Experimental Studies on Effects of Nozzle to Plate Spacing and Jet Inclination on Impingement Heat Transfer with Axisymmetric Water Jets

Nirmal Kumar Kund
Department of Production Engineering
Associate Professor, Veer Surendra Sai University of Technology, Burla, Odisha, India

Abstract-*This paper presents the experimental studies on the heat transfer behavior of an axisymmetric water jet impinging on a heated plate. Various influencing parameters pertaining to the thermal performance of the water jet impingement are identified and their effects on heat transfer characteristics are nozzle to plate spacing (20-35 mm) and jet inclination (30-90°). Additionally, the studies are limited to a constant heat flux situation. The careful observations of the results reveal that the performance of water jet can be optimized with regard to these key parameters. However, with the current experimental conditions, the jet inclination of 60º with the nozzle to plate spacing of 25 mm gives moderate heat transfer characteristics and is the optimum.*

Keywords*-Water jet, Target plate, Nozzle to plate spacing, Jet inclination.*

I. INTRODUCTION

The ever growing needs for faster and smaller electronic components in the electronic industry has resulted in the development of compact electronic equipments with high power densities. Conventional air cooling is insufficient in most cases to help sustain and safeguard the electronics components from the thermal failure. Garimella and Nenaydykh [1] have experimentally investigated the effect of nozzle geometry on the local heat transfer of FC-77 *per-fluorinated dielectric* liquid for normal jet impingement. Roy et al. [2] studied both experimentally and numerically heat transfer of an inclined surface subject to an impinging airflow. Chakraborty and Dutta [3] found analytical solutions for heat transfer during cyclic melting and freezing of a phase change material used in electronic and electrical packaging. Lee et al. [4] have examined the effects of nozzle diameter on impinging jet heat transfer and fluid flow. Nayak et al. [5] developed a numerical model for heat sinks with phase change materials and thermal conductivity enhancers. Eren and Celik [6] investigated the heat transfer characteristic of a heated flat plate by an obliquely impinging air jet. Pavlova and Amitay [7] studied on the heat transfer behavior of electronics with synthetic normal Jet Impingement.

II. LITERATURE REVIEW

Furthermore, Agostini et al. [8] illustrated the state of art of the high heat flux cooling technologies. Adoni et al. [9] developed a thermo-hydraulic model for capillary pumped loop and loop heat pipe used for cooling of electronics. Behera et al. [10] investigated numerically on the heat transfer of interrupted impinging air jets used for electronics cooling. Saha et al. [11] investigated on optimum distribution of fins in heat sinks filled with phase change materials. Sagot et al. [12] studied the jet impingement heat transfer on a flat plate at a constant wall temperature by using gas jets. Sanyal et al. [13] numerically studied on heat transfer from pin-fin heat sink using steady and pulsated impinging air jets. Saha and Dutta [14] developed heat transfer correlations for PCM-based heat sinks with plate fins. Chaudhariet al. [15] examined heat transfer characteristics of synthetic air jet impingement cooling. Narasimhanet al. [16] investigated on thermal management using the bi-disperse porous medium approach. Yu et al. [17] compared a series of double chamber model with various hole angles for enhancing cooling effectiveness by air jets. Besides, Yu et al. [18] also carried out numerical simulation on the effect of turbulence models on impingement cooling of double chamber model. Cheng et al. [19] numerically studied air/mist impinging jets cooling effectiveness under various curvature models. Nguyen et al. [20] investigated cooling effect by sub-zero cold air jet in the grinding of a cylindrical component. Gould et al. [21] studied jet impingement cooling of a silicon carbide module. Zhao et al. [22] also investigated the effect of guide wall on jet impingement cooling.

III. OBJECTIVES OF PRESENT RESEARCH WORK

Careful review and examination of the already stated relevant literature reveals no clear cut and prior theoretical and experimental investigation on the local heat transfer under an obliquely impinging, axisymmetric free surface water jet flow. In addition, to the best of the authors' knowledge, there is not a single comprehensive experimental study pertaining to the effects of the nozzle to target plate spacing and the nozzle inclination on the heat transfer behavior over the heated target

plate previously maintained at a uniform heat flux of 6.25 W/cm2 for investigating the relative importance of the key parameters involved. With this viewpoint, the current paper demonstrates experimental investigations relating to the influence and role of the nozzle to plate spacing (20-35 mm) and the jet inclination (30-90°) on the local heat transfer characteristics over a flat plate heated from the underneath and maintained at a uniform flux of 6.25 W/cm2, for free surface axisymmetric water jet impingements. Additionally, the results thus obtained are analyzed and compared, so as to realize deeply, the heat transfer behavior over the target plate for achieving better cooling effect.

