Flow structure and cooling behavior of air impingement on a target plate

Research Article

Adnan A. Abdul Rasool¹, Faik A. Hamad^{2*}

¹ Mech. Eng. Dept, college of Engineering, Al-Mustansiriya University, Baghdad, Iraq

² School of Science & Engineering, Teesside University, Middlesbrough, TS1 3BA, UK

Received 10 December 2012; accepted 19 March 2013

Abstract: An experimental data of flow field, pressure coefficient and heat transfer of a jet impinging normally on a flat target plate are presented. The measurements of temperatures and static pressures were carried out for flow from three orifices of 5, 10 and 20 mm diameter for orifice-to-target plate distances of 5, 10, 25, 50, 70, 100 and 120 mm from the orifice exit. The axial development of flow structure of the jet from the orifice was investigated by measuring the radial jet velocity distributions at the same axial distances used to measure heat transfer and static pressure. The results show that pressure coefficients distributions on the target plate are similar to the velocity distributions in the impinging jet which indicates the strong relationship between the two parameters. The pressure coefficients from large orifice diameter are higher than the values from the small orifice diameter for same orifice-to-target plate distance. The results also show a nonlinear increase of heat transfer rate with orifice size and the ratio of axial distance to orifice diameter (X/d). The nonlinear behaviour may be attributed to the complex nature of flow structure at the stagnation region. The high velocity gradients at the stagnation zone leads to higher turbulence and comparatively higher values of heat transfer rates for large orifice diameter.

Keywords: Jet impingement • Heat transfer • Pressure coefficient • Wall jet velocity

© Versita sp. z o.o.

1. Introduction

Air impingement is one of the efficient solution used in different industrial applications such as cooling of electronics systems, glass manufacturing industry, gas turbines blade internal cooling, etc. Heat transfer by impingement are also studied as side effects of vertical/short take-off and landing jet devices, for example in the case of direct lift propulsion systems in vertical/short take-off and landing aircraft. Figure 1 shows a schematic of

flow structure and the regions of the jet impinging normal to the target plate. This technique produces a very high rate of heat removal through forced convection especially at the stagnation zone of the impingement.

The flow structure and the mechanism of heat transfer are studied by many researchers. The following literature review gives a summary of some recent published studies in this area. San *et al.* [1] carried out an experimental study of air impingement orthogonally on a flat plate. The air jet is constrained to flow through two exits in opposite directions. Four orifice diameters of 3, 4, 6 and 9 mm were used with jet Reynolds number is in the range of 30,000 – 67,000 and orifice distance to diameter ratio = 2. Their results show that for the same

*E-mail: F.Hamad@tees.ac.uk

Figure 1. Schematic of flow structure of the jet and the different zones on the target plate surface.

Reynolds number a smaller orifice diameter give a lower Nusselt number but for a diameter greater than 6 mm, the influence of jet hole diameter on Nusselt number will tend to decrease. The measured local surface temperature shows that transition from impingement region to a wall jet region take place at (r/d) values close to 1.0. Behina *et al.* [2, 3] used an elliptic relaxation turbulence model to simulate the flow and heat transfer in circular confined and unconfined impinging jet configurations. Their results show that the effect of confinement is only significant in very low orifice distance to diameter ratio < 0.25 and that the flow characteristics in the orifice strongly affect the heat transfer rates, specially in the stagnation region with 30% difference in Nusselt number is obtained with different velocity profiles were used. Goldstien and Bahbahani [4] reported a local heat transfer coefficients measured on a plate exposed to an impinging air jet with tested orifice diameter 12 mm with and without a cross flow of air. The results show that the peak heat transfer coefficients disappear with cross flow at the large orifice to plate spacing while the peak heat transfer coefficients increase at small spacing cross flow. Lee *et al.* [5] examine the heat transfer from a flat plate to a fully developed axisymmetric impinging jet from a nozzle of 11 mm diameter. Variable nozzle to plate distance ratios were tested. Variation of Nusselt number at stagnation point was correlated with $(Re)^n$ show that n is in the range of 0.5–0.7 depending on plate – orifice distance to diameter ratio. The secondary peak of heat transfer rate was observed at the hydrodynamic transition of flow from laminar to turbulent conditions. The secondary peak was observed for all $X/d < 3$ and its location shifts towards the stagnation point with a decrease in X/d . The hydrodynamic transition may be explained by referring to the experimental data reported by Knowles and Myszkowski [6] and O'Donovan and Murray [7]. The results from both papers showed that shear stress distribution along the

