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Experimental Investigations on Influences of Nozzle to Plate Spacing and Nozzle Inclination on cooling characteristics of impinging water jets

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ABSTRACT

The present study involves the experimental investigations of the heat transfer characteristics of axisymmetric water jet impingement on a heated plate. Various parameters affecting the thermal performance of axisymmetric water jet are identified and their effects on heat transfer behavior are also studied. The key parameters considered are nozzle to plate spacing (20-35 mm) and nozzle inclination (30-90°). In addition, all the investigations here are restricted to a constant heat flux condition. Results indicate that the water jet performance can be optimized with respect to these influencing parameters. However, with the present experimental conditions, the jet inclination of 60º 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 one.
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. Chakraborty and Dutta [2] found analytical solutions for heat transfer during cyclic melting and freezing of a phase change material used in electronic and electrical packaging. Nayak et al. [3] developed a numerical model for heat sinks with phase change materials and thermal conductivity enhancers. Pavlova and Amitay [4] studied on the heat transfer behavior of electronics with synthetic normal Jet Impingement.

Furthermore, Adoni et al. [5] developed a thermo-hydraulic model for capillary pumped loop and loop heat pipe used for cooling of electronics. Saha et al. [6] investigated on optimum distribution of fins in heat sinks filled with phase change materials. Sanyal et al. [7] numerically studied on heat transfer from pin-fin heat sink using steady and pulsated impinging air jets. Chaudhari et al. [8] examined heat transfer characteristics of synthetic air jet impingement cooling. Yu et al. [9] compared a series of double chamber model with various hole angles for enhancing cooling effectiveness by air jets. Besides, Cheng et al. [10] numerically studied air/mist impinging jets cooling effectiveness under various curvature models.

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/cm^2 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/cm^2, 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.

II. METHODS AND MATERIAL

It enumerates about the details of a series of rigorous and numerous experiments on axisymmetric water jet impingement cooling concerning the smooth control and regulation of different key parameters of the experimental setup (such as input voltage and current to the heater, nozzle to target plate spacing and nozzle inclination) involved in affecting the heat transfer behavior 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 figure 1.

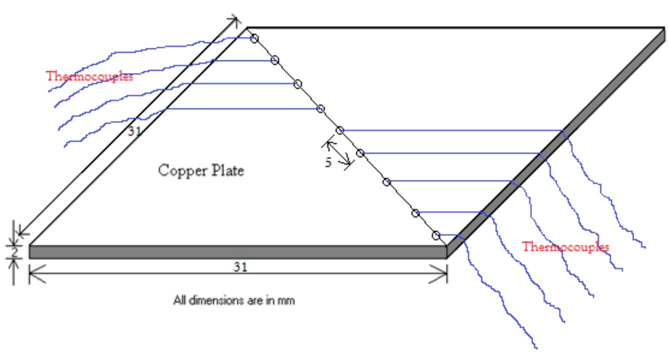

Figure 1. Schematic sketch of the physical model

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 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,

$$h_i = \frac{Q_{out}}{A_h(T_{si} - T_j)}\,;\; Q_{out} = VI \qquad (1)$$

So, average heat transfer coefficient,

$$\bar{h} = \frac{\sum h_i A_i}{\sum A_i} \qquad (2)$$

Average heat transfer coefficient,

$$\bar{h} = \left[\frac{Q_{out}}{A_h^2}\right]\sum\left(\frac{A_i}{T_{si} - T_j}\right) \qquad (3)$$

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

Figure 2 represents the photograph 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.

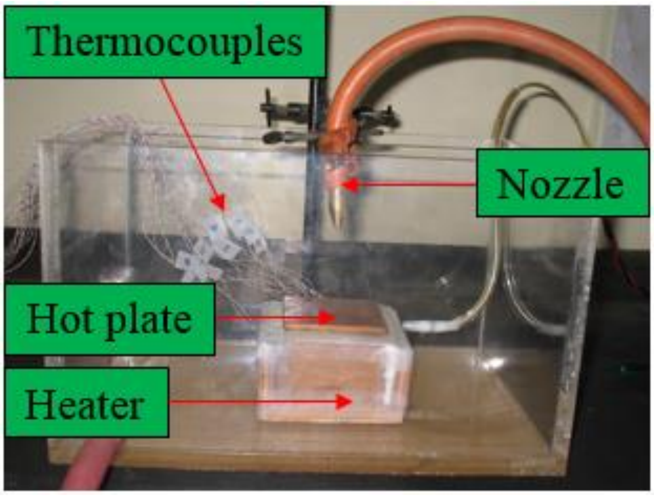

Figure 2. Photograph of the experimental setup

C. Experimental procedure

It involves the measurements of following key 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 is varied by raising or lowering the clamp/heater. The nozzle to target plate spacing is measured by means of filler gauges. The nozzle inclination 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)	Temperature Range (°C)
Copper-Constantan	PTFE coated T-type	0.205	0-200°C

III. RESULTS AND DISCUSSION

Thorough experiments are performed to investigate the effects of nozzle to-target plate spacing, nozzle diameter, jet velocity and jet inclination on the heat transfer distribution over the heated target plate subjected to a uniform heat flux. To begin with, a nozzle of diameter 5 mm with the nozzle to target plate spacing of 25 mm and a normal water jet of flow rate 30 lph. The heat flux of 6.25 W/cm^2 (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.

