Brazilian Journal
of Chemical
Engineering

ISSN 0104-6632
Printed in Brazil
www.abeq.org.br/bjche

Vol. 36, No. 01, pp. 221 - 228, January - March, 2019
dx.doi.org/10.1590/0104-6632.20190361s20170546

# EFFECT OF STEEL PLATE THICKNESSES AND FLUID FLOW CHARACTERISTICS OF AN IMPINGING AIR JET ON HEAT TRANSFER AT THE STAGNATION POINT

K. Jeyajothi[1] and P. Kalaichelvi[1*]

[1] National Institute of Technology, Department of Chemical Engineering, Tamil Nadu, India.
E-mail: kalai@nitt.edu, ORCID: 0000-0001-7396-3500

*(Submitted: October 29, 2017 ; Revised: December 16, 2017 ; Accepted: March 15, 2018)*

**Abstract** - The design and process of heat transfer elements, from its source to heat sink through various media vary according to the product and system concerned. The effect of thickness of a steel plate on heat transfer characteristics with an impinging air jet at the stagnation point was studied. Experiments were carried out with different nozzles, heights, and velocities. It was concluded that the heat transfer increases with the increase in velocity and the increase in Reynolds Number, but decreases with the increase in nozzle height from the impinging point. A maximum stagnation Nusselt number of 309.06 was obtained at the optimum conditions with a plate thickness of 3 mm. The new empirical correlations calculated from the experiments were in reasonable agreement with the equations in the literature and the deviation was less than 15%.
*Keywords*: Impinging Air Jet; Plate thickness; Stagnation point; Nusselt number.

## INTRODUCTION

Air jet impingement heat transfer has a greater ramification and wider applications such as turbine blade cooling, food processing, heat treatment cooling process and of course cooling of electronic devices. Different impinging gas jet configurations grouped according to the number of jets involved (single or arrays) and the geometric characteristics (round or slot) for different flow configurations (cross-flow or in-line) are analyzed to develop empirical fitting data using equations within their ranges of validity. Often, the heat transfer coefficient is reduced into dimensionless form as the Nusselt number and plotted against the Reynolds number with varying height-to-distance to form correlations which represent the experimental data obtained with specific jet configurations (Viskanta, 1993; Gardon and Cobonpue, 1962; Meola, 2009).

Lin et al. (1997) studied the heat transfer behavior of confined slot jet impingement for three Nusselt numbers, with empirical correlation of the effect on air-cooled multi-chip modules. Studies on the influence of local heat transfer characteristics of a circular cylinder subjected to a circular impinging jet in cross-flow were done with the parameters such as jet-to-plate distance, jet velocity and Reynolds number (Wang et al., 2014; Yasaswy et al., 2014). The heat transfer enhancement produced by adding arrays of triangular tabs on a long pipe was analyzed with turbulent round impinging jets (Gao et al., 2003). The role of impinging jet in a cross-flow on heat transfer performance was studied with several parameters including the jet-to-cross-flow mass ratio (2 to 8%), Reynolds number (1434 to 5735) and the jet diameter (2 to 4 mm) (Guoneng et al., 2014).

The thermal performance of a confined impinging slot jet was studied using Nusselt number by varying

* Corresponding author: P. Kalaichelvi - E-mail: kalai@nitt.edu

the Reynolds number and nozzle-to-plate spacing (Zukowski, 2013). The effect of nozzle-to-surface spacing, nozzle diameter and Reynolds number (6000 to 23000) on heat transfer was studied for electronic components (Anwarullah et al., 2012). Studies on heat transfer characteristics of confined twin jets were performed with Reynolds number (30,000 to 50,000), nozzle-to-plate spacing (0.5 to 4) and jet-to-jet spacing (0.5 to 2) (Ozmen, 2011). The heat transfer models and flow characteristics were identified by analyzing the correlation of stagnation and average Nusselt number using Reynolds number from 500 to 20,000 (Roy and Patel, 2003).

Turbulent flow regime confined impinging water jet from a square-edged nozzle was studied with the parameters like nozzle-to-target plate ratio (0.25 to 8.75) and Reynolds number (1350 to 17,300) (Jeffers et al., 2016). The analytical and experimental heat transfer characteristics of single and multiple turbulent jets of air impinging on flat plate surfaces were studied for circular, slot jet, and cross-flow for developing correlations (Livingood and Hrycak, 1973). Effect of nozzle-to-plate space (0.1 to 40) on heat transfer and fluid flow characteristics of submerged jet impinging was studied with air and water as the working fluid, and the correlation was developed for the stagnation point Nusselt number (Choo et al., 2016). Experimental and theoretical studies on convective heat transfer in outward airflow rotor-stator geometries with and without impinging jet were performed by Harmand et al. (2013). The influence of a cylindrical plenum with an in-line array of six jets with different nozzle geometries was studied using CFD by Marzec et al. (2014).

