Abstract

Jet impingement is used in many applications for cooling, the goal of this investigation is to obtain heat transfer distribution experimentally and thus extend the understanding of jet impingement. The main objective of this paper is to investigate the effect of outflow orientation by varying the jet Reynolds numbers \((5000 \leq \text{Re}_j \leq 10000)\) and for a fixed jet-to-plate distance \(S/D_h = 0.67\) with a single array of equally spaced orifice jets of two diameters \(d = 0.5\) and \(1\) cm. Results indicate that the outflow orientation causing crossflow effect significantly affects the Nusselt number distributions on the target surface. Local Nusselt number increases with an increase in jet Reynolds number over the entire impingement surface. The heat transfer observed from jet size \(d = 0.5\) cm is twice more than that observed from \(d = 1\) cm. The highest values of Nusselt numbers are obtained for the outflow orientation where the flow exits in both the directions.

1 Introduction

Impingement heat transfer is considered as a promising heat transfer enhancement technique. Among all convection heat transfer enhancement methods, it provides significantly high local heat transfer coefficient. At the surface where a large amount of heat is to be removed/added, this technique can be employed directly through very simple design involving a plenum chamber and orifices. For instance, in gas turbine cooling, jet impingement heat transfer is suitable for the leading edge of a rotor airfoil. This technique is also employed in turbine guide vanes (stators). Other applications for jet impingement could be combustor chamber wall, steam generators, ion thrusters, tempering of glass, electronic devices cooling and paper drying, etc.

Many researchers have carried out experimental work using the jet impingement mechanism. Chupp et al. [1] has studied the heat transfer characteristics for the jet impingement cooling of the leading edge region of a gas turbine blade. Flourscheutz et al. [2] investigated the heat transfer characteristics of jet array impingement with the effect of initial crossflow. Metzger and Bunker et al. [3] and Bunker and Metzger et al. [4] reported on the interior of the leading edge of a turbine blade – a concave surface. They used the liquid crystal technique for the local heat transfer coefficients. They studied over a wide range of parameters flow rates, jet nozzle orifice spacing, jet nozzle to target plate distance. They found that the jet Nusselt number depends mainly on the jet Reynolds number and they observed that the large Nusselt number gradients appear for smaller jet nozzle spacing. Flourscheutz et al. [5] Goldstein and Behbahani et al. [6], and Goldstein and Timmers et al. [7] summarized their findings related to the parametric effects of turbulence levels, temperature, geometry, crossflow, interference and non-uniformity of jet array on jet impingement heat transfer. Huber and Viskanta et al. [8] studied the heat transfer on a target wall cooled by impinging jets with coolant extracted through additional orifices in the jet wall. They found more uniform heat transfer at close nozzle exit to target wall spacing. Kercher and Tabakoff et al. [9] measured heat transfer coefficients under an
square array of circular air jets. They presented correlation for the heat transfer performance under maximum crossflow scheme. The correlation included the effects of spent air, jet diameter, jet spacing, and jet-to-surface spacing. Obot and Trablod et al. [10] developed an experimental study to investigate the effects of minimum, intermediate, and maximum crossflow schemes on heat transfer distribution. The geometric and flow parameters such as jet-to-surface spacing, open area ratio, and jet Reynolds number were varied during the investigation. They concluded that the heat transfer could be enhanced markedly by having greater number of jets over a fixed area under minimum cross flow scheme. Hollworth and Cole et al. [11] measured the Nusselt number values for staggered arrays of jets under various combinations of jet spacing and Reynolds number. Ekkad et al. [12] also studied the heat transfer characteristics using the jet impingement technique experimentally; they studied the effect of jet impingement with film cooling holes on the target surface. They found that film-cooling holes on the target surface reduce the crossflow which in turn reduces the heat transfer. Van Treuren et al. [13] studied the effect of local heat transfer coefficient and adiabatic wall temperature under the impinging jets with crossflow in one direction. Ekkad et al [14] conducted an experimental investigation to investigate the heat transfer characteristics of a heat target surface with dimples using the jet impingement mechanism. The results presented by them indicate that the dimpled target surface produces lower heat transfer coefficients compared to the flat target surface. Hwang et al. [15] studied the effect of tangential jets impinging on a triangular target surface with three different exit flow orientations generating crossflow. They developed that the outflow orientation would significantly affect the local heat transfer characteristics. Huang et al. [16] studied the effect of impinging orthogonal jets on a flat target surface with three different exit flow orientations. They confirmed that the heat transfer distributions on the target surface are significantly affected by crossflow direction. They have shown that the crossflow effects are minimal when the flow exits in both the directions after impingement. In the studies described above the work is being carried out using multiple arrays of jets.

