Numerical Simulation of Heat Transfer in an Axisymmetric Turbulent Jet Impinging on a Flat Plate

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Abstract-*A computational study of the impingement of a thermally turbulent jet on a solid plate, using k - ε model, is reported. The possibility of improving the heat transfer is carried out according to the characteristic parameters of the interaction jet-wall. The close zone solid wall required a particular treatment using an economic method known as "wall functions". The numerical resolution of the equations is carried out using the finite volume method. For a fixed nozzle–plate distance, the influence of the Reynolds number on the stagnation point heat transfer is investigated. Good agreement with experimental results is observed. The influence of the nozzle–plate distance on the stagnation point Nusselt number is also discussed.*

Keywords: *impinging jets; axisymmetric; two-dimensional; k - ε model; velocity field, heat transfer*

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1. INTRODUCTION

In the last two decades, jet impingement heat transfer has received considerable attention in many industrial and engineering applications (e.g. manufacturing, material processing and electronic cooling, drying paper, textiles and tempering of glass.). This has been due to the high heat transfer rates of jet impingement. A turbulent jet impinging normally on a flat plate is a very effective means to promote high rates of heat exchange. A number of comprehensive reviews of jet impingement are available in the literature (Gaunter et al.(1970), Jones et al. (1972), Launder et al. (1974) Cooper et al. (1993), Amano et al. (1984)). In most applications, a turbulent jet of gas or liquid is directed to the target area.

Wolfshtein (1970) has simulated numerically the plane jet by using the one-equation turbulence model. The results obtained reveal a good agreement with the experiment for the calculation of radial velocity, pressure and skin friction. However, Bower et al. (1977) have shown that the use of this model to the round jet gave unsatisfactory results for short distances between nozzle and plate. According to theses results, researchers interested to the two equation models (k-ε) or low Reynolds models (Yap, (1987), Craft et al.(1993)). Kunugi et al. (1993) carried out a numerical and experimental study of a confined impinging jet and successfully used an anisotropic k-ε model.

The present paper reports computations of the flow and thermal fields for an axisymmetric isothermal fully developed turbulent jet, perpendicular to a uniform heated flat plate. The k - ε model (model with two equations) was used in the calculations. The axisymmetric, incompressible, Reynolds averaged Navier-Stokes equations were solved.

2. MATHEMATICAL FORMULATION

Governing equations

The geometry is depicted in Figure. 1. The turbulent air flow at the nozzle exit, with diameter d, velocity U_0 and temperature T_0, is fully developed. The nozzle is at a distance H from the flat plate maintained at the constant temperature T_w ($T_w > T_0$).

The problem is governed by physical, hydrodynamic and geometrical parameters of the system jet/surface. The principal parameters are the mean velocity U_0 of the jet, dimensions of the jet (the diameter d of the nozzle for a circular jet), the distance H between the nozzle and the surface of impact, generally expressed in the dimensionless form H/d and dimensions of the surface of impact. The geometry and co-ordinate system are shown in figure 1.

