



Thermal and Hydrodynamic Analysis of the Impingement Cooling inside a Backward Facing Step Flow

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Available online at: www.isca.in

(Received 16th September 2011, revised 11th January 2012, accepted 25th January 2012)

Abstract

The heat transfer and fluid flow of multiple confined impinging jets impinge normally to the backward facing step cross flow has been numerically investigated. Different sizes of impinging jets were tested while the channel contraction ratio(SR) was ranged from 0.25 to 0.5. The continuity, Navier-Stokes and energy equations were solved numerically. The discretized form of these equations was obtained by using finite volume method with staggered grid technique. A Fortran built home computer code depending on SIMPLE algorithm was developed to obtain the numerical results. The standard k-ε model is used to treat the effect of turbulence while the wall functions laws were used to treat the regions near the solid walls. The aim of this study is to show how multiple confined impinging jets can be a controlling factor to enhance the rate of heat transfer form the hot surface of the channel backward facing step flow. The conducted results show that the heat transfer is enhanced significantly when using multiple impinging jets. The highest heat transfer was found closer to the region of the facing step. Also the results show that the rate of heat transfer is increased as the jets sizes increase.

Keywords : Impingement cooling, duct flow, backward facing step.

Introduction

Flow separation and reattachment phenomena are widely encountered in multiple engineering applications such as cooling of electronic devices, combustion chambers and cooling of turbine blades. Although the geometry of the channel backward facing step is simple, but the resulting structure of the flow and heat transfer over this step include a high degree of complexity. In the region just after the step, the heat losses occur and this phenomena is extended to reattachment point. So, the present study aims to enhance the heat transfer in this type of the flow geometry even after the reattachment point by using multiple impinging slot jets distributed on the upper channel wall. These jets can remove a large amount of heat transfer from the hot stepped wall. To the knowledge of the author, there is no study documented on this particular flow geometry up to date, Consequently this study will assist to promote the research area and giving new aspects to enhance the rate of heat transfer. There are many studies about the turbulent flow over a backward facing step. Kasagi and Matsunaga¹ investigated the turbulent flow in a channel with a backward facing step. 3-D particle tracking velocimeter was used as a measurement technique. They found that the Reynolds normal and shear stresses had the maximum values upstream of the reattachment. Their study was compared with numerical simulation. Lio an Hwang² performed a numerical study on turbulent flow in a duct with a backward-facing step. The turbulent flow and heat transfer in a channel with rib turbulators was investigated by Lio and chen³, Rau et al.⁴ and

Hane and Park⁵. The main objective of these studies was to obtain the heat transfer characteristics and friction factor. ABE and Kondah⁶ presented a new turbulent model for predicting fluid flow and heat transfer in separating and reattaching flows. The presented model was modified from low-Reynolds number k-ε model. They demonstrated that the used model was efficient in separating and reattaching flows downstream of backward facing step. Jovice and Driver⁷ presented an experimental study on the turbulent flow over a backward facing step at low Reynolds number. The aim was to validate the numerical simulation which was performed by Stanford/NASA center for turbulence research. Ichimiya and Hosaka⁸ performed an experimental study to study the characteristics of impingement heat transfer caused by three impinging jets. They conducted that there was two peaks of the local Nusselt number behind the second nozzle. Zhang et al.⁹ developed a stochastic separation flow model to simulate the sudden expansion of particle-laden flows. Wang et al.¹⁰ used large eddy method and Lagrangian techniques to simulate the turbulent flow over a backward-facing step. The study verified that the particles follow a path when the vorticity of the gas phase is small. Thangam and Knight¹¹ and Nie and Armaly¹² investigated the effect of step height on the separation flow for convective flow adjacent to a backward-facing step. Rhee and Sung¹³ adopted a diffusive tensor heat transfer model instead of the familiar constant Prandtl number model.

In this paper, an attempt is made to incorporate the effect of impinging slot jets flow into the problem of a channel backward-facing step and the turbulent flow and heat transfer of the combined new configuration (figure 1) is numerically simulated. Different parameters, such as contraction ratio (SR), size and number of impinging jets, and jet and channel Reynolds numbers are tested.