The purpose of the present work is to study the effect of the outflow orientation and orifice jet size on the heat transfer characteristics of the impingement heated target surface using a single array of equally spaced jets on an orifice plate in a rectangular duct. The effect of jet-to-plate spacing \( S / D_s = 0.67 \) on the heat transfer characteristics is also studied. Three different outflow orientations are being studied to get different crossflow effects i.e., the exit flow, Hwang et al. [15], as described in figure 2.

Case - 1 (Coincident with the entry flow),
Case - 2 (Opposing to the entry flow),
Case - 3 (Passes out in both the directions).

Tests were conducted with two orifice plates with jet size of diameter 0.5 and 1cm respectively. The heat transfer characteristics are being studied for two orifice jet sizes with jet Reynolds number varying in the following range \( (5000 \leq \text{Re}_j \leq 10000) \) by applying the constant heat flux to the target surface.

2. Description of the experiment

The schematic of the experimental facility is depicted in figure 1. The test rig used to study the heat transfer characteristics was manufactured using Plexi-glass. The test section consists of two channels, impingement channel (10) and the feeding channel (9). Air enters the test section in the feeding channel and is directed onto the heated copper plates in the impingement channel to study the heat transfer characteristics. The target plates made of copper were heated using a constant flux heater, which was controlled by a variable resistance transformer. The other side of the heater was insulated to get the heat transferred only in one direction i.e. onto copper plates. The mass flow rate of the compressed air (1) entering the test section was passed through a settling chamber.
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(2) and was controlled with the help of valves (3) and the pressure regulators (5). The pressure drop across the pressure regulators was measured using the pressure gauges (4). Gas flow meters (7) were used to measure the mass flow rate entering the test rig, which were protected by the air filters (6) of 50µ capacities. The average surface temperature of each of the copper plates was determined from the readings of two T-type thermocouples installed in the grooves drilled at the back surface of the plates to within 1 mm of the surface; the analog signals generated by these temperature sensors were transmitted to the signal-conditioning unit (12), where they were selectively processed (the following operations were performed on the signals generated by the thermocouples: linearization, cold junction compensation, amplification). The resulting analog signals were converted into digital signals by a DAQ card installed on the motherboard of a personal computer and recorded with an application developed in Lab View. The characteristics and accuracy of the instruments used for experiments are summarized in Table 1.

Figure 1 Schematic of the test section.
Table 1 Instrumentation and software used for Data acquisition

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>FMA - 1600 mass flow meter</td>
<td>Accuracy: +1% Omega</td>
</tr>
<tr>
<td>SCXI - 1000 chasis</td>
<td>- National Instruments</td>
</tr>
<tr>
<td>SCXI - 1303 high accuracy</td>
<td>Accuracy: +0.3% National Instruments</td>
</tr>
<tr>
<td>isothermal terminal block</td>
<td></td>
</tr>
<tr>
<td>SCXI - 1100 32-channel isolated analog input module</td>
<td>Offset error +1.5mV/gain, gain error: +0.03% National Instruments</td>
</tr>
<tr>
<td>Labview 7.0</td>
<td>- National Instruments</td>
</tr>
<tr>
<td>SAI - T thermocouples</td>
<td>Accuracy: 0.3% Omega</td>
</tr>
</tbody>
</table>

Figure 2 shows the three-dimensional sketch of the test section. It consists of two channel joined by the orifice plate, which has a single array of equally spaced jets. Two orifice plates of different orifice jet sizes are being studied i.e. $d = 0.5 \text{ and } 1 \text{ cm}$. The jet-to-jet spacing for $d = 1 \text{ cm}$ is twice the orifice jet size and for $d = 0.5 \text{ cm}$ it is taken as 4 times the orifice jet size, as shown in figure 3. The jet orifice plate thickness is 1 cm for both the orifice plates. There are 13 jets on each orifice plate. The length of the test section is 106.54 cm. The width of the feeding channel is considered as $H$, and the width of the impingement channel is taken as $S$. The jet plate geometry was chosen with reference to the previous studies [15, 16]. The test section is made of Plexiglas material. The impingement target surface constitutes a series of copper plates, each with $4.1 \times 4.2 \text{ cm}$ in size, arranged in accordance with the orifice jets, such that the impingement jet hits the geometric center of the copper plate. The copper plates are separated from each other by 1 mm distance to avoid the relative heat conduction, thus dividing the target surface into segments. The thickness of the copper plate is 0.503 cm.
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Figure 3 Orifice plates showing the two orifice jet sizes used in the present investigation.

