THE EFFECTS OF COVERAGE AREA ON THE SPRAY COOLING HEAT TRANSFER PERFORMANCE

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ABSTRACT

An experimental investigation was performed to study the effects of the spray coverage area, the film area, and the impact droplet velocity on the thermal performance of a spray cooling system. All experiments were carried out in a closed loop spray cooling system working with deionized water as a cooling medium. A plain copper surface with a diameter of 15 mm was used as a target surface and tested in the single-phase and two-phase regimes. The coolant mass flux was 15.5 kg/s.m², and the distance between the nozzle and the heated surface was varied from 10 to 16 mm. Due to the change in the nozzle height, the spray coverage area, the film area, and the droplet impact velocity were changed and affected the thermal performance of the spray cooling system. As a sequence of these changes, the experimental results showed that these parameters have a significant effect on the heat transfer characteristics.

KEYWORDS: Spray cooling, Heat flux, Coverage area, Sauter mean diameter.

1 INTRODUCTION

High heat flux cooling schemes have received more attention in recent years due to the rapid miniaturization and integration of electronic components, which increased the heat density [1-2]. Spray cooling is one of the cooling techniques that was introduced to solve this problem in many engineering applications because it is a very effective technique and has the best balance heat flux removing capability, isothermality, and fluid inventory [3]. In recent years, numerous investigations have been performed on the parameters that influence spray cooling performance. Chen et al. [4], [5] experimentally studied the effects of spray parameters (mean droplet size, droplet flux, and droplet velocity) on critical heat flux (CHF) and the heat transfer characteristics. The results showed that the mean droplet velocity (V) had the most dominant effect on CHF and the heat transfer coefficient, followed by the mean droplet flux (N). The Sauter mean diameter (d₃₂) did not affect CHF. Also, increasing the droplet velocity increases both the CHF and the heat transfer coefficient. Furthermore, Wang et al. [1] studied the way in which the spray inclination angle, the mass flux, and the surface temperature influenced the spray cooling heat transfer performance in the non-boiling regime. A swirl semisolid nozzle was used with the mass flux of 15.7 kg/m²·s, 18.1 kg/m²·s and 24.9 kg/m²·s respectively, and the spraying parameters were measured by a Dantec Phase Doppler Particle Analyzer (PDPA). The results indicated that the inclination angle and the film evaporation had a crucial effect on the spray cooling heat transfer performance in the non-boiling regime. Increasing the surface temperature increased film evaporation and enhanced heat transfer performance. Moreover, Cheng et al. [6] experimentally studied the effects of spray characteristics on the spray cooling heat transfer in a system working with distilled water as a working fluid. The study was conducted to investigate the influence of spray flow rate, spray height, and coolant inlet temperature on spray cooling performance in the single-phase regime, and also, to visualize and correlate the spray characteristics at various operating conditions. The results showed that the axial droplet velocity, Sauter mean diameter (SMD), and the droplet numbers density were greater in the mainstream than in the other

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regions. The droplet velocity and SMD were correlated with a mean absolute error of 15%. Also, the heat transfer was correlated with a mean absolute error of 7%.

In addition to the previously mentioned studies, Tao et al. [7] also experimentally studied the effects of the volumetric flow rate, the nozzle height, and coolant inlet temperature on the spray cooling heat transfer performance in the single-phase region. The experiments were conducted in an open loop test system working with deionized water as a coolant. The results indicated that increasing the volumetric flow rate or reducing the coolant inlet temperature improves the heat transfer performance. Also, adding surfactants to working fluid with an appropriate concentration enhances the heat transfer performance. Xie et al. [8] used Particle Image Velocimetry (PIV) and Phase Doppler Interferometry (PDI) to investigate the thermal effects on the spray cone formation. Water was used as a working fluid in this system, and the nozzle inlet pressure and the surface temperature were varied to study their influence on the spray cone. The experimental results showed that the surface temperature has a significant effect on the spray cone formation, whereas the spray cone expands at high surface temperature. Also, increasing the surface temperature increases the diameter of the droplets near the surface, but decreases its velocity. Moreover, Xu et al. [9] experimentally investigated the heat transfer enhancement in a spray cooling system integrated with a refrigeration system utilizing isobutene (R600a) as a working fluid in a closed loop system. The experimental results showed that the mass flow rate is the main influential parameter on heat transfer characteristics, such as the surface temperature, heat flux, and heat transfer coefficient, whereas, the surface temperature uniformity depends on the nozzle inlet pressure and heat flux in addition to the mass flow rate. Hou et al. [10] investigated the spray cooling characteristics of R134a at different volumetric flow rates. The results showed that increasing the volumetric flow rate enhances the critical heat flux because it keeps the surface wet and prevents the dry out phenomenon. The maximum achieved critical heat flux was 117.2 W/cm² at a surface temperature of 319 K and a volumetric flow rate of 0.356 L/min. Recently, Zhou et al. [11] experimentally studied the influences of the spraying parameters such as spray height, heat flux, inlet pressure, and gravitational angle on spray cooling performance to find the most influential parameter. The results indicated that the mass flow rate was the main influential parameter on the spray cooling performance, where increasing the flow rate improves the heat transfer. Also, it was found that the best heat transfer was achieved when the gravitational angle was between 30° and 120°, whereas the worst was 180°.