It was reported that a cooling performance in the stagnation region of 27% and an average of 36% was achieved by varying the experimental parameters (H/D = 0.5 to 3; outlet diameter = 1.2, 1.6 and 2 mm) for a plate of 90 x 200 mm (Rylatt et al., 2013). Local and average heat transfer characteristics of unconfined air jet impingement on a flat plate were studied for different nozzle geometry and Reynolds number for developing a correlation of the Nusselt number (Attalla and Salem, 2013; Trinh et al., 2016). Carlomagno and Laniro (2014) suggested that many experimental thermos-fluid-dynamic studies with short nozzle-to-plate distance would motivate the formation of further benchmarks for computational methods. Liu et al. (2016) performed a study on impingement heat transfer from a synthetic air jet through a diffusion - shaped orifices with angles ($0^o$, $60^o$ and $90^o$) and thickness (1 to 3 mm) having driven frequencies of 400 to 800 Hz. The uncertainty associated with the main features of the experimental data is estimated using the standard single-sample. The uncertainty of the effective thermal conductivity was ± 20%, while the uncertainty in measuring the flow velocity or Reynolds number was 1 to 2% (Kline and McClintock, 1953; Kim et al., 1993; Mahgoub, 2013). Evaporative cooling using air flow was also investigated (Rodrigues et al., 2017).

The present study is the first investigation conducted to analyze the effect of steel plate thickness on heat transfer and fluid flow characteristics of an impinging air jet on a heated flat plate. Though many studies on heat transfer and fluid flow characteristics using various parameters are reported in the literature, extensive information regarding the effect of nozzle diameter, nozzle-to-plate distance, velocity profile of jet and in-line jet configuration with the round long throat shaped nozzle is not available. This study is intended to generate such information and, for this study, an air jet with Reynolds number in the laminar to the transient range was used at the stagnation point. The objectives of this study include: (1) to classify the jet flow characteristics at the nozzle exit; (2) to explore the effects of jet Reynolds number and jet separation distance at the stagnation point; (3) to identify the best suitable plate thickness for achieving the maximum heat transfer under given conditions; (4) to propose new empirical correlations for evaluating stagnation heat transfer characteristics and (5) to compare the present experimental results with the theoretical values for validation.

From our previous study using a similar set up on heat transfer characteristics, namely stagnation, local and average heat transfers with their corresponding Nusselt number (Jeyajothi and Kalaichelvi, 2018), it was observed that maximum heat transfer occurred at the stagnation point. Hence, this study was restricted to stagnation point heat transfer characteristics.

## MATERIALS AND METHODS

### Experimental setup

The experimental setup consists of steel plates with different thicknesses, a heat source system for maintaining the plate temperature, a cooling air jet with a continuous supply of air with various nozzle diameters, an adjustable stand for varying nozzle-to-plate height and instrumentation for monitoring, controlling and measuring the parameters.

Since the majority of the earlier experimentation was carried out with circular plates of one fixed thickness, steel SS304 square plates of 150 mm side, with thicknesses of 2.0, 2.5 and 3.0 mm were chosen for this study (Figure 1). The geometrical center was taken as the air jet impinging – Stagnation – Point.

A Kapton heater (vari-volt type with 10 A capacity) was used as the heating element. It provides uniform heat flux. The target plate and the heater are inserted in an insulator assembly with the top surface of the plate flush with the top surface of the insulator as shown in Figure 2. Fire bricks were used to insulate the assembly of the plate and the heater so that the

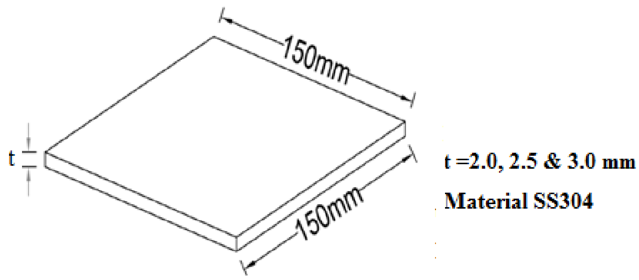


**Figure 1.** Design of flat plate.

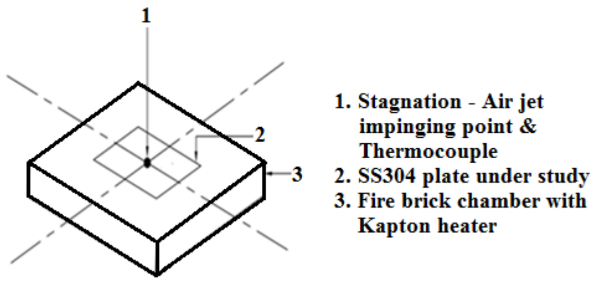


**Figure 2.** Plate insulator–heater box and thermocouple.

energy loss is minimized. The operating range of heater voltage and current were 109 to 113 V and 5.9 to 6.2 A, respectively.