Figure 4 shows the schematic of the three different exit outflow orientations, which are obtained by changing the discharge openings. The upper channel is called as the Feeding channel and the lower channel in which the jets impinge on the target surface is the impingement channel. The flow develops in the open area provided in the feeding channel before entering the jets. The exit of jets in three different outflow orientations from the impingement channel creates different crossflow effects. The three different outflow orientations are represented with case numbers as shown in the figure 4.

Figure 4 Illustration of three exit outflow orientations.
3. PROCEDURE
Tests were carried out using two orifice plates with orifice jet diameters $d = 0.5 \text{ and } 1 \text{ cm}$ with jet Reynolds numbers $(5000 \leq Re_j \leq 10000)$ and for a constant heat flux power input. The procedure followed during the experiment is as follows. Initially the mass flow rate was adjusted to the required value for the experiment to be conducted and the air is blown continuously into the test section. Heat was supplied to the copper plates with electric resistive constant flux heaters (manufactured by Omega) from backside to provide uniform heat flux. The temperature of the copper plates was measured by two thermocouples mounted in a groove of 2.5 mm on the back of the copper plates. Thus the temperature of a particular plate is taken as the average of the reading of two thermocouples. One thermocouple is respectively fixed at the inlet and two at the outlet to monitor the flow temperatures. The temperature of the copper plates, the pressure at the test rig inlet, temperature of the air at the inlet, the mass flow rate of the air was continuously monitored. After the temperature of the copper plates reaches the steady state condition, all the data was collected with LabVIEW program. Heat lost by conduction through the insulation was computed by placing a couple of thermocouples in the wood. Similarly the radiation losses were computed by considering the temperature of the surroundings, measured by placing two thermocouples near the test section. The Nusselt number was then calculated based upon the collected data. The same procedure was repeated for the three outflow orientations described in figure 4, with the jet-to-plate distance $S / D_h = 0.75 \text{ cm}$.

4. Data Reduction and Uncertainty

4.1 Reynolds number Calculations
The procedure to calculate the Jet Reynolds number is described below:

Average velocity of all the orifice jets is calculated using the following expression with total area taken as the summation of areas of all the 13 jets on the orifice plate:

$$V_{avg} = \frac{\sum A_j}{A_{tot}} = \frac{\sum A_j}{\sum \pi d^2}$$  \hspace{1cm} (1)

The Jet Reynolds number is calculated as

$$Re_{jet} = \frac{\rho V_{avg} D_h}{\mu}$$  \hspace{1cm} (2)

The feeding channel hydraulic diameter is calculated based upon the cross-sectional dimensions of the feeding channel as follows:

$$D_h = \frac{4 A_f}{P_f}$$  \hspace{1cm} (2)

4.2 Nusselt number calculations
The total power input to all the copper plates was computed using the voltage and current the former being measured across the heater.

$$Q_{total} = \frac{V^2}{R} = VI$$  \hspace{1cm} (3)

The actual heat supplied to the heater is found by deducting the losses from the total heat supplied to the heater.

$$Q_{actual} = Q_{total} - Q_{losses} = Q_{total} - (Q_{cond} + Q_{rad})$$  \hspace{1cm} (4)

The heat lost by conduction through the wood and to the surrounding by radiation is depicted in figure 5 and is estimated using the following equations.

$$Q_{cond} = k_{wood} A_{cp} \frac{(T_s - T_w)}{t}$$  \hspace{1cm} (5)

$$Q_{rad} = \varepsilon \sigma A_{cp} (T_s^4 - T_{sur}^4)$$  \hspace{1cm} (6)

The local convective heat transfer coefficient of each of the copper plates is calculated using
The average temperature of the heated target surface $T_s$ is obtained as the average of the readings of the two thermocouples fixed in each copper plate. To calculate $h$, $T_{in}$ was considered instead of the bulk temperature or the reference temperature. Based on the literature available, Obot and Trabold [10] discussed that the heat transfer coefficients do not exceed 10% with the adiabatic wall or the reference temperature $T_R$ to replace the inlet temperature $T_{in}$ in heat transfer coefficient equation above. In principle, $T_R$ should be the average recovery temperature measured at the target surface at actual crossflow temperature, but it is difficult to measure in practical experiment. Therefore, in the present paper, the inlet temperature of the air is used as the reference temperature. It is measured at the test section inlet, where the air first enters the feeding channel. Chakroun et al [19] also considered the inlet temperature to calculate the heat transfer coefficient ($h$).