Methodology

Mathematical Formulation: The working fluid (air) is assumed to be Newtonian, Incompressible and the thermo physical properties are constants. The continuity, Navier-Stokes and energy equations were used to model the considered problem. These equations be described as follows.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

$$\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + 2 \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial v}{\partial x} \right) \quad (2)$$

$$\rho u \frac{\partial v}{\partial x} + \rho v \frac{\partial v}{\partial y} = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial v}{\partial x} \right) + 2 \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial v}{\partial y} \right) + \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial u}{\partial y} \right) \quad (3)$$

$$\rho u \frac{\partial T}{\partial x} + \rho v \frac{\partial T}{\partial y} = \frac{\partial}{\partial x} \left(\Gamma_{eff} \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\Gamma_{eff} \frac{\partial T}{\partial y} \right) \quad (4)$$

$$\mu_{eff} = \mu + \mu_t \quad (5)$$

$$\Gamma_{eff.T} = \frac{\mu}{Pr} + \frac{\mu_t}{Pr} \quad (6)$$

where μ_{eff} is the combined laminar and turbulent viscosity and Γ_{eff} is an effective exchange coefficient.

Turbulence Model: The standard turbulence k-ε model proposed by Launder and Spalding¹⁴ is used here to handle the effect of turbulence in the flow. This model includes two transport equations, one for turbulent kinetic energy and the other for the rate of dissipation of turbulent kinetic energy.

$$\rho u \frac{\partial k}{\partial x} + \rho v \frac{\partial k}{\partial y} = \frac{\partial}{\partial x} \left(\Gamma_{eff,k} \frac{\partial k}{\partial x} \right) + \frac{\partial}{\partial y} \left(\Gamma_{eff,\epsilon} \frac{\partial k}{\partial y} \right) + \rho G - \rho \epsilon \quad (7)$$

$$\rho u \frac{\partial \epsilon}{\partial x} + \rho v \frac{\partial \epsilon}{\partial y} = \frac{\partial}{\partial x} \left(\Gamma_{eff,\epsilon} \frac{\partial \epsilon}{\partial x} \right) + \frac{\partial}{\partial y} \left(\Gamma_{eff,\epsilon} \frac{\partial \epsilon}{\partial y} \right) + \rho C_{1\epsilon} \frac{\epsilon}{k} G + \rho C_{2\epsilon} \frac{\epsilon^2}{k} \quad (8)$$

$$\text{Where } G = \mu_t \left[2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right] \quad (9)$$

$$\Gamma_{eff,k} = \mu + \frac{\mu_t}{\sigma_k}, \quad \Gamma_{eff,\epsilon} = \mu + \frac{\mu}{\sigma_\epsilon} \quad (10)$$

the eddy viscosity is obtained by the following formula:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (11)$$

The model coefficients are (σ_k ; σ_ϵ ; $C_{1\epsilon}$; $C_{2\epsilon}$; C_μ) = (1.0, 1.3 , 1.44 , 1.92 , 0.09) respectively. The flow parameters at inlet are described as follows:

$$k_{in} = 0.05 U_{in}^2, \quad T_{in} = T_c = 25^\circ \text{C}$$

$$\epsilon_{in} = k_{in}^{1.5} / \lambda h, \quad \lambda = 0.005$$

$$Re_{in} = \frac{U_{in} h}{\nu} \quad (12)$$

where k_{in} , U_{in} , T_{in} are the turbulent kinetic energy, velocity and temperature at a channel inlet respectively.

At the walls, no slip conditions are imposed; $u = v = 0.$, $k = 0.$,

$$\frac{\partial \epsilon}{\partial y} = 0., \quad T_w = T_h = 50^\circ \text{C}$$

To treat the large steep gradient near the walls of the channel and step, wall function laws used by Versteeg¹⁵ is adopted. The local Nusselt number along the bottom wall is expressed

$$\text{as } Nu = \frac{\partial \theta}{\partial Y}, \text{ at } y=0. \text{ Zero gradients are imposed on the}$$

channel exit for considered variables.

Numerical solution: In this paper, the numerical simulations were performed on non-uniform staggered grid system by using finite volume method (FVM). This gives a system of discretization equations which means that the system of governing elliptic partial differential equations is transformed in to a system of algebraic equations. Then, the solution of these transformed equations is performed by an implicit line by line Gaussian elimination scheme. An elliptic finite volume computer code was developed to attain the results of the numerical procedure by using pressure-velocity coupling (SIMPLE algorithm) according to Versteeg (1995). This code is based on a hybrid scheme. Because of the strong coupling and non-linearity that are inherent in these equations, relaxation factors are needed to ensure convergence. To ensure that the turbulent fluid flow solutions are not significantly affected by the mesh, the numerical simulations are examined under different grid sizes that range from 62×28 to 82×52. Adding grid points beyond 62×28 did not significantly affect the results.