The cooling air jet nozzles used for this study were made of brass and were round, long throat shaped with an inlet diameter of 9 mm and outlet diameter of 0.5, 1.0, 1.5, 2.0, 2.5 and 3.0 mm as shown in Figure 3. An adjustable stand (Figure 4) was used to locate the

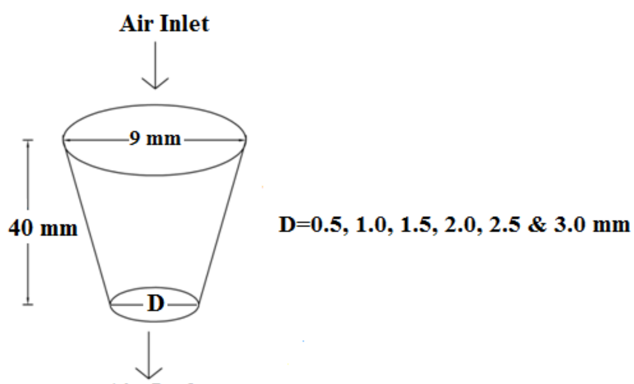


**Figure 3.** Nozzle design.

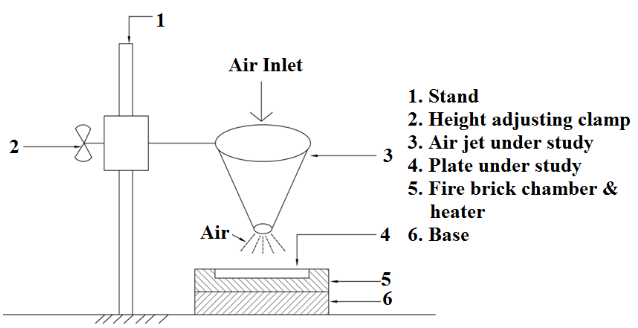


**Figure 4.** Representative picture of the air jet adjusting device.

nozzle at the desired heights of 2, 4 and 8 mm above the plate to understand the effect of air jet distribution on the heat transfer characteristics. An air compressor (CEC model A829) with the specifications of 7 kg/cm$^2$ pressure at 760 rpm speed with a tank capacity of 120 L was provided as an air supply source. The air flow was routed through a series of filters, regulator and also passed through a flow meter and regulating valve for controlling the airflow velocity at 6.4 m/s and 9.2 m/s for all the six nozzles under study.

Instrumentation and gauges were divided into two categories according to their functional use. A set of instruments was used for monitoring and controlling the input parameters such as air flow, pressure, velocity and heating element voltage/current/power as well as the nozzle height from the study plate surface. A thermocouple was used to measure the temperature, which is the important factor determining the Nusselt number and thereby the heat transfer characteristics. The parameters being measured and their corresponding instruments are shown in Table 1.

**Table 1.** Instrumentation used in the experimental setup.

| Parameter | Monitoring/ Controlling device |
|---|---|
| Air flow volume | Rotameter |
| Air jet velocity | Digital anemometer |
| Air temperature | Analog mercury thermometer |
| Air pressure | Pressure gauge |
| Air flow regulator | Flow control valve |
| Heater power consumption | Digital multimeter |
| Heater power control | Rheostat to vary resistance |
| Nozzle height | Calibrated variable stand |
| Temperature of the plate | Thermocouple ofthe Cr - Al type |

## Experimental study

The air jet was derived from the compressor (powered by an electric motor) through an air tank (which acts as a surge suppressor), rotameter, and injected in the center of the steady state heated plate under study. The air jet was projected onto the heated plate after it has attained a steady temperature and then the temperature of the plate was measured and recorded once it reached a stable temperature. Then, the stagnation Nusselt number which indicates the heat transfer characteristics, was derived. This experiment was carried out with the input variable combinations (6 nozzle diameter x 3 nozzle-to-plate height x 3 plate thickness x 2 air velocities) resulting in 108 output data in total. From the analysis and interpretation of the results, the setup with best heat transfer characteristic was determined.

## Flow characteristics and Empirical equations

The Reynolds number, an index of flow characteristics, was derived using Eq. (1). As the

Reynolds number is a function of nozzle diameter and the velocity of the air jet, it was maintained by controlling the volume of air flow with the use of the rotameter.

$$Re_D = \frac{Dv\rho}{\mu} \tag{1}$$

Since the stagnation point was taken for this study, x = 0. Therefore, the stagnation heat transfer coefficient and stagnation Nusselt number were calculated using Eq. (2) and (3). To ascertain the accuracy, three sets of repeat data were recorded under the same experimental conditions

$$h_S = \frac{tq}{\left(T_{w,0} - T_a\right)} \tag{2}$$

$$Nu_S = \frac{h_S D}{K} \tag{3}$$

**Uncertainty**

The overall uncertainty of the experimental data was the cumulative effect of the accuracies of the measuring instruments, i.e., heater input voltage ±2.4%, air speed ±0.5%, temperature of heated plate ±0.75% and plate surface area ±6.78%. Accordingly, the overall cumulative uncertainty was found to be 10.43% in the heat transfer coefficient and Nusselt number. This uncertainty value is less than the 15% deviation (Kline and McClintock, 1953; Kim et al., 1993), which is allowed for this type of experimentation.