The non-dimensional heat transfer coefficient on the impingement target surface is represented by Nusselt number as, with the hydraulic diameter is taken as the diameter of the orifice jet.

$$ Nu = \frac{h D_h}{k_{air}} $$

Detailed uncertainty analysis is performed over different parameters. General uncertainty analysis equations, precision, bias and resultant uncertainty, can be found in Coleman and Steele [17] and ASME PTC 19.1 [18]. The same procedure is applied in the present analysis, handling precision and bias errors independently, maintaining the same confidence level.

4.3 Uncertainty in Nusselt number

The data reduction equation for the Nusselt number is considered along with the heat losses by conduction and radiation.

$$ Nu = \frac{D_h}{k_{air}} \left( \frac{VI - k_{air} (T_s - T_{air}) - \varepsilon \sigma (T_s^4 - T_{air}^4)}{A_{cp} (T_s - T_{in})} \right) $$

Taking into consideration only the measured values, which have significant uncertainty, the convective heat transfer coefficient is a function of:

$$ Nu = f(V, I, T_s, T_{in}, T_{air}, T_{surv}, A_{cp}, k_w, \varepsilon, A_{cp}, k_{air}) $$

Thermal conductivity of wood, temperature of wood, temperature of the surroundings and emissivity are not considered as the experiment is carried out in a controlled environment. Temperature of the wood has a very less effect on the uncertainty of heat transfer coefficient due to the large thickness of the wood and also due to the insulation material attached to the wooden block. Temperature of the surroundings also has a very less effect on the uncertainty as the work is carried out in a controlled environment and the temperature of the surroundings is maintained within a desired limit. Thus the data reduction equation (11) reduces to the following simplified form.
\[ Nu = f(V, I, T_s, T_{in}, A_{ip}, D_h) \]  

(10)

The precision uncertainty in the Nusselt number is calculated using

\[
\left( \frac{U_{p, Nu}}{Nu} \right)^2 = \left( \frac{1}{Nu} \frac{\partial Nu}{\partial V} U_{p,V} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial l} U_{p,l} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial T_s} U_{p,T_s} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial T_{in}} U_{p,T_{in}} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial A_{ip}} U_{p,A_{ip}} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial D_h} U_{p,D_h} \right)^2
\]

(11)

Similarly, the bias uncertainty is written as:

\[
\left( \frac{U_{b, Nu}}{Nu} \right)^2 = \left( \frac{1}{Nu} \frac{\partial Nu}{\partial V} U_{b,V} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial l} U_{b,l} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial T_s} U_{b,T_s} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial T_{in}} U_{b,T_{in}} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial A_{ip}} U_{b,A_{ip}} \right)^2 + \left( \frac{1}{Nu} \frac{\partial Nu}{\partial D_h} U_{b,D_h} \right)^2
\]

(12)

The resultant uncertainty in the Nusselt number is as follows:

\[
\left( \frac{U_{Na}}{Nu} \right)^2 = \left( \frac{U_{p, Nu}}{Nu} \right)^2 + \left( \frac{U_{b, Nu}}{Nu} \right)^2
\]

(13)

### 4.4 Uncertainty in Jet Reynolds number

The data reduction equation for the jet Reynolds number is taken as:

\[
Re = \frac{\rho V_{avg} D_h}{\mu} = \frac{\rho D_h}{\mu} \sum_{i=1}^{13} \pi \frac{D_i}{4} d_i^2
\]

(14)

Taking into consideration only the measured values, which have significant uncertainty, the jet Reynolds number is a function of

\[
Re = f(\forall, D_h, d)
\]

(15)

Density of air and the dynamic viscosity of air is not included in the measured variables since it has negligible error in the computation of the uncertainty in jet Reynolds number.

\[
\left( \frac{U_{p,Re}}{Re} \right)^2 = \left( \frac{1}{Re} \frac{\partial Re}{\partial V} U_{p,V} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial D_h} U_{p,D_h} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial l} U_{p,l} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial T_s} U_{p,T_s} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial T_{in}} U_{p,T_{in}} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial A_{ip}} U_{p,A_{ip}} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial D_h} U_{p,D_h} \right)^2
\]

