3-D NUMERICAL HEAT TRANSFER FOR CONFINED TURBULENT TWIN CIRCULAR JETS IMPINGING ON AN INCLINED MOVING PLATE

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ABSTRACT
The objective of this study is to numerically investigate the 3-D confined turbulent twin circular jets impinging to an inclined moving plate submerged in quiescent water. The effects of different dimensionless plate velocities ($U_p = -0.2, -0.1, 0, 0.1, 0.2$), and the oblique angles ($\theta = 0^\circ, 10^\circ, 20^\circ, 30^\circ$), with $U_p = 0$ and $\theta = 0^\circ$ being the conventional vertical jet impingement to a stationary plate, on the velocity field and heat transfer phenomenon were investigated in detail. It is shown that the plate moving direction has tremendous effect on the local and average Nusselt number distributions. In the case of stationary plate ($U_p = 0$), the average Nusselt number is decreased as the plate oblique angle $\theta$ is increased. For a positive normalized plate velocity ($U_p>0$), the average Nusselt number is decreased with increasing oblique angle $\theta$, while for a negative normalized plate velocity ($U_p<0$), the opposite trend is true.

INTRODUCTION
Impinging fluid jets on a moving surface have been widely used in many industrial processes, such as metal annealing, plastic forming, secondary cooling of cast iron, cooling of gas turbine blade, electronic cooling, and drying processes, etc. due to these superior heat and mass transfer characteristics.

Huang and El-Genk [1] experimentally studied the heat transfer between a uniformly heated flat plate and an impinging circular air jet. Zumbrunnen [2-3] showed that the movement of the impingement surface strongly influences the flow field and heat transfer characteristics. Chattopadhayay and Saha [4] investigated the laminar heat transfer from a moving plate due to an array of impinging slot as well as knife jets with the jet exit Reynolds number range of 100–200 and concluded that, as the plate speed is increased, the span averaged Nusselt number is decreased. Chattopadhayay and Saha [5] used the large eddy simulation technique to investigate the turbulent flow and heat transfer from a slot jet impinging on a moving isothermal plate. The results showed that heat transfer initially increases with non-dimensional surface velocity up to 1.2 and then comes down. Senter and Solliec [6] used particle image velocimetry (PIV) to investigate the flow field of a confined turbulent slot air jet impinging on a moving flat surface. Sharif and Banerjee [7] investigated the heat transfer due to confined slot-jet impingement on a moving plate numerically. The analysis revealed that the average Nusselt number increases considerably with the jet exit Reynolds number as well as with the plate velocity.

Impinging jets with an oblique angle (non-vertical jets) are used in many applications. The imposed complexity attributes to the fact that the oblique angles in the fluid flow and heat transfer could add to the problem of nonuniform heating or cooling. Goldstein and Franchett [8] used the metallic foil heaters with liquid crystals to measure the local heat transfer from an oblique jet to a flat plate, and developed heat transfer correlation for the Nusselt number as a function of the jet oblique angle and jet-to-plate distance. Yang et al. [9] investigated multiple impinging slot jets with an inclined confinement surface numerically. The results showed that, as the inclination angle is increased, the maximum local Nusselt number is decreased. Yan and Saniei [10] studied the heat transfer from an obliquely impinging circular air jet to a flat plate. The results showed that the point of maximum heat transfer shifts away from the geometrical impingement point toward the compression side of the wall jet on the axis of symmetry. The shift is more pronounced with a smaller oblique angle and a smaller jet-to-plate distance. Abdel-Fattah [11] investigated two dimensional impinging circular twin-jet flow
numerically and experimentally. The results showed that the spreading of jet is decreased by increasing nozzle-to-plate spacing. The intensity of recirculation zone between two jets is decreased by increase of nozzle-to-plate spacing and jet angle.

The objective of this study is to numerically investigate the 3-D confined turbulent twin circular jets impinging to an inclined moving plate submerged in quiescent water. The parameters in this study are the plate velocities (\(U_p = -0.2, -0.1, 0, 0.1, 0.2\)), and the oblique angles (\(\theta = 0^\circ, 10^\circ, 20^\circ, 30^\circ\)), with \(0^\circ\) being the conventional vertical jet.