## RESULTS AND DISCUSSION

The effect of the experimental parameters such as plate thickness, nozzle diameter, nozzle-to-plate height and air velocity on heat transfer was analyzed with the calculated stagnation Nusselt number. The relationship between the Reynolds number and $Nu_S$ was determined for the different jet velocities and the correlation was developed for these experimental conditions using empirical equations. The developed correlation was compared and validated with the empirical equations from previous studies.

**Effect of experimental parameters on $Nu_S$**

The relationships between plate thickness, nozzle location, height and stagnation Nusselt number for various nozzle diameters (0.5, 1, 1.5, 2, 2.5, 3 mm) at v = 6.4 and 9.2 m/s are shown in Figure 5 and Figure 6, respectively. $Nu_S$ increases with an increase in nozzle diameter because of increased air flow. For the same nozzle diameter, $Nu_S$ increased for higher air velocity

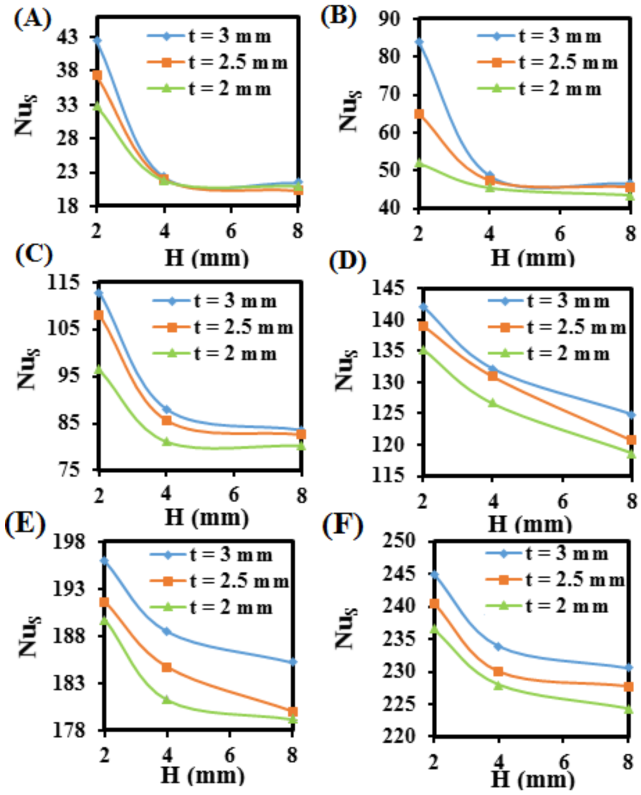


**Figure 5.** Relationship between t, H and $Nu_S$ at v = 6.4 m/s for various nozzle diameters (A to F: 0.5, 1, 1.5, 2, 2.5, 3 mm).

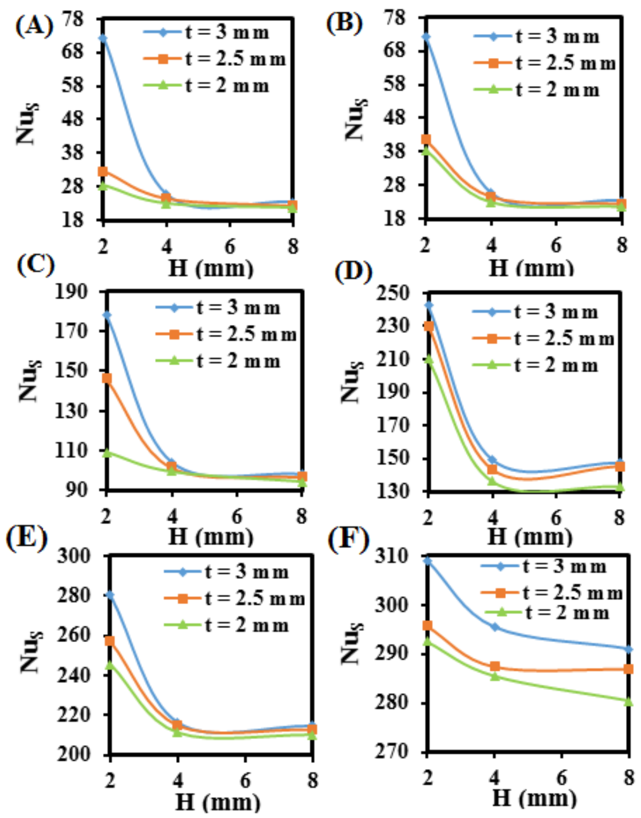


**Figure 6.** Relationship between t, H and $Nu_S$ at v = 9.2 m/s for various nozzle diameters (A to F: 0.5, 1, 1.5, 2, 2.5, 3 mm).

because the quantity of air impinging on the plate increases with velocity.