wall in the radial direction sharply increases for low X/d ratios from r/d of about 1 and reach a peak at around r/d of about 2 and then drop. The turbulent intensities of wall jet at $r/d = 2$ and was highest for $X/d = 2$. Both studies showed that secondary peak in heat transfer correspond in location to the peaks in turbulence fluctuation. Mohanty and Tawfek [8] studied the local heat transfer distribution on a flat plate for three nozzle diameters of 3, 5, and 7 mm and variable values plate – orifice distance to diameter ratio. Their results showed that the heat transfer had a maximum at the impingement (stagnation) point and then decays exponentially with radial distance. The non dimensional heat transfer rate was presented as a function of effective area for the nozzle. Lytle & Webb [9] measured local heat transfer coefficients and static pressure distribution for nozzle diameters of 6.9 and 10.9 mm at low nozzle to plate spacing. Pressures are noticed to decrease monotonically with radius on target plate and reached negative values at outward positions for high jet velocities. They found that the variation of Nusselt number on the target plate were varied with the type of flow encountered at impinging point and the location of flow transition point and turbulence levels in the flow. Mi *et al.* [10] performed an experimental study on the mixing performance for a number of geometries including nozzle, sharp edge orifice and long tube as shown in Figure 2.

The orifice is found to provide the greatest rate of mixing, with the turbulence intensity was low at the orifice centre and higher at the orifice edges. Baydar and Ozmen [11] carried out an experimental and numerical study for very high Reynolds number impingement of air on a target plate ($Re = 30000 - 50000$) with low values of non dimensional nozzle to plate spacing. Their results show that a sub atmospheric region occurs on the impingement plate at nozzle to plate spacing up to 2 for the range of Reynolds numbers studied. The results obtained from their numerical results using the $K - \epsilon$ turbulence model are in agreement with their experimental results except for nozzle to plate spacing less than one. Hofmann *et al.* [12] investigated the flow structure and heat transfer from a single round jet impinging perpendicularly on a flat plate. Based on the experimental results, correlations for heat transfer coefficients were developed. Flow structure in a free jet has also been examined.

The primary aim of this paper is to present results of combined flow field and heat transfer study of jet impinging on the surface. The literature review presented above shows that in spite of the extensive studies on flow structure and heat transfer of the impinging air on a target plate, but there is a need for more research on the combined flow field, pressure coefficient and heat

Figure 2. The different geometries (orifice plate, nozzle and long tube) used to produce an impinging jet.

transfer rate of air jet impinging normally on flat plate. The purpose of the present work is to study the flow structure and heat transfer of impinging jet at the region bounded between orifice of different diameters and the target plate positioned at different axial distances in front of it. The study will lead to an understanding of the relationship between the heat transfer on the target plate and the flow structure in the region between them. The heat transfer from the target plate and the flow structure between the orifice and target plate are investigated for three different orifice diameters of 5, 10 and 20 mm at different axial distances (5, 10, 25, 50, 70, 100 and 120 mm) from the orifice.

2. Experimental facility and instrumentation

Figure 3 shows a schematic diagram of the experimental facility used in this investigation. Air is supplied by a centrifugal blower (1) operated by an electric AC motor of 1.1 kW through a PVC pipe (2) of 100 mm diameter and 1.5 m length to the orifice plate (3). The pipe is fitted with a flow straightener at the entrance to remove any swirl from the flow to ensure that a uniform fully developed flow with low level of turbulence is reached at the pipe outlet where the orifice plate is fitted to generate the air jet. The orifice plate is designed to be fitted firmly at the outlet of the pipe to minimize the effect of movement on the generated jet for each set of experiments. The jet velocity at the orifice exit is controlled by using a gate valve at blower inlet.

A Pitot tube (4) of 1.0 mm diameter is mounted on a three dimensional traversing mechanism designed for this purpose to measure the radial velocity distribution within the jet at different axial positions away from the orifice plate.

Figure 3. Schematic diagram of jet impingement experimental facility. (1) centrifugal blower, (2) PVC pipe, (3) orifice plate, (4) Pitot tube, (5) pressure target plate, (6) static pressure hole, (7) manometer, (8) heat transfer target plate, (9) thermocouples, (10) recorder, (11) digital multimeter, (12) DC power supply.

