

Characterization of Convective Transfer of Heat by Air Impingement using Single Swirling Jet with Insertion of Twisted Tapes by Computational Fluid Dynamics



Rakesh Kumar, Yogesh Kumar

Abstract: This study showed the analysis of heat transfer by computational fluid dynamics (CFD) from a heated target surface by the use of jet of single swirl and impingement by air at various Reynolds numbers. The half-length downstream with insertion of twisted tape is applied in a nozzle body to get the swirl. The distance between twist ratio and nozzle to plate of twisted tape is found to be same as $y = 2.93$ and 21mm ($H/D = 1$). The characteristics of transfer of heat on heated surface are determined with varied Reynolds numbers like 12000, 17000, 22000 and 27000.

Keywords: Computational Fluid Dynamics, Heat transfer, Swirling jet, Reynolds number.

I. INTRODUCTION

The impingement by jet is latest important technique for enhancement of heat transfer. Due to more transfer of heat and other benefits these methods is mainly applied in large applications of engineering and other fields. For example chemical vapour deposition, turbine blades cooling, to dry food products, textiles, films and papers, cooling of electronically used equipment, cooling the outside wall of combustion chamber etc. Various literatures was found on various forms like single and multiple jets with co-relations produced for characteristics of mass and heat transfer connected to this area. The main problem based on impingement by jet is the high non uniformity of the local heat flux (Remie, 2007) [1]. To reduce the limitations of radial non-uniformity different tests were taken by various forms. This problem is solved by introduction of jet in swirl form. Amini Younes (2015)[2] observed the similar radial distribution of transfer of heat is produced by a swirling jet in comparison of non-swirling jet. Different studies were conducted by experimental, empirical and numerical methods. Much literature (Hady, M.F (1997) [3] and (Zuckerman, Neil (2006) [4] was found related to this. Many experiments are needed to get the knowledge of characteristics of transfer of heat by the plate impingement.

Ianiro, Andrea (2012) [5] performed experiments to observe and compare the results of impinging of jets by swirling and multi-channel by the impinging jet of conventional method (CIJ), keeping the radius is same at the similar applications. Qing, Zheng (1997) [6] also studied to know the flow of air in swirl form in a cylindrical hole by the technique of the smoke wires. The picture of the discharge area was not accurate and also not showed the complete discharge field as comparison to the Bhatt, Shibani (2019) [7]. Various methods are implemented by various overall coefficient of heat transfer. The thermo chromic liquid crystal is one out of the various methods. That technique is implemented by Yeranee, K (2017) [8]. Different experiments were conducted at same distance between nozzle and plate at $L=4D$ to study the characteristics of flow and transfer of heat by a jet of swirl impingement. Benmouhoub, Dahbia (2013) [9] adopted a swirl generator prepared by a plug of cylindrical shape to supply swirl to jets of single and multiple with four narrow channels. It has been found that increase of Nusselt number and betterment in radial uniformity of transfer of heat was found by jet of swirl impingement as compared to impingement of jets by multi-channel and conventional impingement of jet.

II. NUMERICAL SETUP

To determine the 3dimensional results of transfer of heat by method of convection on the plate impingement the FLUENT-6.3.26 commercial computational fluid dynamics software was used which is attached with impingement of jets in swirl form with various Reynolds numbers. The computational domain contains the whole assembly of pipe and plate as represent in figure1. The designed was made by flow of air with a diameter of tube (D) 21 mm including twisted tape as in swirl shape (HLDI) and 300 mm length (L) impinge upon a flat plate which is heated at constant 3000 W/m^2 heat flux. The testing of geometry was taken at various Reynolds numbers as 12000, 22000, 17000 and 27000. The design selected for experiment is represented in figure2, where HLDI represents configuration of half-length downstream insert.

Revised Manuscript Received on February 05, 2020.