Higher $Nu_S$ was obtained when the plate thickness increased. For both air jet velocities, it could be observed that the maximum stagnation Nusselt number occurred when the 3 mm plate was used with the nozzle location height of 2 mm for each nozzle diameter. The surface heat transfer distribution depends on the location of the stagnation point (Gao et al., 2003; Guoneng et al., 2014; Zukowski, 2013). At the smallest jet-to-plate distance studied, heat transfer distribution on the compression side of the surface shows a monotonic rapid drop beyond the point of maximum. This is because the kinetic energy at the jet center decreases due to jet spreading. So, a small jet-to-plate distance should be used to achieve maximum heat transfer using the flat plate impinging jet in industrial processes.

### Relationship between $Re_D$ and $Nu_S$

The relationship between the Reynolds number of air flow (192.61 to 1661.26) obtained by varying nozzle diameter (0.5 to 3 mm) and the maximum $Nu_S$ obtained with the plate thickness (3 mm) at 2 mm nozzle location height for the two air velocities is shown in Figure 7. Similar graphs could be constructed for other plates and other nozzle location heights for analysis of the correlation with lesser $Nu_S$. As the nozzle diameter increases the Reynolds Number also increases because of a higher volume of air flow. Reports of the heat transfer analysis at the stagnation in the transient region were explained with fluid dynamic measurements (Zukowski, 2013; Ozmen, 2011; Livingood and Hrycak, 1973).

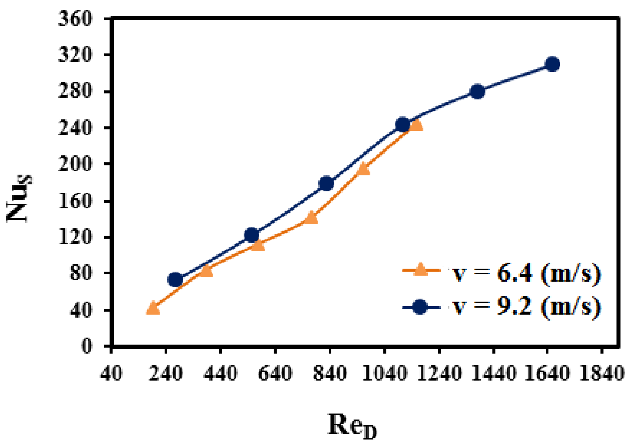


**Figure 7.** Maximum $Nu_S$ obtained for various $Re_D$.

### Correlation between heat transfer characteristics

The heat transfer characteristics represented by the Nusselt number are proportional to the Reynolds number (Livingood and Hrycak, 1973; Gardon and Cobonpue, 1962).

$$Nu \propto Re_D^a \tag{4}$$

The value of ‘a’ lies in between 0.50 and 0.98, depending upon the experimental values of Reynolds Number, Stagnation Nusselt number and nozzle configuration (Gardon and Cobonpue, 1962; Viskanta, 1993; Meola, 2009; Mahgoub, 2013). The equation gets converted into:

$$Nu = C_1 \left(Re_D\right) \wedge C_2 \tag{5}$$

where, $C_1$ and $C_2$ are empirical constants derived from experimental data. This general equation was modified by Gardon and Cobonpue, (1962), Lin et al. (1997) and Meola, (2009) based on their experimental conditions and the following equations were derived with different values of $C_1$ and $C_2$, respectively.

$$Nu_s = 0.3 Re_D{}^{0.68} \tag{6}$$

$$Nu_s = 4.315 Re_D{}^{0.5} \tag{7}$$

$$Nu_s = 0.46 Re_D{}^{0.63} \tag{8}$$

These equations were obtained with the configurations of the experimental setups as indicated in Table 2. Similar to the above set of equations, the following equations (Eq. (9) and (10)) were derived from the experimental values of this study.

$$Nu_s = 0.287 Re_D{}^{0.95} \tag{9}$$

$$Nu_s = 0.628 Re_D{}^{0.84} \tag{10}$$

For the air jet velocity of 6.4 m/s, Reynolds numbers between 192.61 and 1155.66 were used for correlation in Eq. (9) and a correlation coefficient of $R^2 = 0.991$ was obtained. For the air jet velocity of 9.2 m/s, Reynolds numbers between 276.88 and 1661.26 were used for correlation in Eq. (10) and a correlation coefficient of $R^2 = 0.994$ was obtained.