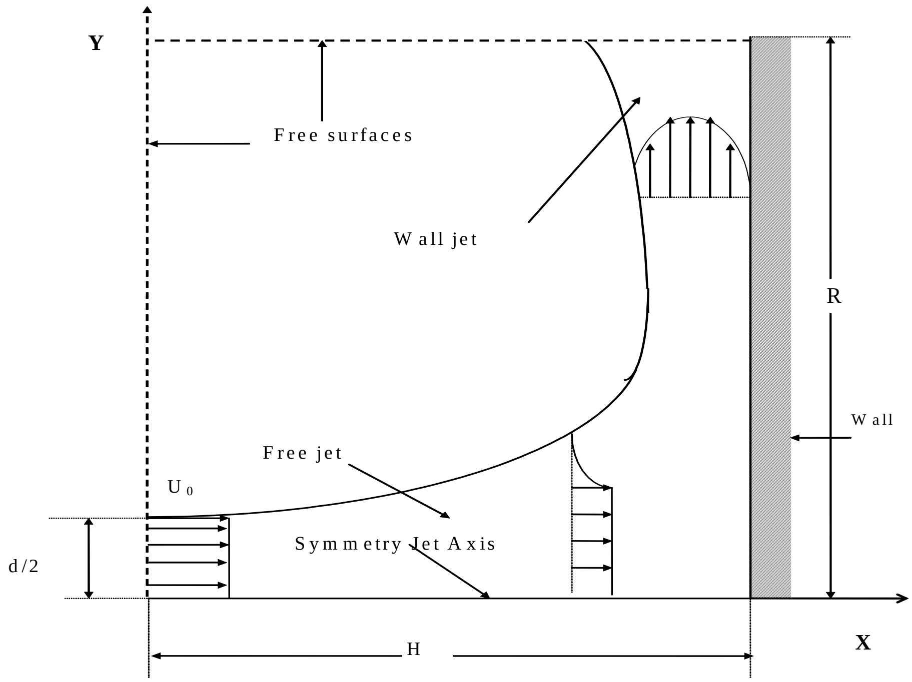

Figure 1: Geometry and computational domain.

The following assumptions are adopted:

- The flow is axisymmetric and incompressible with constant physical properties
- The viscous dissipation is neglected
- A constant temperature condition is applied to the impinge plate (no-slip is considered for velocity)

The equations for the conservation of mass, momentum and energy are obtained by decomposing the primitive variables into mean and fluctuating components using the conventional Reynolds averaging:

$$\overline{\Phi} = \Phi + \Phi'' \text{ and } \overline{\Phi'} = 0 \tag{1}$$

with $\overline{\Phi}$ is the mean value of the dependent variable Φ and $\overline{\Phi'}$ is the fluctuation.

For later convenience and dropping the over bar on the mean variables, the Reynolds-averaged equations can be written in the following generic transport equation form:

$$\frac{1}{r}\left[\frac{\partial}{\partial x}(r\rho U\Phi) + \frac{\partial}{\partial r}(r\rho V\Phi)\right] = \frac{1}{r}\left[\frac{\partial}{\partial x}(r\Gamma_\Phi \frac{\partial \Phi}{\partial x}) + \frac{\partial}{\partial r}(r\Gamma_\Phi)\frac{\partial \Phi}{\partial r}\right] + S_\Phi \tag{2}$$

with the scalar variable Φ, the diffusion coefficient Γ_Φ and source term S_Φ in the respective governing equation are given in table 1.

Table 1: Diffusion coefficients and source terms in the generic transport equation 1

Equation	Φ	Γ_Φ	S_Φ
Continuity	1	0	0
x-Momentum	U	μ_{eff}	$-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}(\mu_{eff}\frac{\partial U}{\partial x})+\frac{1}{r}\frac{\partial}{\partial r}(r\mu_{eff}\frac{\partial V}{\partial x})$
r-Momentum	V	μ_{eff}	$-\frac{\partial p}{\partial r}+\frac{\partial}{\partial x}(\mu_{eff}\frac{\partial U}{\partial r})+\frac{1}{r}\frac{\partial}{\partial r}(r\mu_{eff}\frac{\partial V}{\partial r})-2\mu_{eff}\frac{V}{r^2}$
Energy	T	$\frac{\mu}{Pr}+\frac{\mu_r}{\sigma_T}$	0
Kinetic energy	k	$\mu+\frac{\mu_t}{\sigma_k}$	$G-\rho\varepsilon$
Dissipation energy ε	ε	$\mu+\frac{\mu_t}{\sigma_\varepsilon}$	$C_{\varepsilon 1}\frac{\varepsilon}{k}G-C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k}$

where $\mu_{eff}=\mu+\mu_t$, $\mu_t=C_\mu\rho\frac{k^2}{\varepsilon}$, $G=\mu_t\left[(\frac{\partial U}{\partial r}+\frac{\partial V}{\partial x})^2+2(\frac{\partial U}{\partial x})^2+2(\frac{\partial V}{\partial r})^2+2(\frac{V}{r})^2\right]$

the constants of the model are:

C_μ=0.09, $C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$=1.92, σ_k=1, σ_ε=1.21, σ_T=1

Boundary conditions:

Boundary conditions need to be specified on all surfaces of the computational domain. Boundaries Presented in this study include inflow (Inlet), solid wall and axis of symmetry as shown in Figure 1.

