This study examines the performance of one point closure turbulence models in predicting of heat and momentum transfer of impinging flows. The scope of this paper is limited to impinging jet on a moving wall and heat transfer. The impinging distance is fixed to 8 thickness of the nozzle (8e) for this study. Two parameters are considered: the jet exit Reynolds number (10000≤Re≤25000) and the jet –surface velocity ratio (0≤Rsj≤4). The flow field structure at a given surface-to-jet velocity ratio is independent of the jet Reynolds number, a slight modification of the flow field is observed for low surface-to-jet velocity ratio (Rsj<0.25) whereas at higher ratios Rsj>0.25, the flow field is significantly modified. Good agreement with experimental results is obtained for surface-to-jet velocity ratio 0≤Rsj≤2. The purpose of this paper is to consider the case of higher of surface-to-jet velocity Rsj>2. A further study of heat transfer is achieved and shows that the stagnation points the local heat transfer coefficient have a maximum value. The local Nusselt number at the impinging region tends to decrease significantly when Rsj≤1.5. The evolution of average Nusselt number is correlated according to the surface-to-jet velocity ratios for each Reynolds number.

Key words: impingement jet, moving wall, numerical simulation, turbulence modelling, Nusselt number.

1. Introduction

The interest to the researchers of prediction of heat and momentum transfer of impinging flows is considerable. Heat transfer of impinging jets has received considerable attention in many benchmarks. Such configurations are used in many industrial and engineering applications (e.g. manufacturing, material processing and electronic cooling, drying paper, textiles and tempering of glass). The possibility of improving the heat transfer is carried out according to the parameters of the interaction. Many work have detailed this configuration, as: Abramovich [1], Rajaratnam [2], Gutmark and Wygnanski [3], Ramprian and Chandrasekhar [4], Peng et al [5] and Issa et al [6].

The flow of turbulent jet impingement on a still surface generates three regions: the free jet region, the impingement region and the wall jet region. The addition of surface motion adds one more level of complexity to the problem. Few studies have reported the effect of the impingement surface motion on the flow field and heat transfer. Subba Raju and Schlunder [7] performed an
experimental study on an impinging jet a moving belt. The mean heat transfer coefficient was calculated on the basis of the length of impingement surface for different nozzle-to-surface spacing, several Reynolds numbers and various surface velocity $V_s$ between 0.15 m/s and 5.5 m/s. They found that the mean Nusselt number increases with the surface velocity, and is 1.5-2.0 times larger than that of the stationary wall and then remains constant. Van Heiningen et al. [8] have considered a turbulent plane jet impinging a rotating drum at low tangential velocity less than 2% of the jet exit velocity. They compared their results with those found by Gardon and Akfirat [9] for a still plate and deduced that wall convective heat transfer are not influenced by the low jet–drum velocity ratios. The flow field of confined turbulent slot air jet impinging normally on a flat surface was investigated experimentally by Senter and Sollic[10]. The experiments were conducted for a nozzle-to-plate spacing of 8 slot nozzle widths, at three Reynolds number (5300, 8000, and 10600) and four surface-to-jet velocity ratios (0, 0.25, 0.5 and 1.0). The measurements of the mean velocities and turbulent quantities are presented in the following main characteristic regions of the jet. It appears that the flow field patterns at a given surface-to-jet velocity ratio are independent of the jet Reynolds number between 5300–10600. A minor modification of the flow field is observed for the surface-to-jet velocity ratio of 0.25 whereas at higher ratios of 0.5 and 1, the flow field is significantly affected.

