Experimental investigation of spray cooling performance with enhanced heating surface structures

Long HUANG*, Yu WANG** and Yanlong JIANG*

* Department of Air Conditioning and Refrigeration, Nanjing University of Aeronautics and Astronautics
29 Yudao St. Nanjing, Jiangsu Province, China
E-mail: 15881429391@163.com

** Department of Urban Construction, Nanjing Tech University
30 Pu Zhu St. Nanjing, Jiangsu Province, China.

Received: 29 January 2018; Revised: 22 April 2018; Accepted: 7 May 2018

Abstract
An open loop spray cooling experimental system with ten enhanced groove surfaces was established. The results show that maintaining an unchanged groove width and changing the groove depth from 0.2 mm to 1.6 mm, the surface with a groove depth of 0.8 mm, the largest heat flux enhancement appears when volume flow rate is 0.45 L/min and 0.75 L/min. However, the heat transfer performance increased with an increasing groove depth under the flow rate of 1.25 L/min. Then based on the analysis of the forces acting on the falling droplet, the residual velocity of the droplet is calculated to explain why the optimal heat transfer of enhanced surfaces is different at different volume flow rates. Maintaining a groove depth of 0.8 mm, as the groove bottom width was reduced from 4 mm to 1 mm, the heat transfer coefficient increased by 41%. With the spray flow rate of 1.25 L/min, the heat transfer coefficient increased by only 8.5% and the reason for this phenomenon is explained by the variation of Bond number which reflects the capillary force. Finally, a relevant correlation of the non-dimensional Nusselt number Nu for the grooved surface is proposed in the non-boiling area with water coolant.

Keywords: Spray cooling, Enhanced surface, Heat transfer enhancement, Mechanism, Non-dimensional correlation

Nomenclature

\(A\) surface area (mm\(^2\))
\(a\) droplet acceleration (m/s\(^2\))
\(Bo\) Bond number
\(c\) specific heat (J/(kg-K))
\(D\) diameter of heating surface (mm)
\(d_0\) diameter of nozzle orifice (mm)
\(d_{32}\) Sauter mean diameter (mm)
\(f\) Buoyancy (N)
\(g\) gravitational constant (m/s\(^2\))
\(G_m\) mass flow rate (kg/s)
\(H\) groove depth (mm)
\(h\) heat transfer coefficient (W/(cm\(^2\)-K))
\(L\) groove width (mm)
\(Nu\) Nusselt number
\(p\) spray inlet pressure (kPa)
\(q\) heat flux (W/cm\(^2\))
\(R\) radius of heating surface (mm)
Re    Reynolds number  
T     temperature (°C)  
We    Weber number  
u     droplet velocity (m/s) 

Greek symbols
\( \beta \)    the ratio of the groove depth and width  
\( \delta \)   variance  
\( \lambda \)  heat conductivity coefficient (W/(m·K))  
\( \mu \)   dynamic viscosity coefficient (Mpa·S)  
\( \nu \)   viscosity of water (N·S/m²)  
\( \rho \)   ambient medium density (kg/m³)  
\( \sigma \)  surface tension (N/m)  
\( \zeta \)  evaporation intensity 

Subscript symbols
\( g \)   air  
\( in \)  Spray inlet temperature  
\( l \)   water  
\( sat \)  Saturation temperature  
\( w \)   surface 