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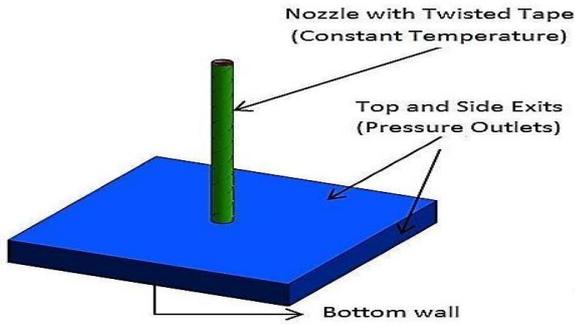


Figure 1 Assembly of pipe and plate

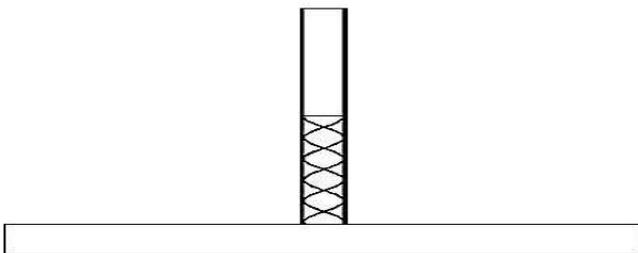


Figure 2 The configuration chosen for study

III NUMERICAL SIMULATION

Figure 3 represents sketch of computational domain for simulation by commercial computational fluid dynamics software FLUENT 6.3.26 and Table 1 shows conditions at the boundary used for this simulation at different parts of the computational domain.

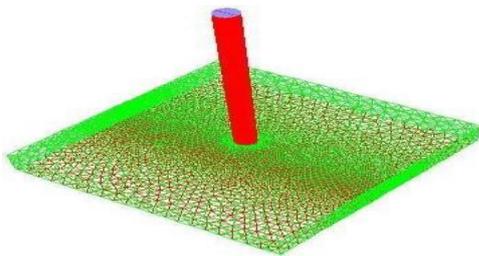


Figure 3 Computational domain used for simulation

To resolve the fluid flow the numbers of cells used for this domain are near about 12, 50,808 which are sufficient. To obtain prediction of heat transfer the fields of discharge and turbulence have to be solved in accurate way. To represent the all terms of heat transfer scheme of second order is used. This scheme is applied for the pressure; turbulence kinetic energy, momentum, specific dissipation rat etc. The standard SIMPLE algorithm is applied for the coupling of pressure and velocity. The way of simulation is used at condition of steady state convergence control is fixed at maximum 1000 trials, criteria of convergence is given as 10E-05 residuals and that can be varied if criteria of convergence is not obtained. The 3D equations of Navier-Stokes and energy are solved by using computational fluid dynamics software (FLUENT 6.3.26) which is attached with momentum and continuity equations to simulate fields of heat and flow of turbulence. There are various types of models available for turbulence but the shear stress transport is the best model for this configuration of flow and is also selected because of its simplicity, wide acceptability and economy.

Table 1. Boundary condition

Geometrical Identity/ Physical Location	Type of Boundary	Color
Outlet / Top Surface Opening	Outlet Pressure	Green
Impingement Plate / Bottom Surface	Wall	Red
Outlet/ Side Opening	Outlet Pressure	Green
Twisted Tape / Centre	Wall	Red
Nozzle Inlet / Inlet	Inlet Velocity	Blue
Nozzle Wall	Wall	Red

IV GOVERNING EQUATIONS FOR TRANSFER OF HEAT

The Newton's law of cooling was used to transfer of heat from solid surface to movement of fluid can be represented by:

$$q = h(T_{\infty} - T_s) \quad (1)$$

in which q is the flux of heat from the solid to the moving fluid, h is convective coefficient of transfer of heat, T_{∞} is the reference temperature and T_s is the for the temperature of the system. The temperature difference is like a tension for convection. The coefficient of transfer of heat (h) is non-dimensionalized using the length scale (D) and thermal conductivity (K_t) which represents the design. This gives a number which is dimensional less called Nusselt's number (Nu):

$$Nu = \frac{hD}{K_t} \quad (2)$$

The Nusselt's number (Nu) is a major feature of the design, the velocity of flow and the fluid's physical properties. Normally various relations are obtained between the Reynolds number (Re), Nusselt's number (Nu), Prandtl number (Pr) and a dimensional less parameter of geometry.