Numerical modelling of turbulent planar air jet on a rectangular duct was performed by Huang et al. [11]. They have found that, when the plate velocity is higher, Nusselt numbers becomes smaller in the vicinity of the wall, when the wall velocity is opposed to the flow and is higher where the surface motion and the flow are in the same direction. Zumbrunnen [12] and Chen et al. [13] have shown that the movement of the impingement surface influences significantly the flow structure and heat transfer characteristics. However, they had considered a numerical model for convective heat transfer within an array submerged planar laminar jets impinging a moving hot surface with uniform heat flux. Zumbrunnen et al. [14] have developed an analytical study for a plane laminar jet impinging a moving surface. They show that the increasing of the plate velocity produce somewhat uniform heat transfer distribution away from the stagnation point. Chattopadhayay et al. [15] investigated numerically turbulent heat transfer of an array of slot jets impinging a moving plate by Large Eddy Simulation (LES), for jet exit Reynolds number ranging between 500 and 3000 and for several surface-jet velocities ratios $R_{t}$: 0, 0.5, 1 and 2. They concluded that the Nusselt number distribution over the plate becomes more uniform when the total heat transfer and plate velocity diminish. Similar conclusions were reported by Chattopadhayay and Saha [16] for laminar heat transfer of an array of impinging slot jets on a moving plate for a jet exit Reynolds number between 100–200. Later, Chattopadhayay [17] by Large Eddy Simulation investigates the turbulent flow and heat transfer by a single slot jet impingement a moving hot plate. They focus their study on the flow structure, velocity profiles and turbulent stresses distribution. The heat transfer induced by impinging jet on a moving wall was investigated numerically by Senter [18] using the standard $k-\varepsilon$ turbulence model. The nozzle-to-plate spacing $H$ is eight slot nozzle thickness ($e$) for jet exit Reynolds number $Re=10600$ and for jet to plate velocities ratios 0, 0.25, 0.5 and 1.0. A good agreement was obtained with Gardon and Akfirat [9] for immobile plate case. This study showed that the local Nusselt number in impact area tends to decrease significantly with the increasing of the plate velocity. Sharif and Benerjee [19] by using CFD Fluent code investigated a numerical study based on the standard $k-\varepsilon$ turbulence model combined to the enhanced wall treatment. Three parameters were considered: the jet exit Reynolds number (5000$\leq$Re$\leq$20000), impinging distance ($6e\leq H \leq 8e$) and the jet–surface velocity
ratio \(0 \leq R_{sj} \leq 2\). They show that the local Nusselt number along the moving plate reaches a maximum value nearby the stagnation point for low velocities ratios which can be explained by the thinning of dynamical and thermal boundary layers. At a given plate velocity, the average Nusselt number along the plate increases when the jet exit Reynolds number increases.

The studies of [20-21] focus on laminar multiples jets impinging a moving wall.

Therefore in this work, all phenomena about the plane turbulent jet impinging normally on a moving flat wall were detailed for jet–plate velocity ratio \(0 \leq R_{sj} \leq 4\).

## 2 Methodology

### 2.1 Turbulence model

Mass conservative equation (eq.1), SRANS (Steady Reynolds Average Navier Stokes) equations (eq. 2) coupled to the averaged energy equation (eq. 3) of the turbulent incompressible flows are written as follow:

**Mass conservative equation**

\[
\frac{\partial U_i}{\partial x_i} = 0
\]

**Momentum conservation equation:**

\[
\rho U_i \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \frac{\partial U_i}{\partial x_j} - \rho u_i u_j \right)
\]

**Energy conservation equation:**

\[
\rho U_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{\mu}{\rho c_p} \frac{\partial T}{\partial x_i} - \rho u_i \theta \right)
\]

where \(P, T\) and \(U_i\) are the mean pressure, temperature and velocity components respectively, \(\theta\) and \(u_i\) are the fluctuating temperature and velocity components, respectively. \(x_i\) is the coordinate direction, and \(\rho, \mu\) are the fluid density and dynamic viscosity respectively.

The Reynolds stress component and correlations between the velocity and temperature fluctuations require a closure. The one point closure turbulent models are based on the concept of Prandtl-Kolmogorov’s turbulent viscosity. Thus, the Reynolds stress tensor and the correlation of the velocity and temperature fluctuations are deduce from Boussinesq assumption:

\[
u_{ij} = \frac{2}{3} k \delta_{ij} - \nu (U_{ij} + U_{ji})
\]

\[
-\rho u_i \theta = \alpha_t \frac{\partial T}{\partial x_i}
\]

By analogy with molecular transport, the turbulent Prandtl number for thermal transport equation can be deduced through eq.6:

\[
\sigma_T = \frac{\nu}{\alpha_t}
\]
The closure energy-specific dissipation (k-ω) one point closure turbulence model is used in this paper.

