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Numerical investigation of Heat Transfer Characteristics of an Axi-symmetric turbulent Impinging Jet on a Flat Plate

R. Suresh Kumar[1], R. Velraj[2] & S. Suyambazhahan [3],*
[1] Jaya Engineering College, Thiruninravur, Chennai – 602 024, Tamil Nadu, India.
sureshrsc@yahoo.com
[2] College of Engineering, Anna University, Guindy, Chennai – 600 025, Tamil Nadu, India.
[3] S.A Engineering College, Poonamalle-Avadi Road, Chennai- 600077, Tamil Nadu, India.

Jet impingement is one of the oldest and most attractive techniques for convective process intensification where convective heating, cooling or drying is applied

The present study highlights the numerical study of heat transfer characteristics of Axi symmetric incompressible Impinging Jet flow. A computational study of the impingement of a thermally turbulent jet on a solid plate has been reported. For a fixed nozzle–plate distance, the influence of the Reynolds number on the stagnation point heat transfer was investigated. The influence of the nozzle–plate distance on the stagnation point Nusselt number has also been discussed.

The possibility of improving the heat transfer is carried out according to the characteristic parameters of the interaction jet and wall.

At higher Reynolds numbers and even for initially laminar jets – the turbulence generated by the jet itself plays an important role in determining the heat-transfer characteristics of impinging jets.

The effect of local mean velocities and turbulence intensities on the heat transfer has been outlined, whereas the objectives of this part is to explore in more detail the influence of the turbulence characteristics of the flow on heat transfer.

In a jet flow, vortices initiate in the shear layer due to Kelvin Helmholtz instabilities. Vortices, depending on their size and strength, affect the heat spread, the potential core length and the entrainment of ambient fluid.

A very important parameter in order to quantify this process is the heat transfer coefficient

The enhancement and reduction of the local heat transfer were related to changes in the flow structure which an impinging jet was forced at different frequencies. It is important, therefore, to understand the unsteady heat transfer characteristics associated with the coherent flow structure.

Numerical Method

The following assumptions are made;

1. The fluid is incompressible with constant properties
2. The flow is Axi symmetric
3. Constant Temperature condition is maintained in the impinging plate and no-slip is for velocity.

The Continuity equation, Momentum and Energy equations are obtained using Reynolds averaging

$$\overline{\Phi} = \Phi + \Phi'' \quad \text{and} \quad \overline{\Phi}' = 0$$

Continuity equation:

$$\frac{\delta(\rho u)}{\delta x} + \frac{\delta(\rho v)}{\delta y} = 0$$

Momentum equation :

$$u\frac{\delta u}{\delta x} + v\frac{\delta u}{\delta y} = -\frac{1}{\rho}\frac{\delta p}{\delta x} + v\left(\frac{\delta^2 u}{\delta x^2} + \frac{\delta^2 u}{\delta y^2}\right) + X$$

$$u\frac{\delta v}{\delta x} + v\frac{\delta v}{\delta y} = -\frac{1}{\rho}\frac{\delta p}{\delta y} + v\left(\frac{\delta^2 v}{\delta x^2} + \frac{\delta^2 v}{\delta y^2}\right) + Y$$

$X = g_x$ and $Y = g_y$

Energy Equation

$$\frac{\delta}{\delta x}\left[\rho u\left\{c_p T + \left(\frac{u^2 + v^2}{2}\right)\right\}\right] +$$

$$\frac{\delta}{\delta y}\left[\rho v\left\{c_p T + \left(\frac{u^2 + v^2}{2}\right)\right\}\right] = k\left(\frac{\delta^2 T}{\delta x^2} + \frac{\delta^2 T}{\delta y^2}\right) +$$

$$\left(\rho u^2\frac{\delta u}{\delta x} + \rho uv\frac{\delta u}{\delta y} + \rho uv\frac{\delta v}{\delta x} + \rho v^2\frac{\delta v}{\delta y}\right) +$$

$$\left(\sigma_x\frac{\delta u}{\delta x} + \tau_{yx}\frac{\delta u}{\delta y} + \sigma y\frac{\delta v}{\delta x} + \tau_{xy}\frac{\delta v}{\delta y}\right) +$$

$$\beta = \frac{-1}{\rho}\left(\frac{\delta p}{\delta T}\right)_p \approx \frac{-1}{\rho}\frac{\Delta p}{\Delta T} = \frac{1}{\rho}\frac{\rho_\infty - \rho}{T_\infty - T} \Rightarrow$$

$$(\rho_\infty - \rho) \approx \rho * \beta\,(T - T_\infty)$$

Boundary conditions

Boundary conditions are specified on the surfaces of the computational domain as shown in Figure 1.