(16)

Similarly, the bias uncertainty is written as

\[
\left( \frac{U_{b,Re}}{Re} \right)^2 = \left( \frac{1}{Re} \frac{\partial Re}{\partial V} U_{b,V} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial D_h} U_{b,D_h} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial l} U_{b,l} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial T_s} U_{b,T_s} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial T_{in}} U_{b,T_{in}} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial A_{ip}} U_{b,A_{ip}} \right)^2 + \left( \frac{1}{Re} \frac{\partial Re}{\partial D_h} U_{b,D_h} \right)^2
\]

(17)

The resultant uncertainty in the Nusselt number is as follows

\[
\left( \frac{U_{Re}}{Re} \right)^2 = \left( \frac{U_{p,Re}}{Re} \right)^2 + \left( \frac{U_{b,Re}}{Re} \right)^2
\]

(18)

Uncertainty in the Nusselt number is calculated based on the ASME-PTC standards 2003 following the procedure mentioned above.

The estimated uncertainty in the Nusselt number in the present study ranges from ± 3% to ± 7% depending upon the jet average velocity.

The estimated uncertainty in the Jet Reynolds number range 5000 ≤ Re ≤ 10000 in the present study ranges from ± 0.9% to ± 2.7% depending upon the jet average velocity.

### 5 Results and Discussions

Jet impingement heat transfer is dependent on several flow and geometrical parameters. The jet impingement Nusselt number is presented as a functional form of

\[
Nu = \left( \frac{h D_h}{k_{air}} \right) = f \left( Re_j, \frac{X}{D_h}, \frac{S}{D_h} \right) \]

(19)
Where, $Re_j$ is the flow parameter, jet spacing to the Hydraulic diameter (considered as the width of the feeding channel) ratio $X/D_h$ and jet-to-plate distance $S/D_h$ are the geometric parameters. The flow exit direction and the jet-to-plate distance are also important parameters having a significant effect on impingement heat transfer.

The X location starts at the end of the feeding channel as shown in the figure 2. As shown in figure 4, for flow orientation 1 i.e., Case 1, air enters the test section at large $X/D_h=14.28$ and therefore the crossflow develops towards the exit. This case is typical of that seen by Flourschetz et al [2]. They also studied the crossflow that is similar to this flow orientation. For case 2, the flow enters at $X/D_h=0$ and as the flow moves forward crossflow develops at low $X/D_h$ and increases towards large $X/D_h$ as the flow exits at the end of the test section where $X/D_h=14.28$. For case 3, the flow exits at both the directions i.e., at $X/D_h=0$ and 14.28 and the crossflow develops towards both the ends of the test section. There is almost less effect of crossflow at $X/D_h=27.9$ because the flow exits from both ends of the impingement channel.

Tests were carried out for three different outflow orientations with Reynolds number ranging between $5000 \leq Re_j \leq 10000$. Two-orifice jet diameters $d=0.5$ and $1cm$ were also used for study the effect on heat transfer along the entire target surface.

All the results presented in this paper are for $0.66 \leq X/D_h \leq 14.28$ over the entire span of the target plates. Where X is defined from the opposite end of the feeding channel of the test section as shown in figure 2. All the results presented below are being extracted using the orifice plate with orifice jets of diameters 0.5 and 1cm.

The local Nusselt number distribution for all the three outflow orientations by varying the jet-to-plate distance is presented in detail, as a function of non-dimensional location $X/D_h$ on the target surface for jet Reynolds number, $Re_j$ and jet-to-plate distance $S/D_h$ in the forthcoming discussion.

Figures 6-11 shows the local Nusselt number distribution as a function of the non-dimensionalized distance along the heated target surface. The distribution of local Nusselt number contains a large amount of information, the extent of which is further enlarged by the presence of the three independent parameters, $Re_j$, $X/D_h$ and the outflow orientation.