The first formulation of k-ω model was developed by Wilcox [22]. It solves two transport equations, for the first one the turbulent kinetic energy $k$ (eq. 7), and second one is the turbulent frequency $\omega$, (eq. 8):

$$
\rho U_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \frac{\partial k}{\partial x_j} \right] + \mu_t \left[ \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right] \frac{\partial U_i}{\partial x_j} - \rho \omega k
$$

(7)

$$
\rho U_j \frac{\partial \omega}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \frac{\partial \omega}{\partial x_j} \right] + C_{\omega 1} \mu_t \left[ \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right] \frac{\partial U_i}{\partial x_j} - C_{\omega 2} \rho \omega^2
$$

(8)

The turbulent viscosity $\nu_t$ is deduced by:

$$
\nu_t = \frac{k}{\omega}
$$

The empirical constants of the above equations are:

$C_\mu = 0.09, C_{ol} = 0.555, C_{\omega 2} = 0.833, \sigma_k = 2.0, \sigma_\omega = 2.0$ and $\sigma_T = 1.0$.

2.2 Numerical method

The steady conservation equations for incompressible two dimensional in Cartesian coordinate are solved with the finite volume method; Patankar [23]. The finite volume method requires the following conservative form (eq. 16):

$$
\frac{\partial}{\partial x_j} \left( \rho U_j \phi \right) = \frac{\partial}{\partial x_j} \left[ \Gamma_\phi \frac{\partial \phi}{\partial x_j} \right] + S_\phi, \quad j = 1, 2
$$

(9)

Where $\phi$ is one of the dependent variables $U, V, T, k$ and $\omega$, $\Gamma_\phi$ and $S_\phi$ are the corresponding turbulent diffusion coefficient and source term, respectively. The terms of $\Gamma_\phi$ and $S_\phi$ are deduced from eq. 9 for each variable $\phi$ (see table 1.)

The convection and diffusion terms are discretized using the POWER LAW schemes. The pressure-velocity coupling is realized by SIMPLEC algorithm. The solution is considered converged when the normalized residual of each variable is less than $10^{-6}$.

The geometry and boundary conditions are summarized in Fig 1. The jet exit velocity $U_j$ is varied giving a Reynolds number, based on the nozzle width $e$ (where $\nu$ is kinematic viscosity) ranging from 10000 to 25000. The moving wall is displaced at the velocity $U_0$ corresponding to the surface-to-jet velocity ratio $R_{sj}$ defined by: $R_{sj} = U_0/U_j$ . The impinging distance $H$ is fixed to 8e in this study. The jet at the exit and the upper plate are maintained at ambient temperature $T_C$ but the moving one is heated at $T_H$ ($T_C = 293$ K and $T_H = 313$ K).
Table 1: Diffusion coefficients and source terms in the generic transport equation (9)

<table>
<thead>
<tr>
<th>Equation</th>
<th>$\phi$</th>
<th>$\Gamma_\phi$</th>
<th>$S_\phi$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuity</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>x-Momentum</td>
<td>$U$</td>
<td>$\mu_{eff} = \mu_h + \mu$</td>
<td>$\left[ -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left( \mu_{eff} \frac{\partial U}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu_{eff} \frac{\partial V}{\partial x} \right) \right]$</td>
</tr>
<tr>
<td>y-Momentum</td>
<td>$V$</td>
<td>$\mu_{eff} = \mu_h + \mu$</td>
<td>$\left[ -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left( \mu_{eff} \frac{\partial U}{\partial y} \right) + \frac{\partial}{\partial y} \left( \mu_{eff} \frac{\partial V}{\partial y} \right) \right]$</td>
</tr>
<tr>
<td>Energy</td>
<td>$T$</td>
<td>$\frac{\mu}{P} + \frac{\mu_l}{\sigma_T}$</td>
<td>0</td>
</tr>
<tr>
<td>Kinetic energy</td>
<td>$k$</td>
<td>$\mu + \frac{\mu_l}{\sigma_k}$</td>
<td>$\mu_l \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right)^2 + 2 \left( \frac{\partial U}{\partial x} \right)^2 + \left( \frac{\partial V}{\partial y} \right)^2 \right) - \rho \omega k$</td>
</tr>
<tr>
<td>Specific dissipation rate</td>
<td>$\omega$</td>
<td>$\mu + \frac{\mu_l}{\sigma_{\omega}}$</td>
<td>$C_{\omega l} \mu_l \left( \frac{\partial U}{\partial y} + \frac{\partial V}{\partial x} \right)^2 + 2 \left( \frac{\partial U}{\partial x} \right)^2 + \left( \frac{\partial V}{\partial y} \right)^2 \right) \omega - C_{\omega l} \rho \omega^2$</td>
</tr>
</tbody>
</table>

