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Experimental Investigations on Influences of Nozzle Size and Jet Reynolds Number 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 jet Reynolds number (800-3200) and nozzle diameter (3-6 mm). 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 nozzle diameter of 5 mm together with the jet Reynolds number of 2400 gives moderate heat transfer behavior and is the optimum one.

***Keywords*: Water jet, Target plate, Nozzle diameter, Jet Reynolds number**

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. The current trend of miniaturization of electronic components together with the increasingly high circuit densities has given rise to alarmingly high power densities. This trend towards miniaturization involves high heat flux in various applications and has provided motivation, during the past several years, for significant volume of research related to the design of effective cooling schemes.

Electronic cooling needs have grown at a tremendous pace since the development of integrated circuit technology as well. Conventional cooling techniques used in the past such as free and forced convection of air are no longer adequate for the high heat flux applications. An alternative form of cooling which has captured much attention in recent years is the use of liquid jet impingement to circumvent the problem of high thermal resistance associated with the above-mentioned techniques.

II. LITERATURE REVIEW

Conventional air cooling is insufficient in most cases to help sustain and safeguard the electronics components from the thermal failure. Roy et al. [1] studied both experimentally and numerically heat transfer of an inclined surface subject to an impinging air flow. Lee et al. [2] have examined the effects of nozzle diameter on impinging jet heat transfer and fluid flow. Eren and Celik [3] investigated the heat transfer characteristic of a heated flat plate by an obliquely impinging air jet. Additionally, Agostini et al. [4] illustrated the state of art of the high heat flux cooling technologies. Behera et al. [5] investigated numerically on the heat transfer of interrupted impinging air jets used for electronics cooling.

Furthermore, Sagot et al. [6] studied the jet impingement heat transfer on a flat plate at a constant wall temperature by using gas jets. Saha and Dutta [7] developed heat transfer correlations for PCM-based heat sinks with plate fins. Narasimhan et al. [8] investigated on thermal management using the bi-disperse porous medium approach. Besides, Yu et al. [9] also carried out numerical simulation on the effect of turbulence models on impingement cooling of double chamber model. Nguyen et al. [10] investigated cooling effect by sub-zero cold air jet in the grinding of a cylindrical component.

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 for Reynolds number of range 800-3200. In addition, to the best of the authors' knowledge, there is not a single comprehensive experimental study pertaining to the effects of the jet velocity rate and nozzle diameter 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 jet Reynolds number (800-3200) and jet diameter (3-6 mm) 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.

III.TEST APPARATUS AND METHOD

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 jet flow rate, input voltage and current to the heater, etc) 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.

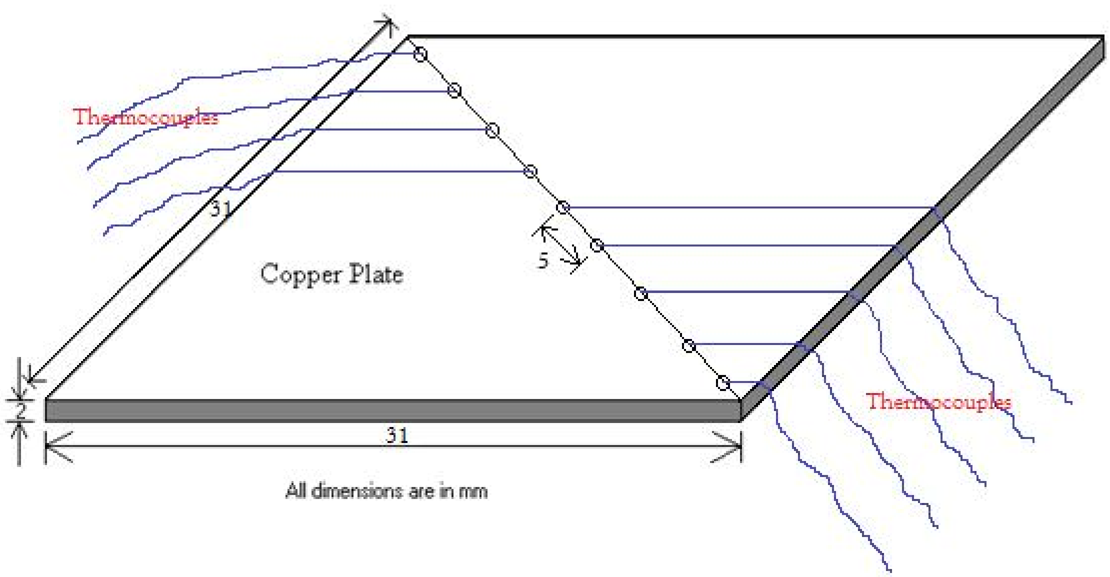

Figure 1. Schematic sketch of the physical model

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 (i.e. the target plate) 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, $h_i = \dfrac{Q_{out}}{A_h(T_{si} - T_j)}$; $Q_{out} = VI$ (1)

