Investigation of the Effect of Inducing Impingement Wall Velocity on the Flow Field and the Heat Transfer Process in a Semi Confined Impinging Configuration

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ABSTRACT

Heat transfer at an impingement wall in a semi-confined impinging jet configuration depends mainly on the flow structure (vortices) in the region between confinement wall and impingement plate. Impingement wall motion introduces vorticity which changes the flow field between the confinement and the impingement wall, hence the heat transfer process at impingement wall. Numerical study using control volume technique is adapted using Gambit 2.2.3 software for mesh generation and Fluent 6.2 as a CFD solver at Reynolds number 500. The spacing between the confined and impingement wall (H/B) is 2, 3 and 4 nozzle width. The impingement wall velocity starts from 5% up to 20% with respect to the jet velocity. The motion of the wall is expressed in a non-dimensional flow parameter R. The parameter presents the wall relative velocity between the jet and the wall. The effect of the flow parameter and H/B ratio on the average and local Nusselt number as well as the flow field are presented. For R<1, as R decreases the location of vortex at impingement surface moves outward downstream and it stretched on larger area. This will lead to the heat transfer process enhancement at the impingement hot wall up to 20% compared to stationary wall case. For R>1, as R increases, the retardation flow is enhanced at the impingement wall, hence the average Nusselt Number decreases. The results are correlated. The enhancement heat transfer is correlated with R and H/B for R<1. The correlation formula is:

\[ \frac{N_{\text{avg}}}{N_{\text{avg,s}}} = 0.984 R^{-0.417} (H/B)^{0.022} \]

Key words: Air jet; moving plate; conveyer and forced convection

Introduction

Jet impinging is one of the major methods used to produce high cooling rate at impingement surface. It is used in quenching products in steel and glass industries after or during rolling. In gas turbines, the moving blades are cooled using an impinging jet. Thermal deformation of products can be reduced or limited by using impinging jet. In addition the cooling of electronics (i.e. computers processor) could be enhanced using impinging jet as reported by Zuckerman and Lior (2006) and Sarkar et al., (2004). Impinging jet flow according to Polat et al. (1989) is characterized by potential core region (I), Jet flow region (II), stagnation region (III) and wall jet flow region (IV) as shown in Fig. 1. The potential core region (I) is a part of the flow where the flow velocity is uniform and remains equal to the nozzle exit velocity. The end of the potential region is determined by the rate of growth of the two shear layers originated at edges of nozzle region. Jet flow region starts where the two shear layers meet. The jet flow region is characterized by the decaying of the jet exit velocity and spreading of the jet. The stagnation region is characterized by an increase of the static pressure as a result of the sharp decrease of axial velocity. In the wall jet region, a boundary layer develops along the impingement surface. According to the spacing between nozzle and the impingement plate, the flow may display one or more of mentioned regions. According to Gardon and Akffirat (1966), the slot jets with Re<2000 could be considered as laminar. Comprehensive studies of the mean fluid flow characteristics of both free and impinging jets have been presented by Jambunathan et al. (1992) and Yan et al. (1992). Liu and Sullivan (1996) showed that controlling the vortices within free impinging jet flow has the potential to enhance the overall surface heat transfer. Hwang and Cho (2003) and Hwang et al., (2001) have used acoustic excitation to control the development of vortices in free impinging jet flow and report the influence of acoustic excitation on the surface heat transfer. Several investigators report secondary peaks in the heat transfer distribution at low nozzle to impingement wall spacing. Miroslaw Zukowski (2013) investigated the thermal performance of a confined impinging slot jet of air. Influence of nozzle width, Reynolds number and nozzle to plate spacing on the Nusselt number was investigated by Chiriac and Ortega (2002). They investigated the steady and unsteady flow and heat transfer for a confined two-dimensional slot jet impinging on an isothermal plate using a numerical finite-difference for different Reynolds numbers. They found that the stagnation Nusselt number is directly proportional to the jet Reynolds number at the steady state condition, whereas those at the unsteady condition are less dependent on the Reynolds number.

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Numerical investigations were carried out by Lee et al. (2005) for two-dimensional laminar slot jet phenomena in the presence of an applied magnetic field to control vortices and enhance heat transfer. The visualization and measurement of heat transfer characteristics are conducted by Hsieh et al. (2006) for the low jet Reynolds number. They reported that flow at even higher Reynolds numbers becomes unstable due to the flow inertia. Experimental investigation at low Reynolds numbers up to 10,000 at various nozzle-to-plate spacing were conducted by Baydar (1999). A sub-atmospheric region, vortex, is observed on the impingement plate for Re > 2700 and the nozzle-to-plate spacing ratio less than 2 and that there existed a linkage between the sub-atmospheric region, (vortex) and the peaks in local heat-transfer coefficients.

