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Non-Boiling Heat Transfer Characteristics of Water Sprays Impinging on a Heated Surface

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Abstract
An experimental study has been carried out to investigate the non-boiling heat transfer characteristics of water spray impinging on a heated surface. The spray was generated by using full-cone commercial water spray nozzles of different diameters. The heat transfer characteristics of water spray were investigated at fixed surface temperature boundary condition (50ºC). Experiments were conducted to investigate the effect of mass flux of spraying water on heat transfer coefficient. It was found that heat transfer increases with increase in mass flux because of increase in liquid velocity resulting in increase in perturbation of thin liquid layer due to impinging droplets. Nozzles having smaller orifice diameter show higher in heat transfer coefficient because of combined effect of evaporation and forced convection. A strong splashing and droplet rebounding occur in case of smaller orifice diameter nozzle at higher mass fluxes. In case of large orifice diameter nozzle, a thicker liquid film is developed over the heated surface and heat transfer is due to turbulence created by striking droplets on this thick-film. Nozzles having larger orifice diameter show less dependence on evaporation and whole the heat transfer process was due to forced convection. Finally, non-dimensional correlations were developed for Reynolds number and Nusselt number for local heat transfer coefficient.

Keywords: Water spray, nozzle, heat transfer, non-boiling, Reynolds number, mass flux, sepration distance

I. Introduction

The need for higher heat flux cooling techniques is driven considerably by advancements in microelectronics and semiconductor industry. According to Moore's Law [1], continuing advances in semiconductor industry allow the device feature size to shrink and the transistor density and switching speed to double every one and half to two years. The heat dissipation from chip increases in same proportion if there is no change in semiconductor technology. As number functions are increased and devices shrunk, heat density increases. High heat fluxes created by high performance electronics have offered great challenges to the engineering community. Advanced liquid immersion cooling techniques such as boiling and spray cooling are particularly effective for addressing these types of high heat density problems. Two-phase systems utilizing boiling or liquid evaporation have long been recognized as having the potential to remove large amounts of heat at low temperature difference. But on the other hand spray cooling, in which an atomizing nozzle provides a flow of liquid droplets directed at a hot surface in non-boiling regime, provides an excellent option for tackling with high heat transfer rate requirements. The primary disadvantages of spray cooling systems include large weight, cost, and complexity. However, attainment of exceptionally high heat transfer rates has made this technique a still lucrative one compared to other single phase or even two-phase systems. Spray cooling has wide range of applications such as solar panel cooling, fuel cells, electric vehicles, electronic devices, high power electronics and the building sector [2-3].

Extensive literature is available covering various aspects of spray impingement cooling where majority of studies are done in boiling heat transfer regime [4-5]. Very few numbers of studies are available in single phase spray cooling. Kim [6] reviewed the spray cooling mechanisms and outlined the areas where additional research is needed in electronic cooling. Major reasons for high heat fluxes in the spray cooling are thinning of thermal boundary layer at large fluid flows. Further the impact of the droplets onto the film can also agitate the liquid making thinning of thermal boundary layer locally.

Numbers of experimental studies have been carried out to investigate the effect of mass flux of coolant on heat transfer characteristics [4, 7-13]. Dense sprays have low evaporation tendency than dilute sprays and CHF increases with increase in mass flux and also with increase in sub-cooling. Other parameters which determine the heat transfer such as surface roughness, the droplet size and droplet velocity are of less importance in direct contact heat transfer where spray density are high enough to have considerable interaction between droplets. Spray impingement cooling can provide the same heat transfer as jets at a significantly lower liquid mass flux because of combined effect of evaporative cooling from the film along the impingement surface and the unsteady thermal boundary layer expected in spray impingement [8, 11]. Cheng et al. [13] noticed that lighter spray has higher efficiency than dense spray however its CHF is higher than dilute spray. The other important parameter which affects the spray impingement heat transfer is injection pressure. Heat flux decreases when the injection pressure increases [14]. Excessively high nozzle inlet pressure after the optimal value has no contribution to the increase of CHF but increase the coolant consumption [15].

Heat transfer during spray impingement cooling strongly depends upon the droplet diameter. Smaller the droplet diameter higher will be the heat transfer coefficient [4, 7]. But there are contrary studies also available like, Ciofalo et al. [16] noticed that heat transfer coefficient and maximum heat fluxes was dependent on mass flux and mean droplet velocity and have negligible effect on droplet size independently. Spray cooling heat transfer can also be enhanced by altering the spraying height [13]. Tay and Sivanand [17] showed that decreasing the chip to nozzle height increases the heat transfer coefficient until the impact area is not reduced to minimum value. Experiments conducted by Wang et al. [12] revealed that there was an optimal orifice-to-surface distance where heat transfer performance is best.