Results and Discussion

The computed results are presented for two-dimensional turbulent flow through a channel backward-facing step along

with orthogonal impinging slot jets flow. Different contraction ratios and sizes of impinging jets are investigated for different Reynolds numbers. The effect of contraction ratio on the distribution of streamlines for three impinging slot jets and $H/B=11$ is demonstrated in figure 2. It is evident that the incoming cross flow affects the behaviour of trajectories of the multiple impinging slot jets because the potential core of each jet is distorted. The trajectories of the impinging slot jet forced the cross channel flow towards the bottom wall of the channel and reduced the reattachment length just after the edge of the step. The flow struck the bottom wall in the region between the step and the first impinging jet and enhanced heat transfer, as shown in figure 4. This phenomenon occurred for all the studied cases and is enhanced as the contraction ratio increase (the size and re-attachment lengths are increased).

Figure 3 illustrates the non-dimensional axial velocity at different stream wise stations. It is evident that the dimensionless axial velocity increases as Re increases because of increasing the inertia forces but this increase is affected by the regions of the presence of impinging slot jets or the facing step.

The effect of increasing the size of the impinging jets on the distribution of the local Nusselt number is found in figure 4. It is clear that the local Nusselt number increases as the slot jet width increases (the ratio H/B decreases) because the strength and size of the recirculation regions are larger behind each jet and the step. The larger recirculation regions increase the impinging flow towards the hot wall besides the turbulence effects, which increases the rate of heat transfer.

Figure 5 demonstrate the comparison of Nusselt number variation between the case of the backward facing step flow with and without impingement cooling in addition to test of the effect of number of impinging slot jets on the mentioned variation. As the figure shows, the heat transfer from the bottom hot wall is significantly increased when incorporating impingement cooling to the backward facing step flow. It can be seen that the number of the impinging slot jets equal to three achieved the maximum rate of heat transfer. When the number of impinging slot jets exceeds three, the rate of heat transfer is significantly decreased beyond $x=0.1$.

To validate the present numerical code, a test on some of the published studies is performed and a good agreement was obtained. Figure 6 represents an example for this comparison

Conclusion

A computational study for thermal and hydrodynamic analysis of the impingement cooling inside a channel backward facing step flow has been numerically performed. It is found that the rate of the heat transfer is significantly enhanced due to the presence of impinging slot jets compared

with conventional backward-facing step problem. The impinging slot jets affected the size of the recirculation regions and the re-attachment length behind the facing step. Also it is found that the heat transfer rate increases as the jet size increases. The strength of recirculation regions and the rate of heat transfer after the step and between the impinging slot jet is enhanced as Reynolds number increases.

Nomenclature

B = slot jet width, m, G = generation term, $\text{Kg/m}\cdot\text{sec}^3$, H = height of the channel, m, k = turbulent kinetic energy, m^2/s^2 , L = length of the channel, m, Nu = local Nusselt number, p = pitch, m, P = pressure, N/m^2 , Pr = Prandtl number, -, Re = Reynolds number, -, s = step height, m, SR = contraction ratio (s/H), -, T_c = cold wall temperature, \dot{C} , T_h = hot wall temperature, \dot{C} , U_{in} = velocity at a channel inlet, U_j = velocity at a slot jet inlet

Greek symbols: ϵ = turbulence dissipation rate, m^2/s^3 , μ = dynamic viscosity, $\text{N}\cdot\text{s}/\text{m}^2$, μ_t = turbulent viscosity, $\text{N}\cdot\text{s}/\text{m}^2$, ρ = air density, Kg/m^3 , Γ_{eff} = effective exchange coefficient, $\text{kg}/\text{m}\cdot\text{s}$

