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Numerical Analysis of Fully Developed Flow and Heat Transfer in Channels with Periodically Grooved Parts

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ABSTRACT

To obtain a higher heat transfer in the low Reynolds number flows, wavy channels are often employed in myriad engineering applications. In this study, the geometry of grooves shapes is parameterized by means of four angles. By changing these parameters new geometries are generated and numerical simulations are carried out for internal fully developed flow and heat transfer. Results are compared with those of rectangular grooved channel. Two different Prandtl numbers, i.e. 0.7 and 5, were investigated while Reynolds number varies from 50 to 300. An element-based finite volume method (EBFVM) is used to discretize the governing equations. Results reveal that that both heat transfer performance and average Nusselt number of rectangular grooved channel were higher than those of other geometries.

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1. INTRODUCTION

Heat transfer enhancement in laminar flow has been special attention in different engineering sectors. In fact, the cases are compact heat exchangers, microelectronic equipment packages, medical and biochemical engineering. Enhancement techniques can be separated into two categories [1]: passive and active. The passive methods require no direct application of external power. On the other hand, active schemes required external power for operation. The passive methods are preferred over the active methods because of those are more realistic and inexpensive. The channel with variable streamwise cross-sections is one of such passive methods that can be used to promote heat transfer. It is well known that increase in heat transfer rate is accompanied by an even larger pressure drop. Therefore, the two main objectives are aiming to maximize the walls heat transfer and minimize pressure drop.

In previous studies, different wall corrugation shapes were used such as sinusoidal, arc-shape, V-shape, trapezoid and rectangular grooved. In addition, the corrugations on the top and bottom walls of the channel can be shifted in space relative to each other. For each case, previous numerical studies considered 1laminar, transitional, and turbulent flow regimes, two-dimensional and three-dimensional domains, steady and unsteady solution approaches and different boundary conditions. In follow, some of those studies are mentioned.

Sinusoidal: Nishimura et al. [2, 3] performed experimental and numerical analysis of channel with sinusoidal wavy walls. The Reynolds number for experiments was up to 10000 and for numerical study was up to 300. Rush et al. [4] experimentally investigated laminar and transitional flows in sinusoidal wavy passages with or without shifting between up and down walls. The Reynolds number for experiments was up to 1000. Wang and Vanka [5] studied numerically the fluid flow and heat transfer through sinusoidal-shaped channels. According to their findings, flow do not provide significant heat transfer if operated in steady regime. After a critical Reynolds, self-sustained oscillatory flow is observed and a relevant increase of the heat transfer rate is reported. Niceno and Nobile [6] have analyzed 2D steady and unsteady laminar flow in sinusoidal and arc-shaped channels. In the arc-shaped channels, flow reaches its unsteady mode in lower Reynolds number compared with sinusoidal channels.

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Metwally and Manglik [7] and Zhang et al. [8] numerically investigated the effects of waviness configuration; its spacing and Reynolds number on flow and heat transfer characteristics in channels with sinusoidal channel for steady and laminar flow regime. Ramgadia and Saha [9] solved time-dependent Navier–Stokes and energy equation through a wavy channel. Effect of geometry, i.e. minimum and maximum height between two wavy walls, on fluid flow and heat transfer characteristics elaborated at a Reynolds number of 600. Pashaie et al. [10]developed an adaptive neuro-fuzzy inference system to determine Nusselt number along a sinusoidal wavy wall in a lid-driven cavity.

V-shape: Wirtz et al. [11] and Hamza et al. [12] reported performance of the channels having a series of V-grooves formed on two and one walls. Zimmerer et al. [13] studied effects of the geometric parameters such as inclination angle, the wavelength, and amplitude in a channel having V-corrugated upper plates. Islamoglu et al. [14-16] and Naphon [17] studied numerically and experimentally laminar and turbulent flow in V-shape channel. Deylami et al. [18] numerically investigated the effects of profiles of V-shaped corrugated channels on the heat transfer and friction characteristics.

Trapezoid: Farhanieh and Sunden [19] investigated numerically heat transfer and fluid flow for channels with trapezoidal grooved parts for laminar and steady state flow. Naphon [20] showed that the sharp edge of trapezoid grooved part has a significant effect on the flow structure and heat transfer enhancement.

Rectangular grooved: Ghadder et al. [21], Sunden and Trollheden [22] and Pereira and Sousa [23] analyzed convective heat transfer in channels with rectangular grooves on one plate. They showed complex flow patterns such as separation, re-attachment and deflection. Especially, Ghadder et al. found the presence of self-sustained ossilationary flow. Adachi and Uehara [24] studied fluid flow and heat transfer in channel with contracted and expanded grooves set up both symmetrically and asymmetrically with the centerline of the parallel plates. Li et al. [25] investigated the fully-developed flow and heat transfer in channels with periodically rectangular grooved parts using an unsteady model. Some investigation has devoted to analyze the flow and heat transfer in channels with irregular-shaped parts (Fabbri, [26] and Nobile et al. [27]).

