Steady State Jet Impingement Heat Transfer from Axisymmetric Flat Plate

Arpit Dwivedi
Department of Mechanical Engineering
Indian Institute of Technology, Bombay
Mumbai, India
Email:200100032@iitb.ac.in

***Abstract*—The report presents the Numerical computations performed in ANSYS Fluent(3D) over a cylindrical jet impinged over a circular plate to analyze the cooling effect of jet impingement. The jet flow is in the range of complete turbulent flow with a Reynolds number of 23,000 and the nozzle-to-plate spacing(z=H) ratio to the nozzle diameter(D) is 2. The experimental data by Fenot et al[1] and Numerical Data of ANSYS(2D-Axisymmetric) by Kannan B T and Senthilkumar Sundararaj[2] have been used for validating the Nusselt number obtained from computations. Two different cases are considered for the Nusselt number calculation one constant wall temperature and another uniform wall heat flux. The Nusselt number is calculated from two approaches one directly provided by ANSYS solutions and another by calculating the temperature gradient near the wall and substituting it in the basic definition of Heat flux to get the heat transfer coefficient and then the Nusselt Number. The Turbulence Intensity has been analyzed in the radial direction at different heights. i.e..(z/D) also along the direction normal to plate at different radial points. i.e..(r/D) in order to figure out the location where flow transitioned from laminar to turbulent. The paper also discusses why the use of $k-\omega$ SST over $k-\epsilon$ STD, $k-\epsilon$ RNG and Transition-SST is better for turbulence modelling.**

Keywords:Jet impingement, Heat Transfer, Nusselt number, Constant Wall flux, Constant Wall temperature, Temperature gradient, Laminar and Turbulent flows.

NOMENCLATURE

ϵ	Turbulent Kinetic Energy dissipation rate
μ	Molecular Viscosity
μ_t	Eddy Viscosity
ω	Specific turbulent Kinetic Energy dissipation rate
ρ	Density
D	Nozzle Diameter
H	Nozzle-to-plate spacing
h	local convective heat transfer coefficient
K	Turbulent Kinetic Energy
k	Thermal conductivity
Nu	Nusselt Number
q	Heat Flux
r	Radial distance from centre of plate
Re	Reynolds Number
T_j	Jet Temperature
T_w	Wall Temperature
$wall$	Wall or Plate used interchangeably
z	Distance from plate

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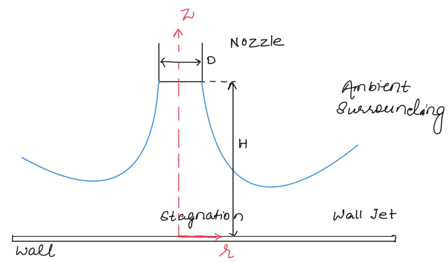

Fig. 1: Configuration of Impinging Jet

I. INTRODUCTION

Impinging jets on walls have very high heat transfer performances hence these configurations are widely used in the industrial realm where a high-performance heat transfer configuration is needed. The configuration can be varied a lot by varying geometrical parameters such as H/D, number of jets, the geometrical arrangement of jets, plate orientation with respect to jet, using grooved plates, and many more. The efficiency and feasibility are checked before using the configuration in the desired situation in Industries. The application of jet impingement is broadly separated into two categories: heating/cooling configurations either at a constant wall temperature (quench cooling) or at a uniform wall heat flux (electronic components cooling).
The current paper discusses one of the basic and simple configurations[Fig.1]. The H/D value for the geometry chosen is 2 with H = 4 cm and D = 2 cm with the Reynolds number 23,000 (completely in the range of turbulent flow). The paper has computational results of both categories: constant wall temperature and uniform wall flux. The results of $k-\omega$ SST turbulence model was used for comparison of the two cases.

A. *Uniform wall heat flux*

Under uniform wall heat flux(q) condition radial distribution of temperature($T(r)$) for fluid just above the wall is obtained from ANSYS computational results. Now as

$$q = h(T(r) - T_j) \tag{1}$$

$$h(r) = \frac{q}{T(r) - T_j} \tag{2}$$

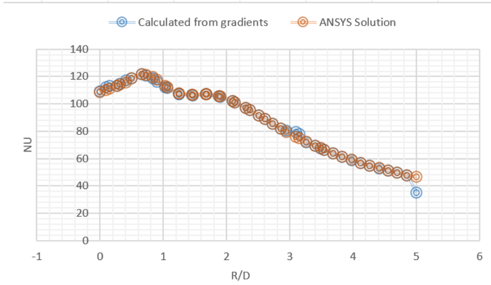

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Fig. 2: Gradient Calculation vs Solution provided by ANSYS

The local convective heat transfer coefficient($h(r)$) is calculated from the Temperature field($T(r)$).Now as

$$Nu = \frac{h.D}{k} \tag{3}$$

Therefore, local Nusselt number can be calculated from:

$$Nu(r) = \frac{h(r).D}{k} \tag{4}$$

The Surface Nusselt number is directly provided by ANSYS computational results but we can verify it using above rigorous definitions and formulae[Fig.2].