V. GRID DEPENDENCY TEST

This test influences the simulation results directly as represented in table 2. The test of dependency of Grid was conducted to compare the deviations in largest values of Nusselt's number. These calculations were conducted at Reynolds number 12000 by without insertion of twisted tape and at a distance of jet to plate 2 ($H/D=2$). These calculations are shown in the table 2. It was found from this table that with 12, 50,808 cells provide accurate calculation at less computation time to resolve the fluid flow the amount of cells used in this domain are near about 12, 50,808 these are adequate.

Table 2. Results of dependency of Grid

Sr. No.	Number of cells	Nusselt's number

1	08,21,751	89.4
2	09,11,627	69.5
3	9,13,630	101.7
4	11,27,328	124.7
5	12,50,808	133.6
6	13,39,937	135.5
7	14,34,630	133.4
8	15,15,536	131.3

VI. SIMULATION OF SWIRLING IMPINGING JET (SIJ) AND ITS RESULTS

To study the influence of Re on the characteristics of transfer of heat the simulation of the fluid flow done for various Re at same twist ratio and separation distance. The separation distances (H/D) applied for this design is H/D=1 with twist ratio of twisted tape insert $y = 2.93$. Re is changed as Re 12000, 17000, 22000 and 27000. The influence of Re on Nusselt's number and coefficient of heat transfer is represented in fig. 4 and 5 respectively.

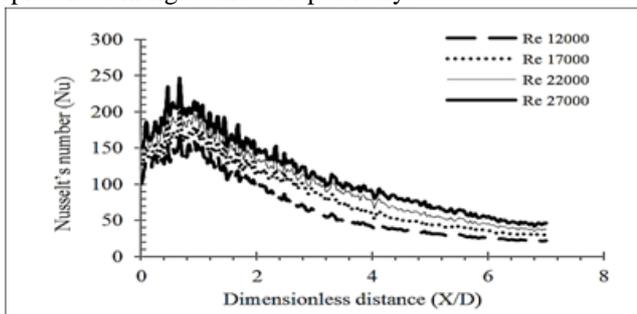


Fig. 4: Variation of Nusselt's number w.r.t. X/D for different Re at H/D = 1 and for $y = 2.93$

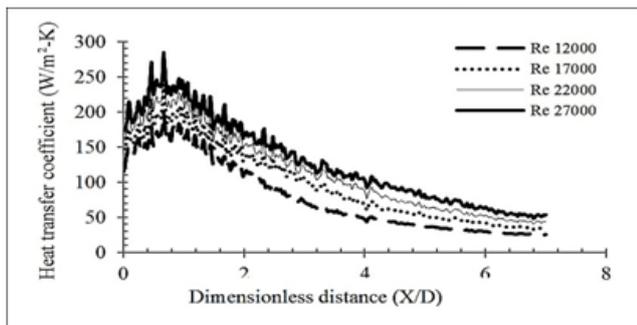


Fig. 5: Variation of coefficient of transfer of heat w.r.t. X/D at H/D = 1 and for $y = 2.93$

Influence on Nusselt's number:

It was shown that the increase in Nusselt's number with the increase in Reynolds number. The values of Nusselt's number at different Reynolds number for similar configuration and separation distance are given as under: At value of H/D=1 and for the half-length downstream configuration with the insertion of twisted tape along with twist ratio of 2.93 the Nusselt's number is 100.21 at peak point i.e. X/D = 0 for Reynolds number = 12000 and dimension will improved to 127.86 for Reynolds number=17000, 141.51 for Reynolds number =22000 and 150.34 for Reynolds number = 27000. The dimension of Nusselt's number is obtained 174.18 at X/D = 0.66 for Re = 12000

and at this value of X/D Nusselt's number is improved to 207.86 for Reynolds number = 17000, 229.17 for Reynolds number = 22000 and 247.05 for Reynolds number = 27000 this is the largest dimension of Nusselt's number in this form.