The following are wall boundary conditions

1. No slip boundary conditions for Wall
2. Turbulent Kinetic energy is assumed as zero
3. The temperature at impinging plate is maintained at 313 K

The inlet boundary conditions are described by

$U_{in} = U_0$

$V = 0$

$T = T_0$

$k_{in} = 0.005\, U_0^2$

$\varepsilon_{in} = k_{in}^{3/2} / 0.3D/2$

At higher Reynolds numbers, the jet flow becomes three-dimensional and turbulent before impinging on the wall. The primary vortices emanating from the jet shear layer are clearly seen, which is the characteristic of unsteady jet flow.

The interaction of the primary vortices with the wall shear layer gives rise to unsteady vertical motions.

The axi symmetric viscous and potential-flow simulations include the nozzle in the solution domain, allowing nozzle-wall effects.

Results and discussion

Figures 2 and 3 illustrates the variation of Nusselt Number with respect to Reynolds Number for H/d=6. It is observed that the increase in Reynolds number leads to an increment of Nusselt number. Also the coefficient of heat transfer would be maximum in the stagnation point, then decreases quickly when the radial distance increases. The presence of a maximum of transfer at the stagnation point is due to the existence of a quasi-uniform axial velocity profile in this zone.

For a Reynolds number equal to 23000, the influence of H/d on the average heat transfer was studied (Figure 4). Beyond H/d<10 value, the average Nusselt number decreases quickly when H/d increases. However, the heat transfer presents a maximum for H/d = 6. For high values of

H/d, the velocity profiles spread out as the jet progress, which causes the decrease of the normal velocity and the heat transfer as well. Figure 5 shows the variation of local Nu for Re=23000 for various H/d. It illustrates that increase in H/d leads to decrease in Nusselt Number.

The increase in Reynolds number leads to an increase of the heat transfer (Figure 6). This evolution of heat transfer coefficient is completely expected because Nu is determined from the mean temperature gradient, while the mean temperature field is governed by a convection-diffusion equation. The higher the velocity, the more important convection is, compared to the diffusion. The logarithmic variation of the local Nusselt number at the stagnation point with Reynolds number was reported in the Figure 7. The range of Reynolds number was varied from 5 x10^3 to 2 x10^5 and H/d=6. Nu_0 is correlated with the Reynolds number as:

$$Nu_0 = 0.743\ Re_d^{0.523}$$

The results show that when R/d is lower than 1.2 (stagnation point), the average Nusselt number is practically independent of the radius of the surface of transfer. For R/d>1.2 (Wall jet region), the average heat transfer coefficient decreases quickly when R/d increases. The evolution of Nu with R is coherent with the spatial distribution of the local heat transfer coefficients.

The temporal nature of both the fluid flow and thermal oscillation would be investigated for various Reynolds numbers ranging from 23000 to 70000 and for stand- off ratios from 0.5 to 20 in continuation of this study.

The symbols r (R), d (D), H and X are the geometrical parameters.

