International Journal of Thermal Sciences 89 (2015) 294–304

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

# Inclination of an impinging jet on a moving wall to control the stagnation point location

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A R T I C L E I N F O

*Article history:*
Received 17 December 2013
Received in revised form
5 November 2014
Accepted 8 November 2014
Available online

*Keywords:*
Impinging jet
Inclined slot jet
Fluid phenomena
Heat transfer
Nusselt number correlation

A B S T R A C T

The present work is devoted to the numerical study of the interaction of an inclined plane turbulent jet with a moving horizontal isothermal hot wall. The inclination of the jet allows the control of the stagnation point location. The numerical predictions based on statistical modeling are achieved using second order Reynolds stress turbulence model coupled to the enhanced wall treatment. For a given impinging distance $H$ ($H = 8e$), the problem parameters are: (a) jet exit Reynolds number (Re, based on the thickness of the nozzle: $e$) ranged from 10,000 to 25,000, (b) surface-to-jet velocity ratio $R_{sj}$ from left to right; range of 0–1.75 and (c) enhanced inclination angle of the jet between 0° and 25°. The calculations are in good agreement with the available data. The numerical results show that the heat transfer is greatly influenced by the velocities of the jet and the moving wall. The local Nusselt number decreases with increasing surface-to-jet velocity ratios (until $R_{sj} = 1$). However, the accurate inclination of the jet enhances heat transfer and recovers the stagnation point location. The distribution of average Nusselt number is correlated according with some problem parameters.



## 1. Introduction

Impinging jets have several industrial applications including the tempering of glass plate, annealing of metal sheets, drying of textile and paper products. In these applications, the jet impinges a moving flat plate. Only few studies have reported the effect of the impingement surface motion on the flow field and the heat transfer. The first work of jet impingement on a moving wall is that of Subba Raju and Schlünder [19]. They have investigated experimentally heat transfer from a single jet on a moving belt. Huang et al. [11] performed numerical modeling of a turbulent, plane air jet on a rectangular duct with surface motion effects. They found that for a high speed of the plate, the Nusselt number is smaller than those at the areas where movement of the surface is opposite to the jet flow, and increases in the region where the movement of the plate and flow are in the same direction. Zumbrunnen et al. [21] investigated analytically the case of a slot laminar jet, impinging an isothermal moving plate subjected to a constant flux. They showed that the heat transfer is more effective due to the slowdown of the development of the boundary layer away from the jet due to the effect of the plate motion. Chattopadhyay et al. [5] examined numerically using Large Eddy Simulation (LES), the heat transfer of an array of plane jets impinging a moving plate; for a jet exit Reynolds number ranging between 500 and 3000 and for several surface-jet velocities ratios $R_{sj}$ of: 0, 0.5, 1 and 2. They confirm that the distribution of the Nusselt number on the plate becomes uniform when the plate velocity decreases. Later, Chattopadhyay and Saha [6] using a large eddy simulation LES, studied the turbulent flow and heat transfer, generated by a single jet impinging a heated moving isotherm plate. They presented the results for only one Reynolds number of 5800, with a single perpendicular jet impinging a moving plate at velocities ratios between 0 and 2. The normalized impinging distance was 10. They mainly focused on the details of the flow structure and the profiles of the velocity and those of the stresses. The confined flow field of a turbulent slot air jet impinging perpendicularly a flat plate, was investigated experimentally by Senter [17] and Senter, J, and C. Solliec [18]. The experiments were conducted for a nozzle-to-plate spacing of 8 slot nozzle widths, for three Reynolds number (5300, 8000, and 10,600) and four surface-to-jet velocity ratios (0, 0.25, 0.5 and 1.0). It appears that the flow patterns for a given surface-to-jet velocity ratio, are independent of the jet Reynolds number in the range of 5300–10,600. A slight modification of the flow field occurs for the

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http://dx.doi.org/10.1016/j.ijthermalsci.2014.11.007
1290-0729/

case of surface-to-jet velocity ratio of 0.25, but for higher ratios of 0.5 and 1, the streamlines are strongly deformed. At a given Reynolds number of 10,600 and for normalized plate velocities 0, 0.25, 0.5 and 1.0. Senter [17] has investigated numerically the local Nusselt number using CFD code Fluent, using the $k-\varepsilon$ WCP turbulence model. This study showed the influence of the jet on the local Nusselt number around the impinging area. He found that local Nusselt number decreases significantly when the plate velocity increases. After that, Sharif et al. (2009) have also used the CFD code Fluent, using the standard $k-\varepsilon$ turbulence model coupled to enhanced wall treatment. They have shown that the local Nusselt number along the moving plate exceeds the values near the impinging region for low plate velocities, which are due to the thinning of dynamical and thermal boundary layers. These excess decreases for higher plate velocities are due to the domination of the driven parallel flow by the plate motion. At a given plate velocity, the average Nusselt number increases while the average skin friction coefficient decreases when jet Reynolds number augments. On the other hand, at a given Reynolds number of 10,600, both of these quantities decrease slightly initially and then increase sharply for a faster plate velocity. By the RANS $k-\omega$ turbulence model, Benmouhoub and Mataoui [4] investigated the turbulent flow and heat transfer of a plane jet impinging a moving hot wall, for a Reynolds number between 10,000 and 25,000, a nozzle to plate spacing of 8e, and velocity ratio ranging between 0 and 4. They presented correlation for average Nusselt number and average skin friction coefficient in terms of Reynolds number and surface-to-jet velocity ratio.

