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The Effect of Flow Deflector Shapes on the Heat Transfer Over Multiple Heated Blocks Installed in a Horizontal Channel

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Abstract: *A numerical investigation is carried out to explore convective heat transfer in a horizontal channel featuring multiple heated blocks. The configuration incorporates flow deflector positioned downstream of each block. The cooling of the system is achieved using air (Prandtl number = 0.71) with constant thermal properties. Throughout the study, the dimensions and placement of the flow deflectors remain unaltered. The computations focus on a Reynolds number of 400, with variations in the geometries of the flow deflectors. Employing the finite volume method through Ansys Fluent © software, the governing mathematical equations of the thermal phenomenon are solved numerically. The outcomes reveal a negligible influence of the flow deflectors shapes on both fluid flow patterns and heat transfer characteristics across the heated blocks.*

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

1. INTRODUCTION:

The challenge of temperature increase is a prevalent issue across diverse engineering domains, encompassing nuclear power, electronics, mechanics, and more. Temperatures increase not only compromise the optimal functioning of engineering products but also curtail their operational lifespan. To counter this challenge, thermal engineers are compelled to innovate and devise novel technics and strategies that facilitate superior cooling processes. This conundrum has sparked extensive research efforts, yielding a corpus of scholarly contributions. For instance, Bergles et al. [1] delineated thirteen techniques geared towards enhancing heat transfer efficiency in industrial systems. Yeh [2], meanwhile, succinctly summarized cooling techniques deployed in the realm of electronics. There is a lot of other research articles investigated various methods for enhancing convective heat transfer in channels. The goal of these studies is to improve heat dissipation in applications like electronic cooling and heat exchangers. Focusing on configurations with obstacles, deflectors, and flow control devices, Herman and Kang (2002) [3] and Luviano-Ortiz et al. (2008) [4], investigated heat transfer enhancement in grooved channels with curved vanes. The curved vanes were employed to alter the flow pattern and increase heat transfer efficiency. Lorenzini-Gutierrez et al. (2015) [5] conducted numerical and experimental analyses of heat transfer enhancement in a grooved channel with curved flow deflectors. Both computational and practical investigations were carried out to optimize heat transfer performance. Rosas et al. (2017) [6] conducted an experimental investigation into convective heat transfer enhancement in an electronic module using curved deflectors. There findings, demonstrate that the implementation of curved vanes enhances heat transfer efficiency over heated blocks. Oztop et al. (2009) [7] explored heat transfer and fluid flow control using a triangular bar in heated blocks located within a channel. The study aimed to optimize both heat transfer and fluid dynamics. The find that the insertion of a triangular cross-sectional bar enhanced heat transfer for all Reynolds numbers tested. The best heat transfer was observed when the bar was positioned at y = 3.5 in the channel. Beig et al. (2011) [8] investigated the optimal positioning of a triangular vortex generator in a blocked channel to enhance heat transfer for electronic chips. The goal was to optimize heat dissipation in confined spaces. The optimization results showed that the optimal position of the vortex generator was found to be above the first block when uniformity was neglected, but it shifted to the top of the second block when uniformity became more important. The optimal position of the vortex generator was found to be independent of the Reynolds number. Placing the vortex generator between the second and third blocks resulted in a more uniform temperature distribution in the channel compared to other positions. Wu and Perng (1999) [9] studied the effect of an oblique plate on heat transfer enhancement and mixed convection over heated blocks in a horizontal channel. The results showed that the installation of an oblique plate above an upstream block effectively enhanced the heat transfer performance of mixed convection in the horizontal channel flow. Based on the previous studies, it's clear that many researchers have focused on improving heat transfer over heated blocks using different types and shapes of flow deflectors. In this work, the goal is to explore new possibilities by investigating various new shapes and forms of flow deflectors that haven't been investigated before.

2. PHYSICAL MODEL MATHEMATICAL FORMULATION:

A. Setups and Description:

Figure 1 illustrates the physical model examined in this study. The geometric configuration of the system consists of two parallel plates (2D), in which five heated blocks are arranged. All dimensions are dimensionless, in relation to the width of the channel H. The dimensions of the heated blocks are equivalent (w=h=H/4=0.25). In addition, multi-shaped flow deflector is installed behind each block. The shape and the dimensions of the flow deflectors are presented in the figure 2. a deflector is positioned with a distance of h/2 after each block in the longitudinal direction and with a distance h in the transversal direction. The channel walls are adiabatic, with the exception of the base of the heated blocks, which are subjected to uniform heat flux. A forced flow is applied at the channel inlet (u_{inlet}). The length of the channel before the blocks and after the blocks are chosen to be L_{in}=3 and L_{out}=20, respectively. The third dimension (the direction normal to the figure) is too large compared with the other dimensions, so we consider the problem to be two-dimensional (2D).

