Heat transfer modelling of slot jet impinging on an inclined plate

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Abstract

A numerical study has been conducted to investigate the heat transfer in the turbulence, unconfined, and submerged flat and inclined plate impinging jet discharged from a slot nozzle by using the commercial finite volume code FLUENT. The Reynolds number in the range of 4000-16000, the nozzle exit-to-plate spacing (H/D) in the range of 4-10 and the inclination angle of air jet and plate in the range of 40-90 have been considered. The constant heat flux of 100 W/m² is set to the impinging plate. Two $k-\varepsilon$ RNG and Reynolds stress models and an Enhanced Wall Treatment for near wall turbulence modelling were applied in all cases and the local Nusselt number on the impinging plate has been compared with experimental results. The Enhanced Wall Treatment by solving the fully turbulence region and sublayer and using a single function improves the effect of pressure gradients and thermal effects. In an inclined impinging jet the stagnation point and maximum Nusselt location move to the uphill side and decreasing of the Nusselt number is more gradual on the downhill side. The movement of maximum Nusselt point to the uphill side increases by increasing the Reynolds number, H/D and by decreasing the inclination angle. The numerical results predict the heat transfer rate in a flat plate impinging jet by less than 10% error, however for an inclined impinging jet with different H/D the results are 5-20% different in comparison with experiments.

Keywords: inclined impinging jet, unconfined, slot nozzle, Nusselt number, Reynolds stress, K-epsilon RNG, Enhanced Wall Treatment.
1 Introduction

The impinging jet has received considerable attention because of the high heat/mass transfer rates in the impinging region. Heating, cooling, drying, and material processing are some applications of the impinging jets.

This numerical study investigates flat and inclined, slot nozzle, unconfined, and submerged impinging plate heat transfer and the numerical results have been validated with recent experimental work of Abdalmonem et al [1].

Obot et al. [2] reported that the confinement reduces the heat transfer rate from 5 to 10% when the nozzle exit-to-plate spacing, H/D<12. The entrainment of ambient air and pressure distributions are two main parameters which affect confinement of the submerged impinging jets [3].

In comparison slot nozzle impinging jets give an even distribution of heat transfer coefficient while round nozzles give higher heat transfer rate in the stagnation region.

The value and distribution of the heat transfer coefficient is also a function of H/D, Reynolds number based on nozzle hydraulic diameter and nozzle exit average velocity, nozzle exit turbulence and velocity profile, roughness of the heated plate, nozzle diameter, amount of heat flux in high Reynolds numbers and confinement.

Baughn and Shimizu [4], Yung-Yang et al. [5], Lytle and Webb [6] experimentally obtained the amount of heat transfer coefficient on the impinging jet of a round jet and studied the effect of various H/D, Reynolds number and amount of heat flux from the impinging plate on Nusselt number.

The effect of inclination has been studied by Beltaos [7], Stevens and Webb [8] and Yan and Saniei [9]. Abdalmonem et al. [1] concluded that the maximum heat transfer region in the inclined slot jet shifts towards the uphill side of the plate and the maximum Nusselt number decreases as the inclination angle decreases. Also the heat transfer distribution in the uphill side becomes less sensitive to jet exit-to-plate spacing.

The turbulence modelling of the impinging jet because of its complexity has become a challenge test case in numerical studies. Craft et al. [10] achieved poor agreement for a low Reynolds $k-\varepsilon$ model and basic Reynolds Stress (RS) models while the modification of the wall reflection effect improved the results.

Durbin [11] discussed the large growth of the turbulence kinetic energy in stagnation flows which causes the overestimation of the Nusselt number in two equation turbulence models. Using the $k-\varepsilon-\nu^2$ model of Durbin [12], Behnia et al. [13] improved the Nusselt number prediction in the impinging jet. Another study by Behnia et al. [14] compared an confined and unconfined impinging jet using the $\nu^2 f$ model. The results showed that the confinement decreases the average heat transfer rate but the local stagnation heat transfer coefficient remains unchanged.

Morris et al. [15] used a basic RS model by FLUENT and validated the numerical results with the flow pattern of a confined impinging jet and achieved
good agreement compared with the poor prediction of the $k-\varepsilon$ standard and RNG models.

Park and Hyung [16] modified the near wall conditions of the $k-\varepsilon-f_\mu$ model and achieved good prediction of the heat transfer rate on a flat plate for the round jet impingement.

Roy and Patel [17] applied the $k-\varepsilon$ RNG model by using FLUENT for a three dimensional two inclined impinging jet and discussed the flow-thermal characteristics of the results.