Solid wall boundary conditions: For the present model, along the solid wall, the non-slip boundary condition for velocities, zero value for turbulent kinetic energy, and zero gradient for energy dissipation rate were used. Temperature at the plate is prescribed as 313 K.

Inlet boundary conditions: the velocity-inlet boundary conditions imposed at the nozzle are described by:

$$\left.\begin{array}{l} U_{in} = U_0 \\ V = 0 \\ k_{in} = 0.005\,U_0^2 \\ \varepsilon_{in} = k_{in}^{3/2} \big/ 0.3\,d/2 \\ T = T_0 \end{array}\right\}$$

Axis of symmetry: the radial velocity component V, and the gradient of the other dependent variables (U,ε, k, T) were equal to zero.

3. NUMERICAL PROCEDURE AND COMPUTATIONAL METHODOLOGY

The governing differential transport equations were converted to algebraic equations before being solved numerically. The finite volume scheme which involves integrating the governing equations about each control volume yielding discrete equations that conserve each quantity on a control volume basis, was applied to equation (2).The governing equations were discretised using the second-order upwind scheme to achieve the best accuracy. A non-staggered variable arrangement was adopted with the momentum interpolation technique to avoid the pressure-velocity decoupling. The coupling between pressure and velocity was achieved by the SIMPLER algorithm (Patankar (1980)). Non-uniform, orthogonal and axisymmetric grid was used with a high resolution near solid boundary.

4. COMPUTATIONAL RESULTS

The reduced axial velocity prediction was compared to experimental values of Tani and Komatsu (1966) obtained for the same conditions, Figure 2. Good agreement with the experiment is obtained.

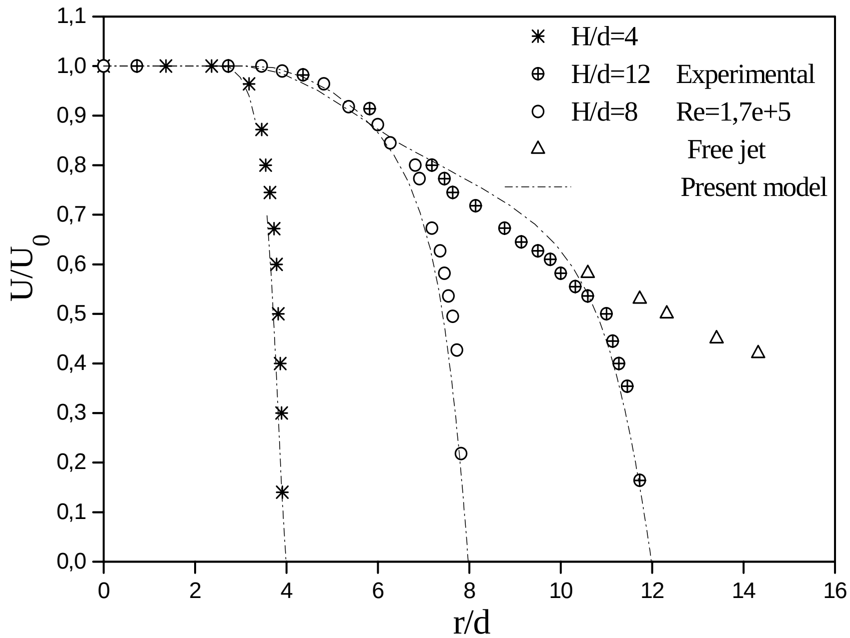

Figure 2: Reduced axial velocity profile along the jet axis.

4.1 Influence of nozzle to wall distance

In figure 3, the profiles of local wall Nusselt number are shown for three values of nozzle to plate distance (H/d =6, 10 and 14). For the three values of H/d, one notes that the coefficient of heat transfer presents a maximum in the vicinity of the stagnation point, then decreases quickly when the radial distance increases. The distributions of the Nusselt number show that the local heat transfer is attenuated when H/d is higher than six especially in the area surrounding the stagnation point, and the effect of H/d on the heat transfer coefficient decreases when r/d 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 as shown in figure 2.