Figure 1. Configuration and boundary conditions
A two dimensional structural non uniform grid has been generated. A sufficiently fine grid is managed near the jet and the wall, where prevails a very high gradient of each variable (Fig 2).

**Figure 2. Grid arrangement**

Impinging jet flows are mainly influenced by the near-wall turbulence behaviour. The local turbulence Reynolds number $y^+$ is used to show the near wall behavior. Qualitatively, Fig. 3 shows similar profiles to those of BEAUBERT. At several section, the tangential velocity $U^+$ profiles are characterized by three zones:

- Viscous sublayer ($y^+ \leq 5$)
- Buffer layer or blending region ($5 \leq y^+ \leq 26$)
- Fully turbulent or log-law region ($y^+ \geq 26$)

In the vicinity of the wall, viscous damping reduces the tangential velocity fluctuations, while kinematic blocking reduces the normal fluctuations. Towards the outer part of the near-wall region, the turbulence grows rapidly by the production of turbulent kinetic energy due to the large gradients in mean velocity.

**Figure 3. Near wall treatment for $y/e=2$, 4, 6 and 8**

3 Results and discussion
3.1 Validation of numerical models

The mean velocities components \((U/V_j)\) and \((V/V_j)\), are in good agreement with experimental data obtained by PIV (Senter. J[18]). The profile of the figure 4 for the case (a), relative to the still wall case are symmetric. For \(R_{sj}=0.5\), the effect of the moving wall is visible by the asymmetry of velocity profiles.

![Diagram showing mean velocities components](image)

**Figure 4.** Mean velocities components \(U/V_j\) and \(V/V_j\), \((H/e = 8, Re=7850; (a), R_{sj} = 0, (b) R_{sj} = 0.5)\)**
The computed streamlines contours are plotted in Fig. 5, corresponding to surface-to-jet velocities ratios ($R_{sj}=0$ and 0.5) for $Re=10600$. The results are compared to experimental measurements (PIV) (fig. 5) and numerical prediction by $k$-$\varepsilon$ model of Senter [18]. Good agreement is obtained. The location of the stagnation point was found with a good precision by calculation. For $R_{sj}=0$, corresponding to the stationary plate. Besides, the center of the recirculation obtained by calculation is slightly shifted to that of the experiment. The first case (fig. 5) presents a typical turbulent slot jet confined and impinging on a stationary surface. Two identical eddies are observed. The second case ($R_{sj}=0.5$), a third contra-rotating eddy appears at the left part of the moving wall. This new recirculation compresses the size of the left principal vortex which deflects the jet towards the right part of the mobile wall.

<table>
<thead>
<tr>
<th></th>
<th>$R_{sj} = 0$</th>
<th>$R_{sj} = 0.5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIV</td>
<td>![Streamlines contours]</td>
<td>![Streamlines contours]</td>
</tr>
<tr>
<td>$k$-$\varepsilon$</td>
<td>![Streamlines contours]</td>
<td>![Streamlines contours]</td>
</tr>
<tr>
<td>$k$-$\omega$ present study</td>
<td>![Streamlines contours]</td>
<td>![Streamlines contours]</td>
</tr>
</tbody>
</table>

Figure 5. Streamlines contours comparison. $H/e = 8$, $Re=10600$

The variation of the local Nusselt number along the impingement plate $Nu(x)$ is calculated from the temperature gradient along the hot plate by:
\[ Nu(x) = \left( \frac{e}{T_{wall} - T_C} \right) \left( \frac{\partial T}{\partial y} \right) \]

and the average Nusselt number along the moving wall is also deduced from eq. 11:

\[ \overline{Nu} = \frac{1}{L} \int_{0}^{L} Nu(x) \, dx \]

Figure 6(a) represents the computed local Nusselt number for the case of the stationary impingement plate (\( R_{sj} = 0 \)) compared to the experimental results of Gardon and Akfirat [9] and to the numerical results of Senter [18] based on the k-\( \varepsilon \) model for \( H/e = 8 \) at \( Re = 11000 \). According to these data, we can say that the prediction of Nusselt number by the standard k-\( \omega \) model is more accurate than k-\( \varepsilon \) model.