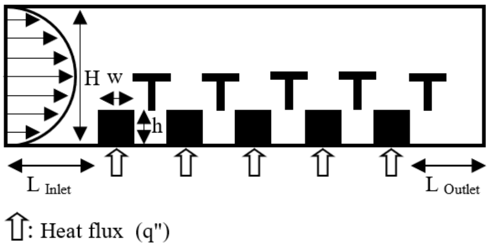

Fig. 1. Configurations of the studied problem.

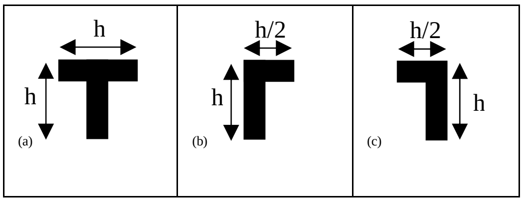

Fig. 2. Geometric shapes of the deflectors used in thisstudy: (a): T ; b): L’Right; (c): L’Left.

Presuming a steady, laminar and incompressible flow, the fluid is characterized as adhering to Newtonian principles with constant thermo-physical properties. Disregarding the effects of buoyancy and viscous dissipation, the ensuing mathematical expressions for the physical model in a non-dimensional format can be structured as follows:

:

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

x-momentum:

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

y-momentum:

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

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:

$$x=\frac{x^*}{H^*};\ y=\frac{y^*}{H^*};\ u=\frac{u^*}{u_m^*};\ v=\frac{v^*}{u_m^*};\ \theta=\frac{(T-T_0)}{(q''.H^*/k_f)};\ p=\frac{p^*.H}{\mu_f.u_m^*} \qquad (6)$$

And the relevant non-dimensional numbers are:

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

B. Boundary Conditions

Table 1. succinctly presents the key boundary conditions

Table 1. Boundry 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:

By applying the finite volume method, the numerical solution of the governing equations of physical model is obtained. Ansys Fluent® software is chosen, to implement the Simple algorithm. the local Nusselt values are used in verifying grid autonomy and ensuring calculation accuracy across various scenarios. Through rigorous analysis in previous works [10], it's established that the grid layout of 1,350 ×110 satisfactorily meets the demands for Re=400. Moreover, the comparison with the findings of Young and Vafai reveals a high level of concurrence with a negligible deviation below 3%.

3. RESULTS AND DISCUSSION:

Various deflectors shapes, namely T, L'Right, and L'Left, are tested through computational calculations at a Reynolds number of 400. The findings encompass streamline distributions, temperature contours, and the mean Nusselt number (as defined in equation 9).

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

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

"hc" denotes the heat transfer coefficient, and "n" specifically represents the normal coordinate in the context of this investigation.

A. Streamlines

The effect of deflector shapes is illustrated in Figures 3. As depicted in the figures, the flow structure does not change based on the shape of the deflectors, except for the vortices located on the upper face of the blocks. As it appears a vortex is appeared above each block except the first. For the all-deflectors shapes, the vortex situated above the second block is noted as the biggest one then it sizes decrees for the rest of blocks. The vortexes are occupying all space of upper faces. The change of positions and sizes of the vortices are due to the flow orientation controlled by the shape of the deflectors. And as shown in Figure 3, the effects only occur for the vortices located above the blocks and the vortices located after the first and the last deflectors. For the cavities between the blocks, the flow penetrates this zone totally which provide a better heat transfer unlike the first cavity after the first block a big vortex is created. These phenomena lead to heat accumulation in this zone.

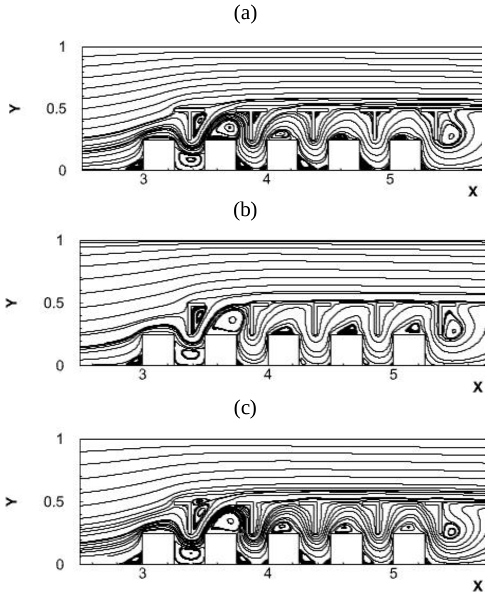

Fig. 3. Stream lines for Re=400 and Various deflectors shapes: (a): T; (b): L'Right; (c): L'Left.