1. Introduction 

Spray cooling is an innovative cooling technology due to its high heat exchange coefficient efficiency (Cheng et al. (2011)). The working principle of spray cooling is a spray cooling medium impelled through a small nozzle orifice, which shatters into a dispersion of fine droplets before hitting the heat source (Gao et al. (2017)). The droplets can diffuse on the surface to form the thin liquid film, removing large quantities of heat not only due to the latent heat of evaporation but also the substantial convection effects (Horace et al. (2005), Bostanci et al. (2018)). In fact, spray cooling technology has been demonstrated to be an attractive method that is adopted for some high heat flux applications to take heat flux away from heat surfaces with low droplet impact speed, low surface superheat and heat removal uniformity (Hsieh Shou-Shing et al. (2015), Karwa et al. (2007), Kim et al. (2007), Xiao et al. (2017)). Recently, spray cooling researchers have paid more attention to improving the performance of heat transfer by using enhanced surfaces, and various studies have been conducted. Liu et al. (2018) investigated the effects of spray cooling heat transfer performance on surface roughness. Their research shows that surfaces with different roughnesses lead to enhanced heat transfer and a broadened high-efficiency heat transfer range. Wang H et al. (2016) studied the effects of spray cooling on enhanced surfaces and indicated the reason for enhanced performance was the large wetted area and higher boiling site density. Coursey et al. (2006) used pf-5060 as a spray cooling medium to cool the different structural parameter surfaces; the results show that surfaces with groove channel heights of 1 mm and 3 mm are best for heat exchange, and the maximum heat flux reached was 124 W/cm². Zhang Zhen et al. (2014) investigated the spray characteristics and the different influences of spray cooling on one smooth and twelve enhanced silicon surfaces with micro-structures. They noted that there is an optimal groove depth corresponding to a given droplet parameter, groove width and stud size that gives the best heat transfer performance. Fukuda and Nakata et al. (2016) studied the influence of surface roughness on water spray cooling; the authors found that heat flux increases with surface roughness, which enhanced the cooling rate during film boiling. Hsieh Shou-Shing et al. (2006) tested spray cooling performance on the surface of cubic fins, pyramids and straight fins. They found that among the three enhanced surfaces, straight fins performed best and produced virtually 80% higher heat flux values compared with those of flat surfaces. Zhang Yu (2016) investigated the spray cooling characteristics for flat, straight fin and porous tunnel surfaces under acceleration conditions, and the results show that volume flow rate as well as the nozzle height influence the spray cooling performance under the acceleration condition for all three surfaces, similar to under the stationary condition. Sodtke and Stephan (2007) showed that compared with smooth surfaces, enhanced surfaces led to significantly better heat transfer performance due to the efficient thin film evaporation. Yang et al. (2013) examined three enhanced surfaces and a smooth surface; the results indicated that heat transfer efficiency on the enhanced surfaces was much better than that on the flat surface at high surface temperatures, and nucleate boiling dominated the
heat transfer in these conditions. Hsieh Cheng-Chieh et al. (2006) indicated that the Bond number of the micro enhanced surface was a key factor to explicate the heat transfer enhancement of evaporative spray cooling on silicon surfaces. Ulson de Souza et al. (2012) studied spray cooling performance of copper-foam test surfaces. The results show that all enhanced surfaces performed worse than those of plain surfaces in all cases.

It can be concluded from the above studies that changing the heating face structures has a direct influence on spray cooling heat transfer performance. However, more accurate and detailed descriptions of the physical insights into the processes that are associated with enhancement mechanism are required, especially in millimeter-level groove surfaces. Therefore, in this work, nine enhanced surfaces were proposed to increase the spray cooling performance. Tests were conducted in an open loop system with water as the working fluid, using pressure atomized spray nozzles. The heat fluxes, heat transfer coefficients and surface temperatures under different liquid volume flow rates with straight-finned surfaces and smooth surfaces were obtained. This paper attempts to discover the optimum structured surface that has the maximum heat transfer enhancement under different flow rates. The results can promote an understanding of water spray cooling and provide alternative surface design to improve heat dissipation in real applications.

2. Experimental set up

2.1. Spray cooling system

The schematic diagram of the open-loop experimental bench is shown in Fig. 1. The test facility comprises the following major components: a spray chamber, a Nozzle, an electrical heating control system, a data acquisition equipment and test heater with enhanced surfaces.

During spray cooling, nitrogen is first used as the driving force pressing the cooling water out of a stainless steel water tank. Then, the water passes through a flow control valve, a filter and a turbine flowmeter in rapid sequence. Finally, the water sprays through a nozzle onto the heating surface. After exchanging heat, the fluid is drained to the reservoir of spray chamber.