B. *Constant wall temperature*

Under constant wall temperature(T_w) condition, radial distribution of temperature gradient, normal to the wall is calculated and since

$$q = -k\frac{\partial T}{\partial z} \tag{5}$$

the term $\frac{\partial T}{\partial z}$ is approximated to $\frac{\Delta T}{\Delta z}$ where ΔT is calculated by obtaining the difference in Temperature at two nodes separated by Δz normal to the wall. Then again obtaining the radial distribution of temperature($T(r)$) for fluid just above the wall and using (1), (2) and (4) local Nusselt number is obtained.
The Surface Nusselt number is directly provided by ANSYS computational results but we can verify it using above rigorous definitions and formulae.

II. Setup in ANSYS

A. *Geometry*

The geometrical setup had a cylindrical pipe of diameter(D=2 cm) and length(l_1 = 4 cm) inside which the fluid flows(Re=23000) in negative z direction. The nozzle is at a distance of 4 cm above the circular plate whose diameter has been chosen to be 20 cm[See Fig.3].

B. *Mesh*

Three types of mesh[Fig.4] with different inflation layers near the wall, element size of wall face and number of divisions in the edge of bottom wall face have been used for computation. The following table contains the information regarding the Grids. The average orthogonal quality of G1, G2 and G3 are 0.99216, 0.99317 and 0.99265 respectively.

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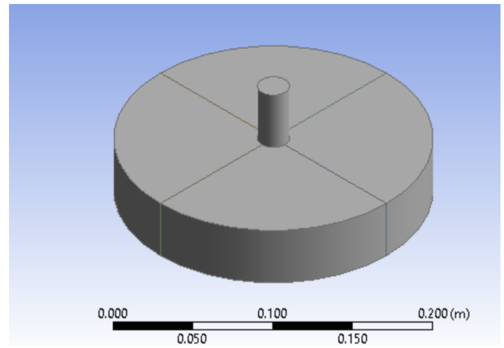

Fig. 3: Geometrical Setup

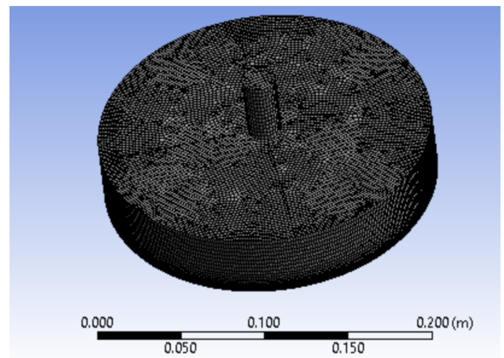

Fig. 4: Mesh

The smoothness level of each mesh has been set to medium and the Multi-Zone Hexa/Prism mapped mesh type is used for computations.

- *Inflation Layers:* Inflation layers are used for capturing the variations of different functions near wall as the gradients are large in viscous sub-layer. The layers are used with growth ratio 1.2. For $k-\omega$ SST we want $y^+ \approx 1$ therefore N(Number of Layers) ≥ 20, so that total height of layers $y_T \geq \delta_{99}$ (Boundary Layer Thickness). I will recommend not to use greater number of inflation layers as it will reduce the size of element closest to wall reducing the y^+ value to order of 10^{-2} which is not recommended for $k-\omega$ SST model. While meshing we need to refine mesh near wall so that we can increase the number of nodes there for results with better accuracy. In the three different type of grids used, the parameters were maneuvered to increase the number of nodes near wall and keeping y^+ nearly 1.