Figure 6(b, c and d) shows the comparison of the computation based on standard k-\( \omega \) model to those of Senter [18] of the local Nusselt number evolution along the moving wall for two the surface-to-jet velocity ratios, \( R_{sj} = 0.25, 0.5 \) and 1 at \( Re = 10600 \). A good agreement between the two predictions is obtained.
3.2 Effect of the surface-to-jet velocity on the local Nusselt number evolution

Figure (7) shows the local Nusselt number evolution along the moving wall, for various values of surface-to-jet velocity ratios \( R_{sj} \) (0 \( \leq R_{sj} \leq 4 \)) at \( Re=10600 \). Figure 7(b) is an enlargement of the figure 7(a). From these figures we can give the following explanations:

- For \( R_{sj}=0 \): the curve has the shape of a symmetrical bell, it characterizes the distribution of the local number of Nusselt of the flow of the plane jet impinging perpendicularly an immobile plate. The maximum value matches exactly with the stagnation point of the jet, on both sides by the jet axis, then it decreases progressively and symmetrically.

- For 0.25 \( \leq R_{sj} \leq 1.5 \) one notes that:
  - When the velocity ratio \( R_{sj} \) increases, the maximum value of the local Nusselt number decreases and the stagnation point (\( X_o/e \)) is shifted in the same direction of the movement of the mobile plate.
  - A reduction of the local Nusselt number is observed in the left part of the plate for each velocity ratio \( R_{sj} \), this decrease is related to the creation of the new recirculation (separation of the fluid of the wall) (Senter [18] and Sharif et al [19]).

  The Nusselt number distribution at the left edge of the moving plate (\( X/e=-50 \)) is characterized by a pronounced peak which increases when \( R_{sj} \) augments. This maximum can be explained by the discontinuity of the configuration located between mobile surface and the immobile left wall (Senter [18]). This phenomenon doesn’t appear at the right edge (\( X/e=50 \)) because the jet flow is driven in the same direction of the plate motion, contrarily to the left edge (\( X/e=-50 \)) where the flow is opposed the plate movement.

  - The Nusselt number increases on the right part of the plate when \( R_{sj} \) increases by the effect of the gradual jet deflection

- For 1.75 \( \leq R_{sj} \leq 4 \); the curves of the local Nusselt number distribution are practically linear (absence of obvious peak) because the flow leaves entirely the plate.

  The locations and the maximum Nusselt number values for the various surface-to-jet velocity ratios \( R_{sj} \) are referred in tab. 2.

![Figure 7](image-url)

(a) Along hot moving wall (b) for -20 \( \leq x/e \leq 20 \)

Figure 7. Distribution of local Nusselt number with \( H/e=8 \) at \( Re=10600 \), 0 \( \leq R_{sj} \leq 4 \)
Table 2. Effect of surface-to-jet velocity on the maximum values of the local Nusselt number and the position of stagnation point. Re=10600, H/e=8

<table>
<thead>
<tr>
<th>Rsj</th>
<th>X/e</th>
<th>Nu&lt;sub&gt;max&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>61.047</td>
</tr>
<tr>
<td>0.25</td>
<td>0</td>
<td>58.796</td>
</tr>
<tr>
<td>0.5</td>
<td>0.434</td>
<td>56.000</td>
</tr>
<tr>
<td>1</td>
<td>0.784</td>
<td>47.849</td>
</tr>
<tr>
<td>1.25</td>
<td>1.837</td>
<td>36.796</td>
</tr>
<tr>
<td>1.5</td>
<td>3.450</td>
<td>25.945</td>
</tr>
</tbody>
</table>

3.3 Effect of the Reynolds number on the evolution of the local Nusselt number

The effect of the Reynolds number on the distribution of the local Nusselt number is represented in fig. 8, for jet exit Reynolds number, ranging from 10000 to 25000 and for two cases of surface-to-jet velocity ratios (Rsj=0.5 and 1).