The above experimental procedure is applied for each test heater with enhanced surfaces. Then the experimental results are compared to obtain the effects of different surface structures on spray cooling heat transfer under the same experimental conditions.

![Fig. 1: Schematic of the experimental system](image)


2.2. Spray cooling chamber

In the entire system, the key component is the spray chamber, which is comprised of a height adjuster device, a sight window, a heat source block, and thermal insulation material (see Fig. 3 and Fig. 4).
In the entire system, the key component is the spray chamber, which is comprised of a height adjuster device, a sight window, a heat source block, and thermal insulation material (see Fig. 3 and Fig. 4).


Fig. 3: Schematic of the spray chamber

Fig. 4: Physical map of the experimental system
2.3. Enhanced heat surfaces

All the Enhanced heat face round surfaces have a radius of 12 mm and a basal area of 452 mm$^2$. To obtain the enhanced surface that experiment needs, the line cutting technique was used to generate different groove depths and widths on the top face of the copper heating block, and part of enhanced surface’s typical diagram are shown in Fig. 5 and all the detailed structure parameters are listed in Table 1. \(H\) and \(L\) are the groove depth and width, respectively.

![Diagram of the enhanced heat surfaces](image)

<table>
<thead>
<tr>
<th>Tested surfaces number</th>
<th>(L) [mm]</th>
<th>(H) [mm]</th>
<th>Wetted area [mm$^2$]</th>
<th>Area enhancement [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 1</td>
<td>0</td>
<td>0</td>
<td>452.16</td>
<td>0</td>
</tr>
<tr>
<td>No. 2</td>
<td>2</td>
<td>0.2</td>
<td>496.253</td>
<td>9.75</td>
</tr>
<tr>
<td>No. 3</td>
<td>2</td>
<td>0.4</td>
<td>540.34</td>
<td>19.5</td>
</tr>
<tr>
<td>No. 4</td>
<td>2</td>
<td>0.6</td>
<td>584.427</td>
<td>29.25</td>
</tr>
<tr>
<td>No. 5</td>
<td>2</td>
<td>0.8</td>
<td>628.53</td>
<td>39.0</td>
</tr>
<tr>
<td>No. 6</td>
<td>2</td>
<td>1.2</td>
<td>716.72</td>
<td>58.51</td>
</tr>
<tr>
<td>No. 7</td>
<td>2</td>
<td>1.4</td>
<td>760.807</td>
<td>68.26</td>
</tr>
<tr>
<td>No. 8</td>
<td>2</td>
<td>1.6</td>
<td>808.9</td>
<td>78.9</td>
</tr>
<tr>
<td>No. 9</td>
<td>1</td>
<td>0.8</td>
<td>689</td>
<td>52.38</td>
</tr>
<tr>
<td>No. 10</td>
<td>3</td>
<td>0.8</td>
<td>594.92</td>
<td>31.57</td>
</tr>
<tr>
<td>No. 11</td>
<td>4</td>
<td>0.8</td>
<td>501.21</td>
<td>25.5</td>
</tr>
</tbody>
</table>

2.4. Nozzle

The real photo of spray nozzle and jet projection is shown in Fig. 6, and the spray nozzle parameters are listed in Table 2. (Wang Yu et al. (2016)).

![Real photo of spray nozzle and jet projection](image)
3. Experimental data processing and uncertainty analysis

3.1. Data handling
During the spray process, tested surfaces are covered with water, so it is difficult to take accurate measurements on its surface. All the thermocouples were positioned below the tested surface to measure the temperature distribution, as shown in Fig. 7. The distance between thermocouple 1 and the tested surface is 17 mm. Thermocouples 2–4 are positioned below thermocouple 1, and the distance between the adjacent thermocouple is 8 mm.

Considering the actual heater structure, the whole copper except the top heating surface are surrounded by aluminum silicate fiber cotton. The heat conductivity coefficient of aluminum silicate fiber cotton is only 0.034W/m·K, and the heat conductivity coefficient of copper is 401W/m·K. One-dimensional heat transfer was demonstrated by simulation, as shown in Fig. 8.