TABLE I: Grid Info

Grid_ID	**Inflation Layers**	**Face Sizing(m)/Edge Division**	**Nodes**
G1	25	0.003	131388
G2	30	0.002/300	270855
G3	35	0.002/400	403056

TABLE II: Convergence criteria

Residuals	Criteria	Residuals	Criteria
continuity	1e^{-3}	k	1e^{-3}
x-velocity	1e^{-3}	omega	1e^{-3}
y-velocity	1e^{-3}	Energy	1e^{-6}

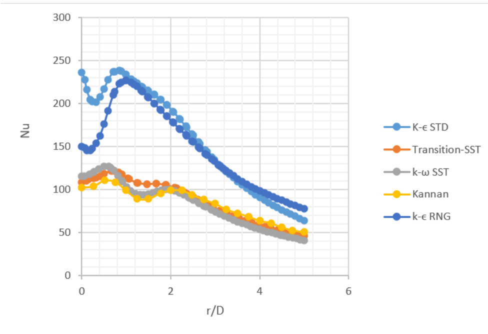

Fig. 5: Nusselt Plot for different models used

III. Computational Results

A. *Results of Different Models*

The computations were done using steady-state, Pressure based(in-compressible fluid) and absolute velocity formulation type solver.The model used were $k-\omega$ SST, $k-\epsilon$ STD, $k-\epsilon$ RNG and Transition-SST. The Nusselt plot obtained from all these when compared with the Nusselt plot of Kannan B T and Senthilkumar Sundararaj[2][Fig.5] depicted that $k-\omega$ is better for modeling the turbulent flows. The $k-\epsilon$ is not accurate in predicting boundary layer with adverse pressure gradient and therefore the results from it are deviating from the experimental values significantly. The convergence criteria used was show in table.

In above computations, uniform wall flux($q = 6000W/m^2$) condition was used. The Jet temperature was taken to be 300K and Reynolds number of 23,000. The SIMPLE (Semi-Implied Method for Pressure linked equations) algorithm was chosen as it gives better results for Pressure based solver for in-compressible fluid flow problems.

B. *Grid Independence Test*

The computations were performed for three different meshes. i.e.G1, G2 and G3 [Fig.6]. There was very little variation in Nusselt numbers, about 0.1% to 6.8%, between G2 and G3. Therefore, changing the grid was not recommended as it was not going to change the results significantly. The Nusselt plot of final model SST $k-\omega$ with Grid 3 were compared and plotted with the Experimental data by Fenot et al[1] and Numerical Data of ANSYS(2D-Axisymmetric) by Kannan B T and Senthilkumar Sundararaj[2] in Fig.8. Validation shows that for H/D=2 secondary peak is at the exact location. The computations were also performed for Re = 40,000 with two H/D ratios(2 and 6) and the results were compared with the experimental data of R. J. Goldstein et. al.[Fig.9].