IV. TEST APPARATUS AND METHOD

It enumerates about the details of a series of rigorous and numerous experiments on jet impingement cooling starting from the descriptions of various components of the experimental setup to the smooth control/regulation of different key parameters (such as jet flow rate, input voltage and current to the heater, nozzle to target plate spacing and nozzle inclination) involved in affecting the heat transfer during the jet impingement. The basic attempt of this work is to measure the equilibrium temperatures (corresponding to the steady state condition) at different points on the target plate subjected to constant heat flux during the jet impingement at uniform rate, while setting the stated parameters at appropriately suitable and predetermined values.

A. Description of the physical problem

In order to achieve the said objective, the model is selected accordingly and the schematic sketch of the physical model is illustrated in Fig. 1.
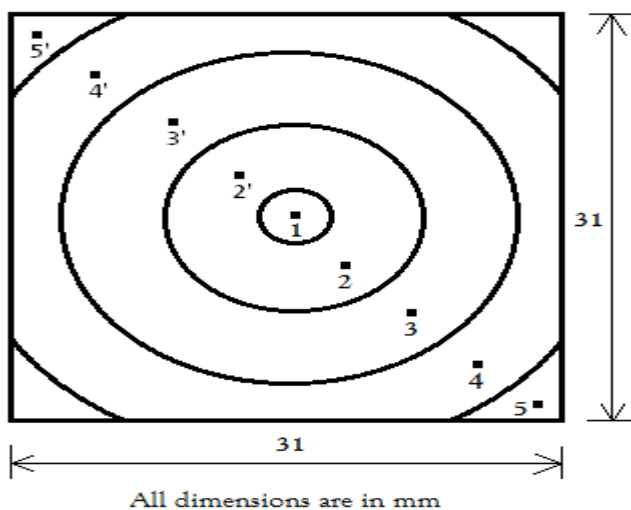

All dimensions are in mm

Fig. 1. Schematic sketch of the physical model divided into annuli

The physical problem consists of a copper plate (also termed as target plate) of dimensions 31×31×2 mm on which T-type thermocouples are mounted along one of the diagonal lines with spacing of 5 mm between two consecutive thermocouples. This target plate is mounted on a heater, whereas, these thermocouples are connected to a data acquisition system to record temperature data continuously while conducting experiments.

The schematic sketch of the physical model (of the target plate) is divided into different annuli corresponding to the different thermocouples. The temperature is assumed to be constant over an annulus. Hence the temperature variation on the plate is assumed to be a step function. The assumption is only used for calculating the average heat transfer coefficient or the average Nusselt number for the normal jet impingement.

Here, local heat transfer coefficient,

$$h_i = \frac{Q_{out}}{A_h (T_{si} - T_j)} \; ; \; Q_{out} = VI$$

(1)

So, average heat transfer coefficient, $\bar{h} = \dfrac{\sum h_i A_i}{\sum A_i}$

(2)

Average heat transfer coefficient,

$$\bar{h} = \left[\frac{Q_{out}}{A_h^2}\right] \sum \left(\frac{A_i}{T_{si} - T_j}\right)$$

(3)

Hence, local Nusselt number, $\mathrm{Nu}_i = \dfrac{h_i d}{k}$

(4)

Now, average Nusselt number, $\overline{\mathrm{Nu}} = \dfrac{\bar{h} d}{k}$

(5)

B. Description of the experimental setup

Fig. 2 represents the schematic of the complete assembly of the experimental setup consisting of a heater kept in a rectangular Plexiglas box, a nozzle connected to a rotameter via a flexible pipe and a copper target plate mounted on the heater. The heater consisting of tungsten filament (with heater wire diameter of 0.576 mm and heater element resistance of 4.3 Ω) is connected to a dual supply D.C. power source. For particular values of current and voltage the heat flux to the heater remains constant. A digital multimeter (Keithley 2700 model) is used to measure the voltage, whereas, current is directly measured from the