It is known that changing the distance between the nozzle and the target surface changes the coverage area and the droplet momentum at the same time. Therefore, in the present study, the effects of the spraying coverage area and droplet momentum on the heat transfer performance was investigated to determine the most influential parameter on the spray cooling performance surface temperature uniformity.

2 EXPERIMENTAL SETUP AND PROCEDURE

2.1 Test Facility

Figure 1 shows a schematic diagram of a closed loop spray cooling system utilizing deionized water as a working fluid, which was designed and built to perform the experiments. The system mainly consists of a spraying system, a testing chamber, a heating system, a data acquisition system, and a cooling system. The experimental results of this setup were compared with previous work for validation; the comparison showed that the results of this set up are valid as shown in details in [12].

2.2 Spray System

In the spraying system, the coolant was driven from the tank by a positive displacement bypass diaphragm pump, which could supply a maximum pressure of 60 psi and a maximum flow rate of 1.4 GPM. First, the pumped deionized water passed through a filter, which removed all impurities. Then, a positive displacement flow meter, a coaxial heat exchanger to control the inlet temperature, a pressure transducer, and a temperature transducer, which were placed before the nozzle’s inlet.

2.3 Heater Assembly

The target surface was heated by a 1000 W cartridge heater, which was inserted in a super-conductive 101 copper block, as shown in Figure 2. A stainless-steel pipe with inner and outer diameters of 4.8 cm and 6 cm respectively surrounded the copper block. The gap between the copper block and the pipe was filled with fiberglass thermal insulation to reduce heat losses to the environment. All sides of the copper block were insulated except the top side (the test surface), which exposed to the spray nozzle, and integrated into the spray chamber.
Figure 1 Schematic diagram of the experimental setup.

Figure 2 Schematic diagram of the heater assembly.
2.4 Experimental Procedure

The deionized water was driven from the water tank by a positive displacement pump, and water was filtered through a strainer to remove all impurities from the working fluid. Before each experiment, the target surface was cleaned by Nitric acid solution with a concentration of 32.5%, which was bought from Sigma Aldrich, then rinsed with deionized water to remove the copper oxide layer. In the beginning, the pump was turned on for 25 minutes to let the flow rate, the pressures, and the temperatures of the system reach the steady-state condition. Also, the cooling system (chiller) was turned on, which used to control the water inlet temperature. Then the heat was supplied to the system by using a variable transformer (Variac), and the power was increased gradually with small increments to avoid the burning out problem and reach the steady-state condition quickly. When the system reached the steady state condition, the data was recorded for 30 minutes. The same steps were repeated in all experiments to investigate the effects of different parameters, the distance between the nozzle and the test surface, the spray coverage area, and the liquid film area.

3 DATA ANALYSIS

3.1 Data Reduction

Eight thermocouples were embedded below the target surface to measure the temperature gradients. The thermocouples (T₁, T₂, T₃, &T₄) and (T₅, T₆, T₇, &T₈) were positioned below the target surface at 4mm and 8mm respectively as shown in Figure 3.

![Figure 3 CAD view of shows the thermocouples positions on the copper block.](image)

Since the copper has a high thermal conductivity and is well insulated from all sides except the top surface, one-dimensional heat conduction was assumed to calculate the heat by using Fourier’s law, as shown in the following equation:

\[ q^* = -k \frac{dT}{dy} \] (1)

\( T_A \) is the average of four thermocouples (T₁, T₂, T₃, &T₄), and \( T_B \) is the average of four thermocouples (T₅, T₆, T₇, &T₈). Also, the surface temperature was calculated based on Fourier’s law by using the following equation:

\[ T_S = T_A - \frac{ka^*}{L} \] (2)

The heat losses to the environment due to the effectiveness of the thermal insulation was calculated by the using the following equation:

\[ Heat \ losses = \left( \frac{Q_{in} - q^* A}{Q_{in}} \right) \] (3)

The input power was calculated by using the below equation:

\[ Q_{in} = I \cdot V \] (4)
The calculations showed that increases with the increase of the input power and the average percentage of heat losses were 11%.