θ = dimensionless temperature $\left(\frac{T - T_c}{T_h - T_c} \right)$, -

σ_k ; σ_c = turbulent Schmidt numbers, -

References

1. Nobuhide Kasagi Akio Matsunaga, Three- Dimensional particle – Tracking – Velocimeter velocimetry measurement of Turbulence Statistics and Energy Budget in a backward-Facing Step Flow, *Int. J. Heat and Fluid Flow*, **16**, 477-485 (1995)
2. Lio T., Hwang J., Developing Heat Transfer and Friction in a Ribbed Rectangular Duct with Flow Separation at Inlet, *ASME. J. Heat Transfer*, **114**, 546-573 (1992)
3. Lio T.M., Hwang G.G. and Chen S.H., Simulation and Measurements of Enhanced Turbulent Heat Transfer in Channels With Periodic Ribs on One Principal Wall, *International Journal of Heat Mass Transfer*, **36**, 507-507 (1993)
4. Rau G., Cakan M., Moeller D. and Arts T., The Effect of Periodic Ribs on The Local Aerodynamics and Heat Transfer Performance of A Straight Cooling Channel, *ASME Journal of Turbomachinery*, **120**, 368-375 (1988)
5. Han J.C, Heat Transfer and Friction Characteristics in Rectangular Channels With Rib Turbulators, *ASME Journal of Heat Transfer*, **110**, 91-98 (1988)

6. Abe K. and Kondoh T., A new Turbulent Model for Predicting Fluid Flow and Heat transfer in Separating and Reattaching flows, **37**, 139-151 (1994)
7. Srba Jovice and David M. Driver, Backward Facing Step Measurements at Low Reynolds Number, $Re_h=500$, NASA, California 94035-1000 (1994).
8. Ichimiya K. and Hosaka N., Experimental Study of Heat Transfer Characteristics Due To Confined Impinging Two Dimensional Jet Exp. Thermal and Fluid Science, **5**, 803-807 (1992)
9. Zhang H.Q., Chan C.K. and Lau K.S, Numerical Solution of Sudden Expansion Particle-Laden Flows Using an Improved Stochastic Flow Model, *Numerical Heat Transfer*, Part A, 4089-102 (2001)
10. Wang B., Zhang H.Q. and Wang X.L., Large Eddy Simulation of Particle Response to Turbulence along its Trajectory in a Back Word-Facing Step Turbulent Flow, *Int. J. Heat Mass Transfer*, **49**, 415-420 (2006)
11. Thangam S. and Knight D., Effect of Step Height on The Separated Flow Past a Backward Facing Step, *Phys. Fluids*, **3**, 604-606 (1989)
12. Nie J.H. and Armaly B.F., Three Dimensional Convective Flow Adjacent to a Backward Facing Step-effects of Step Height, *Int. J. Heat Mass Transfer*, **45**, 2431-2438 (2002)
13. Rhee G.H. and Sung H.G., Enhancement of Heat Transfer in Turbulent Separated and Re-Attachment Flow by Local Forcing, *Numerical Heat Transfer*, Part A, **137**, 733-735 (2000)
14. Jones W.P. and Lunder B.E., The Prediction of Laminarization with a Two Equation Model of Turbulence, *J. Heat Mass transfer* (1972)
15. Versteeg H.K. and Malalasekera W., An Introduction of Computational Fluid Dynamics, Hemisphere Publishing Corporation, United State of America (1995)