display unit of power supply.The rotameter is connected to a water supply tap by means of a flexible pipe with brass ball valve arrangement for regulating water flow. The copper plate mounted on the heater have got grooves underside in order to accommodate thermocouples connected to data acquisition system. The surface of the copper plate is polished with sand paper and then is cleaned with acetone before conducting experiments. The nozzle is kept perpendicular or inclined to the target plate by means of a vertical stand with clamp arrangement. The test fluid (water) discharges out through the outlet of the test chamber after impinging on the target plate.
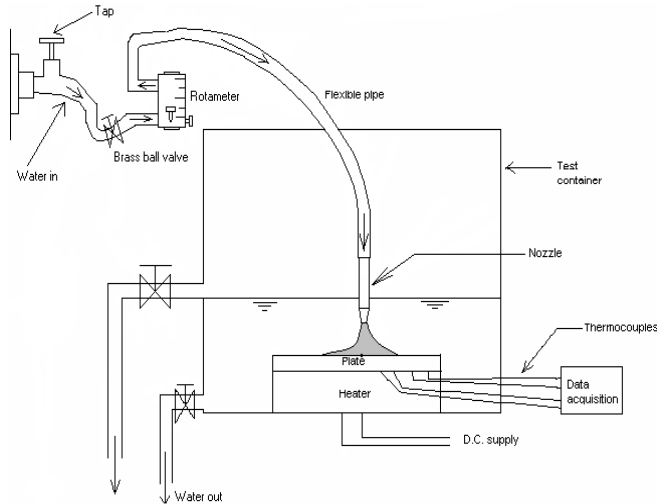

Fig. 2. Schematic of experimental setup

C. Experimental procedure

It involves the measurements of following key process parameters.

1. *Measurements of nozzle to target plate spacing and nozzle inclination*

Basically, the nozzle is fitted to a clamp attached to the vertical stand whereas the target plate is mounted on the heater. Hence, the distance between nozzle and target plate spacing (Z) is measured by means of filler gauges. The nozzle inclination (θ) with respect to horizontal target plate is checked and adjusted by using a protractor attached to the clamp.

2. *Flow measurement*

During the jet impingement the flow rate of water in the tube connecting the tap to the nozzle is measured by means of a rotameter (having measuring range of 0-120 lph). The measurement uncertainty of rotameter is ±0.01 times the flow rate in lph. In order to avoid fluctuations and

to get the desired flow rate of water, the fine tuning of rotameter is also done. The average jet velocity exiting the nozzle is calculated from the jet flow rate. And also, the corresponding jet Reynolds number is calculated from this jet velocity.

3. *Temperature measurement*

Here, as mentioned in Table 1, polytetrafluoroethylene (PTFE) coated T-type (Copper-Constantan, manufactured by T C Ltd, UK) thermocouples (of diameter 0.205 mm with measuring range of 0-200° C) are used to measure temperature at different locations on the target plate during the impingement of water jet. These thermocouples are calibrated well against a platinum resistance thermometer. A Julabo FH40-MH circulation bath is used for this purpose. As per the manufacturer's specification of thermocouples, the response time is 0.8 s and the measurement uncertainty is ±0.004T, where T is the measured temperature in degrees Celsius. As already mentioned, these temperature data are periodically recorded by an interface computer through a data acquisition system. The data acquisition system consists of a 40-channel Keithley thermocouple plug-in card with "T" type 30-gauge Teflon coated copper-constantan thermocouples monitoring the thermal evolution.

Table 1
Specifications of thermocouples

Material	Class	Size (mm)	Range (°C)
Copper-Constantan	PTFE coated T-type	0.205	0-200°C

V. RESULTS AND DISCUSSION

Rigorous experiments are performed to study the effects of nozzle diameter, nozzle-to-target plate spacing, jet velocity and jet inclination on the heat transfer behavior over the heated target plate subjected to a uniform heat flux. At the outset, a base case of nozzle diameter 5 mm with the nozzle to target plate spacing of 25 mm and a normal water jet of flow rate 30 lph corresponding to Reynolds number of 2400 is considered. The heat flux of 6.25 W/cm2 (corresponding to 30 V and 2 A of D. C. power source associated with copper target plate of size 31 mm × 31 mm) is applied in all investigations.

Besides the said base case, the preliminary study also involves four more different observations concerning nozzle to target plate spacing of 30 mm, jet flow rate of 10 lph (corresponding to Reynolds number of 800), nozzle diameter

of 3 mm and nozzle inclination of 30º in line with the stated experimental conditions. The results thus obtained are compared to investigate further the role and effect of various key process parameters involved.