The heat transfer in a semi-confined configuration experimentally and numerically was investigated by Baydar and Ozmali (2005). The study showed the dependence of the heat transfer on flow structure (vortices) at impingement wall and confined plate. They carried out their study at Re=30000 for nozzle to plate spacing ratio between 1 and 6. Ye-zheng Yu et al. (2014) conducted an experimental study related to the heat transfer produced by single row of impinging jets inside a confined channel with different tab orientations of the triangular tabs at the jet exits. The effects of the tab number, tab orientation angle and tab penetration length on trends and differences observed in the heat transfer measurements. The results showed that the presence of tabs increases the jet core velocity and induces array pairs of vortices, and hence enhances the heat transfer in the impingement region over the no-tab case. Yousefi-Laflouraki (2014) investigated numerically a confined impinging slot jet in a converging channel. The numerical simulations were performed for different angles of 0°–5° in order to consider the effect of converging angle on the flow and temperature fields in the channel. The results showed that the intensity and size of the vortex structures increase with raising jet-impingement surface distance ratio, Reynolds number and also converging angle. Also, as the converging angle increases, the average Nusselt number and skin friction increase due to raising velocity gradient along the channel. In some applications the products, which need to be cooled, are carried on moving conveyors. However, there is lack of research work regarding the effect of the moving wall on the heat transfer as well as the flow field for semi-confined jet configuration.

The main objective of the present study is to investigate the effect of the impingement wall motion on flow structures, vortices, in a semi confined impingement jet configuration at low Reynolds number, hence the heat transfer process at the impingement wall.

**Governing Equations:**

Fig. 1 shows the computational domain and coordinate system for the two-dimensional confined impinging jet considered in the present study. The governing equations solved are defined as:

**Continuity equation :**
\[ \nabla \cdot \mathbf{u} = 0 \]  
(1)

**Momentum Equation :**
\[ \nabla \cdot \rho \mathbf{u} = -\nabla P + \frac{1}{Re} \nabla^2 \mathbf{u} \]  
(2)

**Energy Equation:**
\[ \nabla T = \frac{1}{Re Pr} \nabla^2 T \]  
(3)

The dimensionless variables in the present simulation are defined as:

\[ Re = \frac{\rho u}{\mu} \]  
(4)

\[ C_p = \frac{\frac{\rho}{0.5 \rho U^2}}{} \]  
(5)

\[ T^* = \frac{T - T_w}{\frac{T_j - T_w}{q}} \]  
(6)

\[ h = \frac{h}{\frac{T_j - T_w}{q}} \]  
(7)

\[ Nu = \frac{h B}{k_{air}} \]  
(8)

\[ Nu_{av} = \frac{1}{2L} \int_L^L N \nu \, dx \]  
(9)
Where $Re$, $Pr$, $C_p$, $T^*$, $Nu$, $Nu_{av}$ represent Reynolds number, Prandtl number, pressure coefficient, dimensionless temperature, local and average Nusselt number respectively. In the present simulations, $Pr$ has been taken to be 0.7. Incompressible two-dimensional flow with constant thermodynamic and transport properties for the fluid are investigated. The boundary conditions as shown in Fig.1 can be outlined as follows:

Inlet velocity: $u = 0, v = 0, T^* = 1$

Confined wall: $u = v = 0, \frac{\partial T^*}{\partial y} = 0$

Impingement wall: $u = 0, v = 0, T^* = 0$

Lateral exit: $\frac{\partial u}{\partial x} = 0, \frac{\partial v}{\partial x} = 0, \frac{\partial T^*}{\partial x} = 0$

Numerical methodology

The grid distribution used in the present study is illustrated in Fig. 2 for the half of the domain only because of symmetry. The total number of grid points used are 650 (in x direction) and 200 (in y direction). The computational domain is subdivided into two sub domains. Domain 1 from $x = -0.5B$ to $0.5B$ with 350 grid points in the x direction. Domain 2 from $x = 0.5B$ to $x = 40B$ and from $x = -0.5B$ to $x = -40B$, the number of x grid points are 300 with the expansion ratio 1.02 in the outlet direction. The Y grid points in both domains are 200 with expansion ratio=1.02 in the two directions. The number of grid point and distribution as well as the number of iterations used in this study is the result of optimum grid resolution calculation performed for $H/B = 3$ and $Re = 500$ and $L = 40B$. Numerical study is carried out using Gambit 2.2.3 software for mesh generation and Fluent 6.2 as a CFD solver. Numerical study is carried out for fixed value of $Re=500$ and the spacing ratios $H/B$ are 2.3, and 4. The induced impingement wall velocity ratios are $u_n/v_j$ = 5%, 10%, 15%, and 20%. The flow
the heat transfer. At the location of the vortex at the impingent wall the temperature gradient increases at larger portion of the wall which improves the enhancement of the heat transfer process. The correlation formula is developed using the least square method with the correlation coefficient of 0.98 and the increase for $H/B=4$. The enhancement of the heat transfer process is correlated with $R$ and $H/B$ for $R<1$. The $H/B=3$ values are higher than those of $H/B=4$ due to the decrease in the velocity near the wall because the flow area increases as $R$ decreases up to $0.786$.

