# The Effect of Baffles Length on the Forced Convection Heat Transfer over Multiple Heate Blocks Installed in a Horizontal Channel

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*Abstract—A numerical study of convection heat transfer over multiple heated blocks installed in a horizontal channel is performed. The channel contains long baffles in the upper wall. The air (Pr=0.71) is used as a colling fluid. The thermal physical properties are considered constants. The width and position of the baffles are kept constant during the study. The calculations are run for Re=100 and different baffles lengths. The mathematical equations governing the physical phenomenon are solved numerically using the finite volume method with Ansys Fluent © software. The calculations are executed in the steady state. The results demonstrate that increasing the length of the baffle changes the fluid flow structure and improves the heat transfer over the heated blocks.*

*Keywords—Electronic cooling, Finite volume method, heated blocks, heat transfer enhancement.*

## I. Introduction

The temperature increase presents a big common problem for all engineering fields such as nuclear power, electronics, mechanics, etc. The increase of the temperature to high values puts the engineering product in a bad functioning state and decreases its lifetime. To prevent this problem, thermal engineers have to invent and develop new strategies that provide a better cooling process. A lot of papers were written to treat this problem. Bergles et al [1] introduced 13 techniques to make heat transfer better in industrial systems. Yeh [2] briefly write a summary of the cooling techniques used in the electronic field. An important research effort about heat transfer over multiple heated blocks is studied in [3-8]. Herman and Kang [9] used curved deflectors to conduct the flow into space between the blocks and force the trapped warm air there to move away. Their results show a significant heat transfer enhancement but the price of pressure loses about two to three times higher than the case without curved deflectors. Besides, they find that the heat transfer increase with the increase of Reynolds number. The same shape of curved deflectors is tested numerically and experimentally by Lorenzini-Gutierrez et al. [10] and Luviano-Ortiz et al. [11]. Their results confirmed that the heat transfer is improved with the curved deflectors but with

the price of pressure loss. In another study, a solid and porous bar is used by Perng and Wu [12] and Perng et al. [13], respectively. The bar is installed above the first block. Their results demonstrate that a solid or a porous bar plays a significant role in heat transfer enhancement. Mebarki et al [14] studied the effect of fixing a short bar in the upper wall of a horizontal channel contains multiple heated blocks. Their results show that the bar changes the flow structure and improves the heat transfer over the heated blocks. In this work, a long baffles are used to increase the convection heat transfer over multiple heated installed in an horizontal channel. The objective is to investigate the effect of the baffle length as a new parameter on the fluid flow structure and heat transfer.

## II. Ease of Use Physical Model and Mathematical Formulation

### A. Setups and Description

The physical domain investigated in this research paper is presented in Fig.1. The configuration is based on a 2D horizontal channel with five heated blocks of similar size (w=h=0 25). The inlet and outlet distances are chosen to be $L_{in}=3$ and $L_{out}=20$, respectively. A long baffle is fixed above each block at the top wall of the horizontal channel. The horizontal distance between the baffle vertical centerline and the rear face of the next block is fixed at L=0.125. The baffles width is w=0.025. A uniform heat flux "q=1" and a forced flow with a parabolic profile are applied at the base of each block and the inlet of the channel, respectively.
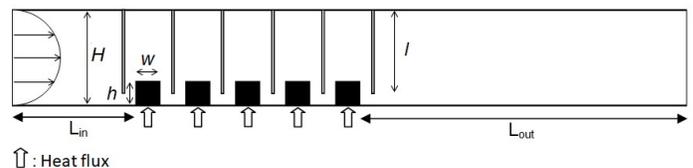


Fig. 1  Physical domain channel.

The flow is assumed to be steady, laminar and incompressible; the fluid is Newtonian with constant thermo-

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physical proprieties. The buoyancy and viscous dissipation are neglected. Thus, the mathematical equations of the physical model in the non-dimensional form can be written as follows:

Mass:

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

x-momentum:

$$Re\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \qquad (2)$$

y-momentum:

$$Re\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \qquad (3)$$

Energy:

The fluid phase:

$$P_e\left(u\frac{\partial \theta_f}{\partial x} + v\frac{\partial \theta_f}{\partial y}\right) = \left(\frac{\partial^2 \theta_f}{\partial x^2} + \frac{\partial^2 \theta_f}{\partial y^2}\right) \qquad (4)$$

The solid phase:

$$\frac{k_{sl}}{k_f}\left(\frac{\partial^2 \theta_{sl}}{\partial x^2} + \frac{\partial^2 \theta_{sl}}{\partial y^2}\right) = 0 \qquad (5)$$

Non-dimensional variables:

$$\begin{cases} x = \frac{x^*}{H^*}; \ y = \frac{y^*}{H^*}; \ u = \frac{u^*}{u_m^*}; \ v = \frac{v^*}{u_m^*}; \\ \theta = \frac{(T - T_0)}{(q". H^*/k_f)}; \ p = \frac{p^*.H}{\mu_f.u_m^*} \end{cases} \qquad (6)$$

And the relevant non-dimensional numbers are:

$$Re = \frac{\rho_f.u_m.H}{\mu_f}; \ Pr = \frac{\mu_f.c_{p_f}}{K_f}; \ Pe = Re.Pr \qquad (7)$$