So, average heat transfer coefficient, $\bar{h} = \dfrac{\sum h_i A_i}{\sum A_i}$ (2)

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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 = $Nu_i = \frac{h_i d}{k}$ (4)

Now, average Nusselt number, $\overline{Nu} = \frac{\bar{h}d}{k}$ (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 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.

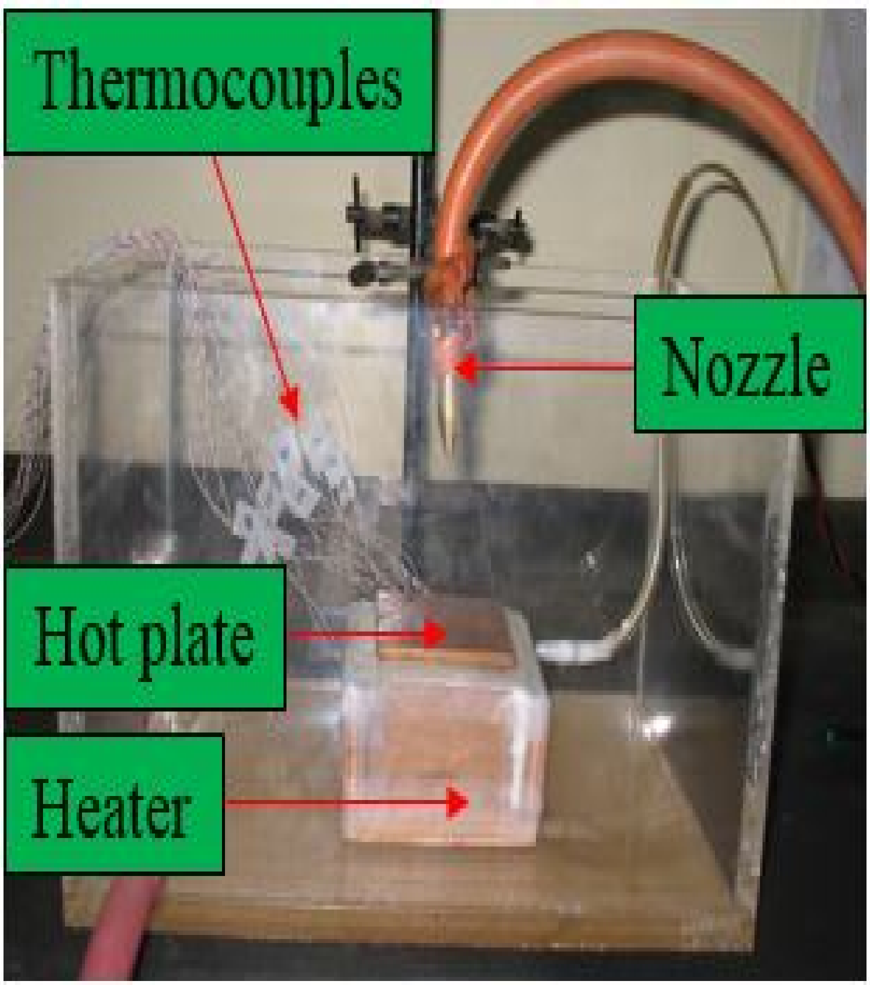

Figure 2. Photograph of the experimental setup

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C. Experimental procedure

It involves the measurements of following key parameters.

1). 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.

2). 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

Item	Material	Class	Size (mm)	Temperature Range (°C)	Temperature uncertainty	Quantity
Thermocouple	Copper-Constantan	PTFE coated T-type	0.205	0-200°C	±0.004T°C	9

IV. 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 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/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 diameter

Apart from the stated case involving nozzle diameter of 5 mm, in this study three more nozzle diameters of 3, 4 and 6mm are taken into account with the said experimental conditions. The results thus obtained are compared to investigate the role and effect of the nozzle diameter.

