# Numerical Investigation on Turbulent Fluid Flow and Heat Transfer in a Channel with a Corrugated Wall

**Vahid Farhangmehr, Behzad Rossoli, Saeed Karam Javani Azar**

*Abstract*— **In this paper, a numerical simulation of the steady two-dimensional incompressible air flow with forced convective heat transfer in the turbulent regime was done in a channel with a corrugated wall. The Navier-Stokes and energy equations in conjunction with two equations of the realizable k-ε turbulence model were solved by the finite volume method. To discretize the governing equations, a second-order upwinding scheme and to couple the pressure and velocity fields, the SIMPLE algorithm were applied. It is found that the height of channel, corrugation angle, and Reynolds number affect the flow and heat transfer. It is found that the optimum heat transfer is achieved in channel height of 1.5cm, corrugation angle of 60°, and Reynolds number of 4000. By using the corrugated channels in heat exchangers, it is possible to enhance the heat transfer and decrease their size.**

*Index Terms*— **Convective heat transfer, Corrugated channel, Navier-Stokes equations, Realizable k-ε turbulence model**

## 1) INTRODUCTION

There are different active and passive techniques in the literature to enhance the forced convective heat transfer in the internal flows. The active techniques, such as the suction or injection of fluid from the physical boundaries, the injection of micro bubbles or micro particles into the flow, the acoustic excitation of the flow, and exerting the electromagnetic force on the fluid, need the use of an external energy. The passive techniques, without needing the use of an external energy for carrying out the same task, are much simpler and cheaper in comparison with the active techniques. In these techniques, the geometry of physical boundaries is appropriately altered by employing the flow disturbing ribs, cavities, or grooves on their surface or by corrugating the physical boundaries. These alterations have the ability to increase the turbulence of flow and generate the secondary flows, and hence lead to a heat transfer enhancement. In the literature, the passive techniques have been extensively studied theoretically, experimentally, and numerically by many researchers. Gradek et al. [1] did an experimental investigation on both the laminar and turbulent internal flows with the forced convective heat transfer. They observed that the heat transfer in a two-dimensional channel with the sinusoidal wavy walls is greater in comparison with the same channel with the straight walls. Wang and Chen [2] studied numerically the forced convective heat transfer for a flowing fluid through a two-dimensional sinusoidally curved channel in a laminar regime. The effects of Reynolds number, Prandtl number, and the wavy geometry on the skin friction and Nusselt number were analyzed in their study. Islamoglu and Parmaksizoglu experimentally[3] and numerically[4] did two investigations on the forced convective heat transfer for respectively the laminar and turbulent flows of air through a two-dimensional corrugated channel. Their numerical study was based on the finite elements method. The results showed that an increase in the channel height leads to an increase in the Nusselt number and the friction coefficient. Eslamoglu and Kurt [5] compared the results obtained by the Artificial Neural Networks (ANNs) with the available experimental data for the forced convective heat transfer for the laminar air flow through a two-dimensional corrugated channel. Their results showed that the application of ANNs approach is capable to provide acceptable results for the Nusselt number. Kanaris et al. [6] studied numerically the forced convective heat transfer for the turbulent flow of water in a two-dimensional channel with a corrugated wall. They proposed a nonlinear correlation between the mean Nusselt number, Prandtl number, and the Reynolds number besides a nonlinear correlation between the mean friction coefficient and the Reynolds number. Naphon experimentally[7, 8]and numerically[9, 10]studied the forced convective heat transfer for the air flow in a two-dimensional channel with the opposite corrugated walls in the laminar and turbulent regimes. He applied the finite volume method in his numerical studies. It is found that the corrugation angle and the height of channel have remarkable effects on the pressure, temperature, and the velocity fields. Lin et al. [11] studied the forced convective heat transfer for the laminar flow of air in a two-dimensional wavy channel, experimentally. It was found that the Reynolds number and the non-dimensional curvature radius of the walls influence the heat transfer. Fernandes et al. [12] by employing the finite elements method did a numerical investigation on the forced convective heat transfer for the laminar flow of an incompressible fluid in a two-dimensional chevron-type channel. Elshafei et al. [13] studied the forced convective heat transfer characteristics and the pressure drop for the turbulent flow of air in a two-dimensional corrugated channel, experimentally. It was observed that the corrugated channel has the ability to increase the convective heat transfer coefficient and the friction coefficient respectively from 2.6 and 1.9 (for the same channel with the straight walls) to 3.2 and 2.6 which implies the effects of both the channel height and the corrugation angle on the flow and the convective heat transfer. Naphon and Kornkumjayrit [14] studied the forced

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convective heat transfer for the turbulent flow of air through a two-dimensional channel with a wavy wall. They applied the finite volume method for the numerical modellings and found that the convective heat transfer is enhanced by corrugating one wall of the channel.