B. Isotherms Contours

For the different forms of deflectors and as shown in Figure 4, the change in the shape of the deflectors has a negligible effect on improving the cooling state of the heated blocks. The isothermal contours of the L'Right and L'Left shaped deflectors are approximately the same, but there is a negligible change when compared to the T-shaped deflectors; the temperature of the blocks increases slightly. It is important to note that the L'-right and L'-left shaped deflectors exhibit the best cooling state; approximately, both (L'-right and L'-left) offer the same cooling state.

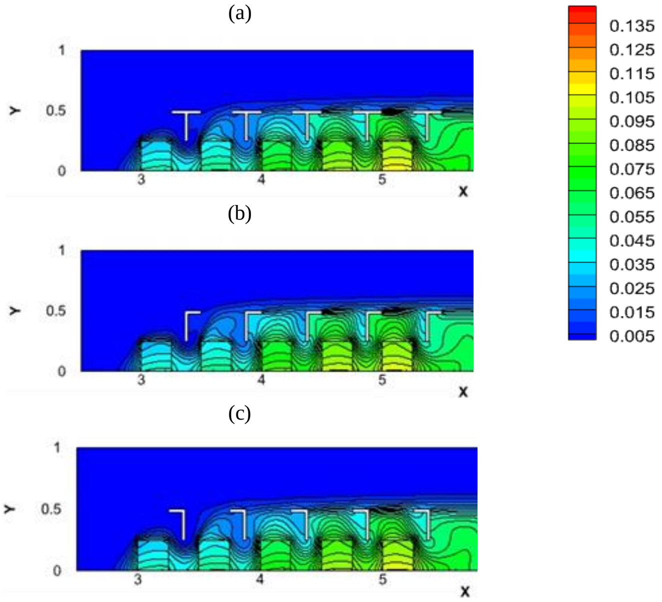

Fig. 4. Isotherms contours for Re=400 and Various deflectors shapes: (a): T; (b): L'Right; (c): L'Left.

C. Mean Nusselt Number

As depicted in the figure 5, there is a decreasing trend in the mean Nusselt number as we progress through the blocks, with the highest Nusselt number observed for the first block and the lowest for the last block. The difference in the mean Nusselt number between the last two blocks for L'Right and L'Left shapes is negligible. For the effect of deflector shapes on the mean Nusselt number, the maximum value of the average Nusselt number is recorded for T-shaped deflectors, while for the last block, the minimum Nusselt number is obtained for T-shaped deflectors. For the other two shapes, it appears that they reach approximately the same mean Nusselt number for all the heated blocks.

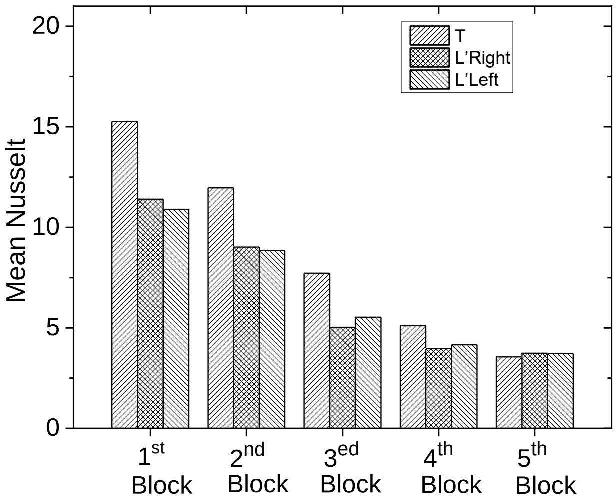

Fig. 5. Mean Nusselt numbers for each block, at Re=400 and various flow deflector shapes.

4. CONCLUSIONS:

In this study, the forced convection heat transfer over five heated blocks with flow deflectors installed in horizontal channel, is studied numerically. The calculations are performed for Re=400 at deferent deflectors shapes width. The main observations are briefly summarized as follows: The effect of deflector shape on streamlines is negligible; deflector shapes only affect the vortices above the blocks. It is important to note that vortices are created on the upper faces of the heated blocks. Also, the effect of deflector shape is insignificant; the isotherms are identical for all the deflectors shapes. The geometry of the deflectors reduces flow mixing, leading to poor heat transfer for the last three blocks. Regarding the impact of deflector shapes on the mean Nusselt number, T-shaped deflectors yield the highest mean Nusselt number. The remaining two shapes seem to achieve a similar mean Nusselt number.

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