Fig. 3 demonstrates the local Nusselt number variation with radial distance from the stagnation point, together for all the stated physical situations, for the comparative study. It is observed that decrease in the jet Reynolds number from (base case value of) 2400 to 800 causes decrease in the stagnation Nusselt number from 51 to 30. Correspondingly, the local Nusselt number values slowly decrease from the stagnation point resulting in the local Nusselt number profile relatively flat, as compared to the base case. In addition, it is also observed that increase in the nozzle to target plate spacing from (base case value of) 5 to 6 causes decrease in the stagnation Nusselt number from 51 to 43. Correspondingly, the local Nusselt number values gradually decrease from the stagnation point resulting in the local Nusselt number profile relatively flat, in comparison with the base case. Besides, it is observed that decrease in nozzle diameter from (base case value of) 5 mm to 3 mm causes increase in the stagnation Nusselt number from 51 to 73. Correspondingly, the local Nusselt number values moderately decrease from the stagnation point resulting in the local Nusselt number profile relatively flat, as compared to the base case. Furthermore, it is observed that decrease in the jet inclination (i.e. increase in obliquity) from 90☐ to 30☐ causes notable asymmetry in the local Nusselt number profile resulting in the sharpening of the peak in the Nusselt number profile leading to the significant increase in the stagnation Nusselt number from 51 to 85.
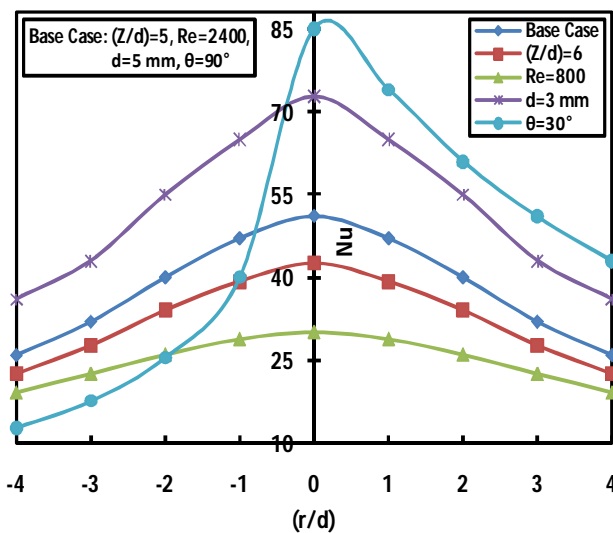

Fig. 3. Variation of Nusselt number with radial distance

A. EFFECT OF NOZZLE TO PLATE SPACING

Apart from the said base case involving nozzle to target plate spacing of 5 times the nozzle diameter, in the present study three more nozzle to plate spacings of 4, 6 and 7 times the nozzle diameter are taken into consideration with the stated experimental conditions. The results so observed are compared to investigate the role and effect of the nozzle to plate spacing. Fig. 8 depicts the variation of both stagnation and average Nusselt numbers with the nozzle to plate spacing.

Fig. 4 shows the said variation for four different jet Reynolds numbers of 800, 1600, 2400 and 3200. With the jet Reynolds number of 800, the stagnation Nusselt numbers are 40, 30, 26 and 22 besides the average Nusselt numbers of 33, 25, 22 and 20 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Likewise, with the jet Reynolds number of 1600, the stagnation Nusselt numbers are 54, 43, 37 and 31 besides the average Nusselt numbers of 42, 33, 28 and 24 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Similarly, with the jet Reynolds number of 2400, the stagnation Nusselt numbers are 64, 51, 43 and 37 besides the average Nusselt numbers of 48, 37, 32 and 27 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Furthermore, with the jet Reynolds number of 3200, the stagnation Nusselt numbers are 75, 60, 50 and 43 besides the average Nusselt numbers of 33, 25, 22 and 20 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively.
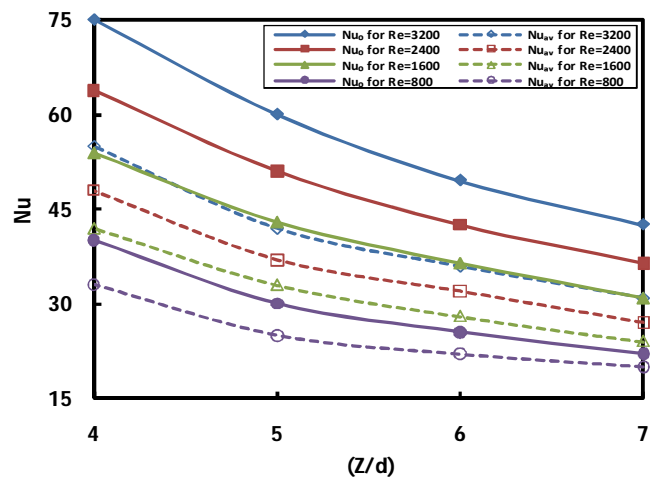

Fig. 4. Variation of Nusselt number with nozzle to plate spacing at different Reynolds numbers