In the present work, three new grooves geometries for channel were studied and compared with simple rectangular grooves, shown in Figure 1. The numerical simulations were carried out to investigate the fully developed flow and heat transfer characteristics in such channels. The flow regime was laminar and two types of fluids with different Prandtl numbers were investigated. The results showed that the channel with rectangular grooves had better thermal performance compared with others.

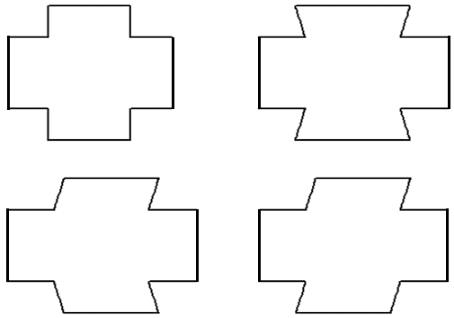


Figure 1. Configuration of evaluated channels

The geometry parameterization and formulation of the problem is presented in Section 2. Brief explanations regarding the flow solver used in this study and validation of solver are given in Section 3. This is followed by computational results in Section 4. Last, in Section 5 the conclusions of this work are presented.

2. PROBLEM STATEMENT

The problems considered in this article are the investigation of fluid flow and heat transfer in convective channels at fully-developed flow and heat transfer conditions. For the periodically developed flow conditions, as treated by Patankar et al. [28], the computational domain is concerned to a typical cycle of the entire geometry. In this work we have used the same conditions expressed in literature [28]. We have used steady and laminar flow regime. The range of Reynolds number is chosen for the simulation up to 300 and 200 for Prandtl numbers 0.7 and 5, respectively.

2. 1. Geometry Modeling

The parameters required for the definition of the channel geometry are showed in Figure 2. L denotes the period of the channel, S is the length of grooves, h is the height of the grooves from the channel wall and H is the height of the parallel plane channel. ψ_1, ψ_2, ψ_3 and ψ_4 are angles between grooves lines with longitudinal axis. The total length of each period section, length and height of grooves are fixed. By changing the angles, seven geometries are obtained as shown in Figure 3 which are classified in four sections, as follows:

Section 1: $\psi_1 = 90^\circ$, $\psi_2 = 90^\circ$, $\psi_3 = 270^\circ$ and $\psi_4 = 270^\circ$

Section 2: ψ_1 and $\psi_2 < 90^\circ$, ψ_3 and $\psi_4 > 270^\circ$

Section 3: $\psi_1 < 90^\circ$, $\psi_2 > 90^\circ$, $\psi_3 < 270^\circ$, $\psi_4 > 270^\circ$