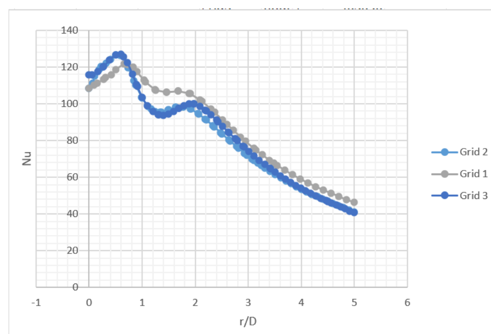

Fig. 6: Grid Independent study for SST $k-\omega$

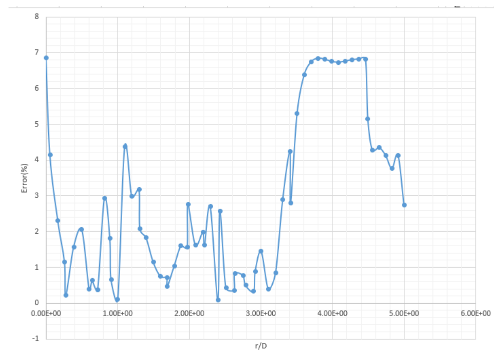

Fig. 7: Error between G2 and G3

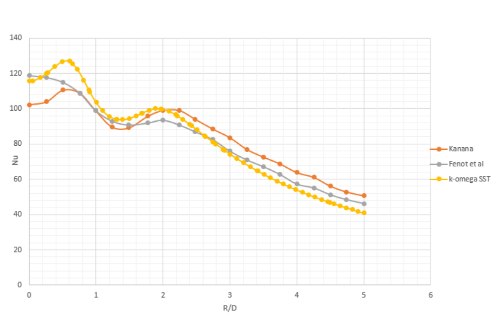

Fig. 8: Validation with Experimental data

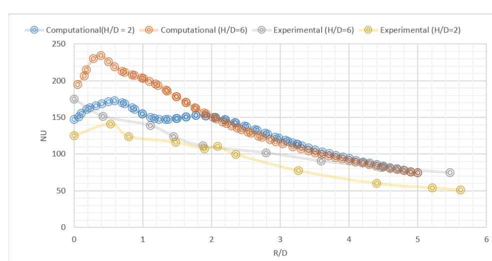

Fig. 9: Results for Re = 40,000

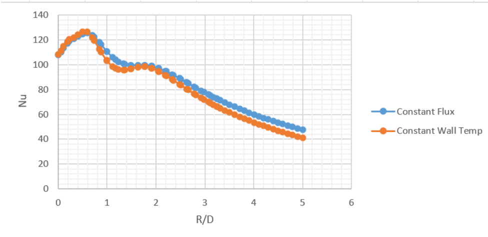

Fig. 10: Nusselt Plot for Uniform Heat Flux and Constant Wall Temperature

The boundary layer thickness is lowest at the centre of plate and hence the local convective heat transfer coefficient is maximum so the Nusselt number while the rise in Nusselt number occur at $r/D = 1.8$ because the flow transitioned from the laminar to turbulent and hence the Advection adds up to the Convection causing rise in local convective heat transfer coefficient.

C. Constant Wall Temperature

Under the constant wall temperature condition: the temperature of the plate is kept fixed at 400 K and jet temperature of 300 K is impinged over the plate keeping the H/D and Re same as that for uniform flux condition. Now ANSYS solution provides the Temperature at all the nodes positioned Δz=0.01 mm above the x-axis. These temperature values are used to calculate the temperature gradient normal to the wall and putting it in (5) we get the heat flux. The rigorous extraction of the Nusselt number and the one provided by ANSYS comes out to the same with an error of less than 1%(which is mainly due to gradient approximation). The Nusselt number obtained from the constant wall temperature comes out to be less than the uniform wall flux condition[Fig.10] which is in accordance with the experimental results of B. Sagot[3].

D. Uniform Flux Condition

The boundary condition of uniform heat flux of 6000 W/m^2 had been set to the wall. For calculating the Heat flux the temperature of fluid at two nodes : just at the wall and another at Δz above the wall, will be needed. Then,

$$q = -k\frac{\partial T}{\partial z} \tag{6}$$

$$= -k\frac{\Delta T}{\Delta z} \tag{7}$$

$$= -k\frac{T(z(=0)+\Delta z) - T(z(=0))}{\Delta z} \tag{8}$$

The heat flux obtained from (8) is then substituted in (1) to get the local convective heat transfer coefficient and finally using (4) Nusselt number is obtained.The error[Fig.11] for the Heat flux obtained from the above method is less than 2%(which is mainly due to gradient approximation).
The Nusselt is plotted along three different lines in XY plane of wall surface of circular plate and they completely overlap due to axis-symmetric nature of problem[Fig.12].

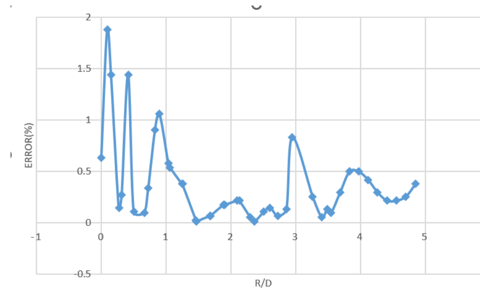

Fig. 11: Error in Flux calculated from Temperature gradient

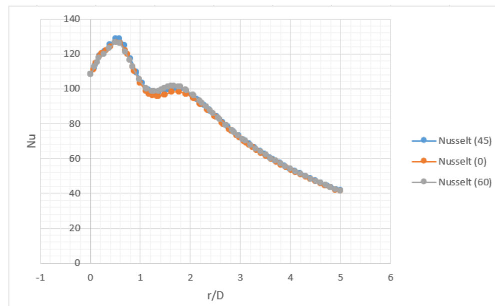

Fig. 12: Nusselt at different angles in XY plane

E. Turbulence Intensity

Turbulence intensity is defined as the ratio of standard deviation of fluctuating fluid velocity to the mean fluid speed, and it represents the intensity of fluid velocity fluctuation.

$$I = \frac{u'}{U} \tag{9}$$

where u' is root mean square of the turbulent velocity fluctuations and U is mean velocity.

$$u' = \sqrt{\frac{1}{3}({u'_x}^2 + {u'_y}^2 + {u'_z}^2)} \tag{10}$$

$$= \sqrt{\frac{2}{3}K} \tag{11}$$

where K is turbulent kinetic energy.