A. Effect of nozzle to plate spacing

Apart from the stated case involving nozzle to target plate spacing of 25 mm, in this investigation three more nozzle to plate spacings of 20, 30 and 35 mm are taken into consideration with the said experimental conditions. The results so observed are compared to study the role and effect of the nozzle to target plate spacing.

Figure 3 illustrates the variation of local Nusselt number with radial distance from the stagnation point for the said nozzle to plate spacings altogether for the comparative investigation. For the axisymmetric water jet with nozzle to plate spacing of 35 mm, the local Nusselt number decreases from 37 at the centre to 20 at the edge of the target plate. Likewise, for the axisymmetric water jets with nozzle to plate spacings of 30, 25 and 20 mm, the local Nusselt numbers decrease from 43, 51 and 64 at the centre to 23, 26 and 33 at the edge of the target plate, respectively. It is observed that the local Nusselt number profile is relatively flat for the bigger nozzle to plate spacing. This is because of slow cooling associated with the impinging jet.

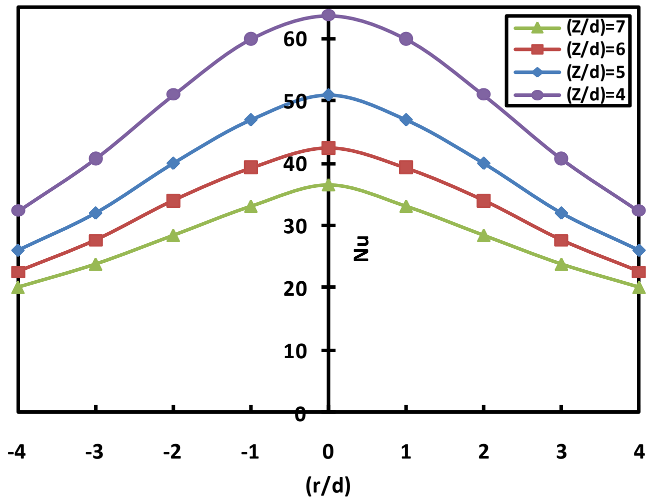

Figure 3. Variation of local Nusselt number with radial distance

Figure 4 depicts the variation of both stagnation and average Nusselt number with the nozzle to plate spacing. The stagnation Nusselt numbers are 30, 43, 51 and 60 for the nozzle to plate spacings of 20, 25, 30 and 35 mm, respectively. Likewise, the average Nusselt numbers are 64, 51, 43 and 37 for the nozzle to plate spacings of 20, 25, 30 and 35 mm, respectively. From both the cited figures, it is clear that the local, stagnation and average Nusselt numbers decrease with nozzle to plate spacing. This result as expected is on account of the higher nozzle to target plate spacing causing the taking away of the heat from the target plate at relatively slower rate. From figure 4, it is also apparent that the variation of both stagnation and average Nusselt numbers with the nozzle to plate spacing is approximately linear.

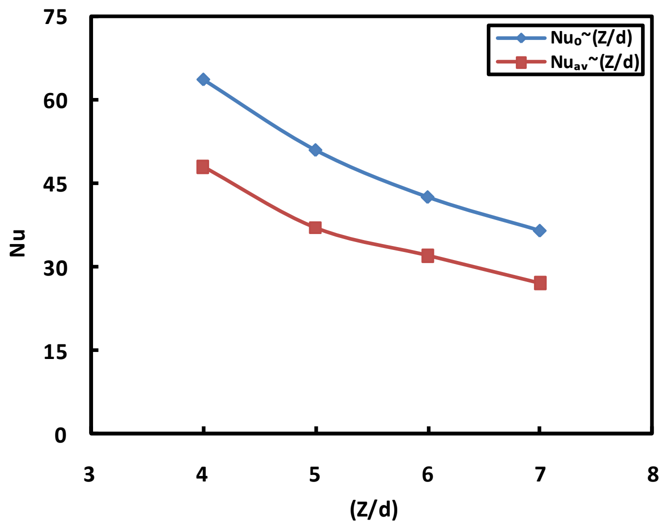

Figure 4. Variation of Nusselt number with nozzle to plate spacing

B. Effect of jet inclination

Apart from the stated case involving jet inclination of 90°, in this investigation four more jet inclinations of 30, 45, 60 and 75° are carefully considered with the said experimental conditions. The results observed are analyzed and compared to study the role and effect of the jet inclination.