Section 4: ψ_1 and $\psi_2 < 90^\circ$, ψ_3 and $\psi_4 < 270^\circ$

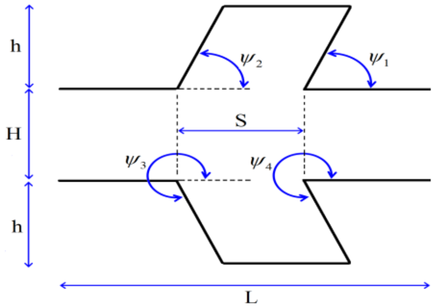


Figure 2. Geometrical configuration of one periodic unit of a channel

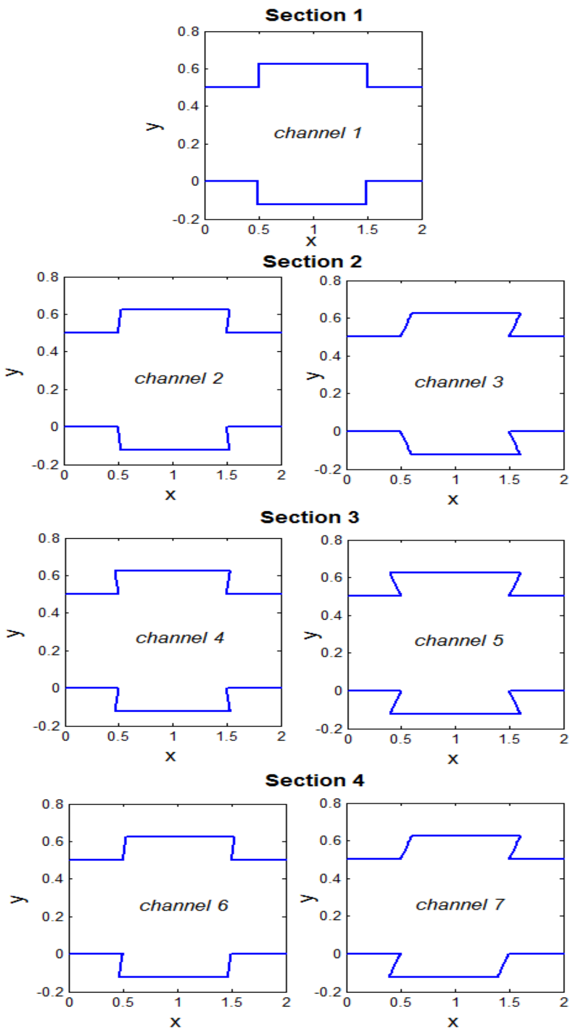


Figure 3. Geometry of evaluated channels

2. 2. Governing Equations The flow is assumed to be two-dimensional, incompressible and constant thermo-physical properties. With these assumptions, the conservation equations for mass, momentum and energy become, respectively:

$$
\begin{aligned}
&\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \\
&\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \\
&\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \\
&u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)
\end{aligned}
\tag{1}
$$

where, u=(u,v) is the velocity field, p is pressure, T is temperature, ρ is the fluid density and μ and α are the kinematic viscosity and thermal diffusivity, respectively.

2. 3. Boundary Conditions and Computational Details For a streamwise periodic geometry, the flow is expected to attain a periodic fully developed regime. In other words, the flow pattern repeats itself from module to module. It is sufficient to analysis only one module of the geometry. This condition can be applied at the inflow and outflow boundaries as follows:

$$
\begin{aligned}
&\mathrm{u}(\mathrm{x},\mathrm{y})=\mathrm{u}(\mathrm{x}+\mathrm{L},\mathrm{y}) \\
&\mathrm{v}(\mathrm{x},\mathrm{y})=\mathrm{v}(\mathrm{x}+\mathrm{L},\mathrm{y})
\end{aligned}
\tag{2}
$$

The pressure is subdivided into two components,

$$\mathrm{p}(\mathrm{x},\mathrm{y})=-\beta\mathrm{x}+\mathrm{P}(\mathrm{x},\mathrm{y}) \tag{3}$$

where, β is constant. Then βx term is related to the global mass flow and P(x,y) is related to detailed local motions. It is evident that P is periodic:

$$\mathrm{P}(\mathrm{x},\mathrm{y})=\mathrm{P}(\mathrm{x}+\mathrm{L},\mathrm{y}) \tag{4}$$

The streamwise momentum equation can be written as:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)+\beta \tag{5}$$

In order to formulate the concept of the thermally developed flow, a dimensionless temperature introduce as:

$$\theta=\frac{T(x,y)-T_w}{T_{b,x}-T_w} \tag{6}$$

where, T_w is wall temperature and $T_{b,x}$ is the bulk temperature that defined as:

$$T_{b,x}=\frac{\int|u|Tdy}{\int|u|dy} \tag{7}$$

Similarly, thermal periodic condition is expressed as:

$$\theta(\mathrm{x,y}) = \theta(\mathrm{x+L,y}) \tag{8}$$

Finally, the energy equation becomes as follow:

$$u\frac{\partial\theta}{\partial x} + v\frac{\partial\theta}{\partial y} - \alpha\left(\frac{\partial^2\theta}{\partial x^2} + \frac{\partial^2\theta}{\partial y^2}\right) = \sigma \tag{9}$$

where,

$$\sigma = \left[2\alpha\frac{\partial\theta}{\partial x} - u\,\theta\right]\frac{dT_{b,x}/dx}{T_{b,x} - T_w} + \alpha\theta\frac{d^2T_{b,x}/dx^2}{T_{b,x} - T_w} \tag{10}$$

The Reynolds number in this study is defined as follow:

$$\mathrm{Re} = \frac{\rho u_{av} D_h}{\mu} \tag{11}$$

where, u_{av} is mean velocity in the channel, D_h is the hydraulic diameter, defined as twise the average channel height (H_{ave}). The local Nusselt number is defined as:

$$Nu_x = D_h \frac{\left.\frac{\partial T}{\partial n}\right|_{wall}}{T_{b,x} - T_w} \tag{12}$$