The influence of reduced height of the jet, expressed by H/d, on the average heat transfer was studied for a Reynolds number equal to 23000. Figure (4) shows a quasi-constancy of the coefficient for H/d<10. Beyond this value, the average Nusselt number decreases quickly when H/d increases. However, the heat transfer presents a maximum for H/d = 6. One can say that the reduced height of the jet has a little influence on the average heat transfer for H/d less than 10, which corresponds to the zone of the potential cone. 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.

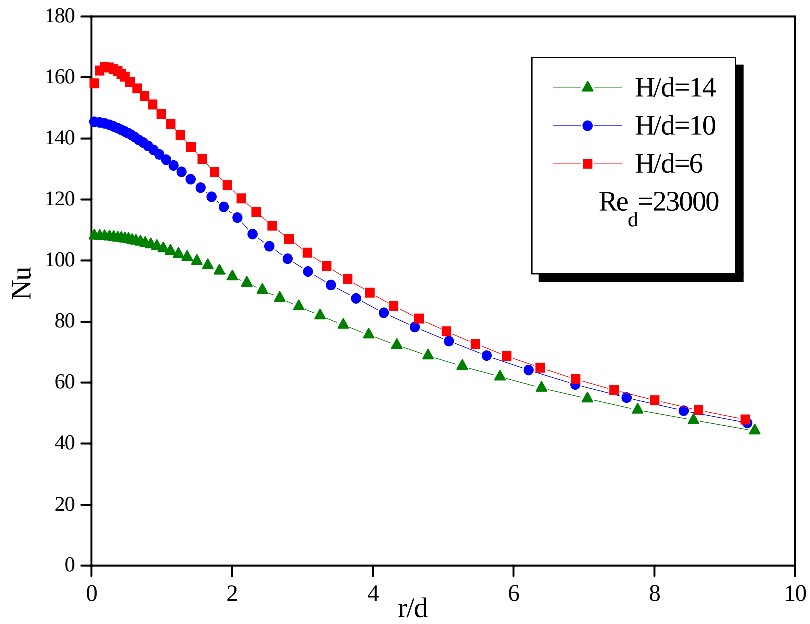

Figure 3: Local Nusselt number profiles at the flat plate.

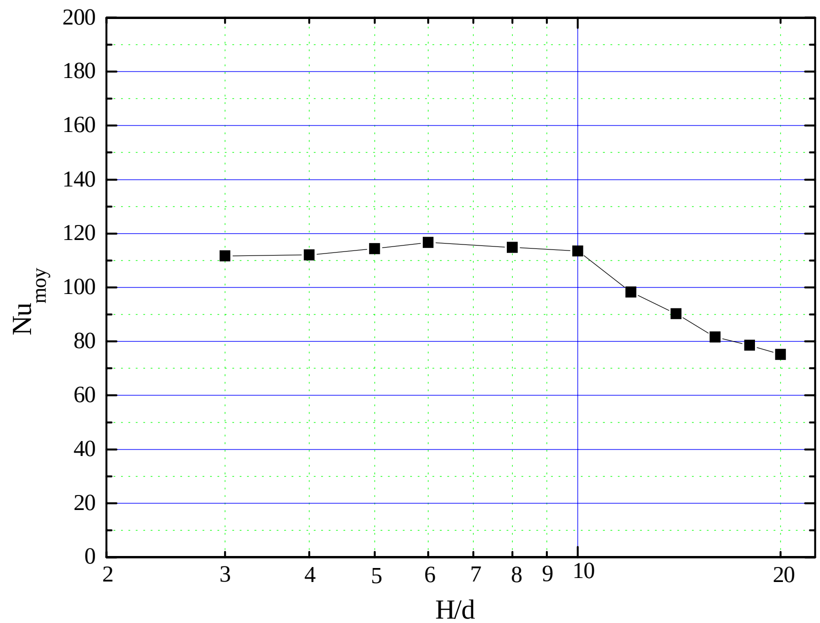

Figure 4: Average Nusselt number profiles vs. nozzle-to-plate distance H/d.