### 5.1 Effect of jet Reynolds number on Nusselt number

Figure 6 shows the local Nusselt number distribution for case 1 for both the orifice jet sizes with the jet-to-plate distance $S/D_h=0.67$. It is clearly observed that the magnitude of Nusselt number obtained from the target surface is twice more for jet size $d=0.5cm$ compared to orifice jet size $d=1cm$. The peak of local Nusselt number at $X/D_h=2.87$ is observed, this may be due to the recirculation of the incoming air in the feeding channel, thus increasing the mass flow rate through the third jet and decreasing the mass flow rates from the first two jets, similar behavior was observed by Pramanick et al [20] in their computational fluid dynamics simulation with the same arrangement of the orifice plate. Another peak of local Nusselt number at large $X/D_h=10$ is also observed due to less crossflow effect, as the flow enters the impingement channel at large $X/D_h$ ($X/D_h=14.28$). As the jet Reynolds number increases from 5000 to 10000 there is a corresponding increase in local Nusselt number. For orifice jet size, $d=1cm$, the local Nusselt number behavior for $Re_j=10000$ and 7500 is almost similar,
whereas for orifice jets of size $d = 0.5cm$ the local Nusselt number behavior for $Re_j = 10000$ and 7500 is completely different with a maximum difference of 33% in the magnitude.

For orifice jets of size, $d = 1cm$, a maximum difference of 21% is observed in local Nusselt number for $Re_j = 5000$ and $Re_j = 7500$, whereas for orifice jets of size, $d = 0.5cm$ a maximum difference of 29% is observed in local Nusselt number for same Reynolds numbers. The reason for the increase in local Nusselt number may be due to the increase in strength of the jets with an increase in jet Reynolds number. The local Nusselt number behavior for both the orifice jets, decreases from $X/D_h = 10$ to $X/D_h = 3.9$ this is due to crossflow dominating the strength of the impinging jets.

![Figure 6 Local Nusselt number distribution for Case – 1 as a function $X/D_h$ for all jet Reynolds numbers.](image)

(Filled symbol: $d = 1cm$ and Unfilled symbol: $d = 0.5cm$)

Figure 7 shows the local Nusselt number distribution for Case -2 for jet-to-plate distance $S/D_h = 0.67$. It is noted that the heat transfer distribution observed for Case – 2 is less compared to Case – 1 with a maximum difference of 20% in magnitude of local Nusselt number. For Case – 2, the flow enters at low $X/D_h$ ($X/D_h = 0$) and leaves the impingement channel at large $X/D_h$ ($X/D_h = 14.28$), which gives more crossflow towards large $X/D_h$ where the flow exits the impingement channel. Decreasing the orifice jet size from $d = 1cm$ to $d = 0.5cm$ leads into one and half times more heat transfer from the target surface. The local Nusselt number distribution for both the orifice jets, at $3.9 \leq X/D_h \leq 9.02$ is uniform with a constant magnitude across the length of target surface.

Figure 8, shows the local Nusselt number distribution for case – 3. The heat transfer from the target surface with a orifice jet size of 0.5cm is one and half times more compared to the heat transfer with a jet size of $d = 1cm$. In Case 3, flow leaves the impingement channel from both the directions i.e. at low $X/D_h$ ($X/D_h = 0$) and large $X/D_h$ ($X/D_h = 14.28$), therefore strong crossflow is expected at both the exits of the impingement channel. Similar to case –1 and case – 2 a peak of local Nusselt number at $X/D_h = 2.87$ is observed in both the orifice jet sizes with $d = 0.5$ and 1cm. This may be due to the recirculation of air in the feeding channel at low $X/D_h$, similar behavior was reported by Pramanick et al [20] as explained previously. The magnitude of heat transfer at $3.9 \leq X/D_h \leq 9$ is more for case – 3 compared to the magnitude of local Nusselt number of case 1 and 2.

5.3 Effect of outflow orientation on Nusselt number

Figure 9–11 shows the heat transfer characteristics for all the outflow orientations comparing the two-orifice jet sizes studied ($d = 0.5$ and 1cm respectively) for all the jet Reynolds number $5000 \leq Re_j \leq 10000$. 

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Figure 7 Local Nusselt number distribution for Case – 2 as a function $X/D_h$ for all jet Reynolds numbers.
(Filled symbol: $d = 1 cm$ and Unfilled symbol: $d = 0.5 cm$)

Figure 8 Local Nusselt number distribution for Case – 3 as a function $X/D_h$ for all jet Reynolds numbers.
(Filled symbol: $d = 1 cm$ and Unfilled symbol: $d = 0.5 cm$)