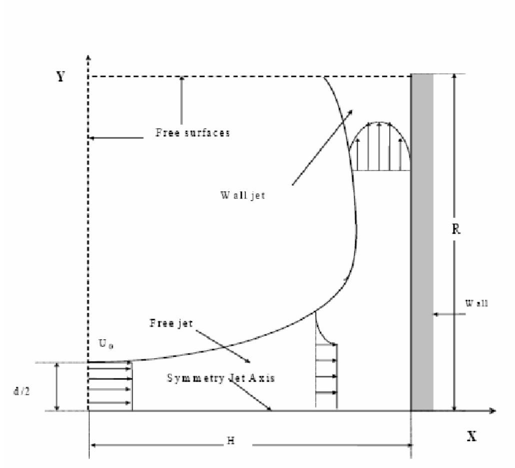

Figure 1: Geometry and Computational domain

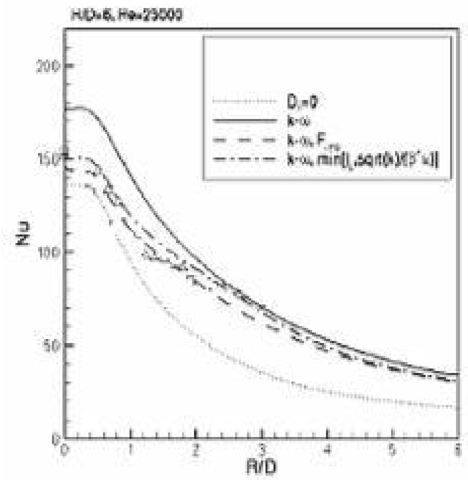

Figure 2: Nusselt number distribution for H/d = 6 and Re = 23000

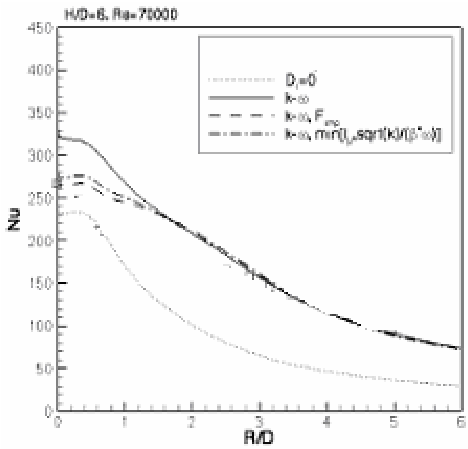

Figure 3: Nusselt number distribution for H/d = 6 and Re = 70000**.**

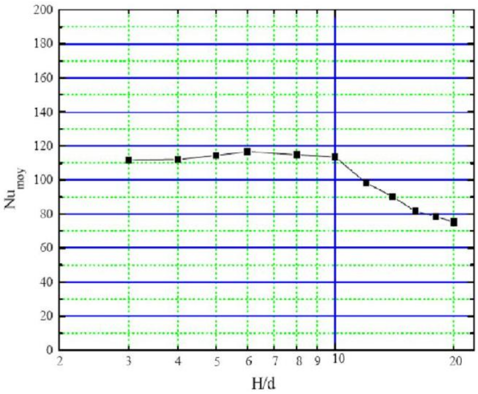

Figure 4 Average Nusselt number Profiles for various H/d on the flat plate

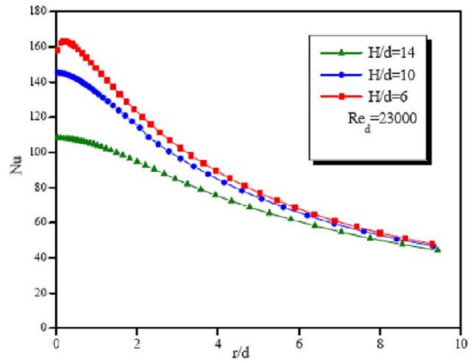

Figure 5 Local Nusselt number Profiles for Re = 23000 on the flat plate for various H/d

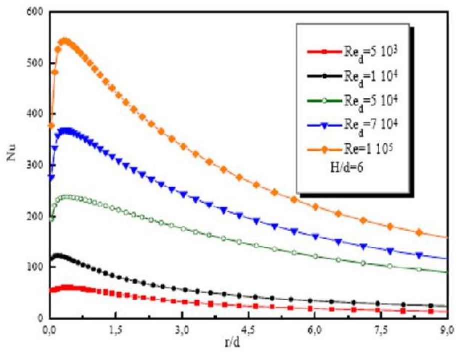

Figure 6: Local Nusselt number Profiles for different Reynolds numbers

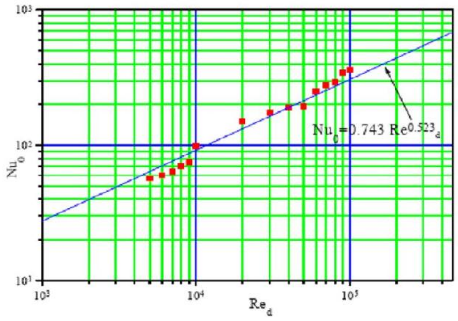

Figure 7 Stagnation point Nusselt number Profiles for different Reynolds numbers

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