4.2 Influence of Reynolds number

Turbulence models are sometimes fitted to a given test-case, at a given Reynolds number and might give much worse results when flow conditions are changed. The influence of the Reynolds number Re_d is illustrated in figures (5), (6) and (7). This influence is strongly related to the dimensions of the transfer surface and its position compared to the different positions of the jet. Figure (5) shows the evolution of the local heat transfer coefficient along the wall, for various values of Reynolds number. The analysis of this figure shows that the increase in the Reynolds number leads to an increase of the heat transfer. This evolution is completely expected because Nu is determined from the mean temperature gradient, while the mean temperature field is governed by a convection–diffusion equation (2). 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 is reported in figure (6). The range of Reynolds number was varied from 5 10^3 to 2 10^5 and H/d=6. Nu_0 is correlated with the Reynolds number as:

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

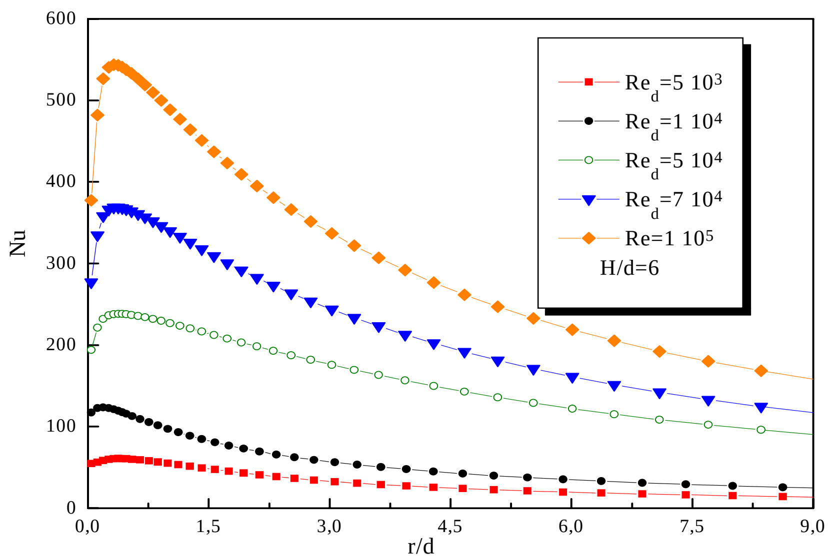

Figure 5: Local Nusselt numbers vs. Z/d at different Reynolds numbers.

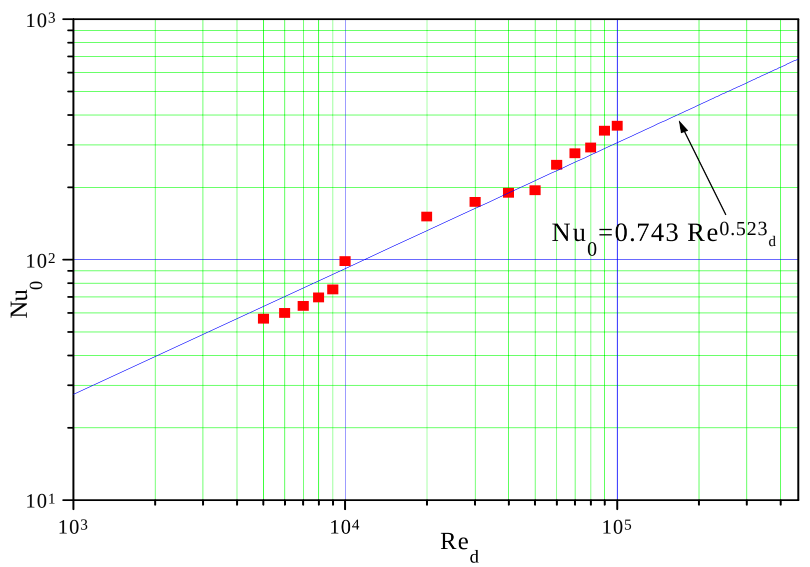

Figure 6: The stagnation point Nusselt number vs. Reynolds number.