Therefore, the main heat transfer mode is axial heat conduction. Thus, the one-dimension Fourier law is used along

### Table 2: Spray nozzle parameters

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8GG-SS1</td>
<td>1/8</td>
<td>1</td>
<td>0.79</td>
<td>200</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.62</td>
</tr>
</tbody>
</table>

Fig. 7: Thermocouples layout

Fig. 8: Temperature contours inside the heater section
the axial direction. Generally, the heat flux \( q \) (W/cm\(^2\)) is calculated by the temperature gradients on the axial direction according to the Fourier conduction low. It is described as follows:

\[
q = -\lambda \frac{\partial T(y)}{\partial y} = -\lambda \frac{T_{i+1} - T_i}{\Delta y}
\]

where \( T_i \) and \( T_{i+1} \) are the temperature of the adjoined two layers in the heater (°C), \( \Delta y \) is the distance between the adjacent two layers (mm), and \( \lambda \) is the thermal conductivity of heat copper [W/m·K].

The surface temperature \( T_w \) can be calculated by:

\[
T_w = T_4 - \frac{\partial T(y)}{\partial y} \cdot y_4
\]

where \( \frac{\partial T(y)}{\partial y} \) is the temperature line fitting, \( T_4 \) is the temperature of the fourth measuring point, and \( y_4 \) is the distance between the fourth temperature measuring point and heat surface (mm).

The heat transfer coefficient \( h \) can be calculated by:

\[
h = \frac{q}{(T_w - T_{in})}
\]

where \( T_w \) represents the heating surface temperature (°C), and \( T_{in} \) represents the water inlet temperature (°C).

### 3.2. Uncertainty analysis and calculation

The thermocouples and thermocouple reader used to gauge the temperature of the adjoined layers in the heater are calibrated with a \( K \) type thermocouple and the maximum uncertainty \( \delta T \) is ±0.8 °C. The spray water inlet temperature is captured by a PT100 platinum resistor, and the maximum uncertainty \( \delta T_{in} \) is ±0.15 °C. The maximum uncertainty of thermal gradient on the heat source \( \Delta T \) is ±0.6 °C. The distance between thermocouples is decided by processing technology, and the uncertainty \( \Delta y \) is ±0.1 mm.

Based on error transfer functions (Kline et al. (1953)) on this experiment bench, the uncertainty of heat flux, surface temperature and heat transfer coefficient can be expressed as follows:

\[
\frac{\delta q}{q} = \sqrt{\left(\frac{\delta \lambda}{\lambda}\right)^2 + \left(\frac{\delta \Delta T}{\Delta T}\right)^2 + \left(\frac{\delta \Delta y}{\Delta y}\right)^2}
\]

\[
\delta T_w = \sqrt{\delta T^2 + (\delta \Delta T)^2}
\]

\[
\frac{\delta h}{h} = \sqrt{\left(\frac{\delta q}{q}\right)^2 + \left(\frac{\delta T_w}{T_w - T_{in}}\right)^2 + \left(\frac{\delta T_{in}}{T_w - T_{in}}\right)^2}
\]

The error in the heat flux, surface temperature and heat transfer coefficient are calculated as ±5.5%, ±4.9% and ±2.8%, respectively.

### 4. Results and discussion

#### 4.1. Influence of groove width

The input power is adjusted to 1000 W, the spray cooling performances of the enhanced surfaces (No. 1, No. 5, No. 9, No. 10, and No. 11) are tested, and each surface corresponds to a groove width of 0 mm, 1 mm, 2 mm, 3 mm and 4 mm, respectively. The trends of the heat transfer coefficient and the surface temperature are shown in Figs. 9-10, respectively. For the case of the three-volume flow rate, the spray cooling performance of all enhanced surfaces is better than that of the smooth surfaces. We can also see from the figures that the enhanced surface heat transfer coefficient of enhanced surfaced decreases with the increase of groove width, and the corresponding surface temperature also increases.