Xia et al. [18] mentioned that a cooling non-uniformity prevails in case of spray impingement on large surfaces. Different methods were explored to attend cooling non-uniformity (CNU) at the impingement surface like, changing the separation distance, impingement angle and by adding liquid jet at the center of the spray. Yang et al. [19] found that increasing the spray angle contributes to improving uniformity of surface temperature distribution, surface equivalence stress distribution and reduction in maximum surface equivalent stress. Silk et al. [20] also investigated the effects of enhanced surfaces and spray inclination angle on heat transfer during spray cooling. Schwarzkof et al. [21] noticed that the cooling capability of spray dropped off significantly when the angle exceeded 40º. Sahu et al. [22] found that impinging spray jet on the LED module maintains its efficacy even the power supply exceeds 112%. Kansy et al. [23] studied experimentally the heat transfer from the impinging jet and the spray on a vertical surface and found that heat transfer to the surface in case of solid cone is greatly influenced by the pressure compared to the separation distance.

Cooling effect can be improved by adding surfactant with an appropriate concentration and it plays an important role in the spray cooling performance. Jia and Qiu [24] found that surfactant addition provides additional safety for heat transfer device to avoid burnout. Ravikumar et al. [25] added different types of surfactants at various concentration levels to air atomized water spray for enhancement of ultrafast cooling rate. Lin et al. [26] mentioned that surface wettability influenced heat transfer performance during spray cooling. In the non-boiling regime, the surface with higher hydrophilicity exhibited higher heat transfer performance owing to the larger wetting area available for heat transfer. Many studies focus on enhancement of heat transfer with change in surface characteristics (surfaces with micro-structures and the surface roughness) and compared the results with corresponding flat surface [27-28]. Xie et al. [29] used enhanced surfaces including micro-, macro- and multiscale-structured shapes along with a referenced smooth flat surface were tested in a closed loop system with R134a as the working fluid.

Literature review revealed that spray impingement cooling is a viable solution to do with the situations where cooling of very high heating densities are involved. In the present study an investigation has been carried out to evaluate the heat transfer characteristics of commercial spray nozzles (Make: Spraying Systems Inc.). Effect of change mass flux and nozzle diameter has been evaluated for three different types of nozzles TG0.3, TG0.4 and TG0.5. Heated surface temperature was kept constant to 50ºC.

II. EXPERIMENTAL FACILITY

Figure 1 shows the detailed schematic of experimental setup. The experimental setup consists broadly of two sections: heat generation part i.e., impingement surface (heat flux calorimeter) and heat removal part i.e., water spray generation part. Spray is generated with specially designed commercial spray nozzles (Make: Spraying Systems Inc.). Sufficient pressure within the line is required for generating the spray at the nozzle and for this purpose a reciprocating plunger pump has been used. Pressure transmitter is used for measuring and controlling the pressure in the line. To desist any impurities present and also to avoid clogging of the dust particles a micron filter is put in the water line before the nozzle. The micron filter has ability to arrest the particle sizes up to 5 microns. As plunger pump is put in use for generating spray the flow coming out the pump is pulsating. In order to minimize the pulsation of water flow through the nozzle, a pulsation damper is installed.

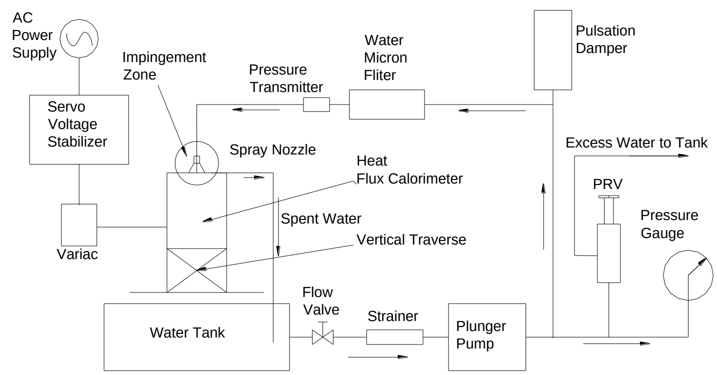

Figure 1 Schematic diagram of experimental setup.

Figure 2 Direct photograph showing the impingement zone during spray impingement.