An equivalent Nusselt number can then be defined as the average of Nu_x :

$$Nu = \frac{1}{L}\int_w Nu_x ds \tag{13}$$

The friction factor computing according to its standard definition:

$$f = \frac{\beta.H_{ave}}{2\rho u_{av}^2} \tag{14}$$

Since, improvements in heat transfer are accompanied by increases in the frictional losses, two different factors defined to analyze the performance of proposed channels using the information of Nusselt number and friction factor. The thermal performance factor (TPF) [9], is defined as follows:

$$TPF = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} \tag{15}$$

where, Nu_0 and f_0 are nusselt number and friction factor in a laminar fully-developed flow between parallel isothermal plates with a similar mass flow rate. The flow area goodness factor (j/f) [29], the ratio of Colburn factor (j) to friction factor (f), is second factor, expressed as:

$$j/f = (Nu.\mathrm{Pr}^{1/3}/f.\mathrm{Re}) \tag{16}$$

3. NUMERICAL METHOD

Several numerical techniques have been used for analysis the flow and heat transfer of corrugated channels. Among them, different discretization methods such as finite difference method [23, 29], finite volume method [7, 21] and finite element method [16, 27, 28] are used for discretizing equations. In this study we used element based finite volume method (EBFVM). This type of discretization is used for structure grid.

The computational grid consists of quadrilateral elements with a node located at each element corner, as shown in Figure 4. A control volume is formed for every node. The surface of each control volume consists of planar panels. Integration points are located at the center of each panel where subscript "ip" denotes integration point. Discretization based on EBFVM results in an algebraic balance equation for each control volume. Proper interpolations are then needed to estimate the flux functions at the control surface integration points.

The velocity and pressure fields are linked by the Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm and numerical solver are implemented in a Matlab program.

3. 1. Validation of the Numerical Simulation To guarantee the reliability of the numerical simulations performed in this work, a validation process is carried out. The calculated local Nusselt number distribution along the wall of the sinusoidal shape channel is compared with the numerical results of Wang and Vanka [5] and experimental results of Nishimura [3] in Figure 4. The dimensions of the reference channel are: L=2.8 m, a=0.35 m and $H_{\max}$=2 m. The fluid Prandtl number is 0.7 and the flow Reynolds number in this validation test case is 50. The friction coefficient calculated in this study will 0.505 providing that reference value is 0.49.

4. RESULT

4. 1. Section 1 The first numerical simulation is carried out for fully-developed flow and heat transfer in channel 1 with $\psi_1 = \psi_2 = 90^\circ$ and $\psi_3 = \psi_4 = 270^\circ$. The Prandtl numbers are 0.7 and 5. For Pr=0.7, the Reynolds number is up to 300 and for Pr=5 the Reynolds number is up to 200. The streamlines and isotherms of the flow and temperature field are showed in Figure 6. The vortex grows larger as the Reynolds number increases. The Nusselt number variations on channel wall are shown in Figure 7. As flow reaches to groove a large variation in Nusselt number was observed.

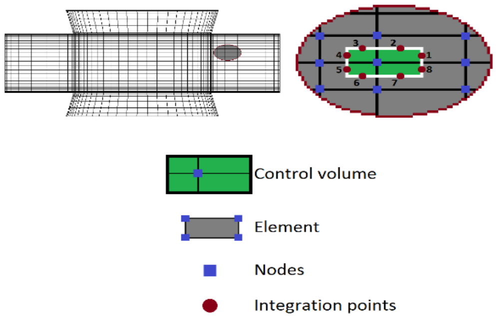


Figure 4. Computational grid

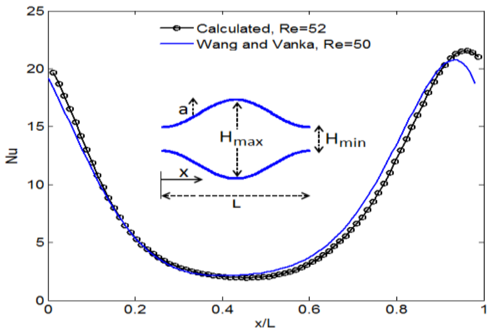


Figure 5. Local Nusselt number along the walls of a Sine-shaped channel

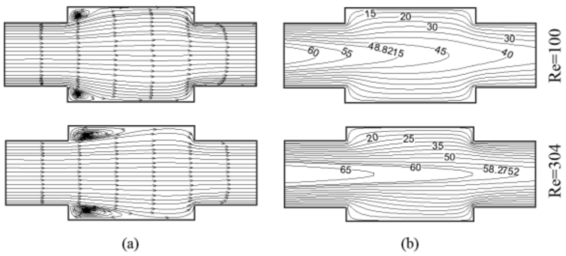
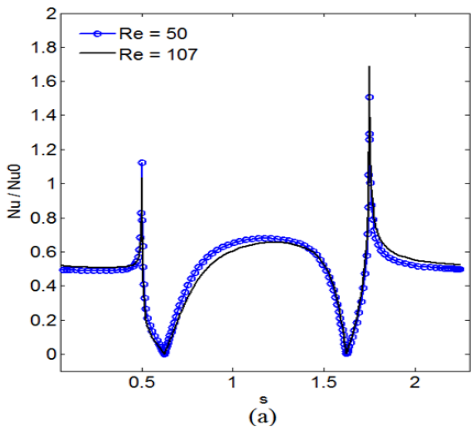


