Heat Transfer Enhancement in channel with obstacles

M. Nazari
Department of Mechanical Engineering
University of Tehran

M. H. Kayhani
Department of Mechanical Engineering
Shahrood University of Technology

Y. Khazraii
Department of Mechanical Engineering
Shahrood University of Technology

Abstract: Analysis of fluid flow and heat transfer in an obstacle channel has lots of industrial applications. In this paper, incompressible fluid flow in a channel has been studied two dimensionally and the effects of obstacle arrangement on heat transfer have been investigated. The heights of channel and obstacle size are variables to find out the more properly model for description of heat transfer's improvement in a channel. The numerical finite volume method has been developed and the Mean Nusselt number has been calculated through the channel incorporating the effects of obstacle geometry. The numerical results have good agreement with former experimental and numerical data.

Key Words: Convective Heat Transfer, Enhancement, CFD

Nomenclature

D_h = Channel hydraulic diameter, m
h = Obstacle height, m
h_c = Convective heat transfer coefficient, $W/m^2.K$
H = Channel height, m
k = Thermal conductivity, $W/m.k$
L = Obstacle streamwise length, m
Nu = Nusselt number
q'' = Heat flux, W/m^2
Re_{D_h} = Reynolds number,
s = Obstacle spacing, m
u = Velocity, m/s
W = Channel width, m
μ = Dynamic viscosity, $(N.s)/m^2$
θ = Dimensionless temperature

Subscripts
e = Entrance
f = Fluid
s = Solid
m = Mean

Introduction

Heat transfer enhancement in channel has extensive engineering applications including heat exchanger design and cooling technology. The forced convective cooling of a two-dimensional array of multiple heated obstacles located upon one wall of an insulated channel was experimentally investigated by Vafai et al. [1]. For an air flow range of 800<Re<13000 and input heat flux of 950<q" <20200 W/m^2, the different geometric arrangements that were employed to study the effects upon the heat transfer include changes in channel height and use of an individual obstacle or an array of similar obstacles. The characterization of turbulent flow and convective heat transport of single isolated two and three-dimensional obstacles in a channel were performed by Roeller et al. [2]. Larger obstacle widths increase the flow acceleration by blocking more channel flow area while smaller widths have more intense three-dimensional transport effects. The use of an odd-sized rectangular obstacle within a three-dimensional array of square obstacles, with maximum heat fluxes of 6700 W/m^2, was found to enhance the heat transfer up to 40 percent by Jubran et al.[3]. Sparrow et al. [4] also investigated the effects of height differences within three-dimensional arrays of square obstacles and found, using the naphthalene sublimation technique, heat transfer enhancements of up to 80 percent compared with an array of uniform height. The two-dimensional conjugate heat transfer problem for laminar flow over an array of three obstacles was solved, utilizing a control volume formulation by Davalath et al. [5].Their analysis included the effects of obstacles spacing.

In this study Heat transfer characteristics in an insulated channel with heated obstacles for developed laminar flow of air are investigated by Finite volume method.

The simulation work has been carried out for variant Reynolds numbers, 600<Re<1400, and the input heat flux to the obstacles ranged from 950 up to 20200 W/m^2. The results are presented in the form of mean Nusselt number at the length of obstacles. In this work affects of different variables as Size and Geometry of obstacles, Channel height, obstacles number, input heat flux and flow rates on Nusslet Number are investigated.

Finding temperature distribution and the convective heat transfer coefficient on the obstacle surfaces for different arrangements of them and measuring the improvement or decrease of heat transfer in the channel are the most important issues resulted from this research.

Data resulted through modelling will be compared with the experimental results and proper comments for improving the heat transfer in channels with obstacle can be offered. This can be a sufficient solution for an accurate designing and construction of compact heat exchanger.

Geometry of problem

The geometry of problem has been shown in Fig. 1. As is shown in this figure the channel walls are completely isolated and the constant heat flux imposed at the bottom surface of obstacle. The length and height of channel are considered L, H. In this study the height of channel (H) is varied between 10-90mm.

To decrease the end effect, channel width has considered $W = 305mm$.