The pressure target plate (5) is made of Plexiglas plate $400 \times 400 \times 4$ mm in size. The plate is designed to be held at its position from its two vertical edges when the air jet leaving the orifice impinges at the centre of the plate. A static pressure hole (6) of 1.5 mm diameter is drilled at the centre of the target plate. Another 12 holes of 1.5 mm are drilled in radial direction with a distance of 6 mm from each other. All these holes are fitted with tubes of 1.5 mm diameter and 30 mm in length to measure the static pressure distribution on the target plate in radial direction. Another traversing mechanism similar to the one used for Pitot tube is used to measure the velocity distribution within the hydrodynamic boundary layer formed on the target plate due to the change of flow direction of the jet from the orthogonal direction to

the radial direction parallel to the plate surface. The traversing mechanism used to move the Pitot tube in axial direction at each radial position on the target plate to measure the radial velocity. The static holes used for wall jet pressure measurement are arranged to cover a radial distance equivalent to about 10 times the orifice diameter. Jet velocity profiles and Pressure coefficients are measured for jet impingement velocity of 20 m/sec.

A sensitive, variable inclination calibrated manometer (7) is used to measure the dynamic head within the impinging jet and the wall jet region. The orifice to plate axial distance X is varied by changing the distance at which the target plate is positioned relative to the orifice. The position of the target plate is changed to give the values of $X = 5, 10, 25, 50, 70, 100$ and 120 mm from orifice. The velocity distribution within the impinging jet is measured at the same axial positions of the target plate from the orifice.

The heat transfer target plate (8) is a second target plate made of Duralumin alloy (contains 95% Al.) of $200 \times 150 \times 1$ mm size fitted with 102 thermocouples (9) of type K connected to the recorder (10) to measure the radial temperature distributions which are used to calculate the local heat transfer coefficient on the plate. Thermocouples are fixed at the centre of the plate and in the radial direction with a distance of 10 mm from each other. The back of the plate is wrapped with 0.2 mm diameter heater silk having a resistance of $15 \Omega/\text{m}$. Heater wrapping is used to ensure uniform distribution of the heat on the plate to provide a constant heat flux conditions. A special thin layer of adhesive paper is used as electrical insulating material to prevent electrical contact of the heater with the plate. The back face of the plate is thermally insulated using a 40 mm thick layer of polyurethane. The local temperature on the plate surface is measured with the assistance of a main selector switch of 20 points and a secondary selector with 6 measuring points. A digital multimeter (11) is used to measure the voltage and current across the resistance of the heating silk wire. The front face of the heated plate is facing the air impinging jet with a centreline alignment relative to the jet. Measurements show that a steady state condition is reached after about one hour of starting the blower and the heater. A DC power supply (12) is used for heating the silk wire in the present work. The electrical power consumed by the heater can be calculated as

$$p = VI \quad (1)$$

The electrical power consumed by the heater is dissipated through the plate by conduction. The total heat transfer

from the plate can be written as

$$q_{total} = q_{conv} + q_{cond} + q_{rad} \quad (2)$$

And

$$q_{rad} = \epsilon A_s (T_s^4 - T_\infty^4) \quad (3)$$

The heat loss by conduction is estimated by measuring the temperature at the outer surface of the insulating layer by a number of thermocouples embedded at the outer face of the insulation layer. The calculations show that the heat lost by conduction and radiation is about 10% of the total input energy respectively. After estimating q_{rad} and q_{cond} , the heat transferred by convection can be calculated from Eq. (2) which can be used to calculate the local heat transfer coefficient as

$$h = q_{conv}/A_s(T_s - T_\infty) \quad (4)$$

The data are also used to calculate the average heat transfer coefficient for the whole plate by averaging the local values.

3. Results and discussions

The experimental facility is designed to carry out the measurements required in this study which include the jet velocity distributions at different axial distances from the orifice using Pitot tube, the static pressure distribution on the target plate using pressure taps and temperature distribution on heat transfer target plate using thermocouples. The distributions of the wall jet velocity at different radial positions on the target plate are also measured. The radial velocity distributions at different axial distances illustrate the jet development and its influence on heat transfer from target plate.