- Contour for the constant wall temperature condition is shown in Fig.13.
- Turbulence intensity and Nusselt number both have their peak nearly at r/D = 1.8[Fig.14]. As the fluid flow transitions from laminar to turbulent, the turbulent intensity rises and the Advection term also started to contribute to heat transfer, and therefore the Nusselt number also rises.
- The turbulence intensity has been plotted along radial direction[Fig.14] at a different height. i.e.z/D ratios[Fig.16]. Two peaks are observed in the plot, the first one is mainly due to the entrainment effect as the jet starts spreading as soon as it leaves the pipe(see the [Fig.13]) while the second one corresponds to the flow transitioning from laminar to turbulent.

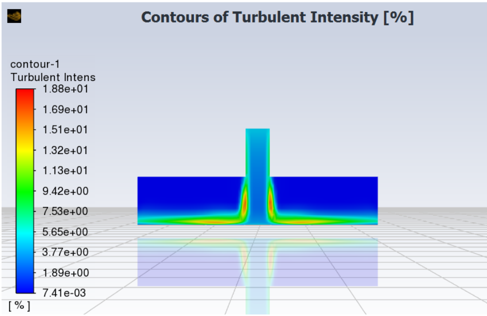

Fig. 13: Turbulence Intensity Contour

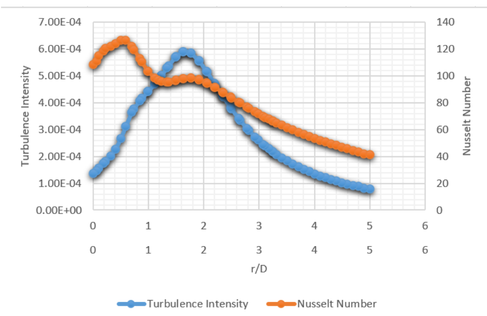

Fig. 14: Turbulence Intensity and Nusselt number

- As z/D increases, the first peak shifts left because the jet expands radially as it comes closer to the wall. i.e..moving in $-z$ direction, so for greater values of z/D, we would have entrainment at small r, compared to smaller values of z/D.
- The turbulent intensity started to drop earlier for smaller z/D as it is much closer to the wall so μ has a dominant effect than μ_t and hence a higher rate of dissipation of ϵ so turbulent kinetic energy(K) dissipates more rapidly($I \propto \sqrt{k}$).

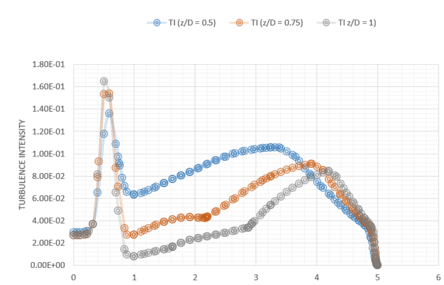

Fig. 15: Turbulence Intensity at different z/D

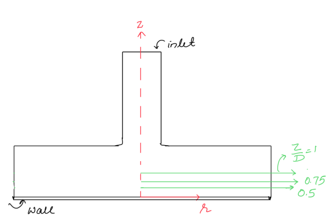

Fig. 16: Radial directions at different z/D

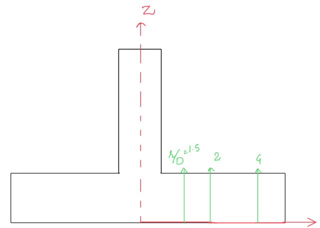

Fig. 17: Geometrical location

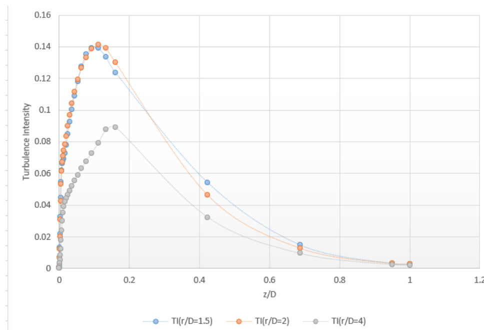

Fig. 18: Turbulence Intensity at different r/D

- The variation of Turbulence intensity at different r/D[Fig.17] are also plotted and in all the plots the peak is observed nearly at z/D = 0.18 [Fig.18].

IV. Conclusions

Following are the conclusions made from the analysis performed :

- The assessment of various turbulent models for Jet impingement Heat transfer shows that $k-\omega$ SST is acceptable for the modeling as its result are more close to experimental data than the other models used.
- The Nusselt number obtained from constant wall temperature conditions is slightly less than that obtained from uniform heat flux condition
- Turbulence intensity and Nusselt number had their peaks nearly at the same r/D.

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