To achieve the variable mass flux of water coming out from the nozzle, pressure in the water line was changed by adjusting the pressure through the pressure relief valve (PRV) and set to desire value by noticing it from pressure transmitter display. An additional dial type pressure gauge is installed in the water line to cross check the pressure in the line. A large size water storage tank is installed to feed the water to the pump. A strainer and flow control valve were put in between the water tank and plunger pump to arrest any bigger particles to enter into the pump. Storage tank also collects the excess water coming out from the pressure relief valve and the spent water coming out from the calorimeter. Other major section of the spray impingement test rig is heat generation part. It consists of target surface with arrangement of heating the target surface with controlled value of the heat flux. The entire unit in the present study is referred as calorimeter. Figure 2 shows the direct photograph of close-up view of the impingement zone.

III. EXPERIMENTAL PROCEDURE

Before conducting every experiment, the target surface was cleaned properly with emery paper followed by cleaning with acetone to produce a fresh oxide free surface which improves the wetting ability of the surface. The pump was kept running for some time without the nozzle to allow the water to flow through the piping before each experiment. This help in removal of any loose particle present in the piping which could be the cause of clogging of nozzles if not removed. After this nozzle was attached at the nozzle holder and spray was initiated. The separation distance between the nozzle and the target surface is adjusted so that the whole spray cone covers the complete target surface. Then the heater supply was turned on and supply to the heaters was adjusted in such a way that temperature of the target surface was attained to 50ºC. All the temperature readings were noted under the steady state conditions which correspond to no change in thermocouple readings for period of 15-30 minutes. The steady state for each run was obtained in about 4-5 hours. Experiments were conducted for three different nozzles: TG0.5 (d = 610 µm), TG0.4 (d = 559 µm) and TG0.3 (d = 508 µm) at various mass flux rates. For each nozzle ten readings were taken at ten different mass fluxes. The mass flux to the target was varied by changing the injection pressure to the nozzle. To check the repeatability of results, three test runs were conducted under fixed operating conditions. The maximum and minimum standard deviation in heat transfer coefficient from average value of heat transfer coefficient was found to be 4.75% and 1.00% respectively. The average value of standard deviation from average value of heat transfer coefficient was found to be 2.60%. In present experimental study the heat flux and surface temperature was measured by using J-type thermocouple wire. The uncertainty in temperature measurement was ±1ºC. The uncertainty in variation of mass flow rate from pump is 5% as described by manufacturing company.

I. DATA REDUCTION

Heat flux to the impingement surface is evaluated using heat conduction through the copper block using Fourier's equation (Eqn. 1).

$$q'' = k\frac{\Delta T}{\Delta x} \tag{1}$$

Heat transfer coefficient (h) was evaluated using Newton's law of cooling as given in Eqn. 2 using T_f as spray fluid temperature.

$$h = \frac{q''}{\left(T_s - T_f\right)} \tag{2}$$

Knowing the value of heat transfer coefficient Nusselt number (Nu) is calculated using Eqn. 3 with D as diameter of the heated target surface.

$$Nu = \frac{hD}{k_f} \tag{3}$$

Here k_f if thermal conductivity of the water at bulk fluid temperature. The spray Reynolds number is calculated using Eqn. 4 with G as mass flux of water spray and μ is coefficient of viscosity of water at mean fluid temperature.

$$\mathrm{Re} = \frac{GD}{\mu} \tag{4}$$

II. RESULTS AND DISCUSSION

Heat Transfer Characteristics

Figure 3 shows the dependence of heat transfer coefficient and heat flux on the mass flux of water spray impinging on the target surface for nozzle TG0.3 (d = 508 µm). It has been observed that with increase in the mass flux of the water, the heat flux as well heat transfer coefficient is increasing continuously. A larger fluid flows or higher liquid velocities result in thinning of boundary layer over the surface. Also, the impact of the droplets onto the thin layer agitates the liquid on the impingement surface [6]. This in combination gives significant enhancement in the heat transfer.

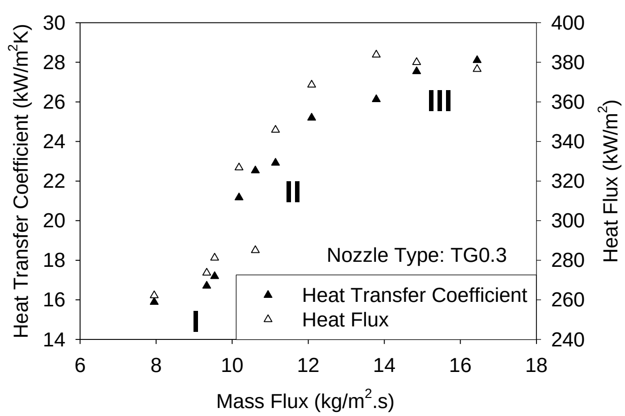

Figure 3 Heat transfer coefficient and heat flux variations at different values of mass fluxes (for nozzle of type TG0.3).