3.1. Velocity measurements

The radial velocity distribution of the impinging jet from orifices of 10 and 20 mm for a jet velocity of 20 m/sec using the Pitot tube at different axial distances from the orifice are measured. The Pitot tube is traversed in a radial direction at each axial position using the 3-D traverse mechanism to measure the total pressure that can be used to calculate the local velocity. Figure 4 presents the velocity profiles for the orifice diameters of 10 and 20 mm at axial positions of 5, 50 and 100 mm. It can be observed that radial velocity distributions of the jet passes through several distinct regions. The jet emerging from the orifice is characterized by a high velocity with a fairly flat

Figure 4. Radial distributions of jet velocity at different distances from the orifice for two orifice diameters, $d = 10$ & 20 mm (jet velocity = 20 m/s).

profile reflecting the existence of the potential flow close to the outlet of the orifice. As the jet moves further in an axial direction it converts into a free jet zone with a lower velocity and broader where the positional flow condition becomes smaller and vanishes in the region of the free jet zone. The free jet zone was explained in detail by Zuckerman and Lior [13] as follows: The velocity gradients in the jet create a shearing layer at the edges of the jet which transfers momentum laterally outward, pulling additional fluid along with the jet and raising the jet mass flow. In this process, the jet loses energy and the velocity profile is widened in spatially and decreases in magnitude along the sides of the jet. If the shearing layer expands inward towards the centre of the jet prior to reaching the target plate, a region of core decay will be formed. The core decay is the region where the emerging jet becomes sufficiently far from the impingement surface to behave as a free submerged jet as shown in Figure 1. The significant effect of orifice diameter on the development of radial velocity distributions can be observed by comparing the results from orifice of $d = 20$ mm (Figure 3) with results from orifice of $d = 10$ mm (Figure 3). The results show that the bigger orifice diameter of 20 mm gives a longer potential zone with higher velocity relative to the smaller orifice size of 10 mm at the same axial distance from the orifice. The results also show a steep decrease in radial velocity distribution for $d = 10$ mm compared to gradual change for $d = 20$ mm.

As the flow approaches the target plate, the flow is subjected to a strong curvature and very high strain due to the presence of the target plate. The flow experiences an unfavourable static pressure gradient due to rapid deceleration near the wall and following the impingement, it accelerates along the target plate. The axial momentum of the flow impinges on the target plate is converted into radial momentum and the flow resembles a wall jet. The air enters a wall jet region where the flow moves radially

outward parallel to the wall of the target plate. The boundary layer within the wall jet begins in the stagnation region, where it has a typical thickness of no more than 1% of the jet diameter (Martin, [14]). The wall jet has a shearing layer influenced by both the velocity gradient with respect to the stationary fluid at the target plate wall (no-slip condition) and the velocity gradient with respect to the fluid outside the wall jet. Figure 5 present the ratio of wall jet velocity/jet velocity profiles on the target plate at different radial positions for orifice diameters of 10 and 20 mm. The results show that the wall jet velocity/jet velocity is higher for orifice diameter of 20 mm compared to 10 mm for $X = 50$ and 120 mm. The wall jet velocity decreases at larger radial positions as the jet spread away from the centre which can be explained by applying the mass conservation law at different radial positions. Figure 6a, Figure 6b, Figure 6c and Figure 6d show the ratio of wall jet velocity/jet velocity distributions and the velocity gradient at different radial positions for different distances from the target plate surface.

Examining these profiles show that the larger orifice diameter of 20 mm gives lower values of velocity gradients near the stagnation zone compared to the orifice diameter of 10 mm. The relationship between velocity gradients and turbulence levels was discussed by Nashino *et al.* [15]. Their results show that the maximum turbulent kinetic energy generated near the solid wall of the air impingement on a plate was as a result of high velocity gradient. This was explained as a result of the change in flow direction at the stagnation zone which experiences high normal and shear stresses in the deceleration region that strongly influences the local transport properties. The resulting flow pattern stretches vortices in the flow and increases the turbulence which increases the heat transfer rate. Fleischer *et al.* [16] showed that turbulent intensity increases with orifice diameter from 2.4% to 7.2% and 14% as the orifice diameter increases from 3.5 to 9.5 and 21 mm for $Re = 10000$ and $X/d = 6$. Their data also showed that turbulent intensity increased by 3.5 times as the X/d changed from 0 to 6 for $d = 21$ mm. The increase in turbulent intensity with axial distance was also observed by Hofmann *et al.* [12].