Figure 8A and 8B shows the comparison of $Nu_S$ values obtained using empirical equations from the literature (Eq. (6) to (8)) with the values obtained using Eq. (9) and Eq. (10), respectively, for different Reynolds Number. The variation in the values for different equations was due to changes in experimental configurations. From Figure 9A, it could be observed that the average deviation between the experimental $Nu_S$ values (at v = 6.4 m/s) and the $Nu_S$ values obtained using Eq. (9) (which was derived at v = 6.4 m/s) was 6.4%, while the maximum deviation was 11.58 %. When Eq. (10) (which was derived at v = 9.2 m/s) was

**Table 2.** Experimental Configurations of various studies.

| Reference | D (mm) | H/D | $Re_D$ | Jet configuration | Nozzle geometry | Remarks |
|---|---|---|---|---|---|---|
| Lin et al. (1997) | 5 | 1-20 | 190-1537 | SA | OP | Eq. (6) |
| Gardon and Cobonpure (1962) | 1.58-12.7 | 4-20 | 250-15000 | SA | R-LTN | Eq. (7) |
| Meola (2009) | 2-12 | 1.6-20 | 200-100000 | SA-IN-CR-OP | RN & RectO | Eq. (8) |
| Present study | 0.5-3 | 0.67-16 | 192.61-1155.66<br>276.88-1661.26 | IN | R-LTN | Eq. (9)<br>Eq. (10) |

Abbreviations: SA = square array; IN = inline; CR = cross flow; OP = jet orifice plate; RLTN = round long throat nozzle; RN = round nozzle; RectO = rectangular opening.

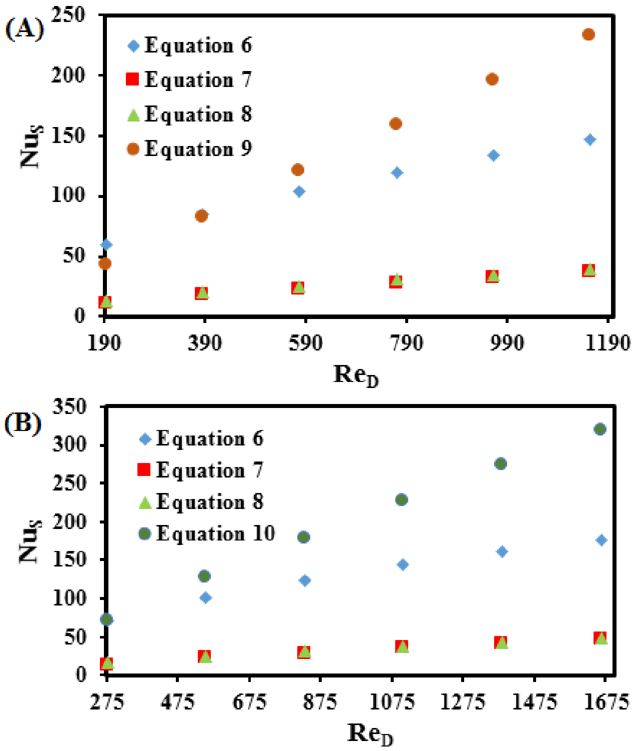


**Figure 8.** Correlation between $Re_D$ and $Nu_S$ obtained from empirical equations (A) v= 6.4 m/s; (B) v = 9.2 m/s.

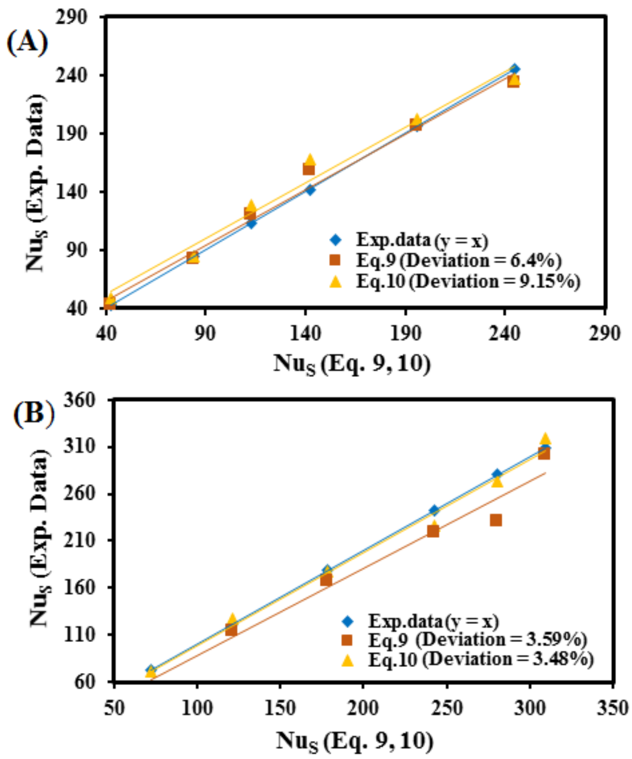


**Figure 9.** Deviation between $Nu_S$ values obtained from the experimental data and from the derived empirical equations (A) v= 6.4 m/s; (B) v = 9.2 m/s.

applied for the experimental values at v = 6.4 m/s, the average deviation was 9.15%. From Figure 9B, the average deviation between experimental values at v = 9.2 m/s and values using Eq. (10) was 3.48%, while the maximum deviation was 6.55%. When Eq. (9) (which was derived at v = 6.4 m/s) was applied for the experimental values at v = 9.2 m/s, the average deviation was 3.59%.