Figure 3 demonstrates the local Nusselt number variation with radial distance from the stagnation point for the stated nozzle diameters altogether for the comparative study. For the axisymmetric water jet with nozzle diameter of 3 mm, the local Nusselt number decreases from 73 at the centre to 36 at the edge of the target plate. Likewise, for the axisymmetric water jets with nozzle diameters of 4, 5 and 6 mm, the local Nusselt numbers decrease from 60, 51 and 45 at the centre to 30, 26 and 23 at the edge of the target plate, respectively. It is observed that the local Nusselt number profile is relatively flat for the bigger nozzle diameter. This is because of low water velocity associated with the impinging jet.

Figure 4 illustrates the variation of both stagnation and average Nusselt number with the nozzle diameter. The stagnation Nusselt numbers are 73, 60, 51 and 45 for the nozzle diameters of 3, 4, 5 and 6 mm, respectively. Likewise, the average Nusselt numbers are 51, 42, 37 and 32 for the nozzle diameters of 3, 4, 5 and 6 mm, respectively. From both the quoted figures, it is obvious that the local, stagnation and average Nusselt numbers decrease with nozzle diameter. As expected, this result is owing to the higher nozzle diameter (causing the lower jet velocity or jet Reynolds number) giving rise to the carrying away of the less heat from the target plate at relatively slower rate. From figure 10, it is also evident that the variation of both stagnation and average Nusselt numbers with the nozzle diameter is almost linear.

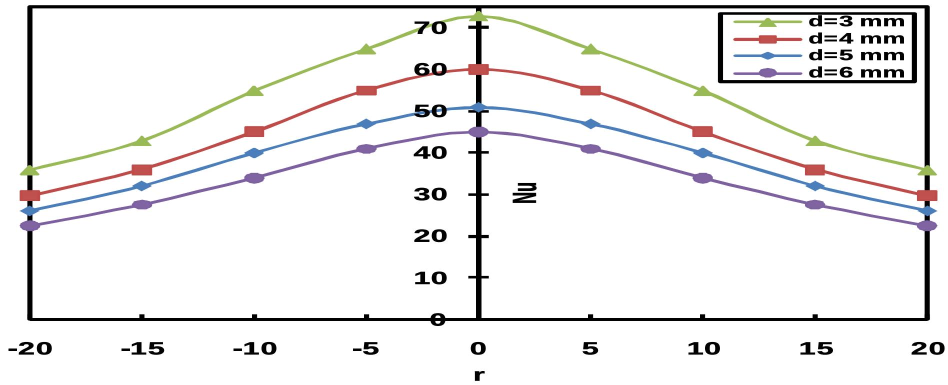

Figure 3. Variation of local Nusselt number with radial distance

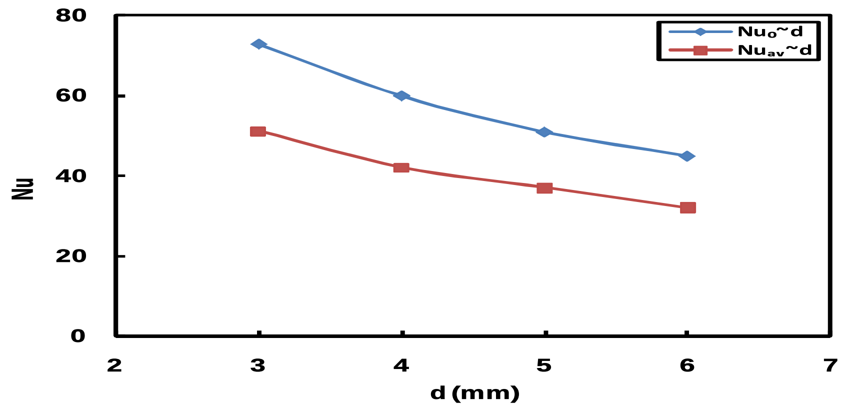

Figure 4. Variation of Nusselt number with nozzle diameter

B. Effect of jet Reynolds number

Apart from the stated case involving jet Reynolds number of 2400, in this study three more jet Reynolds numbers of 800, 1600 and 3200 (corresponding to the jet flow rates of 10, 20 and 40 lph respectively) are considered with the said experimental conditions. The corresponding results are analyzed and compared to investigate the role and effect of the jet Reynolds number.