Figure 6. (a) Streamline, (b) Isotherms for the channel 1



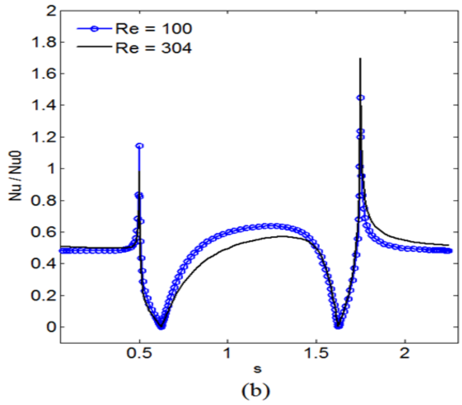


Figure 7. Variation of local Nusselt number as a function of arc length for (a) Pr=5, and (b) Pr=0.7

4. 2. Section 2 In this section numerical simulation are carried out for two channel configurations. In Channel 2, $\psi_1=\psi_2=75^\circ$, $\psi_3=\psi_4=285^\circ$ and in channel 3, $\psi_1=\psi_2=50^\circ$ and $\psi_3=\psi_4=310^\circ$.

The streamlines and isotherms of the flow and temperature field for two channels are shown in Figures 8 and 9. The vortex of channel 2 is larger than channel 3 for two Reynolds number. The vortex grows larger as the Reynolds number increases. The Nusselt number variations of two channel walls are shown in Figures 10 and 11.

4. 3. Section 3 In this section numerical simulation are carried out for Channel 4, $\psi_1=75^\circ$, $\psi_2=105^\circ$, $\psi_3=255^\circ$, $\psi_4=285^\circ$ and channel 5, $\psi_1=50^\circ$, $\psi_2=130^\circ$, $\psi_3=240^\circ$, $\psi_4=310^\circ$. The streamlines and isotherms of the flow and temperature field for two channels are showed in Figures 12 and 13. The vortex of channel 5 is larger than channel 4 for two Reynolds numbers. The vortex grows larger as the Reynolds number increases. The Nusselt number variations of two channel walls are shown in Figures 14 and 15.

4. 4. Section 4 In this section numerical simulation are carried out for Channel 6, $\psi_1=\psi_2=75^\circ$, $\psi_3=\psi_4=255^\circ$, and channel 7, $\psi_1=\psi_2=50^\circ$, $\psi_3=\psi_4=230^\circ$. The streamlines and isotherms of the flow and temperature field for two channels are showed in Figures 16 and 17. In both channels, flow is asymmetrical about the horizontal centerline. The vortex grows larger as the Reynolds number increases. The Nusselt number variations of two channel walls are shown in Figure 18.