## B. Boundary Conditions

The boundary conditions are summarized in table 1:

TABLE I BOUNDARY CONDITIONS.

| | Border of the geometry | Boundary conditions |
|---|---|---|
| **Hydrodynamic conditions** | Inlet | $\frac{\partial p}{\partial x} = 0$; $U_{inlet} = 6y(1 - y)$; $v = 0$ |
| | Outlet | $\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = 0$; The pressure is equal to the ambient pressure |
| | Channel walls Block bases Solid-Fluid interface | $u = v = 0$ |
| **Thermal conditions** | Inlet | $\theta_f = 0$ |
| | Outlet Channel walls | $\frac{\partial \theta_f}{\partial x} = 0$ |
| | Block bases | $q" = 1$ |
| | Solid-Fluid interface | $\theta_f = \theta_{sl}$; $k_f\frac{\partial \theta_f}{\partial n} = k_{sl}\frac{\partial \theta_{sl}}{\partial n}$ |

## C. Numerical Solution and Validation

The governing equations of the aforementioned physical model are solved numerically through the application of the finite volume method. The simulation is conducted using Ansys Fluent® software, utilizing the Simple algorithm developed by Patankar [15]. The second-order upwind and central second-order differencing schemes are chosen for the convective and diffusive terms respectively. The grid independence is checked for four grid sizes 590×45, 935×75, 1350×110, and 1950×145 at Re=1000. The choice of the grid distribution of 1350×110 is judged to be sufficient for the Reynolds number investigated here, where the difference between local Nusselt values obtained using the two mesh sizes 1350×110 and 1950×145 does not exceed 2% (fig. 2a). The validity of the calculations is investigated by comparison with the work of Young and Vafai [4]. As it appears in fig. 2b, a great agreement is obtained and the maximum deviation is less than 3%. All calculations are run iteratively in the steady state until reaching the convergence state. The residuals for all independent variables are fixed at $10^{-6}$.
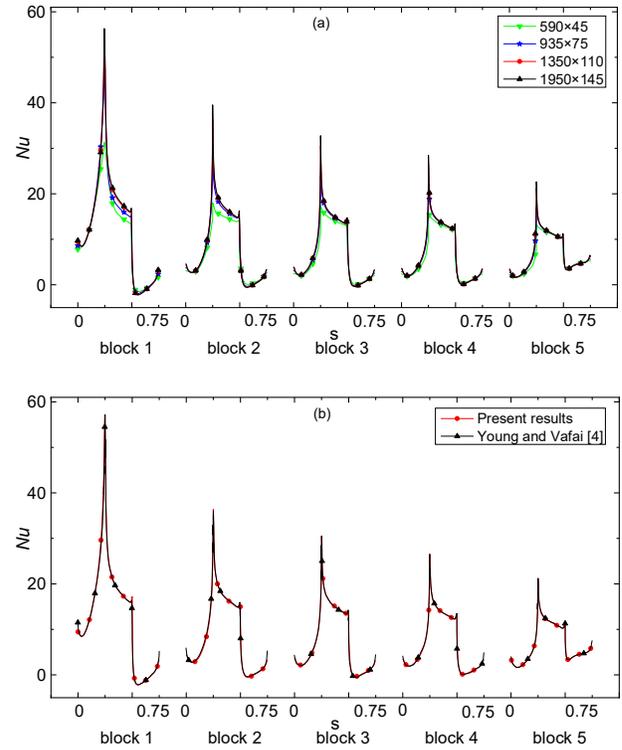


Fig. 2 Validation and grid independence: (a) Grid independence (b) Comparison of present results with those of Young and Vafai [4].

## III. RESULTS AND DISCUSSION

The calculations are performed for different baffles length l =0.625, 0.750 and 0.875 at Re=100. The results are summarized as streamlines temperature contours and mean Nusselt number (eq. 9).

$$Nu_x = \frac{h_c.H^*}{k_f} = -\frac{1}{\theta_s}.\frac{\partial \theta_f}{\partial n} \qquad (8)$$

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$$\overline{\mathrm{Nu}} = \frac{1}{A} \int_A \mathrm{Nu}_s \, ds \qquad (9)$$

$h_c$ is heat transfer coefficient and "$n$" denotes the normal coordinate.

## A. Streamlines

The effect of the length of the baffles is elucidated by simulations for three values of the length, l=0.625, 0.750, and 0.875 at one Reynolds number value (Re=100). The streamlines for the three lengths at Re=100 are given in fig. 3. As it appears, increasing the length of the baffle causes the flow to penetrate further into the cavity between the blocks, thus reducing the amount of trapped fluid in these areas. For l=0.875, only a small recirculation zone is observed in the vicinity of the block back stream and a smaller one on the front stream lower corner of the next block. Moreover, the vortex observed on the front stream of the first block has completely vanished and the size of the vortex located behind the last block is considerably reduced. On the other hand, increasing the length of the baffles from l=0.625 to l=0.750 gives rise to a recirculation zone above the blocks, which increases more in size for l=0.875.