Figure 5 illustrates the local Nusselt number distribution over the entire target plate surface for the already stated jet Reynolds numbers altogether for the comparative study. For the axisymmetric water jet with 3200 jet Reynolds number, the local Nusselt number decreases from 60 at the centre to 28 at the edge of the target plate. Likewise, for the axisymmetric water jets with 2400, 1600 and 800 jet Reynolds numbers, the local Nusselt numbers decrease from 51, 43 and 30 at the centre to 26, 23 and 19 at the edge of the target plate, respectively. As expected, the local Nusselt number profile appears to be symmetric with peak at the stagnation point. It is due to the monotonic increase in the local temperature from centre to edge of the target plate. This is owing to the continuous heat gain by water from the target plate while flowing on it. Besides, it is also observed that the local Nusselt number profile is relatively flat for the smaller jet Reynolds number. This is because of less water flow rate associated with the impinging jet.

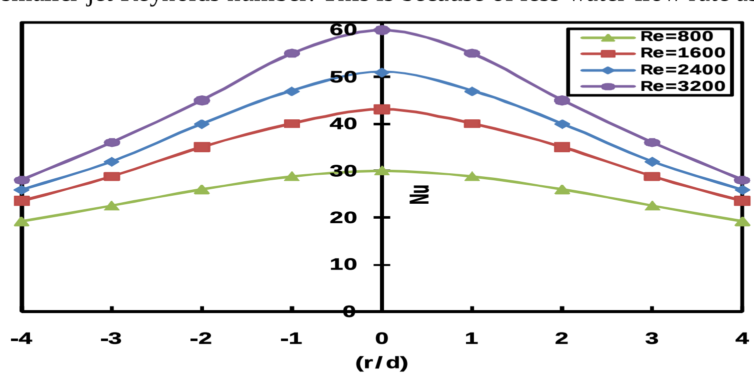

Figure 5. Variation of local Nusselt number with radial distance

Figure 6 shows the variation of both stagnation and average Nusselt number with the Reynolds number. The stagnation Nusselt numbers are 30, 43, 51 and 60 for Reynolds numbers of 800, 1600, 2400 and 3200, respectively. Likewise, the average Nusselt numbers are 25, 33, 37 and 42 for Reynolds numbers of 800, 1600, 2400 and 3200, respectively. From both the stated figures, as expected, evidently the local, stagnation and average Nusselt numbers increase with jet Reynolds number. It is because of the higher jet flow rate (causing the higher jet velocity or jet Reynolds number) results in the carrying away of more heat from the target plate at faster rate. Also, from figure 6, it is quite clear that the variations of both stagnation and average Nusselt numbers with jet Reynolds number is nearly linear.

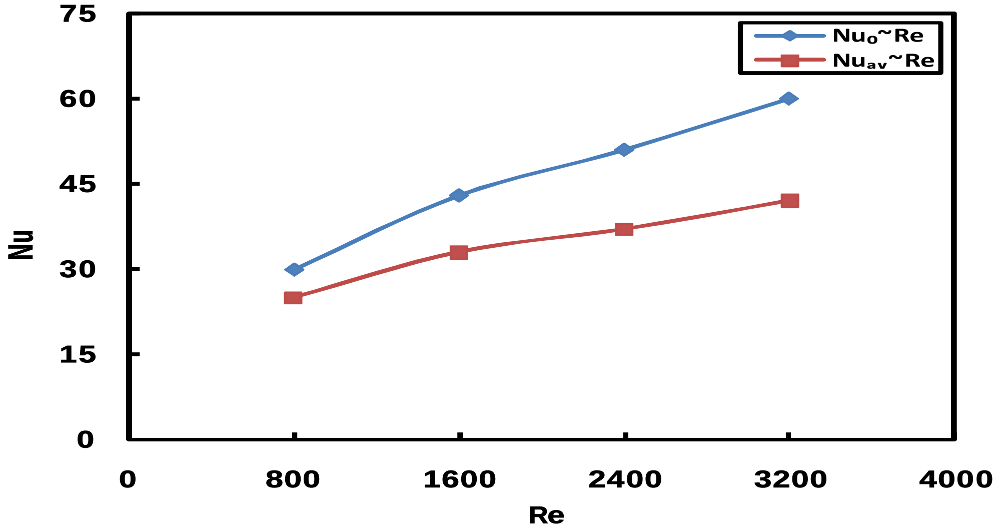

Figure 6. Variation of Nusselt number with Reynolds number

V. CONCLUSIONS

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 jet Reynolds number and nozzle diameter 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. 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 nozzle diameter of 5 mm together with the jet Reynolds number of 2400 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.

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