Fig. 5 shows the said variation for four different nozzle diameters of 3, 4, 5 and 6 mm. With the nozzle diameter of 3 mm, the stagnation Nusselt numbers are 89, 73, 61 and 52 besides the average Nusselt numbers of 65, 51, 43 and 36 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Likewise, with the nozzle diameter of 4 mm, the stagnation Nusselt numbers are 75, 60, 51 and 44 besides the average Nusselt numbers of 55, 42, 36 and 31 for the

nozzle to plate spacings of 4, 5, 6 and 7, respectively. Similarly, with the nozzle diameter of 5 mm, the stagnation Nusselt numbers are 64, 51, 43 and 37 besides the average Nusselt numbers of 48, 37, 32 and 27 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Furthermore, with the nozzle diameter of 6 mm, the stagnation Nusselt numbers are 56, 45, 38 and 32 besides the average Nusselt numbers of 42, 32, 28 and 23 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively.
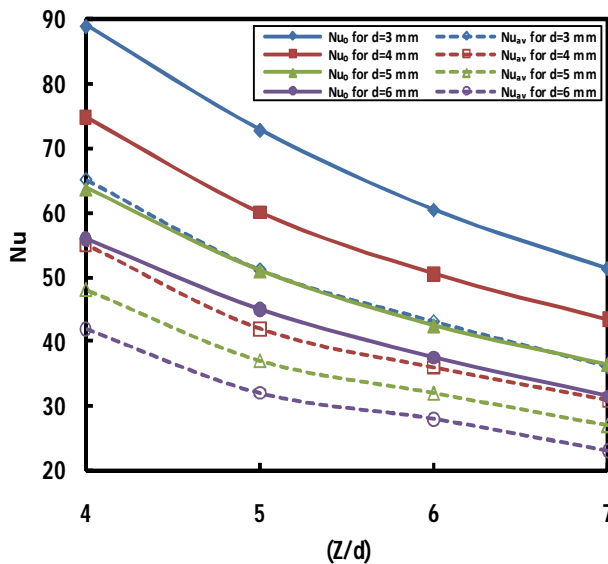

Fig. 5. Variation of Nusselt number with nozzle to plate spacing at different nozzle diameters

Fig. 6 shows the said variation for five different jet inclinations of 30□, 45□, 60□, 75□ and 90□. With the jet inclination of 30□, the stagnation Nusselt numbers are 102, 85, 73 and 63 besides the average Nusselt numbers of 74, 60, 51 and 42 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Likewise, with the jet inclination of 45□, the stagnation Nusselt numbers are 89, 73, 62 and 53 besides the average Nusselt numbers of 65, 52, 44 and 36 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Similarly, with the jet inclination of 60□, the stagnation Nusselt numbers are 79, 64, 54 and 46 besides the average Nusselt numbers of 58, 45, 38 and 31 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Furthermore, with the jet inclination of 75□, the stagnation Nusselt numbers are 71, 57, 48 and 41 besides the average Nusselt numbers of 52, 40, 34 and 28 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively. Additionally, with the jet inclination of 90□, the stagnation Nusselt numbers are 64, 51, 43 and 37 besides the average Nusselt numbers of 48, 37, 32 and 27 for the nozzle to plate spacings of 4, 5, 6 and 7, respectively.
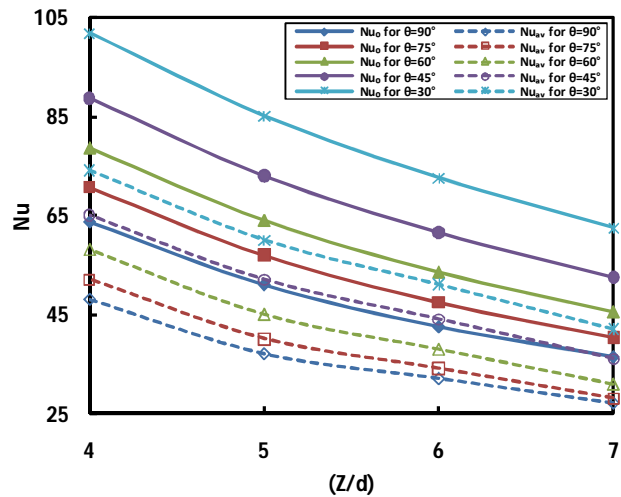

Fig. 6. Variation of Nusselt number with nozzle to plate spacing at different jet inclinations

From the cited figures, it is evident that both stagnation and average Nusselt numbers decrease with increase in nozzle to plate spacing. This result as expected can be on account of the higher nozzle to plate spacing causing the carrying away of heat from the target plate at relatively slower rate. Additionally, for a particular nozzle to plate spacing, both stagnation and average Nusselt numbers increase with Reynolds number (Fig. 4). Likewise, for a particular nozzle to plate spacing, both stagnation and average Nusselt numbers decrease with increase in nozzle diameter/jet inclination (Fig. 5 and Fig. 6). Here also, from Figs. 4-6, it is apparent that the variation of both stagnation and average Nusselt numbers with nozzle to plate spacing is nearly linear.