In the process of spray, droplet impact continuous on groove surface. Both the capillary force and the droplet’s own gravitational force influence the droplet distribution. The droplets in the groove channel are presented with a curved crescent as shown in Fig. 11, which in turn affects the distribution of droplets on top of enhance surface. So the influence of capillary force is an important factor during the spray cooling especially for different structural surfaces.
When maintaining the same groove depth, the decrease in groove width increased the heat transfer area and provided a stronger capillary force for liquid spreading. To characterize the influence of the capillary force on the series of tested enhanced surfaces, a dimensionless parameter Bond number \((Bo)\) (Hsieh Cheng-Chieh et al. (2006)) is adopted as follows:

\[
Bo = \frac{L}{\sqrt{\sigma / \left( \rho_l - \rho_g \right) g}}
\]

(7)

where \(L = 1\, \text{mm} - 4\, \text{mm}, \sigma = 72.1 \times 10^{-3}\, \text{N/m}, \rho_l = 1000\, \text{kg/m}^3, \rho_g = 1.2\, \text{kg/m}^3.\)

The Bond numbers of the tested surfaces are shown in Table 3. An ordinary trend is that narrower grooves (smaller \(Bo\)’s) are more desirable in heat transfer because of the better liquid spreading ability (Hsieh Cheng-Chieh et al. (2006)).

Table 3: Bond number of the tested surfaces

<table>
<thead>
<tr>
<th>Tested surfaces number</th>
<th>(L) [mm]</th>
<th>(Bo)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 9</td>
<td>1</td>
<td>0.369</td>
</tr>
<tr>
<td>No. 5</td>
<td>2</td>
<td>0.738</td>
</tr>
<tr>
<td>No. 10</td>
<td>3</td>
<td>1.107</td>
</tr>
<tr>
<td>No. 11</td>
<td>4</td>
<td>1.476</td>
</tr>
</tbody>
</table>

According to Fig. 12, the heat transfer coefficient decreases with the increase in Bond number. Compared with the volume flow rate of 0.45 L/min, this trend is hardly changed at a volume flow rate of 1.25 L/min, indicating that the effect of surface tension is weakened. Increasing the volume flow rate helps to increase droplet velocity and diminish droplet diameter; therefore, during the spray process, more droplets accumulate to form a moving liquid film on the heater face. As more droplets begin crowding in, the film is rapidly swept away by fresh, cold water and becomes gradually turbulent, removing large amounts of heat flux due to substantial forced convection. Therefore, when the volume flow rate increased, the droplet velocity is an important factor that influences heat transfer, while the benefits of capillary force begin to diminish.
4.2. Influence of groove depth

The input power is adjusted to 1000 W, the spray cooling performances of the enhanced surfaces (No.1, No. 2, No. 3, No. 4, No. 5, No. 6, No. 7 and No. 8) are tested, and each surface corresponds to a groove width changed from 0 mm to 1.6 mm per interval of 0.2 mm. The trends of the groove depths impact on the heat transfer coefficient and the surface temperature are illustrated in Fig. 13-Fig. 14.

It can be concluded from Figs. 13 and 14 that the optimum groove depth that causes the surface to achieve the maximum heat transfer performance is 0.8 mm when the volumetric flow rate is 0.45 L/min and 0.75 L/min. The maximum heat transfer coefficient is 2.85 W/(cm²·K) when the volumetric flux is 0.45 L/min; the maximum heat transfer coefficient is 4.05 W/(cm²·K) when the volumetric flux is 0.75 L/min; the heat transfer coefficient increases by 18.75% over a smooth surface when the volumetric flux is 0.45 L/min, and the heat transfer coefficient increases by 26.5% over a smooth surface when the volumetric flux is 0.75 L/min. However, for the volumetric flux of 1.25 L/min (Fig. 13), the heat transfer coefficient continues to rise with increasing groove depth. The reasons for this phenomenon are then analyzed below.

Through the above analysis, both surface groove depth and flow rates effect heat transfer performance. The volume flow rate influences the spray droplet performance, so when the volume flow rate increases, the droplet average diameter and droplet velocity vary. Thus, spray droplet characteristics should be considered when discussing structured geometric enhancement mechanisms.