Therefore, it could be determined that the empirical equation derived at the specific velocity has good fit to the experimental values at that velocity and, even if the empirical equation obtained for the other velocity was used, the deviation was in the acceptable range (< 15%) (Gardon and Cobonpue, 1962; Lin et al., 1997; Meola, 2009; Ozmen, 2011; Zukowski, 2013). This shows that the derived empirical equations are applicable for Reynolds numbers from laminar to transient range. Further studies should be carried out to analyze the heat transfer characteristics at Reynolds numbers in the turbulent range.

## CONCLUSIONS

As the heat transfer is reduced when the nozzle is shifted away from the surface of the plate, the stagnation Nusselt number was at the peak for the nozzle location height of 2 mm when compared to 4 and 8 mm, for all the nozzle diameters and air velocities. The plate with a greater thickness (3 mm) had better heat transfer characteristics and the increase in air velocity always showed the higher stagnation Nusselt number for different operating conditions. Variations in nozzle diameter and air velocity resulted in air flow conditions from laminar to transient range with Reynolds number from 192.61 to 1661.26. These changes in air flow regime had a major impact on the heat transfer characteristics as the stagnation Nusselt number obtained varied from 21.5 to 309.06 at the same plate thickness (3 mm). Two empirical equations were derived to correlate stagnation Nusselt number with Reynolds number at different air velocities and the calculated values from these equations had < 5% average deviation and < 12% maximum deviation when compared with experimental results. Hence, this experimental configuration could be used to gain optimum heat transfer characteristics by using an impinging air jet on a flat plate.

## NOMENCLATURE

| | |
|---|---|
| a | Empirical power constant |
| $C_1$, $C_2$ | Empirical constants |
| D | Diameter of nozzle, (m) |
| $h_S$ | Stagnation heat transfer coefficient, ($W\ m^{-2}\ K^{-1}$) |
| K | Thermal conductivity of air ($W\ m^{-1}\ K^{-1}$) |
| $Nu_S$ | Stagnation Nusselt number |
| q | Convective heat flux on the top surface of the plate ($W\ m^{-2}\ K^{-1}$) |
| $R^2$ | correlation coefficient |
| $Re_D$ | Jet Reynolds number |
| t | Thickness of the plate (m) |
| $T_a$ | Temperature of air (K) |
| $T_{w,x}$ | Wall temperatures of the plate at a distance x, (K) |
| v | Velocity of air ($m\ s^{-1}$) |

***Greek symbols***

| | |
|---|---|
| μ | Viscosity of air ($m^2\ s^{-1}$) |
| ρ | Density of air ($kg\ m^{-3}$) |

## REFERENCES


Anwarullah, M., Rao, V.V. and Sharma, K.V., Effect of Nozzle Spacing on Heat Transfer and Fluid Flow Characteristics of an Impinging Circular Jet in Cooling of Electronic Components, Int. J. of Thermal & Environmental Engineering, 4, 7-12 (2012). https://doi.org/10.5383/ijtee.04.01.002

Attalla, M. and Salem, M., Effect of nozzle geometry on heat transfer characteristic from a single circular air jet, Appl. Therm. Eng., 51, 723-733 (2013). https://doi.org/10.1016/j.applthermaleng.2012.09.032

Carlomagno, G.M. and Laniro, A., Thermo-fluid-dynamics of submerged jets impinging at short nozzle-to-plate distance: A review, Exp. Therm Fluid Sci., 58, 15-35 (2014). https://doi.org/10.1016/j.expthermflusci.2014.06.010

Choo, K., Friedrich, B.K., Glaspell, A.W. and Schilling, K.A., The influence of nozzle-to-plate spacing on heat transfer and fluid flow of submerged jet impingement, Int. J. Heat Mass Transfer, 97, 66-69 (2016). https://doi.org/10.1016/j.ijheatmasstransfer.2016.01.060

Gao, N., Sun, H. and Ewing, D., Heat transfer to impinging round jets with triangular tabs, Int. J. Heat Mass Transfer, 46, 2557-2569 (2003). https://doi.org/10.1016/S0017-9310(03)00034-6

Gardon, R. and Cobonpue, J., Heat transfer between a flat plate and jets of air impinging on it, International Developments in Heat Transfer. ASME, New York, 454-460 (1962).