B. EFFECT OF JET INCLINATION

Besides the said base case involving jet inclination of 90°, in this investigation four more jet inclinations of 30, 45, 60 and 75° are carefully considered with the stated experimental conditions. The corresponding results are analyzed and compared to study the role and effect of the jet inclination. Fig. 9 describes the variation both stagnation and average Nusselt numbers with the jet inclination.

Fig. 7 shows the said variation for four different jet Reynolds numbers of 800, 1600, 2400 and 3200. With the jet Reynolds number of 800, the stagnation Nusselt numbers are 50, 43, 37, 33 and 30 besides the average Nusselt numbers of 39, 34, 30, 27 and 25 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Likewise, with the jet Reynolds number of 1600, the stagnation Nusselt numbers are 70, 60, 53, 47 and 43 besides the average Nusselt numbers of 51, 45, 39, 35 and 33 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Similarly, with the jet Reynolds number of 2400,

the stagnation Nusselt numbers are 85, 73, 64, 57 and 51 besides the average Nusselt numbers of 60, 52, 45, 40 and 37 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Furthermore, with the jet Reynolds number of 3200, the stagnation Nusselt numbers are 98, 85, 74, 66 and 60 besides the average Nusselt numbers of 70, 60, 52, 47 and 42 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively.
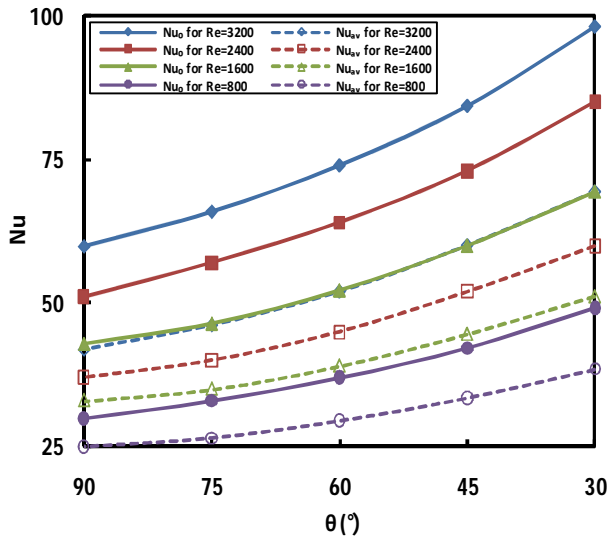

Fig. 7. Variation of Nusselt number with jet inclination at different Reynolds numbers

Fig. 8 shows the said variation for four different nozzle diameters of 3, 4, 5 and 6 mm. With the nozzle diameter of 3 mm, the stagnation Nusselt numbers are 117, 102, 90, 81 and 73 besides the average Nusselt numbers of 81, 71, 62, 55 and 51 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Likewise, with the nozzle diameter of 4 mm, the stagnation Nusselt numbers are 99, 85, 75, 67 and 60 besides the average Nusselt numbers of 68, 59, 51, 46 and 42 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Similarly, with the nozzle diameter of 5 mm, the stagnation Nusselt numbers are 85, 73, 64, 57 and 51 besides the average Nusselt numbers of 60, 52, 45, 40 and 37 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Furthermore, with the nozzle diameter of 6 mm, the stagnation Nusselt numbers are 77, 66, 57, 51 and 45 besides the average Nusselt numbers of 53, 46, 40, 35 and 32 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively.

Fig. 9 shows the said variation for four different nozzle to plate spacings of 4, 5, 6 and 7 (times the nozzle diameter). With the nozzle to plate spacing of 4, the stagnation Nusselt numbers are 107, 92, 80, 72 and 64 besides the average Nusselt numbers of 78, 68, 59, 52, and 48 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Likewise, with the nozzle to plate spacing of 5, the stagnation Nusselt