Figure 9 shows the local Nusselt number distribution for the jet Reynolds number $Re_j = 5000$ for all the cases studied plotted for the jet-to-plate distance $S/D_h = 0.67$. It is clearly observed that the magnitude of the Nusselt number is almost doubled with a decrease in the orifice jet size from $d = 1 cm$ to $d = 0.5 cm$. The local Nusselt number distribution is less sensitive to the outflow orientation as very less variation in the local Nusselt number values is observed for all the jet-to-plate distances studied. A peak of local Nusselt number at $X/D_h = 2.87$ is observed, this may be due to the recirculation of air in the impingement channel, similar recirculation of the air was observed by Pramanick et al [20] as discussed earlier. At $X/D_h \geq 10$, case – 1 has another peak of local Nusselt number due to strong impingement effect at the closed end (i.e., $X/D_h \geq 10$) whereas case – 2 has the least amount of heat transfer due to crossflow obstructing the jets as the flow exits from large $X/D_h$, $X/D_h = 10$ for all the jet-to-plate distances studied. For case – 3 highest values of local Nusselt number are observed at $3.9 \leq X/D_h \leq 9$ this may be due to less effect of crossflow as flow exits from both ends of impingement channel. For $d = 0.5 cm$, the peak of local Nusselt number at large $X/D_h (X/D_h = 10)$ increases in magnitude with decrease in the jet-to-plate distance. For case – 1, gives the highest values of the local Nusselt numbers followed by the other two cases with a lesser magnitude comparatively. Whereas for orifice jet size $d = 1 cm$ there is no such peak of local Nusselt number observed, this may be due to a decrease in the strength of the jets with an increase in the orifice jet size.

Figure 10 shows the local Nusselt number distribution for $Re_j = 7500$ plotted for all the outflow orientations and jet-to-plate distance $S/D_h = 0.67$. With a decrease in the orifice jet size from $d = 1 cm$ to $d = 0.5 cm$ the increase in heat transfer observed is a marginally less compared to the increase in heat transfer observed for $Re_j = 5000$. Similar to the local Nusselt number distribution observed for $Re_j = 5000$ at $3.9 \leq X/D_h \leq 9$, the case – 3 has the highest values of local Nusselt number.
compared to the other two cases for both the orifice jet sizes studied.

Figure 9 Local Nusselt number distribution for \( \text{Re}_j = 5000 \) as a function \( X/D_h \) for all outflow orientations.
(Filled symbol: \( d = 1\text{cm} \) and Unfilled symbol: \( d = 0.5\text{cm} \))

A peak of local Nusselt number at \( X/D_h = 2.87 \) is observed due to the recirculation of air in the feeding channel as explained earlier, similar behavior was observed by Pramanick et al [20]. For orifice jet size, \( d = 1\text{cm} \) the magnitude of local Nusselt number for case – 3 is slightly more compared to case – 1 and case – 2 at \( 3.9 \leq X/D_h \leq 9 \), whereas for orifice jet size \( d = 0.5\text{cm} \), no such variation of magnitude of local Nusselt number is observed with respect to the outflow orientations studied.

Figure 11 presents the local Nusselt number distribution for the jet Reynolds number \( \text{Re}_j = 10000 \) for all the jet-to-plate distance \( S/D_h = 0.67 \) studied and for all the outflow orientations. The heat transferred from the target surface is more for this particular jet Reynolds number as it can be clearly distinguished from the magnitude of the local Nusselt number compared to the jet Reynolds numbers \( \text{Re}_j = 5000 \) and \( \text{Re}_j = 7500 \). As observed in the previously discussed jet Reynolds numbers, case – 3 has the highest magnitude of local Nusselt number between \( 3.9 \leq X/D_h \leq 9 \) compared to the other two cases for both the orifice jet sizes. The magnitude of the local Nusselt number is increased by one and a half times with a decrease in the orifice jet size from \( d = 1\text{cm} \) to \( d = 0.5\text{cm} \).

Figure 10 Local Nusselt number distribution for \( \text{Re}_j = 7500 \) as a function \( X/D_h \) for all outflow orientations.
(Filled symbol: \( d = 1\text{cm} \) and Unfilled symbol: \( d = 0.5\text{cm} \))

Figure 11 Local Nusselt number distribution for \( \text{Re}_j = 10000 \) as a function \( X/D_h \) for all outflow orientations.
(Filled symbol: \( d = 1\text{cm} \) and Unfilled symbol: \( d = 0.5\text{cm} \))
EFFECT OF OUTFLOW ORIENTATION AND ORIFICE JET SIZE ON A HEATED SURFACE IN A RECTANGULAR DUCT USING JET IMPINGEMENT TECHNIQUE

Similar to the local Nusselt number behavior observed in the early discussions a peak of local Nusselt number is observed at $X/D_h = 2.87$, this may be due to the recirculation of the air in the feeding channel, as explained in the previous discussions. Another peak of local Nusselt number at $X/D_h = 10$ is also observed for both the orifice jet sizes studied. This peak of local Nusselt number is high for the orifice jet size $d = 0.5cm$, compared to that for orifice jet size $d = 1cm$, due to the increase in the strength of the jets with a decrease in the orifice jet size. The case – 1 has the highest magnitude due to less crossflow the impingement channel at $X/D_h = 10$ as the flow initially enters the impingement channel at this location whereas the other two outflow orientations, case – 3 and case – 2, have a lesser magnitude of local Nusselt number due to the crossflow obstructing the flow of the jets.

