieved at higher air velocities.

Equation (13) indicates there is a trade-off between air velocities and surface temperatures in affecting the evaporation rates. These phenomena have been reported by several researchers: Raimundo et al. [40] reported that air flow velocity dominates the mass transfer, and temperature differences had much less contributions for a forced air flow ($0 < U_\infty < 0.7 \text{ m/s}$) over a heated pool ($8^\circ\text{C} < (T_s - T_\infty) < 22^\circ\text{C}$). Whereas M. H. Shih [41] numerically presented that the magnitude of evaporative heat flux could be 10 to 25 times greater than that of the sensible heat flux, where the surface temperature increased from 45°C to 75°C with a given Reynolds number. Jiang et al. [7] concluded that the higher the

surface temperature, the better the latent heat transport related with the liquid film evaporation. As a summary, evaporation rates depend on both air velocities and surface temperatures.

With a constant heat flux, higher air velocities result in lower surface temperatures. Despite of various air velocities, as shown in Figure 11 and 12, the saturated wetting conditions are reached with the same dripping rate of 16 ml/h. This indicates that there is a possibility that the equal contribution of the air flow velocity and the temperature differences to the total evaporated mass. The convective mass transfer related to surface temperature is most partially or even fully compensated by the effect of air flow velocity.

4.3 Effects of thermal loads and the widths of grooves

Figure 13 shows the saturated water dripping rates under different thermal loads. There is a linear relationship between the saturated dripping rates and the thermal loads. Assuming all dripping water is completely evaporated, the latent heat flux due to evaporation is approximately 94.53% of total thermal loads. This confirms that the evaporative heat transfer plays a dominant role under the saturated wetting conditions. The surface temperatures under the saturated conditions increases linearly with the thermal loads, as shown in Figure 14. HTC's have a relative stable value of $179.7 \text{ W/m}^2\cdot\text{K}$ for the thermal loads in range of 0.29 to 0.72 W/cm^2 . At a high thermal load, the thermal performance of sweating-boosted air cooling is independent of thermal loads. Abnormal high HTC's (about $252.6 \text{ W/m}^2\cdot\text{K}$) are observed for a low thermal load of 0.14 W/cm^2 . This value is of a large uncertainty (about $41.5 \text{ W/m}^2\cdot\text{K}$) due to the low surface temperature at a low thermal load. The average surface temperature for all grooved surfaces is 27.7°C with a thermal load of 0.14 W/cm^2 , but it reaches 63.6°C with a thermal load of 0.72 W/cm^2 .

The grooves coated with CuO nanostructures significantly improve the surface wetting, as illustrated by the capillary rise experiments and IR images. However, no obvious effects of groove widths are observed for the groove widths of 0.5 mm, 0.75mm and 1.0 mm, respectively. This may be due to the fact that the liquid film covers the grooves at the saturated wetting conditions. This film acts as a lubricating layer that significantly reducing the flow resistance. The unique surface used in this research provides a complete surface wetting.

The water-use efficiency in WEC (η) is defined as the actual cooling delivered divided by the maximum cooling obtained from evaporating a given mass of water. For the direct WECs used in southwestern United States, $\eta = 0.37 \sim 0.45$; while for the indirect WECs, $\eta = 0.17 \sim 0.23$ [42]. In WEC systems, the total consumption of water consists of evaporative water, drift loss, and blow down. By controlling the surface wetting conditions (i.e., operating at the saturated wetting point), the only water consumption within the sweating-cooling boosted air cooling is the evaporative water. Technically, the water-use efficiency in WEC for sweating-boosted air cooling may reach $\eta \approx 1$. Therefore, sweating-boosted air cooling can dramatically reduce the water consumption compared to WEC.

Figure 13. The saturated dripping rates for different thermal loads. ($U_{\infty} = 6 \text{ m/s}$)

Figure 14. The effects of thermal loads on the performance of sweating-boosted air cooling. The solid (■) and hollow (□) symbols indicate the surface temperature and the HTC, respectively. ($U_{\infty} = 6 \text{ m/s}$)

Conclusions

A sweating-boosted air cooling strategy is proposed and experimentally validated. The test samples are featured with macroscale grooves for global liquid delivery, and with superhydrophilic nanoscale CuO wick structures for local liquid spreading. The experiments are conducted in a wind tunnel system for forced convective heat and mass transfer process. Experimental results successfully prove the concept of sweating-boosted air cooling. The main conclusions are drawn as below:

- a) Typical thermal performance of sweating-boosted air cooling shows that there are three wetting conditions associated with the increasing of dripping rates, i.e., dry, partially wetted, and flooded conditions. At the saturated wetting condition, the optimized heat transfer coefficients are achieved.
- b) With a constant heat flux, higher air velocities result in enhanced HTC and lower surface temperatures. High air velocity and low surface temperature have trade-off effects on the evaporation process. This effect results in a constant saturated dripping rates for a given thermal load.
- c) A linear relationship between the saturated dripping rates and the thermal loads confirms that the evaporative heat transfer plays a dominant role in the sweating-boosted air cooling under the saturated wetting conditions.
- d) Great surface wetting is achieved on the designed surfaces. No obvious effects of grooves widths on sweating cooling HTC is observed in various groove widths.
- e) Sweating-boosted air cooling can significantly increase air/fin side HTC in ACC, and dramatically reduce the water consumption compared to current WEC.

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