The average diameter of spray droplets $d_{32}$ is obtained from a correlation suggested by Visaria et al. (2008) as
follows:
\[
\frac{d_{32}}{d_0} = 3.07 \left( \frac{\rho_i^{0.8} \Delta p d_0^{1.5}}{\sigma^{0.3} u} \right)^{-0.259}
\]
(8)
where \(d_{32}\) is the Sauter diameter, \(d_0\) is the nozzle orifice diameter, \(\rho_i\) is the water density, \(\Delta p\) is the pressure drop across the nozzle, and \(\sigma\) and \(u\) are the surface tension and viscosity of the working fluid, respectively.

The droplet velocity \(u\) that comes out of the spray nozzle can be estimated (Ghodbane et al. (1991)) as
\[
u = (V_i^2 + \frac{2\Delta p}{\rho_i} - \frac{12\sigma}{\rho_i d_{32}})^{0.5}
\]
(9)
where \(V_i\) is the mean velocity of water entering the nozzle, and \(\rho_i\) is water density. The droplet initial velocity and average diameter are listed in Table 4.

<table>
<thead>
<tr>
<th>Volume flow rate [L/min]</th>
<th>Average diameter of spray droplets (d_{32}) [mm]</th>
<th>Initial velocity of droplets (u) [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.45</td>
<td>0.15052</td>
<td>30.15</td>
</tr>
<tr>
<td>0.75</td>
<td>0.13187</td>
<td>38.73</td>
</tr>
<tr>
<td>1.25</td>
<td>0.11553</td>
<td>50.22</td>
</tr>
</tbody>
</table>

By calculation and analysis, we found that the larger the volume flow rate, the smaller the droplet diameter and the greater the initial velocity of the droplet. Therefore, the velocity of the droplet arriving on the surface must be different from the values which are listed in Table 4. These calculations about initial diameter and velocity of droplet will provide support for subsequent calculations.

The nozzle to top surface distance is 17 mm. When a droplet comes out of the nozzle, it must move nearly 17 mm until it reaches the top surface and then continue to move to the groove bottom of the heated block. In all moving period, the droplet interacts with an ambient medium. The ambient medium is air before the dividing point (First region) and the ambient medium is water after the dividing point (Second region). The second region stands for the groove channel space (Fig. 15).

![Fig. 15: Schematic of movement of droplets in various region](image-url)

Take a single drop of liquid as a research object. After leaving the nozzle, the drag force \(F_D\), buoyancy \(f\), and gravity force \(F_g\) to which a globule droplet is subjected can be expressed as (Zhang Wei et al. (2013)):
\[
F_D = C_D \frac{\pi d_{32}^2}{4} \rho \frac{u^2}{2}
\]
(10)
\[
f = \frac{1}{6} \pi d_{32}^3 \rho g
\]
(11)
Take a single drop of liquid as a research object. After leaving the nozzle, the drag force to which a globose droplet is subjected can be expressed as (Zhang Wei et al. (2013)):

\[ F_D = \frac{1}{6} \pi d_{32}^3 \rho g \]  

(12)

In Eqs. (10)-(12), \( C_D \) is the drag coefficient, \( u \) is the droplet velocity, \( d_{32} \) is the droplet Sauter diameter, \( \rho \) and \( \rho_l \) are the medium density and water density, respectively. \( C_D \) depends on the Reynolds’s number, and a simple correlation in a broad range of values of \( Re \) (Ciofalo et al. (2007)) is:

\[ C_D = \frac{24}{Re} (1 + 0.048 Re) \]  

(13)

The droplet acceleration \( a \) can be described as:

\[ a = \frac{F_D + f - F_g}{m} = \frac{3}{4} \left( \frac{24y}{ud} + 0.048 \right) \frac{\rho}{\rho_l} u^2 + \frac{1}{d} + \frac{\rho}{\rho_l} g - g \]  

(14)