In a heat transfer study of an impinging jet, the most important factor is the distribution of turbulence heat flux across the sublayer because most of the temperature drop happens across the low Reynolds number sublayer [10]. This phenomena and presented background shows that the near wall turbulence condition in the application of the $k-\varepsilon$ and basic RS models (which have been designed for fully developed turbulence conditions) is a critical parameter in prediction of the Nusselt number. Note that the Nusselt number also depends on the level of turbulence energy prevailing near the edge of the fully turbulence region and thus is quite sensitive also to the turbulence modelling.

This study compares the RS and $k-\varepsilon$ RNG model by using the Enhanced Wall Treatment for near wall turbulence modelling by commercial code FLUENT 6.0 [18], and for the flat and inclined impinging slot nozzle jet. Enhanced Wall Treatment solves the turbulence sublayer instead of using a function to set and simulate turbulence behaviour near the wall.

### 2 Numerical modeling

In the impinging jet, flow along the centre line of a nozzle to the stagnation point and so in the vicinity of the stagnation point is nearly irrotational and has rapid normal strain changes. In the edges of the impinging jet, flow is strongly rotational with stream line curvature. From the stagnation point on the wall, normal stresses change to shear along the wall. Although the $k-\varepsilon$ RNG model, based on the Boussinesq approach, is derived by assumption of isotropic magnitude of turbulence viscosity ($\mu_t$), two modifications in comparison to the standard $k-\varepsilon$ model make it suitable in the impinging jet simulation. By modifying one of the terms in the dissipation equation of the RNG model, this model has become sensitive to changes in flow strain rate. In large strain rate regions, this sensitivity causes the reduction of turbulence kinetic energy ($k$) and thus turbulence viscosity. Also turbulence viscosity in the $k-\varepsilon$ RNG model is modified to take the effect of rotation or swirl in the flow. The modification is done by:

$$\mu_t = \mu_{t0} \cdot f(\alpha_s, \Omega, k / \varepsilon)$$

where $\mu_{t0}$ is the value of original turbulence viscosity, $\Omega$ is a characteristic swirl number, $\alpha_s$ is a swirl constant and $\varepsilon$ is dissipation rate.
In the two dimensional RS model, five equations more than for the $k-\varepsilon$ model are solved in two dimensions which causes a 35-40\% increase in time per iteration of solution. This model improves prediction of anisotropic flows which include stream line curvature, swirl, rotation, and rapid strain changing rate. The modification which has been used in this study is the wall reflection effect which damps the normal stress perpendicular to the wall and enhances stresses parallel to the wall. The effect of this modification in heat transfer is a decrease of Nusselt number in the stagnation region.

For the near wall turbulence simulation in this study, the Enhanced Wall Treatment option has been used. In this model different turbulence near wall layers are simulated by a single wall law for the entire region. This approach allows the effects of pressure gradient and variable properties to be taken into account. The single wall law is formulated by blending the laminar and logarithmic profiles. In the fully turbulence region classical logarithmic law has been modified by the effect of pressure gradients and thermal effects. The laminar sublayer law only includes the effect of pressure gradients and neglects variable properties due to the heat transfer and compressibility. The heat flux in this study was adjusted to a value where temperature change is low and fluid parameters assumed constant.

**Computational domain:** The two-dimensional computational domain consists of the inlet, impinging wall, nozzle walls and outlet boundaries (see Fig.1). The inclined impinging is studied by rotating the flat plate around the fixed intersection of the nozzle symmetry line and plate and nozzle-exit-to-plate spacing is based on the distance of the nozzle from the fixed point on the plate. The nozzle length is 5.5 mm, and the impinging target plate length with heat flux is 200mm as it was in the experiments. The impinging wall is extended by 900mm without heat flux from each side as the domain is extended to prevent influence of the far field boundary condition on the flow and thermal characteristics of the study region.

![Computational Domain](image)

**Figure 1:** Computational domain of an inclined impinging jet.
For simulating the experiment conditions of the inlet boundary condition, a two dimensional duct flow with a fixed length has been solved numerically and outlet velocity, turbulence intensity, and turbulence length scale of duct have been adjusted as inlet conditions of the impinging jet. The turbulence intensity of duct inlet was assumed to be 2% and the turbulence length scale based on duct diameter is adjusted to 0.001. In the range of Reynolds number in this study the velocity profile obtained a fully developed parabolic outlet of the duct (inlet condition of nozzle).

All the walls were imposed to no-slip conditions. Also the study region of the impinging wall was set to 100 $W/m^2$ heat flux from the wall. The pressure in the far-field outlet boundary is adjusted to ambient pressure.