numbers are 85, 73, 64, 57 and 51 besides the average Nusselt numbers of 60, 52, 45, 40 and 37 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Similarly, with the nozzle to plate spacing of 6, the stagnation Nusselt numbers are 71, 61, 54, 48 and 43 besides the average Nusselt numbers of 52, 45, 39, 35 and 32 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively. Furthermore, with the nozzle to plate spacing of 7, the stagnation Nusselt numbers are 61, 53, 46, 41 and 37 besides the average Nusselt numbers of 44, 38, 33, 30 and 27 for the jet inclinations of 30, 45, 60, 75 and 90°, respectively.
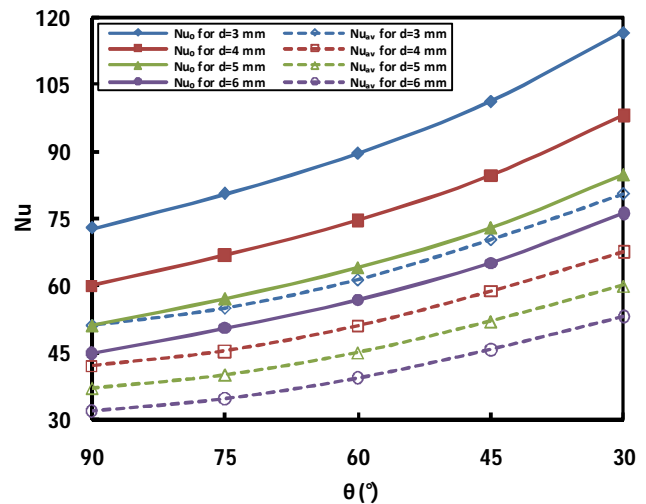

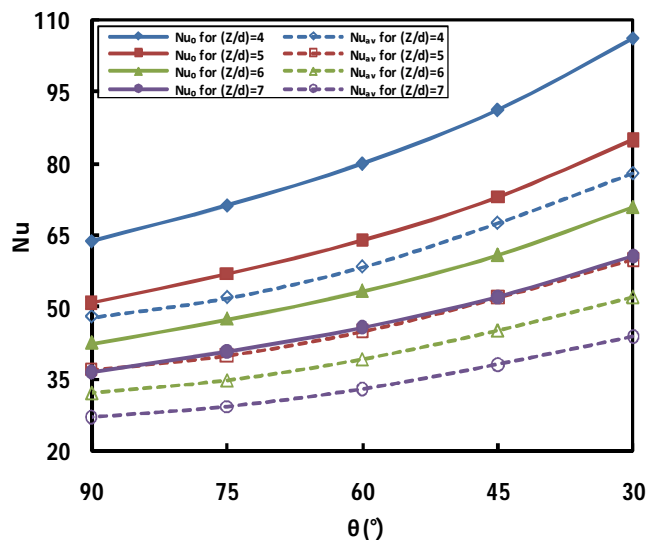Fig. 8. Variation of Nusselt number with jet inclination at different nozzle diameters

Fig. 9. Variation of Nusselt number with jet inclination at different nozzle to plate spacings

From the stated figures, it is apparent that both stagnation and average Nusselt numbers increase with decrease in jet inclination. It may be due to the higher local water flow rates on the downstream side for the oblique jet impingement resulting in the taking away of heat from the

target plate at faster rate. This result is to be expected in view of the positive dependence of the heat transfer rate on the local water flow rates. Also, for a particularjet inclination, both stagnation and average Nusselt numbers increase with increase in Reynolds number (Fig. 7). Furthermore, for a particularjet inclination, both stagnation and average Nusselt numbers decrease with increase in nozzle diameter/nozzle to plate spacing/jet inclination (Fig. 8 and Fig. 9). Moreover, from Figs. 7-9, it is quite evident that the variation of both stagnation and average Nusselt numberswith jet inclination is approximately linear

VI. CONCLUSION

To study the heat transfer behavior, comprehensive experiments are carried out and accordingly measurements are taken for different combinations of interconnected and interdependent parameters concerned with a water jet impinging on a heated plate. Exhaustive experimental investigations on the effects of the jet inclination and the nozzle to plate spacing are conducted as they highly influence the thermal performance of the water jet impingement. In accordance with the measurements taken and the data obtained, the trends of the results pertaining to various parameters are found to be along expected lines. However, the enhancement in the averaged heat transfer characteristics with water jet inclination is newly identified. Besides, the appropriate combinations of the influential and interrelated parameters for which enhancement in the averaged heat transfer from the plate can be expected is also identified. Direct comparison with other experimental/numerical results is not possible due to non-availability of such experimental conditions in the literature. Furthermore, comparison with a numerical model pertaining to the present experimental conditions is planned for the future. In addition, the present study also neglects target plate side heat losses because of lesser thickness comparable to length/breadth. Nevertheless, with the present experimental conditions, the jet inclination of 60° with the nozzle diameter of 5 mm, nozzle to plate spacing of 25 mm and the jet Reynolds number of 2400 gives moderate heat transfer behavior and is the optimum. Hence, the present combination can be used directly in industries to enhance heat transfer and for cooling of electronic systems.

ACKNOWLEDGMENT

REFERENCES

[1] S. V. Garimella, B. Nenaydykh. "Nozzle-geometry effects in liquid jet impingement heat transfer" International Journal of Heat and Mass Transfer. 1996; 39(14): pp. 2915–2923.