In addition to this, evaporation from the surface of thin liquid film also adds to the rate of heat transfer. The effect of evaporation is high at low mass fluxes and it decrease with increase in mass flux, droplet diameter and also with decrease in spray pressure. At low mass flux (dilute sprays), smaller droplet diameter or increase in spray pressure results in more interactions of water droplets with ambient air. This entrapped air escapes from the surface of liquid film and carries some of the fluid and hence gives better heat transfer on the surface.

When the mass flux increased further, the number of impacting droplet increased resulting in increase in coolant surface area. This also creates more turbulence is on thin liquid film. The combined effect can be seen as increase in the heat transfer coefficient with mass flux. The rate of increase of heat transfer coefficient and heat transfer was not constant with increase in water spray mass flux. The rate of increase of heat transfer was lesser in Region-I and Region-III compared to Region-II. The possible cause of this slow increase in heat transfer coefficient in Region-I was low droplet striking velocity and larger droplet size at small pressures of the nozzle. It is worth mentioning here that the mass fluxes of the nozzles in the present study were controlled with the nozzle pressures. This indicates that the droplet diameter was largest in the Region-I resulting in lower heat transfer from the impingement surface. In Region-II, the variation of increase in heat transfer coefficient with increase in mass flux is very sharp as compared to other two regions. This increase indicates that optimum mass fluxes lie in this region. Thus it can be appreciated that in Region-II the large increase in heat transfer rates are because of combined effect of more number of smaller size droplets in the water spray and increased flow velocities at higher mass fluxes. With further increase in the mass fluxes in Region-III, the increase in heat transfer rates again becomes slower. Here with increased mass flow rates the flow velocities become excessively high resulting in splashing of the water droplets of the water spray on the impingement surface. This splashing action reduces the effective mass flux striking on the impingement surface resulting in slow increase in heat transfer rates.

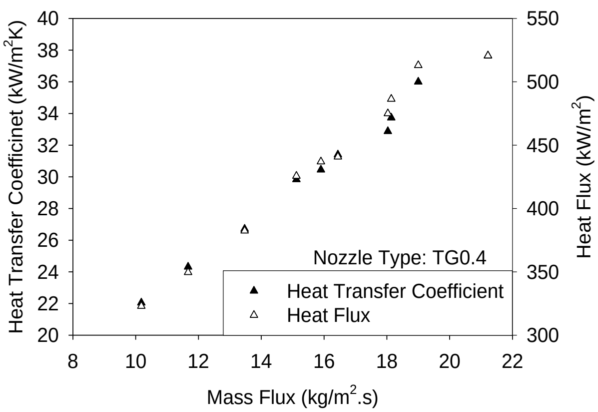

Figure 4 Heat transfer coefficient and heat flux variations at different values of mass fluxes (for nozzle of type TG0.4).

Figure 4 describes the variation of heat transfer coefficient and heat flux with mass fluxes issuing from the spraying nozzle of type TG0.4 (d = 559 µm). The increase in heat transfer coefficient with mass flux is almost linear. Since the temperature of the target surface in the present study is fixed at 50°C, the evaporation effect is less but it cannot be neglected at low mass fluxes. But for nozzle type TG0.3 at lower mass fluxes, the amount of evaporation is more and most of the heat transfer process is carried out by phase convection and evaporation. Since sprays are classified as dense or dilute on the basis of mass fluxes, so the sprays coming out of smaller orifice diameter give less mass flux at same inlet pressure to nozzle as compared to larger diameter nozzle resulting in higher heat transfer on the impingement surface. This is also evident from Figs. 3 and 4. It has also been observed from Fig. 4 that the heat flux shows linear dependence on mass flux up to the value of 18 kg/m^2s after that a slight jump in heat flux (51.3 kW/m^2) at mass flux of 19 kg/m^2s followed by no significant variation in the heat transfer.