Figure 8 illustrates the average $Nu/Nu$ becomes higher over a larger portion of the wall than that of $R>1$ and $R=1$ cases, hence the average $Nu$ becomes higher. Figure 6 depicts the local $Nu$ at the impingement wall for $R=1$, $R=0.74$ and $R=1.35$ at $H/B=2, 3$ and $4$ respectively. For $R<1$, the figures show as $R$ decreases the local $Nu$ values for $R<1$ and $R=1$ at different $H/B$ is discussed in Fig. 5 by investigating separation point locations. The figure shows that as $R$ decreases, the lower wall vortex (LV) moves outward which is reported by Baydar (1999) in the case of changing the $H/B$. The figure shows a linear variation between the vortex location and the flow parameter, $R$. Figure 6 depicts the local $Nu$ at the impingement wall for $R=1$, $R=0.74$ and $R=1.35$ at $H/B=3$. For stationary case, $R=1$, the local $Nu$ is symmetric about $x/B =0$, it has a peak value at stagnation point ($x/B =0$). The local $Nu$ decreases rapidly, the first decay rate zone, close to the stagnation point where the normal velocity turns aggressively to the parallel velocity in the wall jet zone. Then the decay rate decreases, second decay region followed by a sharp decay, until it reaches the separation point. The $Nu$ has a constant low value from separation point to the center of vortex locations. The $Nu$ value starts to increase to reach a second peak value at the trailing edge of the vortex. For $R=0.74$, the $Nu$ has the same values and trend in the stagnation zone as that of the stationary case, $R=1$. However, for $R=0.74$, the second $Nu$ decay rate end location is shifted toward the outlet boundary due to the shifting of the vortex (LV) location. Also, the figure shows that the $Nu$ values are higher over larger portion of the impingement wall than that of the stationary case. For $R=1.35$, the second decay region disappears due to existence of a vortex close to the stagnation zone. Also, the first peak of the $Nu$ value is shifted toward the stagnation point. Figures 7a to 7c show the local $Nu$ distributions along the impingement wall for $R<1$ and $H/B=2$, 3 and 4 respectively. For $R<1$, the figures show as $R$ decreases the $Nu$ values becomes higher over wide portion of the wall than that of $R>1$ and $R=1$ cases, hence the average $Nu$ becomes higher. Figure 8 illustrated the average $Nu/Nu_s$ variation with $R$ for the different $H/B$. For $R=1$ and $H/B=3$, $\frac{Nu_{av}}{Nu_{av,s}}$ decreases as $R$ increases and their values are lower than those of $R<1$. For $R<1$, the $Nu_{av}$ increases as $R$ decreases and decreases up to 0.786, $\frac{Nu_{av}}{Nu_{av,s}}$ increases as $H/B$ change for 2 to 4. For $R<0.785$, $Nu/Nu_s$ values for $H/B = 3$ are higher than those of $H/B = 4$ due to the decrease in the velocity near the wall because the flow area increase for $H/B=4$. The enhancement of the heat transfer process is correlated with $R$ and $H/B$ for $R<1$. The correlation formula is developed using the least square method with the correlation coefficient of 0.98 and the output formula is defined as follows:

$$\frac{Nu_{av}}{Nu_{av,s}} = 0.984 R^{-0.417(H/B)^{0.022}}$$ for $R<1$

The temperature isotherm contours for different wall velocities are depicted in Fig. 9. The figure shows that as the wall velocity increase the temperature gradient increase at larger portion of the wall which improve the heat transfer. At the location of the vortex at the impinging wall the temperature gradient decrease and...
therefore the local Nu at the zone. Figure 10 shows the variation of the shear stress along the impingement wall for different R. At x/B = ± 0.025, the shear stress has its highest value and then decreases downstream. For R = 0.74, the shear stress at jet wall zone (x/B > 10) is mainly due to wall velocity. For R = 1.35, the shear stress has its higher value at x/B = -0.025 than those of R = 1 and R = 0.74 cases. The effect of R on the local shear stress distributions is illustrated in Figs. 11a to 11c for R < 1 and different H/B values. The figures show that as H/B increase, the separation points moves further toward the flow outlet. For R < 1, as the R decreases, the locations of separation point, at which the shear stress is zero, move toward outlet boundary of the flow. Also, the shear stress decreases as R decreases. Figure 12 shows the Cp variation along the impingement wall for stationary and moving wall for H/B = 3, R = 0.74 and R = 1.35. The Cp value is the highest at the stagnation zone for the stationary and moving wall. In the jet wall zone, the CP has a high value at the trailing and leading of vortices at impingement wall. The effect of R on the Cp for different values of H/B is shown in Figs 13a to 13e, It is observed from the figures that the variation of Cp along the impingement wall depend strongly on R. The region near the impingement plate is often referred to as the deflection zone where there is a rapid decrease in axial velocity and a corresponding rise in static pressure. At the stagnation point (x/B = 0), the Cp is maximum. The Cp decreases as the flow accelerates along the impingement wall. The Cp becomes constant till the separation point and starts to increase up to the leading edge of the LV.

![Graph](image1)

**Fig. 3 Local quantities distributions along the impingement wall at H/B = 5**

(a) Pressure coefficient and (b) Nusselt number
Fig. 4 Streamlines distributions at different impingement wall velocities at $H/B = 3$ & $Re = 500$

(a) $\frac{U_1}{U_2} = 0$ (b) $\frac{U_1}{U_2} = 5\%$ (c) $\frac{U_1}{U_2} = 10\%$ (d) $\frac{U_1}{U_2} = 15\%$ (e) $\frac{U_1}{U_2} = 20\%$

Fig. 5 Effect of $R$ on separation points locations at different $H/B$
Fig. 6 Distributions of local Nusselt number along the impingement wall for different R

Fig. 7 Local Nusselt number distributions for $R \leq 1$ at : (a) $H/B=2$  (b) $H/B=3$ (c) $H/B=4$
Fig. 8 Effect of R on $N_u_{av} / N_u_{av_s}$ at different H/B

Fig. 9 Isotherms for H/B=3, Re=500 at different wall velocities
(a) $u_w / v_{j_i} = 0$, (b) $u_w / v_{j_i} = 5\%$, (c) $u_w / v_{j_i} = 10\%$, (d) $u_w / v_{j_i} = 15\%$ and (e) $u_w / v_{j_i} = 20\%$
Fig. 10 Distributions of local shear stress along the impingement wall for different $R$.

Fig. 11 Local shear stress distributions for $R \leq 1$ at: (a) $H/B=2$ (b) $H/B=3$ (c) $H/B=4$
Conclusions

For $R<1$, the motion of impingement wall enhances the heat transfer at impingement wall due to changing the location of the vortex, attached to wall, toward to outlet zone. The $Nu$ increase by 22% for the $R=0.66$ compared to the stationary wall. The average $Nu$ number increases linearly with the decrease of $R$. For $R>1$, the heat transfer is deteriorating as $R$ increase due the enhancement of the generation the vortices, retard flow, at the impingement wall. This leads to the recommendation when cooling objects placed on moving conveyers, the direction of the motion of the conveyers should be set in co-flow mode.

Nomenclature

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$B$</td>
<td>Jet nozzle width (m)</td>
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<tr>
<td>$T$</td>
<td>Temperature (K)</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Pressure coefficient</td>
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<tr>
<td>$T^*$</td>
<td>Dimensionless temperature</td>
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<tr>
<td>$H$</td>
<td>Convective heat transfer coefficient (W/m$^2$K)</td>
</tr>
<tr>
<td>$u,v$</td>
<td>Velocity in x and y direction (m/s)</td>
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<tr>
<td>$H$</td>
<td>Nozzle-to-plate spacing (m)</td>
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<tr>
<td>$x,y$</td>
<td>Cartesian coordinates (m)</td>
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<tr>
<td>$K$</td>
<td>Thermal conductivity (W/m K)</td>
</tr>
<tr>
<td>$L$</td>
<td>Channel length (m)</td>
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<tr>
<td>$\rho$</td>
<td>Density (kg/m$^3$)</td>
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<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
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<tr>
<td>$\theta$</td>
<td>Kinematic viscosity (kg/s')</td>
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<td>Pressure (N/m$^2$)</td>
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<td>$Pr$</td>
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<td>Heat transfer per unit area through impingement wall (W/m$^2$)</td>
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<td>Flow parameter</td>
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<td>Reynolds number based on jet width</td>
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<td>wall</td>
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References