[2] Roy S, Nasr K, Patel P, Abdul Nour B. "An experimental and numerical study of heat transfer off an inclined surface subject to an impinging airflow" Int J Heat Mass Transfer. 2002; 45(8): pp. 1615–1629.

[3] Chakraborty S, Dutta P. "Analytical solutions for heat transfer during cyclic melting and freezing of a phase change material used in electronic and electrical packaging". ASME Journal of Electronics Packaging. 2003; 125(1): pp. 126-133.

[4] Lee DH, Song J, Jo MC. "The effects of nozzle diameter on impinging jet heat transfer and fluid flow". J Heat Transfer. 2004; 126(4): pp. 554-557.

[5] Nayak KC, Saha SK, Srinivasan K, Dutta P. "A numerical model for heat sinks with phase change materials and thermal conductivity enhancers". Int J Heat Mass Transfer. 2006; 49: pp. 1833-44.

[6] Eren H, Celik N. "Cooling of a heated flat plate by an obliquely impinging slot jet". Int Comm Heat Mass Transfer. 2006; 33(3): pp. 372–380.

[7] Pavlova A, Amitay M. "Electronic cooling using synthetic jet impingement". J Heat Transfer. 2006; 128(9): pp. 897-907.

[8] Agostini B, Fabbri M, Park JE, Wojtan L, Thome JR, Michel B. "State of the art of high heat flux cooling technologies". Heat Transfer Eng. 2007; 28(4): pp. 258–281.

[9] Adoni A, Ambirajan A, Srinivasan K, Dutta P. "Thermo-hydraulic modeling of capillary pumped loop and loop heat pipe". AIAA Journal of Thermophysics and Heat Transfer. 2007; 21(2): pp. 410-421.

[10] Behera R, Srinivasan K, Dutta P. "Numerical study of interrupted impinging jets for cooling of electronics". IEEE Transactions Components and Packaging Technologies. 2007; 30(2): pp. 275-84.

[11] Saha SK, Srinivasan K, Dutta P. "Studies on optimum distribution of fins in heat sinks filled with phase change material". ASME Journal of Heat Transfer. 2008; 130(3): pp. 1-4.

[12] Sagot B, Antonini G, Christgen A, Buron F. "Jet impingement heat transfer on a flat plate at a constant wall temperature". Int J Thermal Sciences; 47: pp. 1610–1619.

[13] Sanyal A, Srinivasan K, Dutta P. "Numerical study of heat transfer from pin-fin heat sink using steady and pulsated impinging jets". IEEE Transactions on Components and Packaging Technology. 2009; 32(4): pp. 859-867.

[14] Saha SK, Dutta P. "Heat transfer correlations for PCM-based heat sinks with plate fins". Applied Thermal Eng. 2010; 30: pp. 2485-2491.

[15] Chaudhari M, Puranik B, Agrawal A. "Heat transfer characteristics of synthetic jet impingement cooling". Int J Heat Mass Transfer. 2010; 53(5-6): pp. 1057-1069.

[16] Narasimhan A, Reddy BVK, Dutta P. ":Thermal management using the bi-disperse porous medium approach". Int J Heat Mass Transfer. 2012; 55(4): pp. 538-546.

[17] Yu Z, Xu T, Li J, Ma L, Xu T. "Comparison of a series of double chamber model with various hole angles for enhancing cooling effectiveness". Int Comm Heat Mass Transfer. 2013; 44: pp. 38–44.

[18] Yu Z, Xu T, Li J, Xiu H, Li Y. "Numerical simulation on the effect of turbulence models on impingement cooling of double chamber model". Mathematical Problems in Eng. 2013; Article ID 170317: pp. 1-8.

[19] Cheng P, Yao G, Chen W, Li B, Li J, Xu T, Yu Z, Ma L. "Computational study of air/mist impinging jets cooling effectiveness under various curvature models". Advances in Mechanical Eng. 2014; Article ID 569243: pp. 1-9.

[20] Nguyen T, Liu M, Zhang LC. "Cooling by sub-zero cold air jet in the grinding of a cylindrical component". Int J Adv Manuf Technol. 2014; 73(1-4): pp. 341-52.

[21] Gould K, Cai SQ, Neft C, Bhunia A. "Liquid jet impingement cooling of a silicon carbide power conversion module for vehicle applications". IEEE Trans Power Electronics. 2015; 30(6): pp. 2975-2984.

[22] Zhao QY, Chung H, Choi SM, Cho HH. "Effect of guide wall on jet impingement cooling in blade leading edge channel". J Mech Sci Technol. 2016; 30(2): pp. 525-531.