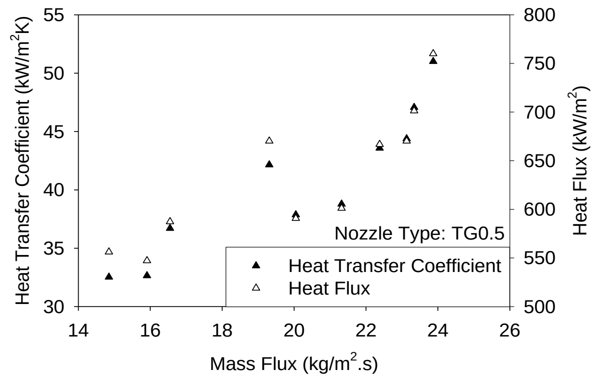

Figure 5 Heat transfer coefficient and heat flux variations at different values of mass fluxes (for nozzle of type TG0.5).

Figure 5 shows the variation of heat transfer coefficient and heat flux with mass flux of water spray on the target surface for nozzle of type TG0.5 (d = 610 µm). In the very beginning for mass fluxes of 14-16 kg/m^2s, a very less change in heat transfer coefficient was observed. This is because of larger the diameter of droplets coming out of nozzle resulting in low value of heat transfer coefficient. After this, for next range of the mass flux up to 19 kg/m^2s, heat transfer coefficient shows a steep increase. This can be attributed to formation of smaller droplet diameters at higher pressures resulting in improvement of heat transfer. Now at this larger mass fluxes a thick water film over the target surface is also started forming. As this film starts growing, a fall in heat transfer is observed in spite of increase in mass flux of the water spray. This fall in heat flux can also be attributed to splashing of the fluid (resulting in decrease in effective mass flux striking on the impingement surface) on the impingement surface. With further increase in mass flux beyond 20 kg/m^2s the heat transfer rate again starts increasing at a much faster rate. The steep rise in heat transfer rate at this range of mass flux can be attributed to change in fluid-dynamics of the thick film because of striking of the droplets at very high velocities. This creates strong perturbations in the thick film formed at the target surface resulting in enhancement of mixing and higher turbulence hence faster heat transfer from the impingement surface.

Comparison between Heat Transfer Characteristics of Different Nozzles

Figure 6 shows the variation of Nusselt number with Reynolds number for spraying nozzles of types TG0.3, TG0.4 and TG0.5. With increase in Reynolds number the Nusselt number increases for all types of nozzles. The difference between variations in heat transfer characteristics of nozzles of different diameters is divided into two parts, first is the general trend shown for each nozzle and second is the difference in heat transfer characteristics at same mass flux for each nozzle. The cause of variation in general heat transfer trend follow by each nozzle is the difference in basic heat transfer mechanism followed by each nozzle. Oliphant et al. [8] mentioned that at lower mass fluxes the evaporation from the thin liquid layer cannot be neglected. As the mass flux increases the evaporation effects starts decreasing from thin liquid layer and whole heat transfer depends upon the mass flux of impinging water. Heat transfer characteristics show sharp growth with mass flux. In case of nozzle TG0.3, the heat transfer characteristics are due to combined effects of evaporation and mass flux of coolant. Here, evaporation is caused by carrying atmospheric air with fine droplets coming out of nozzle. The second part of variation in heat transfer characteristics at same mass fluxes. It is seen that at same mass flux the droplets coming out from nozzle of smaller orifice diameter are smaller in diameter and higher in velocity. Theoretically it should give high heat transfer but actually the heat transfer at same mass flux is less for smaller diameter because of high splashing effect and droplet bouncing at higher droplet velocities. This reduces the effect of increase in mass flux of water.

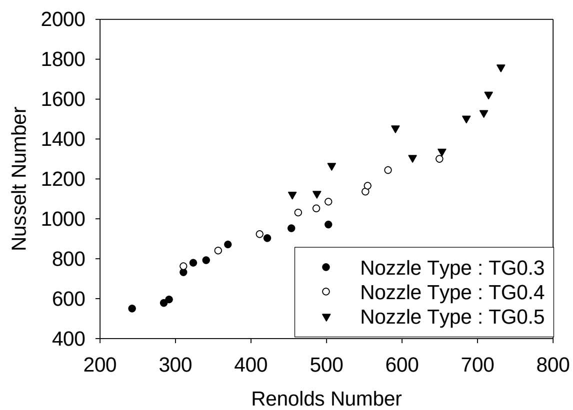

Figure 6 Variation of Nusselt number with Reynolds number for spraying nozzles of types TG0.3, TG0.4 and TG0.5.