5.4 Averaged Nusselt number

Figures 12 shows the average Nusselt number $Nu_{avg}$ as a function of jet Reynolds number $Re_j$, plotted for different outflow orientations. The value of $Nu_{avg}$ is an average of the detailed results on the entire surface of the target plate.

It can be observed from the figure 12 that the outflow orientation 3 has maximum values of average Nusselt number where as outflow orientation 1 has a maximum value of Nusselt number at $Re_j = 7500$, for $d = 1cm$. Moreover it is also observed that outflow orientation 2, where the air exits at large $X/D_h$ ($X/D_h = 2.87$), has the least values of local Nusselt number, the reason being the effect of crossflow which is more dominant over the impingement of jets.

For orifice plate – 2 the average Nusselt number distribution is shown in figure 12. Similar to orifice plate 1 with $d = 1cm$, orifice plate – 2 with $d = 0.5cm$ has the outflow orientation 3 giving maximum values of average Nusselt number where as outflow orientation 1 has a maximum value of Nusselt number at $Re_j = 10000$.

![Figure 12 Average Nusselt number as a function of jet Reynolds number for Orifice plate 1 and 2.](image)

(Filled symbol: $d = 1cm$ and Unfilled symbol: $d = 0.5cm$)

6 Conclusions

The effect of outflow orientation on local Nusselt number using jet impingement technique (single array of jets), for a the jet Reynolds numbers ($5000 \leq Re_j \leq 10000$), and for jet-to-plate distance $S/D_h = 0.75$ is investigated. The conclusions are as follows.

1. Nusselt number is significantly affected by the crossflow caused by the outflow orientation.

2. The effect of crossflow orientation was studied for all jet Reynolds numbers. The local Nusselt number is dependent on the jet Reynolds number. The local Nusselt number increases with an increase in the jet Reynolds number.

3. Orifice jet size also affects the heat transferred from the target surface. For case – 1 the increase in heat transfer is twice with a decrease in orifice jet size from $d = 1cm$ to $d = 0.5cm$, similarly
for case – 2 an increase of 33% and for case – 3 an increase of 50% is observed.

3. Under the same jet Reynolds number, the outflow orientation 3 has the highest averaged heat transfer on the target surface among the three outflow orientations investigated.

4. Under the same jet Reynolds number, the outflow orientation 3 has the highest averaged heat transfer on the target surface among the three outflow orientations investigated.

7 Nomenclature

- $A_{cp}$ - Area of the copper plate
- $A_f$ - Cross-sectional area of feeding channel
- $A_j$ - Area of the jet
- $d$ - Diameter of the orifice jet
- $D_h$ - Hydraulic diameter of the feeding channel
- $h$ - Convective heat transfer co-efficient
- $H$ - Width of the feeding channel
- $I$ - Current supplied to heater
- $k_{air}$ - Thermal conductivity of air
- $Nu$ - Nusselt number
- $Nu_{avg}$ - Average Nusselt number
- $Nu_s$ - Smooth tube average Nusselt number
- $P_f$ - Perimeter of the feeding channel
- $Q_{actual}$ - Actual heat released from target surface
- $Q_{cond}$ - Heat lost due to conduction
- $Q_{rad}$ - Heat lost due to radiation
- $Q_{total}$ - Total heat input
- $Re_j$ - Jet Reynolds number
- $R$ - Resistance of the heater
- $S$ - Jet-to-plane distance
- $T_{in}$ - Inlet temperature
- $T_s$ - Surface temperature
- $T_s$ - Temperature of the surroundings
- $V$ - Voltage supplied to the heater
- $V_{avg}$ - Average velocity of all jets
- $\Phi$ - Volume flow rate
- $X$ - Distance in the x-direction

Greek

- $\varepsilon$ - Emissivity
- $\sigma$ - Stefan- Boltzman constant

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9 References