Figure 5 shows the variation of local Nusselt number with radial distance from the stagnation point for the said jet inclinations altogether for the comparative investigation. For the axisymmetric water jet with 30° jet inclination, the local Nusselt number varies from 13 at the upstream edge (left side) to 43 at the downstream edge (right side) through 85 at the stagnation point (centre) of the target plate. Likewise, for the axisymmetric water jets with 45, 60 and 75° jet inclinations, the local Nusselt numbers vary from 16, 20 and 23 at the upstream edge to 37, 33 and 29 at the downstream edge through 73, 64 and 57 at the stagnation point of the target plate, respectively. It is observed that the decrease in the jet inclination 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 of the stagnation Nusselt number). In other words, with decreasing inclination, the local Nusselt number profiles exhibited an increasing asymmetry around the point of maximum heat transfer (i.e. the stagnation point), with the upstream side of the profile dropping off relatively more sharply compared to the downstream side. This result is to be expected in view of the positive dependence of the heat transfer coefficient on the Reynolds number, as there are higher local flow rates on the downstream side of the profile for oblique impingement.

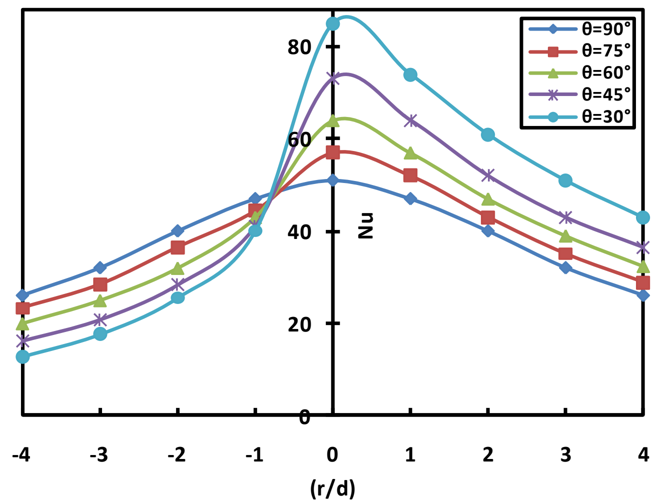

Figure 5. Variation of local Nusselt number with radial distance

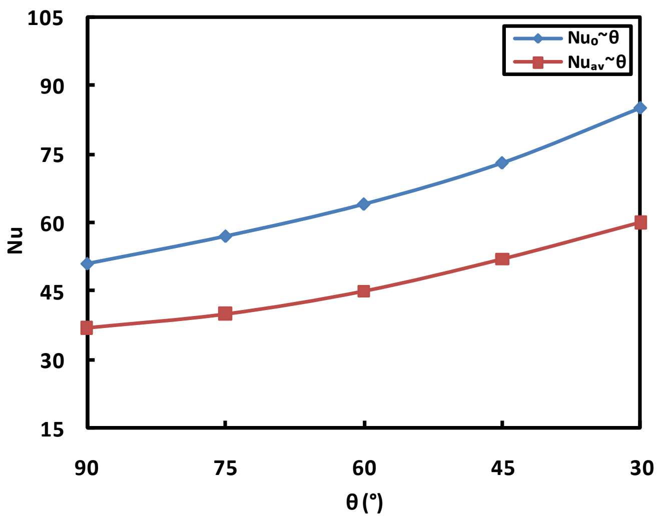

Figure 6. Variation of Nusselt number with jet inclination

Figure 6 depicts both stagnation and average Nusselt number variations with the jet inclinations. The stagnation Nusselt numbers are 51, 57, 64, 73 and 85 for the jet inclinations of 90, 75, 60, 45 and 30°, respectively. Likewise, the average Nusselt numbers are 37, 40, 45, 52 and 60 for the jet inclinations of 90, 75, 60, 45 and 30°, respectively. Furthermore, both stagnation and average Nusselt numbers increase with the decrease of jet inclination (i.e. with the increase of obliquity). It is also apparent that the variations of both stagnation and average Nusselt numbers with jet inclinations are nearly linear.

IV.CONCLUSION

Exhaustive experiments are conducted and measurements are made for various combinations of interdependent and interrelated parameters involved with the water jet impingement on a heated plate to study the heat transfer characteristics. Comprehensive experimental studies on the effects of the nozzle to plate spacing and the jet inclination are carried out as they greatly influence the thermal performance of the impinging water jets. Based on the measurements and the data derived, the trends of the results with regard to various parameters are observed to be along expected lines. However, the enhancement in the stagnation and averaged heat transfer characteristics with water jet inclination is newly identified. In addition, the appropriate combinations of the key interdependent and interconnected 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. However, 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. Nevertheless, with the present experimental conditions the jet inclination of 60º together with the nozzle to plate spacing of 25 mm gives moderate heat transfer behavior and is the optimum. Hence, the present combination can be utilized directly in industries to enhance heat transfer and for cooling of electronic systems.

V. REFERENCES

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