The average heat transfer coefficients $\overline{Nu}$ can be obtained by dividing the average heat flux around the heated surface by the mean temperature between the wall and the ambient. The evolution of the average heat transfer coefficient ($\overline{Nu}$) vs. Reynolds number is illustrated in figure 7. From this figure it can be deduced that the evolution of average Nusselt number can be correlated for R/d>1.5 as:

$$\overline{Nu} = \alpha \, Re_d^{\beta}$$

The superscript β is 0.725 and the coefficient α is ranged between 0.08 and 0.116 according to the dimensions of the impact surface.

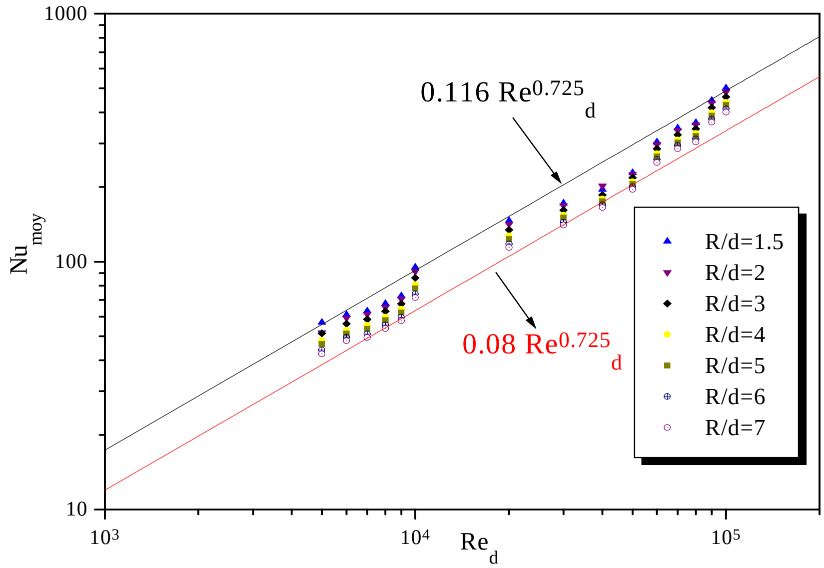

Figure 7: Average Nusselt number vs. Reynolds number.

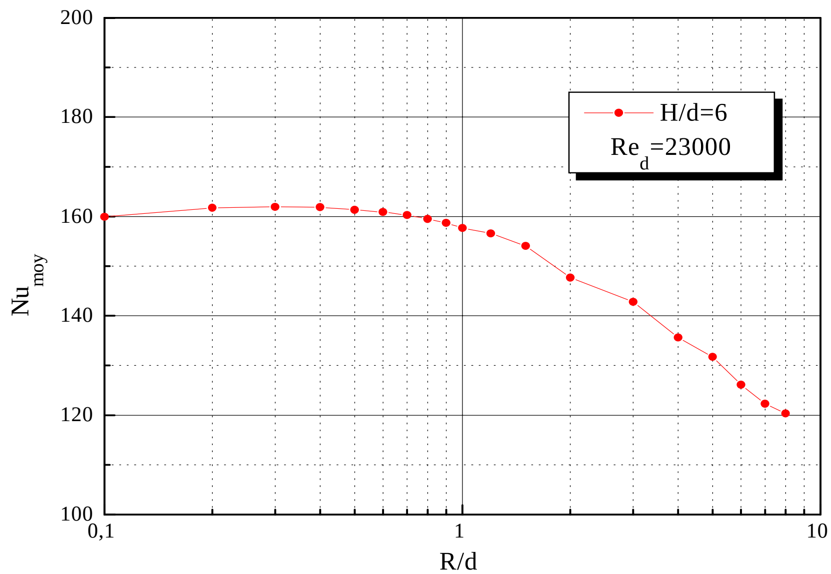

Figure 8: Effect of the wall dimension on average Nusselt number.