Korger and Kized [13], Belataos [3], Beitlmal et al. [2], Tong [20] and Ramezanpour [22] examined the case of a slot jet impinging an immobile inclined wall. They concluded that, the maximum transfer coefficients are highest for the case of perpendicular impingement. They also showed that the maximum heat transfer region (stagnation point) shifts towards the uphill side of the plate and the maximum Nusselt number value decreases as the inclination angle increases. Senter [17], from measurements of the case of a perpendicular jet impingement on a mobile wall, examined the inclined case in order to recover the characteristics of perpendicular impingement on an immobile wall. Indeed, for a Reynolds number of 10,600 and surface-to-jet velocity ratio of 1, Senter [17] confirmed an improvement in the Nusselt average number of 25% for a jet inclination of 8°. Table 1 summarizes some contributions of the main works of impinging jet studies. However, moving surface has also bad influence on heat transfer even in boiling regime, as evidence experimental studies of Filipovic [8], Devynck [7], Gradeck et al. [10].

**Table 1**
Citations of impinging jet studies.

| Authors | Direction of the jet | Wall | Methodology |
|---|---|---|---|
| Beaubert & Viazzo [1] | Perpendicular | Immobile | Numerical (CFD-LES) |
| Beitelmal et al. [2] | Oblique | Immobile | Experimental |
| Beltaos [3] | Oblique | Immobile | Experimental |
| Benmouhoub & Mataoui [4] | Perpendicular | Moving | Numerical (CFD-RANS) |
| Chattopadhyay et al. [5,6] | Perpendicular | Moving | Numerical (CFD-LES) |
| Gardon & Akfirat [9] | Perpendicular | Immobile | Experimental |
| Korger and Kized [13] | Oblique | Immobile | Experimental |
| Tong [20] | Oblique | Immobile | Numerical (CFD-VOF) |
| Ramezanpour [22] | Oblique | Immobile | Numerical (CFD-VOF) |
| Huang et al. [11] | Perpendicular | Moving | Experimental |
| Senter, J. [17] | Perpendicular and oblique | Moving | Numerical (CFD-RANS) Experimental |
| Senter & Solliec [18] | Perpendicular and oblique | Moving | Numerical (CFD-RANS) Experimental |

The present study extends the previous work of a perpendicular jet interaction on a moving wall by examining the effect of the jet inclination (Fig. 1) on the flow structure and heat transfer. The objective of this work is to find the inclination corresponding to the flow structure case of a jet impinging perpendicularly on an immobile wall.

## 2. Methodology

### 2.1. Governing equations

The flow is steady on average and fully turbulent. The fluid (air) is incompressible with constant thermophysical properties. The conservatives averaged equations are: mass (Eq. (1)), momentum (Eq. (2)) and energy (Eq. (3)). These equations are coupled with the equations of the turbulence model:

- Continuity

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{1}$$

- Momentum

$$U_j\frac{\partial U_i}{\partial x_j} = -\frac{\partial}{\partial x_i}\left(\frac{P}{\rho}\right) + \frac{\partial}{\partial x_j}\left(\nu\frac{\partial U_i}{\partial x_j} - \overline{u_i u_j}\right) \tag{2}$$

- Energy

$$U_i\frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\gamma\frac{\partial T}{\partial x_i} - \overline{u_i\theta}\right) \tag{3}$$

### 2.2. Turbulence model

The closure of the averaged equations is achieved by linear strain pressure - Reynolds stress second order model. This model does not require eddy viscosity hypothesis. Six transport equations of each Reynolds stresses components ($\rho\overline{u_i u_j}$) are added to the averaged Equations ((1), (2) and (3)).The differential equation Reynolds stress transport consists of the standard Reynolds stress model based on dissipation equation $\varepsilon$. There are three versions of the standard Reynolds stress models. They are called: LRR-IP, LRR-QI and SSG. Reece and Rodi [14] developed the LRR-IP and LRR-QI models. The pressure-strain expression consists of an anisotropy tensor, mean strain rate tensor and vorticity tensor. The production due to the buoyancy is neglected in this study.

By analogy with molecular transport, for all models (first or second order models), the Simple Gradient Diffusion Hypothesis (SGDH) is used. The following algebraic constitutive law is allows to deduce the velocity–temperature correlation:

$$\overline{u_j\theta} = \Gamma_t\frac{\partial T}{\partial x_j} \quad \text{Where} \quad \Gamma_t = \frac{\nu_t}{\mathrm{Pr}_t} \quad \text{and} \quad \nu_t = C_\mu\frac{k^2}{\varepsilon} \tag{4}$$

Rather significant viscosity effects characterize the flow close to the wall. Therefore, the high Reynolds models are no longer suitable in this area of flow. The wall treatment is required. After several tests, the enhanced wall treatment predicts the flow fields with the best accuracy [12].