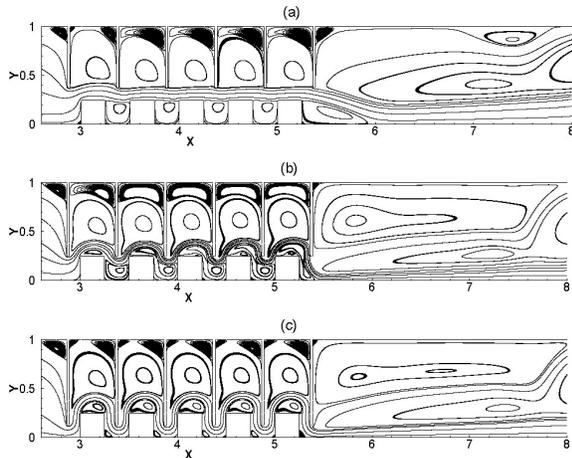Fig. 3 Stream lines for Re=100 and different baffle lengths: (a): l=0.625; (b): l=0.750; (c): l=0.875.

## B. Isotherms contours

As demonstrated by fig. 4, the reduction of the boundary layer produced by the increasing baffles' length is followed by a reduction of the temperature of the heated blocks. In addition, the penetration of the flow towards the cavities allows to increase the convective heat transfer from the hot faces of the heated blocks situated between the blocks. For the cases of l=0.750 and l=0.875, the temperature at the blocks' surface is slightly increased with the block number in the flow direction. it seems like all the blocks have the same temperature; in some way, the good flow mixing provides balanced cooling for all surfaces, unlike the case of l=0.625.
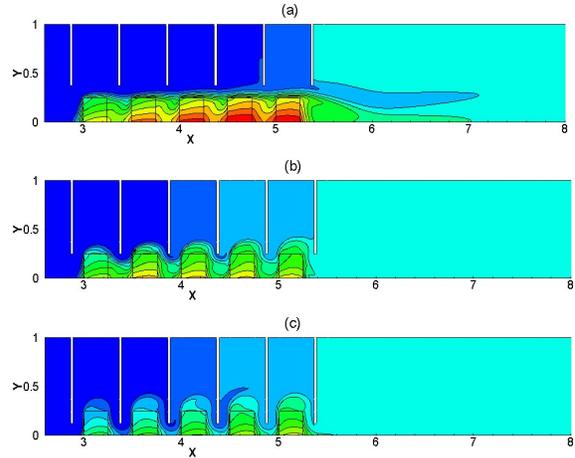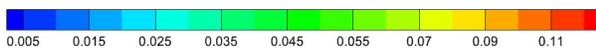
Fig. 4 Isotherms contours for Re=100 and deferments baffles lengths: (a): l=0.625; (b): l=0.750; (c): l=0.875.

## C. Mean Nusselt Number

The mean Nusselt numbers obtained for four different baffle lengths at Re=100 are illustrated in Fig. 5, with additional curves included for the case without baffles at Re=100 and 1000 for comparison.

From this figure, it is evident that increasing the length of the baffles consistently leads to higher Nusselt numbers for each block. For the baffles with a length of l=0.625, the mean Nusselt number surpasses that obtained at Re=1000 without baffles, except for the first block positioned in front of the fresh air. Comparing this with the scenario of Re=100 without baffles, the increase is substantial and occurs consistently across all blocks and baffle lengths. Notably, the greatest increase is observed for the first block, where the mean Nusselt number nearly quadruples between the cases of Re=100 without baffles and Re=100 with baffles of length l=0.875.

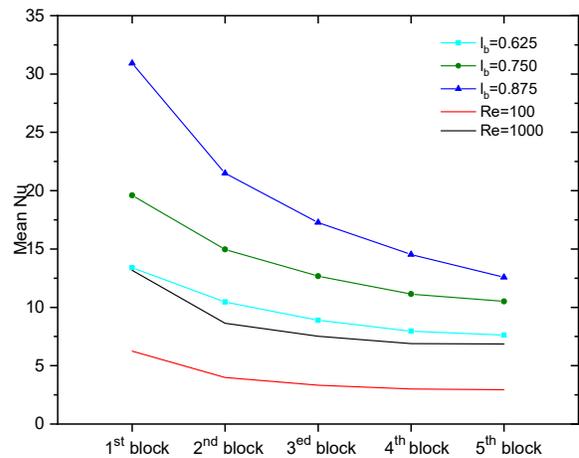Fig. 5 Effect of baffles length on the mean Nusselt number at *Re=100*.

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## IV. Conclusions

In this study, the forced convection heat transfer over five heated blocks with long baffles installed in horizontal channel, is studied numerically. The calculations are performed for Re=100 at constant baffles width (d=0.025) and different baffles length (l=0.625, 0.750 and 0.875). The main observations are briefly summarized as follows: The increase of the baffles length decrease the size of each vortex situated after each block. A new vortex appears above each block at l=0.750 then increase in size for l=0.875. Concerning temperature contours, the temperature in the solid and fluid phases is decreased significantly with increasing the baffles length. For the heat transfer quantification, the Nusselt number of each block increase with increasing the baffles length.

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