Guoneng, L., Youqu, Z., Guilin, H. and Zhiguo, Z., Convective Heat Transfer Enhancement of a Rectangular Flat Plate by an Impinging Jet in Cross Flow, Chin. J. Chem. Eng., 22(5), 489-495 (2014). https://doi.org/10.1016/S1004-9541(14)60060-4

Harmand, S., Pelle, J., Poncet, S. and Shevchuk, I.V., Review of fluid flow and convective heat transfer within rotating disk cavities with impinging jet, Int. J. Therm. Sci., 67, 1-30 (2013). https://doi.org/10.1016/j.ijthermalsci.2012.11.009

Jeffers, N., Stafford, J., Conway, C., Punch, J. and Walsh, E., The influence of the stagnation zone on the fluid dynamics at the nozzle exit of a confined and submerged impinging jet, Exp. Fluids, 57(2), 17 (2016). https://doi.org/10.1007/s00348-015-2092-6

Jeyajothi, K. and Kalaichelvi, P., Augmentation of Heat Transfer and Investigation of Fluid Flow Characteristics of an Impinging Air Jet on to a Flat Plate, Arab. J. Sci. Eng., (2018). https://doi.org/10.1007/s13369-018-3511-9

Kim, J.H., Simon, T.W., and Viskanta, R., Journal of heat transfer policy on reporting uncertainties in experimental measurements and results, J. Heat Transfer, 115(1), 5-6 (1993). https://doi.org/10.1115/1.2910670

Kline, S.J., and McClintock, F.A., Describing uncertainties in single-sample experiments, Mechanical Engineering, 75, 3-8 (1953).

Lin, Z.H., Chou, Y.J. and Hung, Y.H., Heat transfer behaviours of a confined slot jet Impingement, Int. J. Heat Mass Transfer, 40, 1095-1107 (1997). https://doi.org/10.1016/0017-9310(96)00135-4

Liu, Y.H., Chang, T.H. and Wang, C.C., Heat transfer enhancement of an impinging synthetic air jet using diffusion-shaped orifice, Appl. Therm. Eng., 94, 178-185 (2016). https://doi.org/10.1016/j.applthermaleng.2015.10.054

Livingood, J.N.B. and Hrycak, P., Impinging Heat transfer from turbulent air jets to flat plates - A literature survey. Lewis Research Centre, NASA, Washington (1973).

Mahgoub, S.E., Forced convection heat transfer over a flat plate in a porous medium, Ain Shams Engineering Journal, 4, 605-613 (2013). https://doi.org/10.1016/j.asej.2013.01.002

Marzec, K. and Kucaba-Pietal, A., Heat transfer characteristic of an impingement cooling system with different nozzle geometry, J. Phys.: Conf. Ser., 530, 012038 (2014). https://doi.org/10.1088/1742-6596/530/1/012038

Meola, C., New Correlation of Nusselt Number for Impinging Jets, Heat Transfer Engineering, 30(3), 222-228 (2009). https://doi.org/10.1080/01457630802304311

Ozmen, Y., Confined impinging twin air jets at high Reynolds numbers, Exp. Therm Fluid Sci., 35, 355-363 (2011). https://doi.org/10.1016/j.expthermflusci.2010.10.006

Rodrigues, L.G.G., Parisotto, E.I.B., Carciofi, B.A.M. and Laurindo, J.B., Experimental approach to assess evaporative cooling under forced air flow, Braz. J. Chem. Eng., 34(1), 171-181 (2017). https://doi.org/10.1590/0104-6632.20170341s20150433

Roy, S. and Patel, P., Study of Heat transfer for a pair of rectangular jets impinging on an inclined surface, Int. J. Heat Mass Transfer, 46, 411-425 (2003). https://doi.org/10.1016/S0017-9310(02)00295-8

Rylatt, D.I. and O'Donovan, T.S., Heat transfer enhancement to a confined impinging synthetic air jet, Appl. Therm. Eng., 51, 468-475 (2013). https://doi.org/10.1016/j.applthermaleng.2012.08.010

Trinh, X.T., Fenot, M. and Dorignac, E., The effect of nozzle geometry on local convective heat transfer to unconfined impinging air jets, Exp. Therm Fluid Sci., 70, 1-16 (2016). https://doi.org/10.1016/j.expthermflusci.2015.08.006

Viskanta, R., Heat Transfer to Impinging Isothermal Gas and Flames Jets, Exp. Therm Fluid Sci., 6, 111-134 (1993). https://doi.org/10.1016/0894-1777(93)90022-B

Wang, X.L., Motala, D., Lu, T.J., Song, S.J. and Kim, T., Heat transfer of a circular impinging jet on a circular cylinder in cross flow, Int. J. Therm. Sci., 78, 1-8 (2014). https://doi.org/10.1016/j.ijthermalsci.2013.11.005

Yasaswy, N.S., Saroj, S., Hindasageri, V. and Prabhu, S.V., Local heat transfer distribution of an impinging air jet through a cross flow, Int. J. Therm. Sci., 79, 250-259 (2014). https://doi.org/10.1016/j.ijthermalsci.2014.01.006

Zukowski, M., Heat transfer performance of a confined single slot jet of air impinging on a flat surface, Int. J. Heat Mass Transfer, 57, 484-490 (2013). https://doi.org/10.1016/j.ijheatmasstransfer.2012.10.069