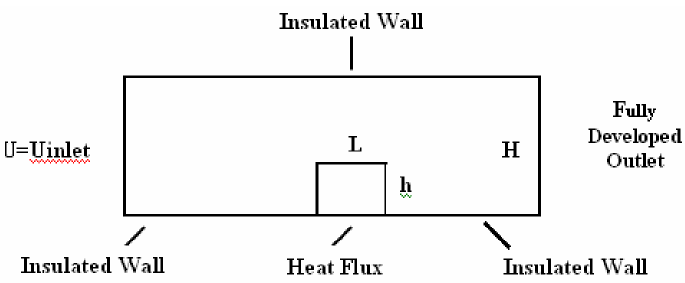


Fig. 1: the geometry of problem

The stramwise length of obstacles, L, was chosen to nondimensionalize the geometric data.

In this study the channel height varied between the range of 1.22< $\frac{H}{L}$ <3.11.

Governing Equations and Boundary Conditions

In numerical analysis of convective heat transfer in a channel with arrays of heated obstacles, Navier-Stocks equations for a laminar flow in a Newtonian fluid, steady incompressible flow with constant thermodynamic properties is solved. Complete conductive heat transfer in solid is also considered.

Governing equation are the conservation of mass, incompressible Navier-Stokes and Energy equation, presented in the following nondimensional form:

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

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = -\frac{dP^*}{dx*} + \frac{1}{\mathrm{Re}_H} \nabla^2 u^* \quad (2)$$

$$u^* \frac{\partial v^*}{\partial x^*} + v^* \frac{\partial v^*}{\partial y^*} = -\frac{dP^*}{dy*} + \frac{1}{\mathrm{Re}_H} \nabla^2 v^*$$

$$u^* \frac{\partial \theta}{\partial x^*} + v^* \frac{\partial \theta}{\partial y^*} = \frac{1}{Pe} \nabla^2 \theta \quad (3)$$

With assuming an incompressible flow and one dimensional velocity profile and constant heat flux on the obstacles surfaces, boundary conditions will be defined as below:

Entrance:

$$u^* = 1, v^* = 0, \theta_f = 0 \quad (4)$$

Outlet:

$$\frac{\partial u^*}{\partial x^*} = 0, \frac{\partial v^*}{\partial x^*} = 0, \frac{\partial \theta_f}{\partial x^*} = 0 \quad (5)$$

Fluid/solid interfaces:

$$u^* = 0, v^* = 0, \theta_f = \theta_s \quad (6)$$

Channel walls:

$$u^* = 0, v^* = 0, \frac{\partial \theta_f}{\partial y^*} = 0 \quad (7)$$

Basic Theory

Local convective heat transfer coefficient is defined as below:

$$h_c = \frac{q}{A_C(T_e - T_m)} \tag{8}$$

Where q is surface heat flux, A_C is wetted surface area, T_e and T_m are surface temperature and mean temperature simultaneously that can be obtained with the following equation:

$$T_m = \frac{1}{UA}\int uTdA \tag{9}$$

The local Nusslet number is defined as $Nu = \dfrac{h_c x}{k_f}$ and the mean Nusselt number is found as the average of the local values,

$$Nu_m = \frac{\int_{A_c} Nudx}{A_C} \tag{10}$$

The Reynolds number was defined as $\mathrm{Re}_{D_h} = \rho_f u_m D_h \big/ \mu_f$ where the mean fluid velocity was found from the volumetric flow rate within the channel and the channel hydraulic diameter is $D_h = 2W.H \big/ (W+H)$.

All thermophysical properties of the air were evaluated at the entrance temperature. In our study that heat flux distribution is constant, fluid mean temperature can be calculated from the energy balance in each selective point of channel.

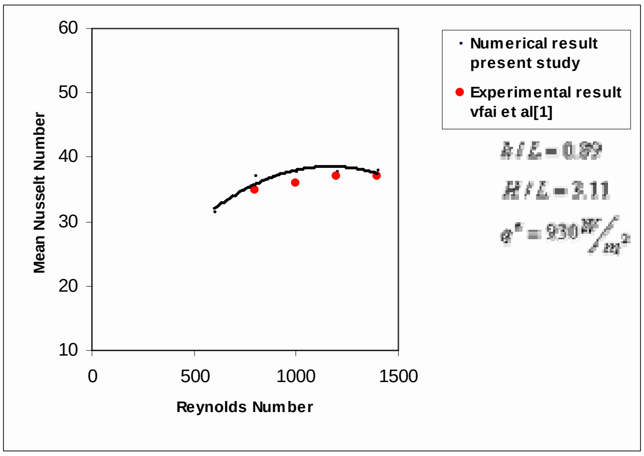
Numerical Procedure

In our numerical analysis, finite volume method and SIMPLE algorithm have been used.