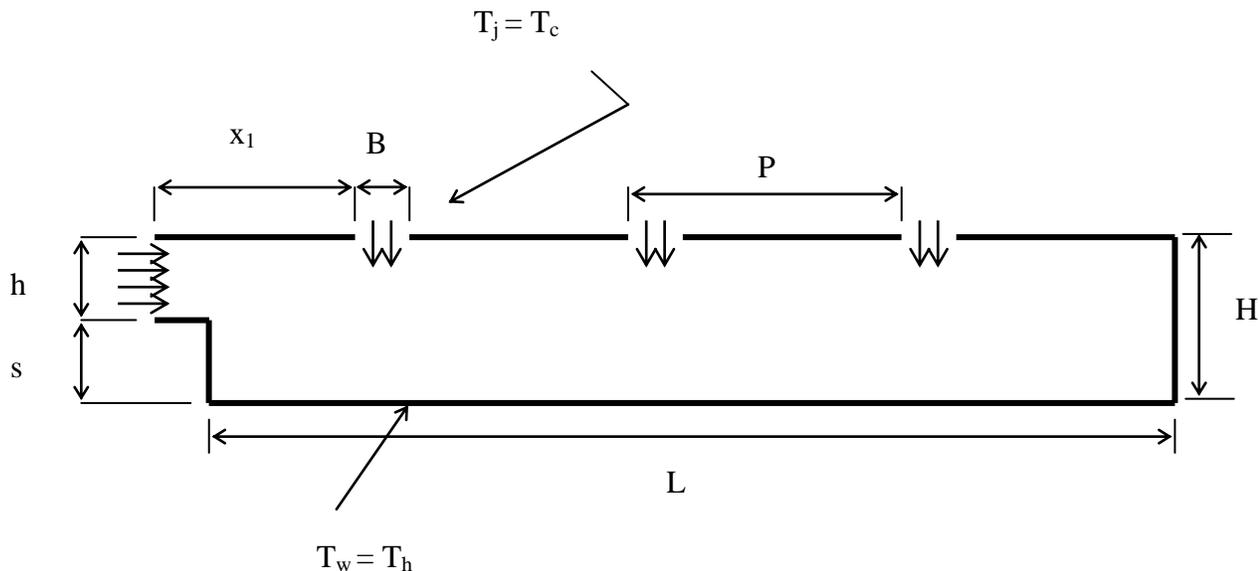
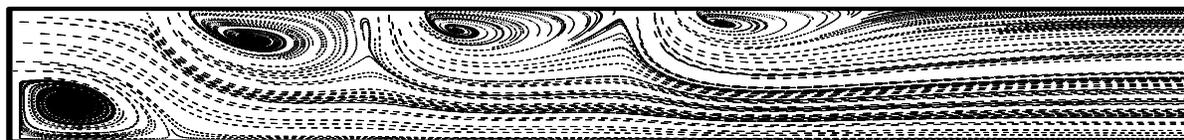
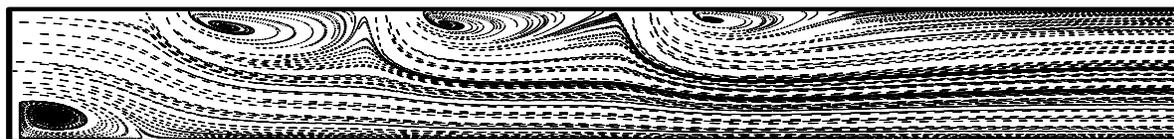


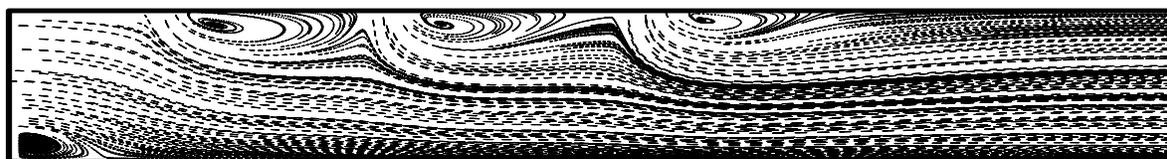
Figure-1
 Schematic diagram of the considered problem, $H=0.05m$, $L=0.4m$, $x_1=0.0492m$, $H/B=11$ and $P/B=4$



a. SR =0.5

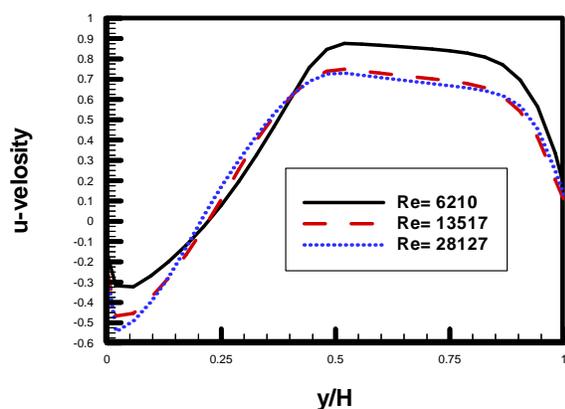


b. SR =0.35

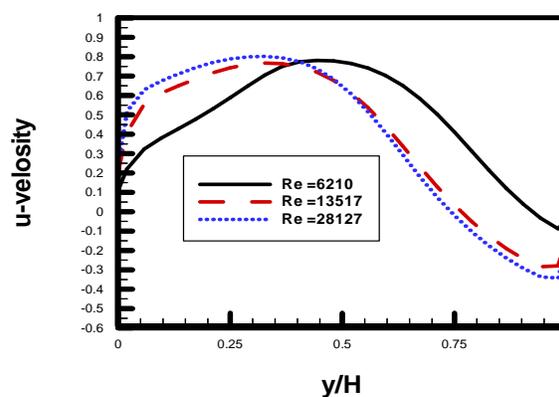


c. SR =0.25

Figure-2
 Distribution of Stream lines for 3 jets and different values of contraction ratio;
 $Re_j=13517$, $Re_{in}=16896$, $H/B=11$ and $P/B=4$