Regardless of the Reynolds numbers range considered in this study, for a given surface velocity Rsj, the evolutions of the local Nusselt number are similar. Their maximum values increase with the Reynolds number but its location does not seem to be influenced by the Reynolds number.

![Figure 8](image)

3.4. Evolution of average Nusselt number

Figure 9 illustrates the average Nusselt number evolution according to the surface-to-jet velocity ratios at several Reynolds numbers, between -20e and 20e in order to take into account only the effect of the impinging region. The average Nusselt number decreases when the surface-to-jet velocity ratios is up to 1.5 (Rsj≤1.5), then it increases with the increase the surface-to-jet velocity ratios.

![Figure 9](image)
The average Nusselt number evolution according to the surface-to-jet velocity ratios for each Reynolds number is correlated by power expression (tab. 3):

\[ \overline{\text{Nu}} = \gamma R_{sj}^\delta \]

For \(0.25 \leq R_{sj} \leq 1.5\): \(\delta\) is equal to -0.166, but \(\gamma\) varies between 17.368 and 27.437 depending on the Reynolds number.

For \(1.75 \leq R_{sj} \leq 4\): \(\delta\) is equal to 0.646 but \(\gamma\) varies between 12.227 and 17.944 depending on the Reynolds number.

Table 3. Correlation of the average Nusselt number according \(R_{sj}\)

<table>
<thead>
<tr>
<th>(R_{sj})</th>
<th>(0.25 \leq R_{sj} \leq 1.5)</th>
<th>(1.75 \leq R_{sj} \leq 4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10000</td>
<td>(\overline{\text{Nu}} = 17.368 R_{sj}^{-0.166})</td>
<td>(\overline{\text{Nu}} = 12.227 R_{sj}^{0.646})</td>
</tr>
<tr>
<td>15000</td>
<td>(\overline{\text{Nu}} = 22.263 R_{sj}^{-0.166})</td>
<td>(\overline{\text{Nu}} = 14.764 R_{sj}^{0.646})</td>
</tr>
<tr>
<td>20000</td>
<td>(\overline{\text{Nu}} = 25.475 R_{sj}^{-0.166})</td>
<td>(\overline{\text{Nu}} = 16.579 R_{sj}^{0.646})</td>
</tr>
<tr>
<td>25000</td>
<td>(\overline{\text{Nu}} = 27.437 R_{sj}^{-0.166})</td>
<td>(\overline{\text{Nu}} = 17.944 R_{sj}^{0.646})</td>
</tr>
</tbody>
</table>

The relative error is deduced thought the difference between the calculate \(\text{Nu}\) value and the computed \(\text{Nu}\) from the correlation, by:

\[ \frac{\overline{\text{Nu}}_{\text{cal}} - \overline{\text{Nu}}_{\text{corr}}}{\overline{\text{Nu}}_{\text{cal}}} \]

The relative error of the correlations of average Nusselt number according to surface-to-jet velocity ratios \(R_{sj}\), compared calculations (k-\(\omega\)) are gathered in tab. 4. The error is great for high value of Reynolds number and the big surface-to-jet-velocity ratios.
Table 4. Relative error of the correlations of average Nusselt number according to $R_{sj}$

<table>
<thead>
<tr>
<th>$R_{sj}$</th>
<th>0.25</th>
<th>0.5</th>
<th>1</th>
<th>1.25</th>
<th>1.5</th>
<th>1.75</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re 10000</td>
<td>1.15</td>
<td>5.01</td>
<td>2.73</td>
<td>2.28</td>
<td>4.30</td>
<td>1.30</td>
<td>2.86</td>
<td>1.34</td>
<td>1.46</td>
</tr>
<tr>
<td>Re 15000</td>
<td>3.51</td>
<td>4.94</td>
<td>2.99</td>
<td>4.42</td>
<td>8.03</td>
<td>1.80</td>
<td>3.30</td>
<td>2.01</td>
<td>7.76</td>
</tr>
<tr>
<td>Re 20000</td>
<td>4.35</td>
<td>1.47</td>
<td>4.54</td>
<td>8.87</td>
<td>10.02</td>
<td>3.96</td>
<td>6.51</td>
<td>1.38</td>
<td>12.89</td>
</tr>
<tr>
<td>Re 25000</td>
<td>7.36</td>
<td>3.09</td>
<td>5.90</td>
<td>10.49</td>
<td>6.69</td>
<td>5.58</td>
<td>6.01</td>
<td>4.04</td>
<td>17.67</td>
</tr>
</tbody>
</table>