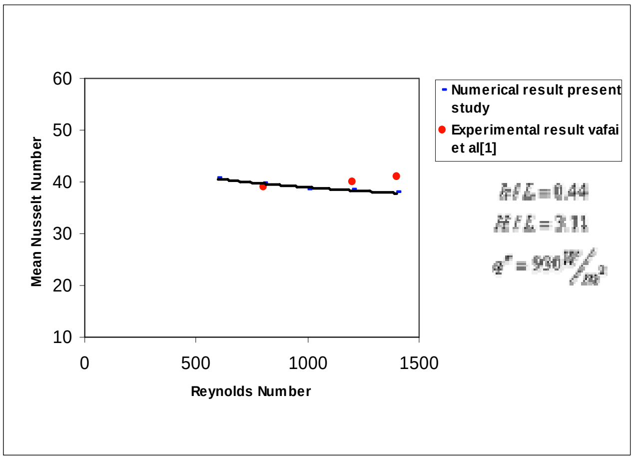
An orthogonal mesh is used through the channel. This mesh is finer near to obstacles to obtain critical properties of regions before, after and on the surface of obstacles and the number of cells is varied from 2000 to 4000 in various steps. It is found that after 3900 cells, further increase in cells has less than 3% variation in Nusselt number value which is taken as criterion for grid independency. In numerical solution of single obstacles in a channel some different sections are defined to calculate fluid mean temperature, Then local convective heat transfer coefficient will be calculated and average of this values result in the h_m in the obstacles length. Calculated Mean Nusselt number is compared with Vafai's data that are shown in figure 2.

Should be noted that figure 2(a) shows the results for h/L=0.89 and H/L=3.11 and figure 2(b) Is also for h/L =0.44 and H/L =3.11.

As shown in the figures, Nusselt values calculated in our paper have a similar trend to Vafai's experimental results.



(Fig. 2-a)



(Fig. 2-b)

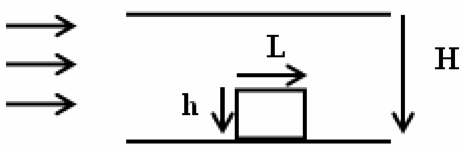


Fig. 2 (a,b): Mean Nusselt number, comparison between the experimental and present numerical results

Multiple Obstacle Arrays:

The numerical analysis was utilized to compare the mean convective heat transfer behaviour for the system–like configuration of an array of three heated obstacle of height h/L =0.44and inter-obstacle spacing s/L=0.44 within a channel of height H/L=3.11. For the experiments the input heat flux was $930 \, {}^{W}\!/_{m^2}$ while the airflow was kept within the laminar regimes. Figure 3 shows for the three obstacles, the mean Nusselt number comparison between the experimental and numerical analyses.

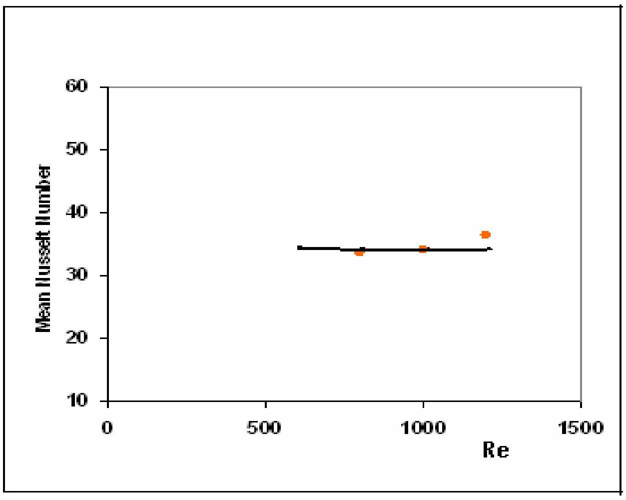


Fig.3a: Mean Nusselt Number, h/L=0.44, s/L=0.44, H/L=3.11, q”=930 W/m^2, heat Flux imposed on obstacle1