**NOMENCLATURE**

- \(H\) distance between jet exit to the plate
- \(h\) heat transfer coefficient
- \(L\) centerline spacing between two circular jets
- \(Nu\) local Nusselt number
- \(\overline{Nu}\) average Nusselt number
- \(P\) mean pressure
- \(Pr\) Prandtl number
- \(R\) radial distance from center origin
- \(Re\) Reynolds number\(=\rho v_0 D/\mu\)
- \(T\) mean temperature
- \(T'\) fluctuating temperature
- \(U_i\) mean velocity components
- \(U_p\) normalized plate velocity\(=V_p/V_{in}\)
- \(u'_i\) fluctuating velocity
- \(V_{in}\) inlet velocity
- \(V_p\) plate velocity
- \(x_i\) coordinate directions

**Greek symbols**

- \(\rho\) fluid density
- \(\mu\) dynamic viscosity of the fluid

**Subscripts and superscripts**

- \(\text{fl}\) fluctuating component of the flow variables
- \(i\) coordinate index
- \(p\) plate

**MATHEMATICAL ANALYSIS**

Figure 1 designates the physical model and computation domain for the impinging twin circular jets. The cuboid geometry consists of a confining adiabatic wall placed opposite to the constant heat flux moving inclined plate with oblique angle \(\theta\) and velocity \(V_p\). The twin-jets are located in the middle of the confining wall. The jet to jet centerline spacing \(L\) is eight times of the jet diameter \(D\), while the jet-to-plate distance \(H\) is eighteen times of the jet diameter \(D\). The dimensionless plate velocity \(U_p\) \((=V_p/V_{in})\) is normalized by the jet inlet velocity \(V_{in}\). When the dimensionless plate velocity \(U_p\) is positive, the plate moves along positive \(y\) (+\(y\)) direction; while as it is negative, the plate moves to the opposite (-\(y\)) direction.

![Figure 1. The physical model](attachment:image.png)

The fluid considered here is water, which is assumed to be incompressible with constant properties and the flow is 3-D steady and turbulent. The governing equations for continuity, momentum and energy equations are as follows:

**Continuity equation:**

\[
\frac{\partial \overline{u}}{\partial x} + \frac{\partial \overline{v}}{\partial y} + \frac{\partial \overline{w}}{\partial z} = 0 \tag{1}
\]

**Momentum equation:**

\[
\frac{\partial \overline{u}}{\partial x} + \frac{\partial \overline{v}}{\partial y} + \frac{\partial \overline{w}}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \frac{\mu}{\rho} \nabla^2 \overline{u} - \frac{\partial}{\partial y} \left( u' v' \right) + \frac{\partial}{\partial z} \left( u' w' \right) \tag{2}
\]

\[
\frac{\partial \overline{v}}{\partial x} + \frac{\partial \overline{v}}{\partial y} + \frac{\partial \overline{w}}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \frac{\mu}{\rho} \nabla^2 \overline{v} - \frac{\partial}{\partial x} \left( v' u' \right) + \frac{\partial}{\partial z} \left( v' w' \right) \tag{3}
\]

\[
\frac{\partial \overline{w}}{\partial x} + \frac{\partial \overline{v}}{\partial y} + \frac{\partial \overline{w}}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial z} + \frac{\mu}{\rho} \nabla^2 \overline{w} - \frac{\partial}{\partial x} \left( w' u' \right) + \frac{\partial}{\partial y} \left( w' v' \right) + \frac{\partial}{\partial z} \left( w' w' \right) \tag{4}
\]

**Energy equation:**
\[
\frac{\partial \overline{T}}{\partial x} + \overline{v} \frac{\partial \overline{T}}{\partial y} + \overline{w} \frac{\partial \overline{T}}{\partial z} = \frac{1}{\rho c_p} \nabla^2 T - \left[ \frac{\partial}{\partial x} (u'^2) + \frac{\partial}{\partial y} (v'^2) + \frac{\partial}{\partial z} (w'^2) \right]
\]  

(5)

where \( \overline{p}, \overline{T}, \) and \( \overline{u}, \overline{v}, \overline{w} \) are the mean pressure, temperature, and velocity components, respectively, \( T' \) and \( u', v', w' \) are the fluctuating temperature and velocity components, respectively and \( \rho, \mu \) and \( c_p \) are the fluid density, dynamic viscosity, and specific heat respectively.

The turbulent Reynolds stresses \( \overline{u'u'}, \overline{u'v'}, \overline{v'w'}, \overline{u'u'}, \overline{v'v'} \) and \( \overline{w'w'} \) in eq.(2), (3) and (4) are calculated by standard k-ε turbulence model [12] with enhanced wall treatment to resolve the wall bounded effects.

Because the governing equations are elliptical in spatial coordinates, the boundary conditions are required for all boundaries of the computation domain.