The temperature difference on the impinging jet region was found to be less than 10K and incompressible air flow with constant fluid parameters was applied.

Using the Enhanced Wall Treatment to solve the viscous sublayer, demands a fine grid size perpendicular to the wall. A boundary layer grid by 20 rows was adjusted to the impinging wall. The first row size on the wall is $1\times10^{-5}$ m and the growing factor of the boundary layer grid is 1.2. Also the number of grids parallel to the wall in the heat flux region is considered to be 200. Two grid independency tests were adjusted by changing the grid size parallel and vertical to the wall. The grid independency test has been done by the $k-\varepsilon$ RNG model and for a flat plate impinging jet of H/D=6 and Re=4000.

By increasing the number of grids parallel to the wall from 100 to 300 the change in Nusselt number was found to be less than 2.5% and the solution assumed grid independence.

<table>
<thead>
<tr>
<th>B.L. Grid first row size (m)</th>
<th>B.L. Grid growing factor</th>
<th>B.L. Grid number of rows</th>
<th>Stagnation Nusselt number</th>
<th>Stagnation-adjacent cell $(y^+)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>5e-6</td>
<td>1.2</td>
<td>36</td>
<td>30.42</td>
<td>0.02</td>
</tr>
<tr>
<td>1e-5</td>
<td>1.2</td>
<td>24</td>
<td>30.97</td>
<td>0.044</td>
</tr>
<tr>
<td>1e-4</td>
<td>1.15</td>
<td>16</td>
<td>32.03</td>
<td>0.447</td>
</tr>
<tr>
<td>5e-4</td>
<td>1.15</td>
<td>10</td>
<td>----</td>
<td>1.71</td>
</tr>
</tbody>
</table>

The results of changing boundary layer grid size in table 1 show that by using Enhanced Wall Treatment for impinging jet simulation in the stagnation-adjacent cell $y^+$ should be less than about 0.5 to have enough grid in the sublayer region [18]. The Nusselt number along the impinging wall for different conditions of table 1 show that $y^+$ above the critical value leads to failure of the prediction. The effect of exceeding $y^+$ from the critical point is seen in the stagnation region in Figure 2.

Below the critical $y^+$, the change in stagnation Nusselt number is less than 5% which assumed grid independence.
Solution parameters: A segregated and explicit solver was chosen to solve flow, energy, turbulence and Reynolds stresses (RS model) equations in two dimensions. The Pressure Staggering option was chosen for pressure discretization which suites steep pressure gradients [18]. The momentum and Reynolds stress equations were discretized by the second order interpolation scheme. The common value of underrelaxation factors for pressure was set to 0.2 in the RS model and 0.3 for the $k-\varepsilon$ RNG model while for other parameters it was set to 0.4 for the RS model and 0.6 for the $k-\varepsilon$ RNG model. In each case, the solution was considered converged when the normalized residual errors were reduced to $1\times10^{-4}$ for continuity, momentum and turbulence and $1\times10^{-6}$ for the energy equation.

Figure 2: Local Nusselt along the impinging wall with different sizes of boundary layer grid.

3 Results and discussion

In the slot nozzle flat plate impinging jet, the maximum Nusselt number on the impinging plate happens at the stagnation point and then the Nusselt number reduces along the wall in the flow direction in both sides. The value of Nusselt number increases by reduction of H/D and increase of Reynolds number.

For the impinging jet to the inclined plate the maximum Nusselt moves to the uphill side of the plate where the stagnation point has been moved. As a qualitative description, by dividing the velocity vector from the nozzle exit to inclined plate into components parallel and vertical to the plate, the parallel component acts as a wall jet on the downhill side of the plate and the vertical component cause the creation of a stagnation point which is on the uphill side of the plate. Fig. 3 shows the streamlines of the impinging jet to the inclined plate (H/D=10, Re=12000, Theta=40) by using the RS model which gives a more accurate prediction of flow field streamlines [15]. On the uphill side of the stagnation point a recirculating toroid is created between the uphill side of the plate on one side and the nozzle wall and nozzle exit streamlines on the other side. The effect of inclination on the downhill side of the inclined plate is strengthening of wall jet flow.

For an inclined impinging jet, the value of Nusselt number on the uphill side of its maximum value because of sudden loss of flow momentum on the wall.
reduces with a sharp slope, however on the downhill side reduction of Nusselt number happens gradually along the wall jet region.

Fig. 4 shows that the numerical prediction of local Nusselt number of the flat plate impinging jet in different Reynolds numbers and its comparison with available experimental data [1].