In Eq. (14), the droplet acceleration \( a \) is upward. By numerical calculation, when the spray volume flow rate is 0.45 L/min, the droplet moves downward across the air, the velocity of the droplet out of the nozzle is 30.15 m/s, and the average diameter of the droplet is 0.15052 mm. The value of the first item is 317.4 m/s², while the other two items are 0.012642 m/s² and 9.8 m/s², respectively. As we can see, the value of the first item is much larger than the values of the other items, so only the value of the first item is considered in the calculation. Although droplet acceleration is large as the droplet moves only 17 mm to the top enhancement heat face, the droplet velocity in the first stage decreases little. At the second stage, the ambient medium is turned into water, and the droplet acceleration is 1,856.21 m/s², which is much larger than that in the air. As already indicated, groove spaces are filled with water because the groove walls prevent water from leaving the spaces. Therefore, when the droplets ultimately reach the groove bottom, the droplet velocity attenuation is quick.

The residual velocity trends of droplet over moving distances are plotted in Fig. 16-Fig. 17. The droplet velocity change trend dividing point occurs on top of the heat block. It can be viewed from Fig. 16 and 17 that the droplet velocity decreases slowly with moving distance until a dividing point, and the velocity then decreases rapidly. The first stage is from the nozzle to the top of the heat block, and the second stage is the groove space, which is fully filled with water. Therefore, the droplets pass through the first stage and then impact the water in the groove until reaching the bottom of groove. The motion resistance to which the droplet is subjected is larger in the water than in the air, because water density is far greater than that of air. Hence, the droplet velocity decreases more quickly in the second stage.

Fig. 16: Residual velocity of droplet before the dividing point  Fig. 17: Residual velocity of droplet after the dividing point

The droplet velocity reaches the top and bottom of every enhanced surface, as shown in Table 5. The residual droplet velocity is 9.52 m/s when spray volume flow rate is 1.25 L/min. However, the spray volume flow rates are 0.45 L/min and 0.75 L/min, and the impingement velocity of the droplets on groove bottom are reduced by 98.24% and 94.18%, respectively, to approximately zero.

After obtaining the residual velocity of the droplet, the enhanced heat transfer mechanisms of enhanced surfaces under different volume flow rates are discussed.

As shown in Fig. 18, the enhanced surfaces include the top surface, the sidewall and the bottom surface of the grooves. Correspondingly, the heat flux of the tested surfaces can be generally divided into the following three parts: heat
The relationship between $q$ and groove depth is more complicated. Originally, the wetting area increases with groove depth; however, when the groove is too deep, the power of the fluid flow nearly equals zero. The convective heat transfer coefficient on the sidewall decreases. Therefore, $q_{\text{side}}$ does not further increase.

Comprehensive consideration is paid to the influence of groove depth on $q_{\text{bottom}}$ and $q_{\text{side}}$; therefore, although the effective wetted area of the enhanced surface is increased following the groove depth, the heat transfer performance decreases after a depth of 0.8 mm for the spray volume flow 0.45 L/min. Meanwhile, the heat transfer performance increases following the groove depth at the spray volume flow 1.25 L/min.

The residual velocity at 0.8 mm for 0.75L/min is greater than that at 1.6 mm for 1.25 L/min (see Fig.17) where the heat transfer performance is still enhanced as shown in Fig.13 and Fig.14. The reason for this phenomenon is that compared with 0.75L/min, more droplets are participated in heat exchange with the flow rate of 1.25L/min. Both the droplet numbers and droplet residual velocity influence heat transfer.

### Table 5: Droplet velocity at different stages

<table>
<thead>
<tr>
<th>Volume flow rate [L/min]</th>
<th>$u_{\text{top}}$ [m/s]</th>
<th>$u_{\text{bottom}}$ [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.45</td>
<td>28.2</td>
<td>0.51</td>
</tr>
<tr>
<td>0.75</td>
<td>37.5</td>
<td>2.15</td>
</tr>
<tr>
<td>1.25</td>
<td>49.1</td>
<td>9.52</td>
</tr>
</tbody>
</table>

Fig. 18: Heat transfer components of enhanced surfaces
4.3. Experimental non-dimensional criteria equations