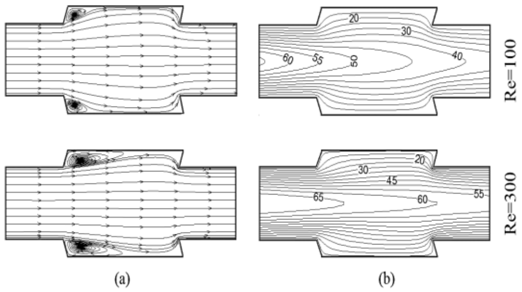


Figure 8. (a) Streamline, (b) Isotherms for the channel 2

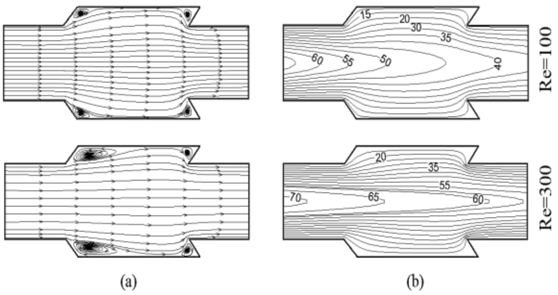


Figure 9. (a) Streamline, (b) Isotherms for the channel 3

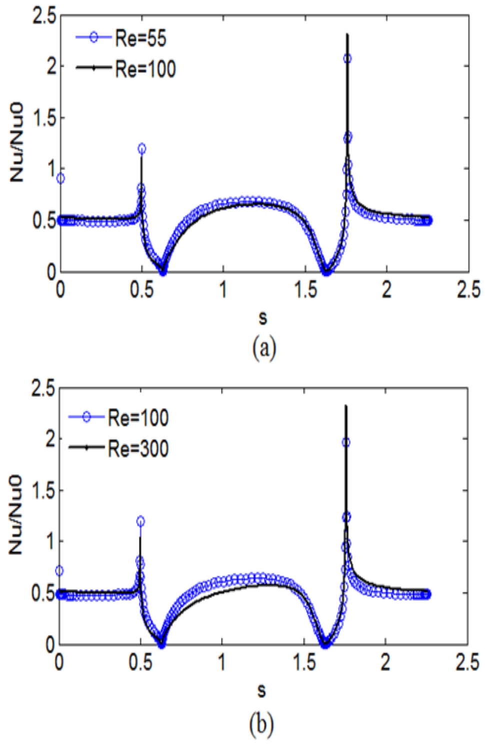


Figure 10. Variation of local Nusselt number as a function of arc length for (a) Pr=5, and (b) Pr=0.7 (channel 2)

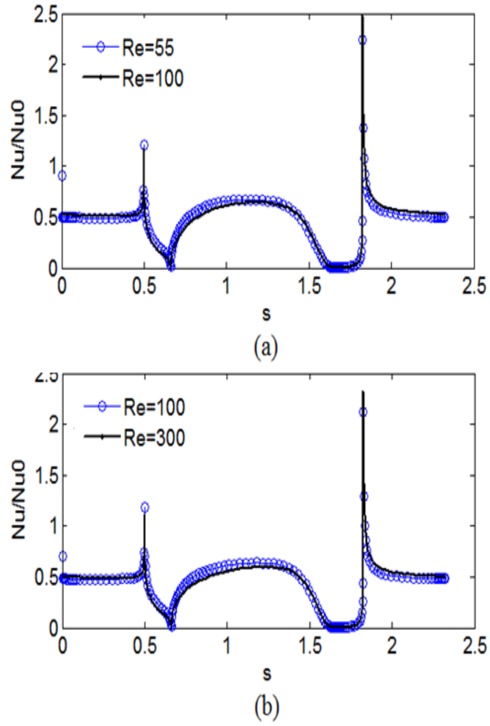


Figure 11. Variation of local Nusselt number as a function of arc length for (a) Pr=5, and (b) Pr=0.7 (channel 3)

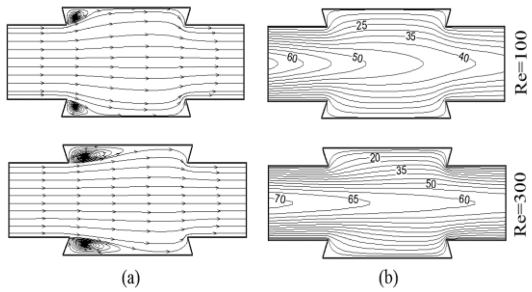


Figure 12. (a) Streamline, (b) Isotherms for the channel 4

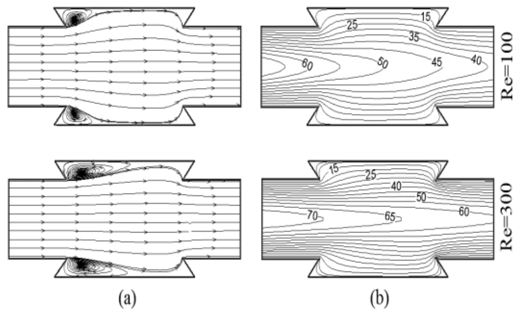


Figure 13. (a) Streamline, (b) Isotherms for the channel 5.

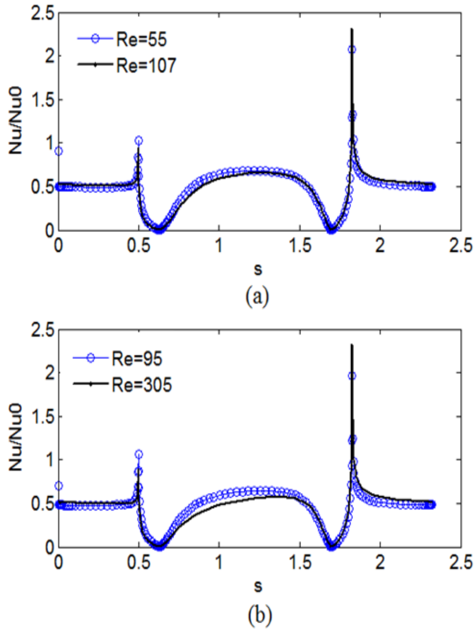


Figure 14. Variation of local Nusselt number as a function of arc length for (a) Pr=5, and (b) Pr=0.7 (channel 4)