4.3 Influence of wall dimensions

The analysis of the influence of wall surface dimensions on the heat transfer was carried out in a systematic way, either by varying the radius R of circular surfaces (average transfer), or by the calculation of the transfer coefficient at different points (case of the local transfer).

Figure 8 shows the evolution of average Nusselt number with the radius of the surface R/d. This figure is referred to the case of a nozzle-to-plate distance of 6 jet diameters (H/d=6) and Re=23000. 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 presented previously.

5. CONCLUSION

The k - ε turbulence model has been used as the basis of predictions of the flow that results from the orthogonal impingement of circular and two-dimensional (2-D) jets on a flat surface. The effect of the physical parameter (Reynolds number) and the geometrical parameter (wall dimension and nozzle-wall distance) on heat transfer have been analysed. It appears from the results that the use of the single jet with the aim of obtaining good heat transfer coefficients is restricted on small surfaces of impact corresponding to the area of the stagnation point and for distances nozzle--wall about the length of the potential cone.

Nomenclature

C_p	specific heat	T	temperature
d	diameter of the jet	T_w	wall temperature
G	production	T_0	inlet temperature
H	nozzle-to-plate distance	U (V)	velocity component
h	wall heat transfer coefficient	U_0	jet inlet velocity
k	thermal conductivity	Γ_Φ	diffusion coefficient
k	turbulent kinetic energy		
Nu	Nusselt number, Nu=hd/k	**Greek symbols**	
P	pressure	ρ	density
Pr	Prandtl number, $Pr=\mu C_p/k$	**ε:**	dissipation rate of turbulence
Pr_t	turbulent Prandtl number	υ	fluid kinematic viscosity.
Re	Reynolds number, Re=dυ/D	υ_t	turbulent kinematic viscosity.
r (x)	radial (axial) radial distance		
S_Φ	source term		

REFERENCES

Gaunter J. W., Livingood J. N. B. and Hrycak P., (1970) Survey of literature on flow characteristic of a single turbulent jet impinging on a flat plate, NASA TN D-5652

Jones W. P. and Launder B. E.,(1972) The prediction of laminarization with a two equations model of turbulence, Int. J. Heat Mass Transfer, Vol. 15, pp. 301.

Launder B.E., Spalding D.B., (1974) The numerical of computation of turbulent flows, Computer Methods in applied Mechanics and Engineering, Vol. 3, pp. 269, 1974.

Cooper D., Jackson D.C., Launder B.E. and Liao G.X., (1993) Impinging jet studies for turbulence model assessment: Flow-field experiments, Int. J. Heat Mass Transfer, Vol. 36, pp. 2675-2684

Amano R. S., Brandt H., (1984) Numerical study of turbulent axisymmetric jets impinging on a plate and flowing into an axisymmetric cavity, Journal of Fluid Engineering, Vol. 106, pp. 410-417

Wolfshtein M., (1970) Some solutions of the plane turbulent impinging jet, ASME Journal of Basic Engineering, pp. 915-922

Bower W.W., Kotanshy D.R. ad Hoffman H.,(1977) Computations and measurements of tow-dimensional turbulent jet impinging flow fields, Proceedings of Symposium on Turbulent Shear Flows , Pen State University, Vol. 3(1)

Yap C.R., (1987) University of Manchester, PHD Thesis, Institute of Science and Technology.

Craft T.J., Graham L.J.W., Lander B.E., (1993) Impinging jet studies for turbulence model assessment: An examination of the performance of four turbulence models, Int. J. Heat Mass Transfer, Vol. 36, pp. 2685-2697

Kunugi, T., Yokomine, T. and Ichimiya, K.,(1993) Numerical and experimental study on heat transfer of an impinging turbulent plane jet with confined wall, ASME HTD, Vol. 246, pp. 25-31

Patankar, SV., (1980) Numerical Heat Transfer and Fluid Flow, Hemisphere, Washington DC, USA

Tani I. and Komatsu Y., (1966) Impingement of a round jet on a flat surface, The International Congress of Applied Mechanics Munich, Springer, Berlin, pp. 672-676