The elementary non-dimensional numbers of the heat transfer performance Reynolds number $Re$, is also vital factors of spray cooling (Zhou et al. (2017)). Through the analysis of the above experimental results, some additional influence factors must be taken into account as follows: the ratio of the groove depth and width $\beta$ and the influence of evaporation intensity $\zeta$. The Weber number $We$ is a non-dimensional number that represents atomized droplet characteristics; the Bond number $Bo$ is a non-dimensional number that represents the liquid spreading ability on surface. The above factors are important in the non-dimensional of Nusselt number $Nu$ fitting, which can be defined by:

$$We = \frac{\mu u^2 d_{32}^2}{\sigma}$$  \hspace{1cm} (16)

$$Re = \frac{G_m d}{\mu}$$  \hspace{1cm} (17)

$$\zeta = \frac{T_w - T_\infty}{T_{sat}}$$  \hspace{1cm} (18)

$$\beta = \frac{H}{L}$$  \hspace{1cm} (19)

The heat transfer coefficient can be presented as:

$$h = \frac{A}{D} Nu(Re, We, Bo, \zeta, \beta)$$  \hspace{1cm} (20)

Comprehensively considering the influence of $Re$, $We$, $Bo$, $\delta$ and $\beta$, the non-dimensional criteria equations are:

$$Nu = 7.1389Re^{-0.0118} We^{-0.083} Bo^{0.556} \beta^{0.1124} \zeta^{0.319}$$  \hspace{1cm} (21)

The application conditions of correlation are as follows: $Re = 356.8 - 936.1$; $We = 832.45 - 3268.6$; $\zeta = 0.21 - 0.76$; $Bo = 0.369 - 1.476$; $\beta = 0.1 - 2.0$; $H = 0$ mm$-1.6$ mm; $L = 1$ mm$-4$ mm. Fig.19 shows that 95% of the experimental data are predicted within a mistake band of $\pm15\%$.

![Graph of Nu vs. Nu](image)

Fig. 19: $Nu$ (Experiment) vs. $Nu$ (Correlation)

In realistic application, when the computational precision is not high, Eq. (21) can be used for enhancing the heat transfer performance prediction.

5. Conclusions

Experiments were conducted to study the effects of volume flow rate and enhanced surfaces on the heat transfer of cooling water spray in an open loop system. The following conclusions can be drawn:

(1) The experimental results show that increasing only the groove depth of a heat surface will not effectively improve heat transfer performance in spray cooling for low flow rates. The optimal groove depth is 0.8 mm when the volume...
flow rate is 0.45 L/min and the heat transfer coefficient is 2.75 W/(cm²·K), which is enhanced by approximately 30.95% relative to that of a flat surface.

(2) By analyzing forces acting on a droplet, the residual droplet velocity at different locations between the nozzle and the surface is obtained. The results indicate that the residual velocity is much smaller for the volume flow rate of 0.45 L/min than the volume flow rate of 1.25 L/min, which indicates that the liquid in the grooves cannot be removed quickly. Thus, the bottom surface of grooves and the sidewalls have relatively lower heat transfer performance. That is the reason that surface with a groove depth of 0.8 mm achieves the largest heat transfer enhancement for a volume flow rate of 0.45 L/min.

(3) While for a volume flow rate of 1.25 L/min, the heat transfer enhancement always improves with groove depth because the droplet residual velocity is much larger at the volume flow rate of 0.45 L/min, so the bottom surface of grooves and the sidewalls heat transfer performance $q_{\text{bottom}}$ and $q_{\text{side}}$ are improved.

(4) For volume flow rates of 0.45 L/min and 1.25 L/min, the heat transfer coefficient decreases with the increase in Bond number. This trend changed slowly at high volume flow rate. Therefore, when the volume flow rate increased, the droplet velocity and wetting area influenced heat transfer, while the benefits of the capillary forces began to diminish.

(5) An experimental non-dimensional correlation of the Nusselt number $N_u$ is presented with water as the coolant considering the influence of the evaporation intensity, Bond number, Weber number and the ratio of the groove depth and width.

References