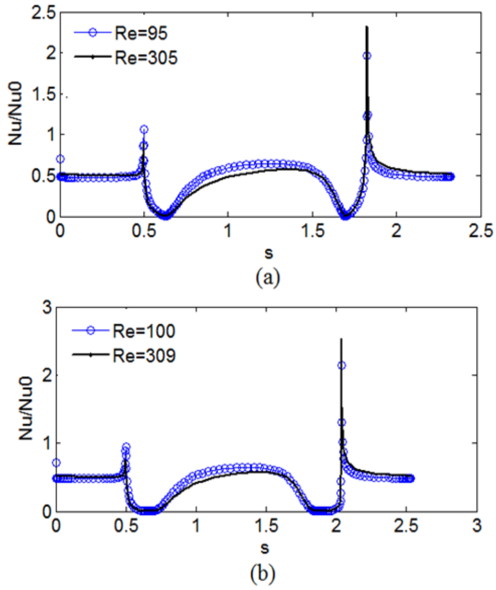


Figure 15. Variation of local Nusselt number as a function of arc length for (a) Pr=5, and (b) Pr=0.7 (channel 5)

4. 5. Compare All Channels As was seen in the previous section, the local Nusselt number was less than parallel-plate channel, in all cases. In order to analyzing the heat transfer and pressure drop together, two parameters thermal performance factor (TPF) and the flow area goodness factor (j/f) are used.

The thermal performance factor and flow area goodness factor of all channels are shown in Figures 19 to 22. In almost all Reynolds numbers, thermal performance factor and flow area goodness factor of channel 1 are higher than others for Pr=0.7 and Pr=5. However, channel 2 and channel 6 have higher TPF for Reynolds numbers greater than 153 at flow with Pr=5. Also, flow area goodness factor of channel 6 is highest for Reynolds numbers greater than 158 at flow with Pr=5. Channel 5 has the lowest efficiency factors among other channels.

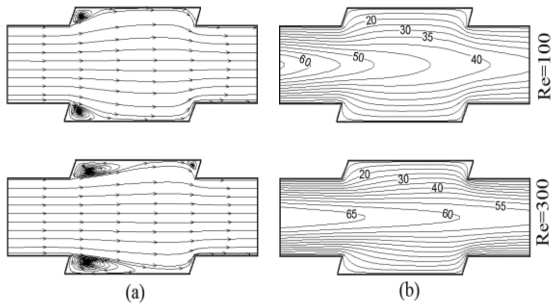


Figure 16. (a) Streamline, (b) Isotherms for the channel 6

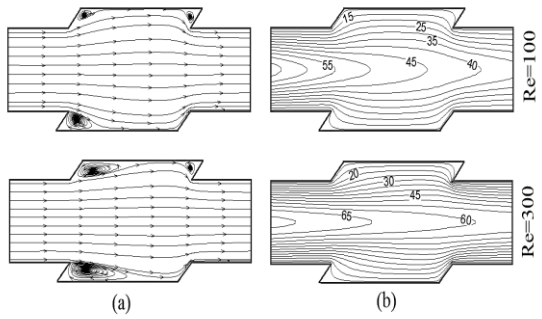


Figure 17. (a) Streamline, (b) Isotherms for the channel 7

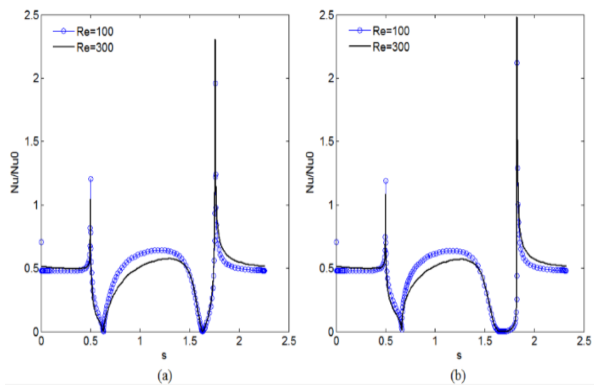


Figure 18. Variation of local Nusselt number as a function of arc length at Pr=0.7 for (a) channel 6, (b) channel 7

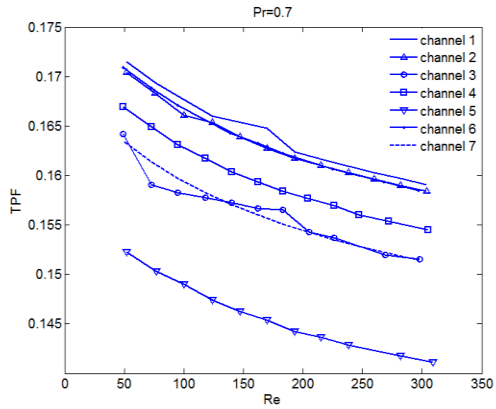


Figure 19. Thermal performance factor of channels (Pr=0.7)