Correlation fit for various nozzles

The biggest challenge in case of experimental results is generalizing the information acquired from the set of experiments. Here in the present study different correlations have been developed for evaluating Nu as a function of Re. Firstly a set of correlations were developed for individual nozzles as given in Eqns. 5-8.

1) Spraying nozzle type: TG0.3

$$Nu = 6.7\,\mathrm{Re}^{0.81} \text{ with } R^2 = 0.89 \tag{5}$$

(For Re 243-503, at surface temperature 50°C)

2) Spraying nozzle type: TG0.4

$$Nu = 10.6\,\mathrm{Re}^{0.74} \text{ with } R^2 = 0.99 \tag{6}$$

(For Re 311-650, at surface temperature 50°C)

3) Spraying nozzle type: TG0.5

$$Nu = 7.2\,\mathrm{Re}^{0.82} \text{ with } R^2 = 0.83 \tag{7}$$

(For Re 455-731, at surface temperature 50°C)

4) Combined results of all spraying nozzles

$$Nu = 2.67\,\mathrm{Re}^{0.97} \text{ with } R^2 = 0.94 \tag{8}$$

(For Re 243-731, at surface temperature 50°C)

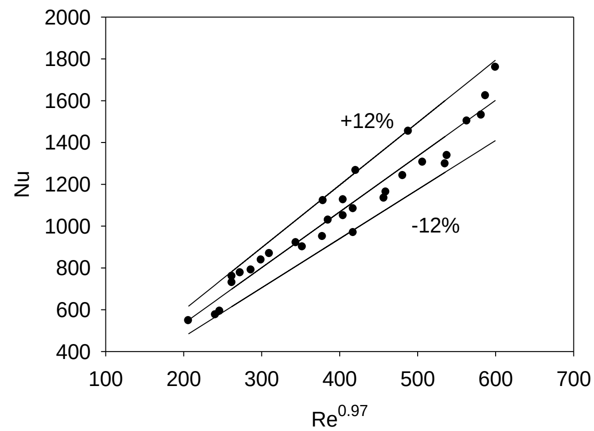

Figure 7 Correlation fit between Nu and Re for various spraying nozzles (TG03, TG04 and TG05).

It has been seen form the different correlations that different confidence levels of curve fits were possible for different set of nozzles. The output of the nozzle of type TG0.4 showed a most satisfactory fit with R2 value of 0.99. It has also been observed that the exponent of Re has different values for different nozzles showing the dependence of Re of different magnitude. A very reasonable fit with R2 value of 0.94 was developed for the overall data. It shows almost linear variation of the Nu with Re as exponent is found to be 0.97. A curve fit to entire data points of all three types of nozzles are shown in Fig. 7. It has been seen that the overall fit predicted the entire experimental data with ±12%.

III. Conclusions

The aim of present study is to investigate the heat transfer characteristics of water spray impinging on a heated surface whose surface temperature is fixed at 50°C. Effects of operating parameters such as mass flux, nozzle diameter on heat transfer have been investigated. With increase in the mass flux of the water the heat flux as well heat transfer coefficient is continuously increasing because of higher liquid velocity over the surface resulting in thinner thermal boundary layer. Also, impact of the droplets onto the film agitates the liquid making local thinning the thermal boundary layer. Rate of increase of heat transfer coefficient and heat transfer rate was not constant with increase in water spray mass flux for TG0.3 and TG0.5 whereas it was almost constant for TG0.4. This is due to different nature of physics happening at different impingement mass fluxes. A phenomenon like strong splashing occurs at high mass fluxes resulting in decrease in effective mass flux striking and hence lesser heat transfer. This creates strong perturbations in the thick film formed at the target surface resulting in enhancement of mixing and higher turbulence hence faster heat transfer from the impingement surface. Heat transfer coefficients and heat flux variations with mass flux follows almost similar trends for all types of nozzles. Different correlations have been developed for evaluating Nu as a function of Re. It has also been observed that the exponent of Re has different values for different nozzles showing the dependence of Re of different magnitude.

Nomenclature

CHF Critical heat flux
D diameter of the heated surface (m)
G mass flux ($kg/s/m^2$)
h heat transfer coefficient (W/m^2K)
k thermal conductivity (W/mK)
k_f thermal conductivity of water at bulk fluid temperature (W/mK)
Nu Nusselt number
q'' heat flux (W/m^2)
Re Reynolds number
T_f fluid temperature (K)
T_s surface temperature (K)
Δx axial distance (m)
ΔT temperature difference (K)

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