The RS model in the impinging jet regions and some parts of the wall predicts higher heat transfer rate value in comparison to the $k-\varepsilon$ RNG model and the reduction of the Nusselt number from its maximum value is more gradual. In this part the trends of experimental results of the local heat transfer rate is in better agreement with the RS model while the trends of the $k-\varepsilon$ RNG model after $|X/D|>3$ are more reliable. The value of the local heat transfer in both numerical models with increase of the Reynolds number, increases slightly related to experimental data. The prediction of Nusselt number by the $k-\varepsilon$ RNG model at the stagnation point for Re=4000 is 7% underestimated and for Re=7900 is 1.5% overestimated while these percentage differences for the RS model are 1% underestimated and 5.2% overestimated respectively. Some part of this slight increase is because of the increasing of $y^+$ value at wall-adjacent cells by increasing of the Reynolds number.

Figure 3: Flow streamlines (impinging jet to inclined plate).

Figure 4: Local Nusselt number on flat plate impinging jet in different Reynolds numbers.
The results of the heat transfer rate for constant inclination angle (70) and Reynolds (4000) in different H/D shows more accurate predictions in low H/D ratios in comparison to higher H/D (fig.6-right). The percentage difference for results of the $k-\varepsilon$ RNG model and experiments varied from 2.7% in H/D=4 to 16% in H/D=8 and H/D=10. The percentages for the results of the Reynolds stress model are slightly higher than the $k-\varepsilon$ RNG model.

Figure 5: Local Nusselt number on inclined impinging jet plate in different H/D (right) and different inclination angles (left).

Since the experimental results have been measured in fixed locations (X/D=0, ± 1.5, ± 3.0,...) hence the correlations in the impinging region with this data can not give accurate results, the exact maximum heat transfer location in an inclined impinging jet is difficult to estimate; however the results of Fig.6 show better prediction for the location of the maximum Nusselt number and the heat
transfer rate distribution along the inclined impinging jet by the $k-\varepsilon$ RNG model. The prediction of maximum Nusselt number location moves to the uphill side of the inclined plate by increasing the H/D in both numerical models. The maximum Nusselt location in fig. 5 for H/D = 4, 6, 8, and 10 by the $k-\varepsilon$ RNG model happens for X/D = 0.28, 0.34, 0.45, and 0.50 respectively.

Fig. 5, left side, shows the local Nusselt number on the inclined impinging plate in different angles. The results show that the RS model is more sensitive to change in inclination angle. The trends of the $k-\varepsilon$ RNG model in the stagnation region and along the inclined wall give good prediction and are more stable. The comparison with experiments for the $k-\varepsilon$ RNG model for an inclination angle of 40, 50, and 70 shows 2.2, 4.4, and 6.7% underestimation for heat transfer rate in X/D = 0. The location of maximum heat transfer rate moves to the uphill side of the inclined plate as the inclination angle decreases. The location of maximum Nusselt number in fig. 5 for Theta = 70, 50, and 40 by the $k-\varepsilon$ RNG model happens in X/D = 0.14, 0.44, and 0.75 respectively. The value of the heat transfer rate by the RS model in different angles gives better results; however prediction of the heat transfer rate is unstable in low inclination angles. The RS model for an inclination angle of 40 fails to predict the heat transfer in the uphill side of the plate. The other study of different inclination angle by H/D = 10 confirms instability of the RS model when Theta = 40 and also reduction of the accuracy in the value of Nusselt number with increasing H/D. The heat transfer rate for H/D = 10 and Theta = 40 shows a 20% difference in comparison to experiments.

4 Conclusions

The numerical simulation was adjusted to predict local heat transfer rate in a flat and inclined impinging jet by using the $k-\varepsilon$ RNG and basic Reynolds Stress models with Enhanced Wall Treatment for near wall turbulence modelling. The value of local Nusselt number in the range of Reynolds number, H/D, and inclination angle was predicted with about 1-3% different from experiments with H/D = 4 and Theta = 70-90 and about 16% different when H/D = 10 and Theta = 70. The accuracy of Nusselt number prediction reduced by increasing the H/D and slightly by decreasing the inclination angle. The trends of local Nusselt number by using the $k-\varepsilon$ RNG model gave good prediction in all ranges of this study; however the value of Nusselt number was underestimated in the stagnation region in high H/D. The Reynolds Stress model predicted the local Nusselt value for lower inclination angles better than the $k-\varepsilon$ RNG model, however the results of the RS model at an inclination angle of 40 showed instability.

References