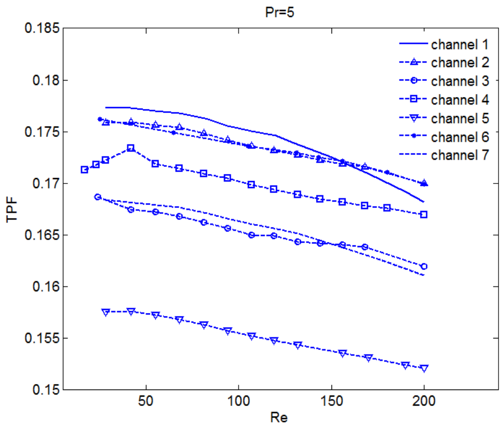


Figure 20. Thermal performance factor of channels (Pr=5)

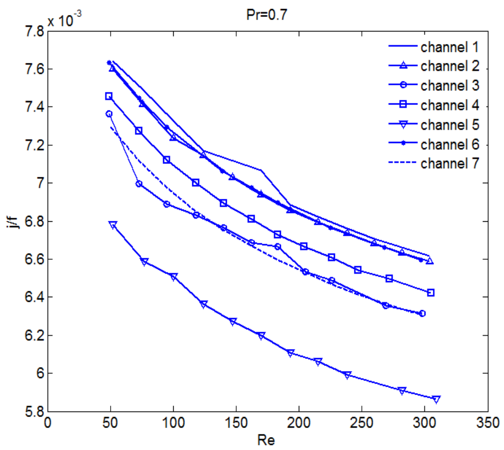


Figure 21. Flow area goodness factor for all channels (Pr=0.7)

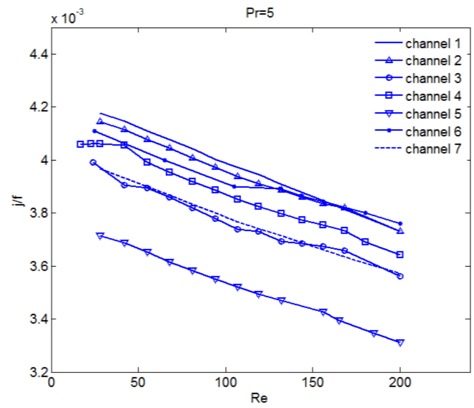


Figure 22. flow area goodness factor for all channels (Pr=5)

5. CONCLUSIONS

Fully developed flow and heat transfer through a series corrugated channels has been simulated numerically. Numerical results for steady laminar flow (50<Re<300), incompressible, constant properties and by two values of Prandtl number (0.7 and 5) are presented. The effect of grooves shape on channel wall on the flow and heat transfer has been considered.

In all cases studied, the average Nusselt number is lower than for the case of parallel-plates channel. Both the thermal performance factor (TPF) and flow area goodness factor (j/f) decrease with the increasing of Re either in Pr=0.7 or Pr=5 fluid flow. The best performances are obtained for channel 1 (rectangular groove shape). Channel 5 has the lowest efficiency factors among other channels. The TPF value of fluid with Pr=5 is higher than other one in all Reynolds number. Also, flow area goodness factor of fluid flow with Pr=0.7 is higher than Pr=5 in all Reynolds numbers.

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Numerical Analysis of Fully Developed Flow and Heat Transfer in Channels with Periodically Grooved Parts

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چکیده

از کانال‌های موجدار به صورت گسترده در حوزه‌های مختلف مهندسی به منظور دستیابی به انتقال حرارت بیشتر در جریان‌های با عدد رینولدز پایین استفاده می‌شود. در این مطالعه، هندسه شکل زائده‌های دیواره کانال به کمک چهار زاویه مدلسازی شده است. با تغییر این پارامترها هندسه کانال‌های جدیدی تولید شده و از حل عددی به منظور شبیه‌سازی جریان و انتقال حرارت توسعه یافته داخلی استفاده شده است. نتایج با یک کانال با زائده‌ی مستطیل شکل مقایسه شده است. عدد پرانتل 0.7 و 5 انتخاب و عدد رینولدز در محدوده 50 تا 300 فرض شده است. از یک روش حجم محدود مبتنی بر المان برای گسسته‌سازی معادلات حاکم استفاده شده است. مطابق نتایج به دست آمده بازده انتقال حرارتی و همچنین عدد نوسلت متوسط کانال با زائده مستطیل شکل نسبت به بقیه شکل‌ها، بزرگتر می‌باشد.

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