3.2. Pressure measurements

The pressure target plate is used to measure the static pressure distribution which is used to calculate the radial distribution of pressure coefficients on the plate. The results for orifice diameters of 10 and 20 mm at axial distances of $5, 10, 25, 50, 70, 100$ and 120 mm from orifice exit for jet velocity = 20 m/s are given in Figure 7. It can be observed that that pressure coefficient is equal to 1

Figure 5. (a) Variation of wall jet velocity/jet velocity ratio with distances from plate surface for $X = 50$ & 120 mm at five different radial positions ($d = 10$ mm); (b) Variation of wall jet velocity/jet velocity ratio with distances from plate surface for $X = 50$ & 120 mm at five different radial positions ($d = 20$ mm).

Figure 6. (a) Variation of wall jet velocity/jet velocity ratio with radial distance for $X = 50$ & 120 mm at three different distances from the plate surface ($d = 10$ mm); (b) Variation of velocity gradient with radial distance for $X = 50$ & 120 mm at three different distances from the plate surface ($d = 10$ mm); (c) Variation of wall jet velocity/jet velocity ratio with radial distance for $X = 50$ & 120 mm at three different distances from the plate surface ($d = 20$ mm); (d) Variation of velocity gradient with radial distance for $X = 50$ & 120 mm at three different distances from the plate surface ($d = 20$ mm).

at the centre line for $X \geq 70$ mm but drops sharply for $X > 70$ mm for the orifice of 10 mm diameter. On the other hand; the pressure coefficients are very close to $X \leq 50$ but drop gradually for $X > 50$ for the orifice of 20 mm diameter. The higher values of pressure coefficient near

the orifice exit indicate the existence of a potential flow zone within the jet. The optimum distance from the orifice is 50 mm ($X/d = 5$) for 10 mm diameter and about 70 ($X/d = 3.5$) for 20 mm diameter. It can be observed that C_p distributions are different for different orifice diameters

Figure 7. Radial pressure coefficients distribution at different axial distances from the orifice (jet velocity = 20 m/s). A) $D = 10$ mm, B) $d = 20$ mm.

and analogous to the velocity distributions. The present data are supported by the findings of Tu and Wood [17] and Guo and Wood [18].

3.3. Heat transfer measurements

The heat transfer target plate is used to carry out a set of experiments to measure the temperature distribution that can be used to calculate the local heat transfer coefficients on the target plate for orifice diameters of 5, 10 and 20 mm and different orifice-to-plate distances.

It is observed from Figure 8 that radial temperature distributions for all cases under investigation are nearly uniform due to conjugate heat transfer involved. Heat conduction within the target plate makes the surface temperature almost uniform due to the high thermal conductivity of aluminium alloy of 147 W/mk compared to convection heat transfer coefficient in the range of 60 – 130 W/m²k. This shows that the heat transfer through the plate is much faster than the convection to the flowing air. Figure 9 shows the corresponding radial distribution of the local Nusselt number. A small peak can be observed at the stagnation point due to a high circulation of the flow at this point. The results also show that orifice-to-target plate distance has a significant effect on Nusselt number (heat transfer removal).

To ensure the accuracy of the present work, Figure 10 shows a comparison of average Nusselt numbers with Reynolds number from the current data and from the literature (Martin [14], Goldestien and Behbahani [4], Chougule *et al.* [19]). It reveals that the present data is slightly lower than the previous published data. The discrepancy between the present data and the literature is due to the sensitive nature of the

Figure 8. Radial temperature distribution on the surface of the target plate.

Figure 9. Radial local Nusselt number distribution for different orifice diameters and distances from the orifice (jet velocity = 30 m/s).

experiments. The data scatter may be partially due to the differences in jet outlet conditions, jet generator geometry (orifice, nozzle), thermal boundary conditions, target plate geometry and experimental techniques. This has hindered the interpretation of the data both with regard to understanding the pertinent physical phenomena, and for developing general heat transfer correlations.