Also, Weber number is defined as the ratio of inertial force to surface force, and the droplet diameter was calculated, based on the Sauter mean diameter (SMD), from the following equations [13]:

\[
We_{d32} = \frac{\rho_l u^2 d_{32}}{\sigma}
\]  

(5)

Where the Sauter mean diameter was calculated by using the following correlation [14]:

\[
d_{32} \frac{d_i}{d_o} = 3.07 \left( \frac{\rho^{0.5}_a \Delta \rho_d^{1.5}}{\sigma^{0.5} \mu_l} \right)^{-0.259}
\]  

(6)

Also, the break up velocity at the nozzle’s outlet was calculated from the following equation [13]

\[
V_z = \sqrt{V_1^2 + \frac{2 \Delta P}{\rho_l} - \frac{12 \sigma}{\rho d_{32}}}
\]  

(7)

The droplet is affected by external forces before hitting the target surface as illustrated in Figure 4. These forces change the impact droplet velocity, which has a significant effect on the spray cooling heat transfer performance. The droplet was assumed to be a sphere, and the buoyancy and the gravity forces were calculated from the following equations[15]:

\[
f = \frac{1}{6} \pi d_{32}^3 \rho_a g
\]  

(8)

\[
G = \frac{1}{6} \pi d_{32}^3 \rho_l g
\]  

(9)

Also, the drag force was calculated by an equation proposed by Ciofalo et al. [16]:

\[
F_D = C_D \frac{\pi d_{32}^3}{4} \rho \frac{u^2}{2}
\]  

(10)

Where the drag coefficient \(C_D\) was calculated from the following correlation:

\[
C_D = \frac{24}{Re_a} \left( 1 + 0.02 Re_a \right)
\]  

(11)

\[
Re_a = \frac{ud_{32}}{\dot{\theta}_a}
\]

Based on the forces balance, the acceleration of a single droplet can be calculated by using Newton’s second law.
\[ \ddot{a} = \frac{f + G + F_D}{m} \]  

(12)

The droplets were assumed to be in rectilinear motion with variable acceleration. Therefore, the local droplet velocity calculated from the flowing equation [17]:

\[ v \, dv = a \, ds \]  

(13)

By integrating the above equation, the droplet local velocity can be calculated from the following equation:

\[ u = \sqrt{v^2 + 2 \ddot{a} H} \]  

(14)

The above equations were used to calculate the droplet impact velocity, which was assumed the local droplet velocity at 99.99% of the distance between the nozzle and the target surface.

3.2 Uncertainty Analysis

Since the heat flux and the surface temperature were calculated by using Fourier’s law and based on experimental data. Some errors were introduced to both quantities resulting from the errors of the thermocouples and the thermal conductivity of the copper. Therefore, the uncertainty of these quantities was calculated by using the following equation [18]:

\[ q = f(x_1, x_2, x_3, ..., x_n) \rightarrow U(q) = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial q}{\partial x_i} U(x_i) \right)^2} \]  

(15)

The maximum uncertainties of the thermocouples were ± 0.4%, and the thermal conductivity uncertainty was ± 0.05 W/m. K [19]. The calculations showed that maximum uncertainties of the heat flux and the surface temperature were ± 7.5%, and ± 0.4 % respectively.

4 RESULTS AND DISCUSSION

Figure 5 illustrates the effect of the distance between the nozzle and the target surface on the heat flux at constant mass flux. It is found that increasing the distance between the nozzle and the target surface decrease the heat flux significantly. Resulting from increasing the spray coverage area and decreasing both the liquid film area and the droplets momentum.
Figure 6 shows that the relation between the spray coverage area and nozzle height, which affect the thermal performance of a spray cooling system. Increasing the distance between the nozzle and the target surface increases the spray coverage area. While, decreasing the liquid film area, which plays a significant role in the spray cooling thermal performance.

Figure 7 illustrates the relation between the nozzle height and Weber number. It is indicated that the average Weber number decreases with the increase in the distance between the nozzle and the target surface due to the increase of the effects of forces exerted on a droplet. Where, the effect of forces increases with the increase of droplet’s path.

Figure 6 the relationship between the nozzle height and the percentage of the coverage area.

Figure 7 the relationship between the nozzle height and the average Weber number.
Also, the effect of average Weber number on the heat flux was investigated, and the results indicated that the heat flux increased with increasing Weber number as shown in Figure 8. Due to the increasing in the droplets momentum and creating more turbulence on the target surface in the liquid film area.

![Figure 8](image)

Figure 8 the effect of Weber number on the heat flux at a temperature difference of 45K.

5 CONCLUSIONS
The present study focused on the effect of spray coverage area on the spray cooling heat transfer performance at a constant volumetric flow rate of 165 ml/min. A plain copper surface was tested with varying distances between the nozzle and the target surface, where the distance ranged between 10-16 mm. The following conclusions can be concluded:

1. Heat flux decreases with the increase in the distance between the nozzle and the target surface.
2. The droplet Weber number has more effect than the spray coverage area on the heat transfer performance.
3. Weber number decreases with the increase in the distance between the nozzle and the target surface.
4. Increasing both Weber number and the liquid film area improve the spray cooling thermal performance.

ACKNOWLEDGMENT
This work was supported by the Office of Naval Research under Grant N00014-14-1-0165. Therefore, the authors would like to thank the Office of Naval Research, the Department of Mechanical Engineering at the University of South Carolina, and the Higher Committee for Education Development in Iraq (HCED) for their support.
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