4 Conclusions

The k-$\omega$ turbulence model has been used to predict the flow resulting from the orthogonal impingement of two-dimensional plane jet on a moving wall. The flow structure simulation turbulence model is compared with experimental results have been obtained using a finite volume numerical method. Results show good qualitative and quantitative agreement with measurements and previous numerical works.

The effect of Reynolds number and surface-to-jet velocity ratio on heat transfer has been analyzed. The influence of the moving surface velocity on the global structure and heat transfer of such a flow has been also captured. The calculation of time-averaged velocities and Nusselt number in this case will provide comparisons with numerical works. The heat transfer on the impingement surface is evidenced by the flow with the increasing of the moving surface velocity. Certainly, the deviation of the location of the stagnation point from the jet exit axis will generate an important variation on the heat transfer. For high velocity ratios, no stagnation point is observed because there is a full detachment of the flow from the wall. Many correlations are proposed for the average Nusselt number evolution according to the surface-to-jet velocity ratios for each Reynolds number.

Nomenclature

<table>
<thead>
<tr>
<th>$e$</th>
<th>Slot nozzle width, [m]</th>
<th>$U_j$ Nozzle jet velocity exit, [ms$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H$</td>
<td>Impinging distance, [m]</td>
<td>$u_i$ Fluctuating velocity components, [ms$^{-1}$]</td>
</tr>
<tr>
<td>$k$</td>
<td>Kinetic energy of turbulence, [m$^2$s$^{-2}$]</td>
<td>$V$ Mean velocity in the y-direction, [ms$^{-1}$]</td>
</tr>
<tr>
<td>$L$</td>
<td>Length of the impingement plate, [m]</td>
<td>$x_i$ Coordinate directions, [m]</td>
</tr>
<tr>
<td>$\text{Nu}_x$</td>
<td>Local Nusselt number, [-]</td>
<td>$y^+$ Non-dimensional distance, [-]</td>
</tr>
<tr>
<td>$\overline{\text{Nu}}_x$</td>
<td>Average Nusselt number, [-]</td>
<td></td>
</tr>
<tr>
<td>$P$</td>
<td>Mean pressure, [Pa]</td>
<td>$\rho$ Fluid density, [Kgm$^{-3}$]</td>
</tr>
<tr>
<td>$P_r$</td>
<td>Prandlt number, [-]</td>
<td>$\mu$ Dynamic viscosity of fluid, [Kg.m$^{-1}$s$^{-1}$]</td>
</tr>
<tr>
<td>$\text{Re}$</td>
<td>Reynold number, [-]</td>
<td>$\mu_t$ Turbulent eddy viscosity, [Kg.m$^{-1}$s$^{-1}$]</td>
</tr>
<tr>
<td>$R_{sj}$</td>
<td>Surface-to-jet velocity ratio, [-]</td>
<td>$\omega$ Specific dissipation rate, [s$^{-1}$]</td>
</tr>
<tr>
<td>$T$</td>
<td>Mean temperature, [K]</td>
<td>$C$ Cold temperature</td>
</tr>
<tr>
<td>$U$</td>
<td>Mean velocity in the x-direction, [ms$^{-1}$]</td>
<td>$H$ Hot temperature</td>
</tr>
<tr>
<td>$U'$</td>
<td>Non-dimensional velocity [-]</td>
<td>$t$ Turbulent</td>
</tr>
<tr>
<td>$U_0$</td>
<td>Velocity of the impingement surface, [ms$^{-1}$]</td>
<td></td>
</tr>
<tr>
<td>$U_i$</td>
<td>Mean velocity components, [ms$^{-1}$]</td>
<td></td>
</tr>
</tbody>
</table>
References