Influence on co-efficient of transfer of heat:

By the study to know the influence of co-efficient of heat transfer with the variation of Reynolds number it was observed that shows that Co-efficient of transfer of heat increase with the increase in Reynolds number. These are shown as below:

- i. At value of H/D=1 and for the form of half-length downstream by the insertion of twisted tape with twist ratio of 2.93 the dimension of co-efficient of transfer of heat at peak point 115.49 at X/D = 0 is and Reynolds number = 12000. The dimension increased to 147.35, 163.08, 173.25 with the increase of Re = 17000, 22000 and Re = 27000.
- ii. The dimension of coefficient of transfer of heat is obtained to be 200.73 at X/D = 0.66 and Reynolds number = 12000 and at this dimension of co-efficient of transfer of heat is improved to 239.53, 264.09, 284.70 for Reynolds number 17000, 22000 and 27000 which is the highest value of co-efficient of transfer of heat. Figure 6 and figure 7 represents the average Nusselt's number and values of coefficient of heat transfer at varied Reynolds numbers.

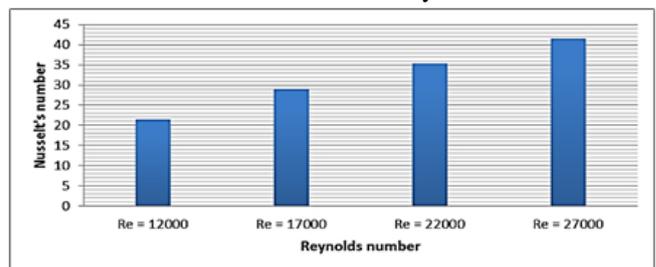


Fig. 5: Variation of average Nusselt's number at varied Reynolds numbers

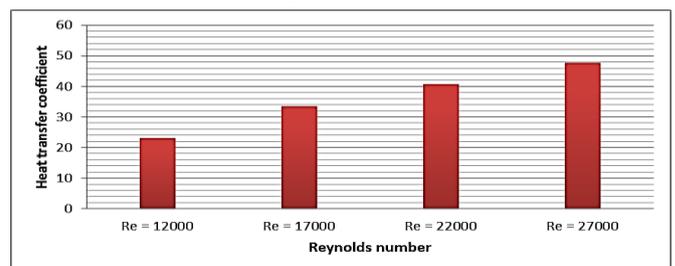


Fig. 6: Variation of average coefficient of heat transfer at varied Reynolds numbers

VII. CONCLUSION

It was concluded that increase of velocity of jet is a natural way to increase intensity of turbulence of the jet and this was achieved by increase of Reynolds number. As the Reynolds number increased the Nusselt's number also increased as well as rates of transfer of heat. Due to this, in turbulent flow, the rates of momentum and heat transfer more in comparison to laminar flow.



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It was also noticed that this increment due to turbulence achieve largest values as the intensity of turbulence increases for a specific Reynolds number. The Figure7 shows the contours of Nusselt's number for varied range of Reynolds number.

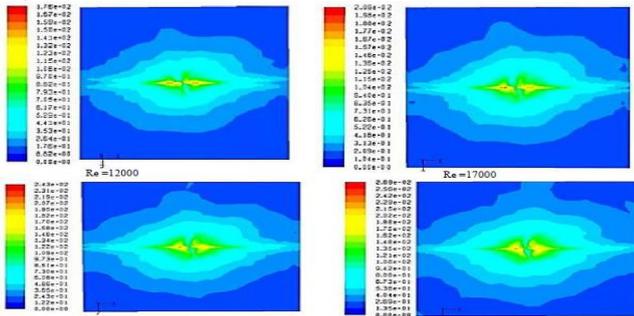


Fig. 7: Map of contours of Nusselt number at varied Reynolds numbers

Nomenclature

D : Nozzle diameter in m.

h : coefficient of transfer of heat, $w/m^2 k$

k : Thermal conductivity in air, $w/m \cdot k$

Nu : Nusselt number

q^* : heat flux in convection

r : Radial distance from the stagnation point, X/D : Dimensionless radial distance

Re : Reynolds number based on the outlet diameter, $Re = \rho u D / \mu$

u : Jet velocity,

$m/s \rho$: Air Density, Kg/m^3

μ : Kinetic viscosity of air, $Kg/m \cdot s$

ACKNOWLEDGMENT

We would like to thank for this work: Dr.Gurinder Pal Singh Brar, and Dr.M.K.Kushwaha.

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