Returning to Figure 9, the results show that the small orifice size of 5 mm gives low values of heat transfer coefficients at $X/d \leq 2$. As the axial distance increases the local values increase reaching the maximum values at $X/d = 6 - 8$ which is associated with the end of potential core as observed from the velocity and pressure coefficient measurements. These results are in agreement with previous published results (Gao *et al.* [20]) for similar cases. This behaviour may be attributed to the higher levels of turbulence at larger axial distances which enhance heat transfer at the stagnation point and the zone beyond it. The results for the orifices of 10 mm and 20 mm diameter which are given in Figure 9 as well show the same trend of higher Nusselt number values at $X/d = 6 - 8$. The results for 20 mm diameter overlap with the results from 10 mm diameter orifice which can be attributed to the similar level of turbulence generated at the stagnation zone and the existence of cross flow which tends to decrease heat transfer coefficients for relatively larger orifice diameter. Figure 11 shows the effect of orifice diameter on the average Nusselt number. For all X/d ratios, the Nusselt number increases as the orifice diameter increases. The non-linear increase of average Nusselt number on the target plate with the increase in orifice diameter reflects the complexity of the flow. The non-linear increase in average Nusselt can be explained as follows: i) from Figure 4, the jet velocity distribution is more uniform for small orifice diameter (10 mm) compared to large one (20 mm). Figure 6b & Figure 6d also show that the velocity gradients are different for different orifice diameters, ii) the effect of cross flow within the stagnation zone is expected to be less for small orifice diameter compared to large one which has a direct influence on heat transfer and iii) the heat transfer area on target plate becomes larger for higher orifice diameter. The combination effects of these parameters resulting in a non-linear change in Nusselt number.

The results in Figure 11 shows that the average Nusselt number for the orifice diameters of 5 and 10 mm has not changed for $X/d \leq 6$. It can be observed that the Nu has the same values when $X/d = 6$ or 8 but has different and larger values when the orifice diameter is 10 & 20 mm.

Figure 12 shows variation of average Nusselt numbers with X/d values for different Reynolds numbers (different orifice diameters). It can be observed that the Nusselt

Figure 10. Comparison of Average Nusselt numbers with the previous work.

Figure 11. Variation of average Nusselt number with orifice diameter at different distances from the orifice (jet velocity = 30 m/s).

number increases slightly with X/d ratio for orifice diameter = 10 and 20 mm. For $d = 5$ mm ($Re = 9700$), a small peak appears at $X/d = 6$ indicating the end of the potential core zone. The present results are supported by the findings of Flisher *et al.* [16].

4. Conclusions

An experimental investigation is performed to study the jet velocity, wall jet velocity, pressure coefficient and heat transfer coefficient for air impingement on target plate for different orifice diameters and orifice-to-plate/orifice diameter ratios. The results are compared well with the previous finding in this field. The expected high rate of heat transfer due to doubling the orifice diameter is gained due to the modification of the complex flow field at the stagnation point and wall jet appeared on the target plate

Figure 12. Variation of average Nusselt number with X/d . (jet velocity = 30 m/sec).

surface. The main conclusions from the present work are:

1. The radial velocity distribution of the jet becomes broader at higher X/d values with a smaller peak at the centreline. The orifice diameter has a significant effect on radial velocity distribution and the centreline peak.
2. The local pressure coefficient distributions on the target plate reflect a distribution similar is to the velocity profiles with high C_p values at stagnation point which decreases monotonically in the radial direction. The larger orifice diameters give high values of C_p .
3. The higher level of turbulence associated with high velocity gradients at the stagnation zone leads to comparatively higher values of heat transfer rates. The present results for orifice of 20 mm diameter show that Nusselt numbers are higher which confirm the importance of higher velocity gradient. The average Nusselt number increase with orifice diameter at all X/d ratio. The rate of increase reduces with the increase in orifice diameter.
4. Present results confirms the non-linear behaviour of the relation between the heat transfer rate and the different orifice diameters used in this investigation including the large orifice diameter of $d = 20$ mm. This is in agreement with previous works which suggests that heat transfer coefficients increase with d^n ($n < 1.0$).
5. Different orifice diameters of $d = 5, 10, 20$ mm are used in this investigation show that heat transfer rate increases as the ratio of orifice-to-plate distance/orifice diameter increases and reach

its maximum value at $X/d = 6 - 8$. The larger orifice diameter gives higher values of heat transfer coefficients compared to smaller orifice diameter at same x/d values. This finding is very important as most of published work in literature focused on small size in the range of 3 - 10 mm.