a. $x/L=0.0315$



b. $x/L=0.2$

Figure-3
 Variation of the dimensionless axial velocity (u/U_{in}) at different jet Reynolds numbers: $H/B=11$, $P/B=4$,
 $Re_{in}=16896$, and $SR=0.5$

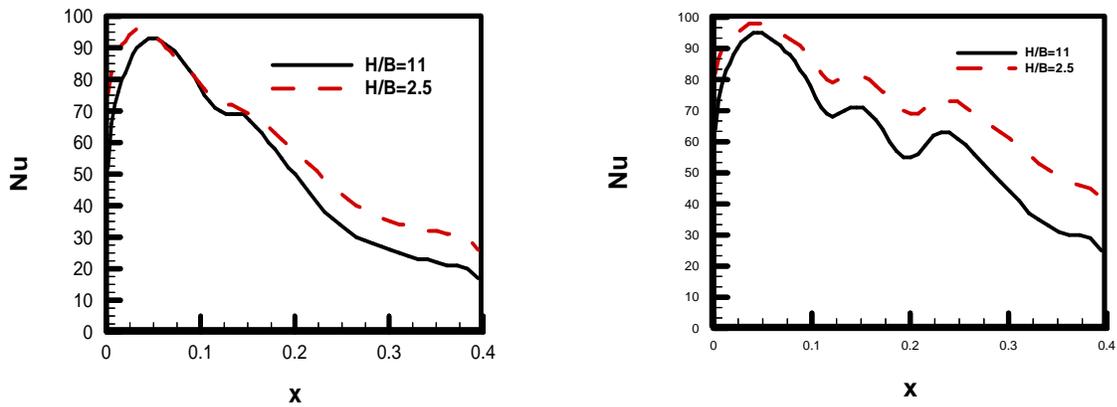


Figure-4
 Variation of local Nusselt number (on the hot wall) for $SR=0.5$, $Re_j=13517$ and $Re_{in}=16896$

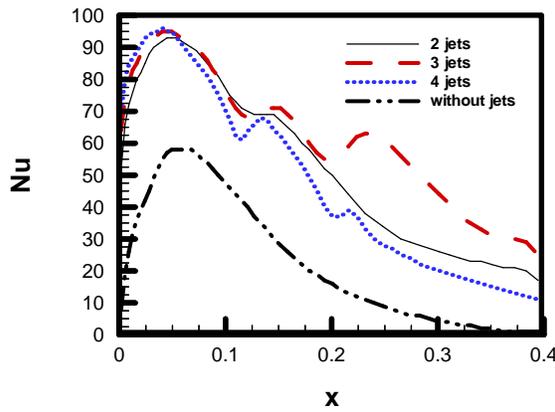


Figure - 5
 Comparison of variation of local Nusselt number (on the hot wall) with and without impingement for $SR=0.5$, $Re_j=13517$, $Re_{in}=16896$, $H/B=11$ and $P/B=4$

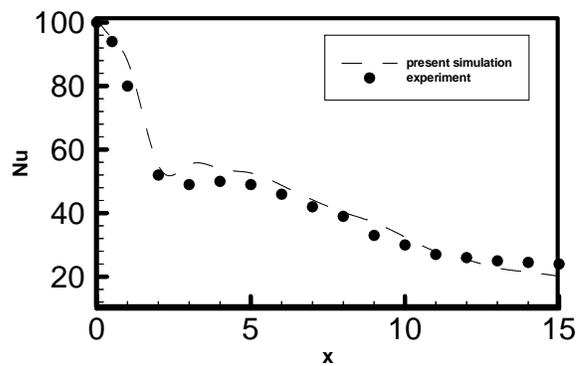


Figure-6
 Comparison of the present results with published results of [8] for $H/B=1$ and $Re_j=6000$