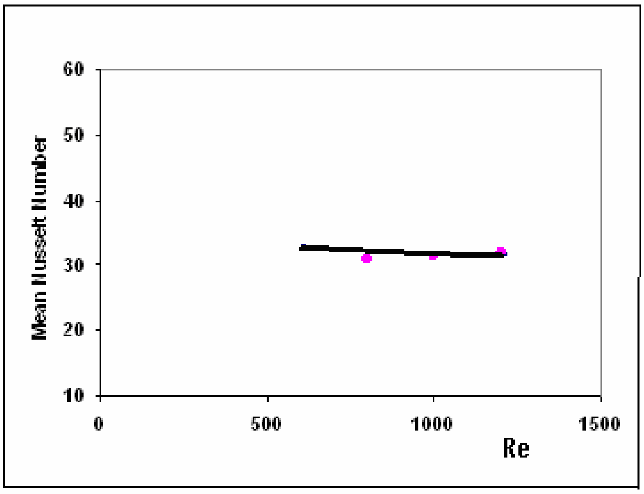


Fig.3b: Mean Nusselt Number, h/L=0.44, s/L=0.44, H/L=3.11, q”=930 W/m^2, heat Flux imposed on obstacle 2

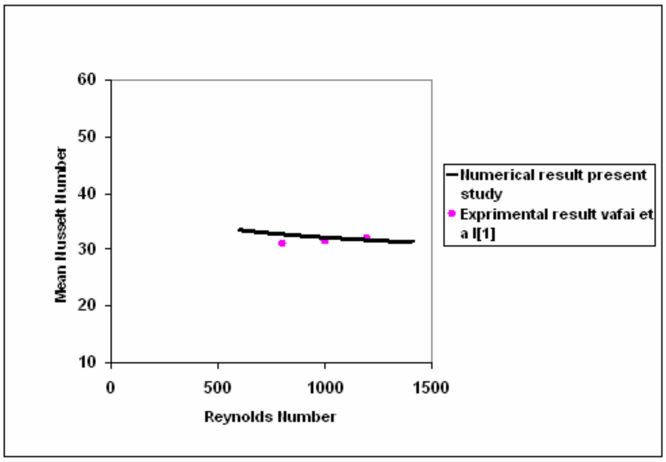
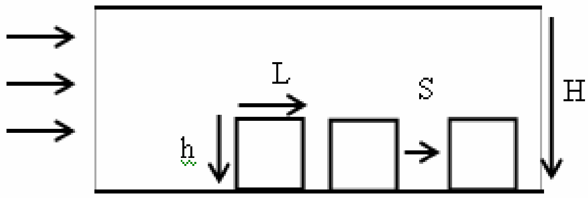


Fig.3c: Mean Nusselt Number, h/L=0.44, s/L=0.44, H/L=3.11, q”=930 W/m^2, heat Flux imposed on obstacle 3



Conclusions

In this paper, incompressible fluid flow in a channel has been studied two dimensionally and the effects of obstacle arrangement on heat transfer have been studied.

The heights of channel and obstacle size are variables to find out the more properly model for description of heat transfer’s improvement in a channel. The numerical finite volume method has been developed and the Mean Nusselt number has been calculated through the channel incorporating the effects of obstacle geometry. The numerical results have good agreement with former experimental and numerical data and the obstacle lead to change the mean Nusselt number in the channel. These numerical results can be used in design and fabrication of heat exchangers and the rate of heat transfer can be predicted exactly.

References

[1]K.Vafai, T.J, Young, *Experimental and Numerical Investigation of Forced Convective Characteristics of Arrays of Channel Mounted Obstacles*, ASME JOURNAL of HEAT TRANSFER,Vol.121, pp.34-42, 1999.

[2] P.T.Roeller, J. Stevens, B.W. Webb, *Heat Transfer and Turbulent Flow Characteristics of Isolated Three-Dimensional Protrusions in Channels*, ASME JOURNAL of HEAT TRANSFER,Vol.113,pp.597-603, 1991.

[3] B.A. Jubran, S.A. Swiety, M. Hamdan, *Convective heat transfer and pressure drop characteristics of various array configurations to simulate the cooling of electronic modules*, Int.J.Heat andMAss Transfer, Vol.39, pp.3519-3529, 1996.

[4] E.M. Sparrow, A.A Yanezmoreno, D.R. Otis, *Convective heat transfer response to height differences in an array of block –like electronic components*, Int.J.Heat and Mass Transfer,Vol.27,pp,469-473, 1984.

[5] J. Davalath, Y. Bayazitoglu, *Forced Convection Cooling Across Rectangular Blocks*, ASME JOURNAL OF HEAT TRANSFER ,Vol.109, pp.321-328, 1987.