**NUMERICAL METHOD**

In this study, the governing equations are solved numerically using the commercial computer program code CFD-ACE+ [13]. This package is developed based on SIMPLEC algorithm and finite-volume method. The solution is considered converged when the normalized residual falls below \( 1 \times 10^{-4} \) for all variables. Prior to computation, a thorough verification of the grid-independence of the numerical solution is performed in order to ensure the accuracy and validity of the numerical results. Three different grid systems, 424,000, 508,800, 678,400 for the study were tested. It was found that the relative errors in the local Nusselt number is less than 1%. Computations were performed on a personal computer with Intel(R) Core(TM) i7-2600 3.4GHz CPU, and typical CPU times were about one hour for each case.

**RESULT AND DISCUSSION**

The predicted twin-jets velocity vectors and plate surface Nu distributions for three different oblique angle (\( \theta = 0^\circ, 15^\circ \) and \( 30^\circ \)), with \( \text{Re} = 6,500 \) and \( U_p = 0 \) are presented in Figure 2.

![Figure 2 Velocity Vector and Nu for different \( \theta \) at \( \text{Re}=6,500 \) and \( U_p = 0 \).](image-url)
One can see that the evolution of the fluid flow and thermal contour with increasing oblique angle $\theta$ is tangible. For the case of stationary horizontal plate, $\theta = 0^\circ$, as the twin-jets impinge onto the plate, two stagnation points are generated on the plate surface and the fountain region appears in the middle of the two stagnation points; As expectedly, for $\theta = 0^\circ$, the Nu distribution on the plate surface is symmetry. When the oblique angles $\theta$ is increased, the velocity field and the corresponding Nu contours are changed significantly and the fountain region shifts close to the jet 2 direction (left side). It is interesting to note that, for $\theta = 30^\circ$, the fountain region is almost disappeared.

Figure 3 illustrates the variation of the local Nusselt number along the centerline ($x=0$) from $R/D = -10$ to 10 for different values of $\theta$ ($\theta = 0^\circ$, $10^\circ$, $15^\circ$, $20^\circ$ and $30^\circ$) at $Re=6,500$, $Up=0$. It is observed that, as the oblique angle $\theta$ is increased from $0^\circ$ to $20^\circ$, the peak Nu value for the jet 1 is increased, while for jet 2, it is decreased. This is due to that fact that as $\theta$ is increased, the jet-to-plate distance $H$ is decreased for jet 1. This results in larger velocity near the plate surface and higher heat transfer rate on the plate surface; while for jet 2, as $\theta$ is increased, the jet-to-plate distance $H$ is increased and this results in lower velocity near the surface and thus lower Nu value.

Figure 4 shows the variation of the local Nusselt number along the centerline ($x=0$) from $R/D = -10$ to 10 for three different plate velocities ($Up = 0$, $0.1$, $0.2$) at $Re=6,500$ and $\theta = 0^\circ$. When the plate velocity $Up$ is positive, it means the plate moves along $+y$ direction. One can see that the peak Nu values for both jets 1 and 2 are increased as the plate velocity is increased.

Figures 5 - 7 present the local Nusselt number along the centerline ($x=0$) for three different values of $\theta = 10^\circ$, $20^\circ$ and $30^\circ$, respectively, at $Re=6,500$. The effects of different $Up$ ($Up = 0.2$, $0.1$, $0$, $-0.1$ and $-0.2$) are also shown in each figure. It is seen that the Nu is strongly dependent on the plate moving direction due to the relative velocity difference between the water jet flow and the moving plate. In addition, as the oblique angle $\theta$ is increased, the peak Nu values for jet 1 and jet 2 differ significantly.

To examine the effect of the plate velocity and plate oblique angle on the overall heat transfer rate, the average Nusselt number at the moving plate is calculated as shown in Figure 8. In the case of stationary plate ($Up = 0$), the average Nusselt number is decreased as the plate oblique angle $\theta$ is increased. For a positive plate velocity ($Up > 0$), the average Nusselt number is decreased with increasing oblique angle $\theta$, while for a negative plate velocity ($Up < 0$), the opposite trend is true.
CONCLUSIONS

In this study a three-dimensional simulation model of the confined turbulent twin circular jets impinging to an inclined moving plate was built. The effects of different normalized plate velocity, ranging from -0.2 to 0.2, and different plate oblique angles, ranging from 0° to 30°, are presented, investigated and compared. The major conclusions are summarized as follows:

1. The Nu is strongly dependent on the plate moving direction due to the relative velocity difference between the water jet flow and the moving plate. As the oblique angle \( \theta \) is increased, the peak Nu values for jet 1 and jet 2 differ significantly.

2. In the case of stationary plate \( (U_p = 0) \), as the plate oblique angle \( \theta \) is increased, the average Nusselt number is decreased.

3. For a positive normalized plate velocity \( (U_p) \), the average Nusselt number is decreased with increasing oblique angle \( \theta \), while for a negative \( U_p \), the opposite trend is true.
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REFERENCES