Nomenclature

A_s	Plate surface area (m^2)
C_p	Pressure coefficient ($P/0.5\rho U_j^2$)
d	Orifice Diameter (mm)
h	Heat transfer coefficient (W/m^2K)
I	DC electrical Current (A)
k_a	Thermal conductivity of air (W/mK)
Nu	Nusselt number (hd/K_a)
P	Static Pressure (N/m^2)
q_{cond}	Heat transfer by conduction (W)
q_{conv}	Heat transfer by convection (W)
q_{rad}	Heat transfer by radiation (W)
r	Radial Distance from the stagnation point (mm)
Re	Reynolds number ($U_j d/\mu$)
T_s	The local surface temperature on the target plate ($^{\circ}C$)
T_{∞}	The ambient temperature of the impinging jet ($^{\circ}C$)
U_j	Jet Velocity (m/s)
V	Voltage (V)
X	orifice-to-plate distance (mm)
μ	Air viscosity ($N s/m^2$)
ϵ	Emissivity of plate surface (0.09)

References

- [1] San J. Y., Huang C. H., Shu M. H., Impingement cooling of a confined circular air jet. Int. J. Heat Mass Transfer, 1997, 40, 1355-1364
- [2] Behina M., Parneix S., Shabany Y., Durbin P. A., Numerical study of turbulent heat transfer in a confined and unconfined impinging jets. Int. J of heat and fluid flow, 1998, 20, 1-9
- [3] Behina M., Parneix S., Shabany Y., Durbin P. A., Prediction of heat transfer in an axisymmetric turbulent jet impinging on flat plate. Int. J of heat and mass transfer, 1998, 41, 1413-1426
- [4] Goldestien R. J., Behbahani A. I., Impingement of circular jet with and without cross flow. Int. J. Heat Mass Transfer, 1982, 25, 1377-1382
- [5] Lee D., Greif R., Lee S. J., Lee J. H., Heat transfer from a flat plate to a fully developed axisymmetric impinging jet. Transactions of the ASME, 1995, 117, 772-776

- [6] Knowles K., Myszko M., Turbulence measurements in radial wall-jet. *Exp. Thermal fluid Sci.*, 1998, 17, 71-78
- [7] O'Donovan T. S., Murray D. B., Jet impingement heat transfer-part 1: Mean and root-mean-square heat transfer and velocity distribution. *Int. J. Heat Mass Transfer*, 2007, 50, 3291-3301
- [8] Mohanty A. K., Tawfek A. A., Heat transfer due to round jet impinging normal to a flat surface. *Int. J. Heat Mass Transfer*, 1993, 36, 1639-1643
- [9] Lytle D., Webb B. W., Air jet impingement heat transfer at low nozzle plate spacing. *Int. J. Heat Mass Transfer*, 1994, 37, 1687-1697
- [10] Mi J., Nothan G. J., Nobers D. S., Mixing characteristics of axisymmetric free jets from a contoured nozzle, an orifice plate and a pipe, *Journal of fluids Engineering. Transactions of the ASME*, 2001, 123, 878-883
- [11] Baydar E., Ozmen Y., An experimental and numerical investigation on a confined impinging air jet at high Reynolds numbers. *Applied thermal engineering*, 2005, 25, 409-421
- [12] Hofmann H. M., Kind M., Martin H., Measurements on steady heat transfer and flow structure and new correlations for heat and mass transfer to submerged impinging jet. *Int. J. Heat Mass Transfer*, 2007, 50, 3957-3965
- [13] Zuckerman N., Lio N., Jet impingement heat transfer: physics, correlations, and numerical modelling. *Advances in heat transfer*, 2006, 39, 565-631
- [14] Martin H., Heat and mass transfer between impinging gas jets and solid surfaces. *Advances in heat transfer*, 1977, 13, 1-60
- [15] Nashino K., Samada M., Kasuya K., Torij K., Turbulence Statistics in the Stagnation Region of an Axisymmetric impinging Jet Flow. *Int. J. Heat and Fluid Flow*, 1996, 17, 193-201
- [16] Fleischer A. S., Nejad S. R., Jet impingement cooling of a discretely heated portion of a protruding pedestal with a single round air jet. *Experimental thermal and fluid science*, 2004, 28, 893-901
- [17] Tu C. V., Wood D. H., Wall pressure and shear stress measurements beneath an impinging jet. *Experimental thermal and fluid science*, 2002, 25, 605-614
- [18] Guo Y., Wood D. H., Measurement in the vicinity of stagnation point. *Experimental thermal and fluid science*, 1996, 13, 364-373
- [19] Chougule N. K., Parishwad G. V., Gore P. R., Pagnis S., *et al.*, CFD analysis of Multi-jet air Impingement on flat plate. *Proceeding of the World Congress on Engineering*, London, U.K., Vol. III, July 6-8, 2011
- [20] Gao N., Sun H., Ewing D., Heat transfer to impinging round jets with triangular tabs. *Int. J. Heat Mass Transfer*, 2003, 46, 2557-2569