In the present numerical work, the turbulent air flow with forced convective heat transfer in a two-dimensional channel with a corrugated wall has been investigated. To the best of our knowledge, there is a limited number of investigations for this type of channels. Our investigation provides great details for the effect of Reynolds number, corrugation angle, and the channel height on the convective heat transfer characteristics and the pressure drop in a channel with a corrugated wall.

## 2) GOVERNING EQUATIONS AND NUMERICAL MODELING

In Fig.1, a schematic representation for the geometry of a two-dimensional channel with a corrugated wall besides the boundary conditions has been shown in which, H is the height of channel. Also in Fig.2, a schematic representation for the corrugated wall has been shown in which, θ is the corrugation angle. Aluminum has been chosen as the material for both the straight and corrugated walls. 300mm for the entrance length ensures the flow to become fully developed before reaching the corrugated wall and 200mm for the exit length ensures the continuous flow of air through the channel. Four values for H have been considered which are 1, 1.5, 3, and 6cm. The height of corrugation in all cases is 5mm. Thus, the ratio of channel height to corrugation height are 2, 3, 6, and 12, respectively. Three values for θ have been considered which are 20°, 40°, and 60°.

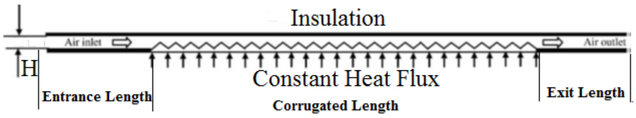


Fig.1. A schematic for the channel with a corrugated wall


Fig.2. A schematic for the corrugated wall

The steady two-dimensional incompressible governing equations are the Navier-Stokes and energy equations besides two equations for the generation and the waste of turbulent kinetic energy as follows [10]:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad \text{i, j=1,2} \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_i}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}(-\rho\overline{u_i' u_j'}) \tag{2}$$

$$\frac{\partial}{\partial x_i}[u_i(\rho E + P)] = \frac{\partial}{\partial x_i}[(K + \frac{C_p \mu_t}{Pr_t})\frac{\partial T}{\partial x_j}] \tag{3}$$

$$\rho\frac{\partial}{\partial x_i}(K u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial x_j}\right] - \rho\varepsilon \tag{4}$$

$$\rho\frac{\partial}{\partial x_i}(\varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] - C_\mu \rho \frac{\varepsilon^2}{K + \sqrt{\nu\varepsilon}} \tag{5}$$

in which ρ is the fluid density, $x_1$ and $x_2$ are x and y Cartesian coordinates, $u_1$ and $u_2$ are x and y components of the velocity, P is the pressure, μ is the dynamic viscosity of fluid, E is the energy, K and ε are the generation rate and the waste rate of turbulence kinetic energy, $\mu_t$ is the dynamic eddy viscosity, ν is the kinematic viscosity of fluid, T is the temperature, and $C_p$ is the heat capacity of fluid. The dynamic eddy viscosity is defined as [10]:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{5}$$

The constants in the realizable K-ε turbulence model are as follows [10]:

$$Pr_t = 0.85;\ \sigma_k = 1;\ \sigma_\varepsilon = 1.3;\ C_\mu = 1.9 \tag{6}$$

The air follows into channel with a uniform velocity $u_{in}$, a constant temperature T=300°K, and a constant turbulence intensity I=10%. K and ε at channel entrance are as follows:

$$K_{in} = \frac{3}{2}(u_{in} I)^2;\ \varepsilon_{in} = C_\mu^{\frac{3}{4}}\frac{K^{\frac{3}{2}}}{L};L = 0.07\ D_h;D_h = 2H \tag{7}$$

The other boundary conditions are the no slip condition on the walls and the constant heat flux ($q=800W/m^2$) on the corrugated wall. The straight walls are insulated.

## 3) NUMERICAL SOLUTION

The governing equations are solved by the control volume method. In order to discretize these equations, a second-order upwinding scheme and in order to couple the pressure and the velocity fields, the SIMPLE algorithm are used. In a smooth structured grid which is clustered appropriately in the vicinity of walls, then the discretized equations are solved by a time marching approach. The grid should be clustered and should have the sufficient cells in the vicinity of walls due to higher velocity and temperature gradients there in comparison with other regions of the flow.

For the mean convective heat transfer coefficient along the corrugated wall, $h_C$, we have [10]:

$$Q_{Ave} = h_c \Delta T_{LMTD} A_C \tag{8}$$

in which $Q_{Ave}$ is the mean convective heat transfer rate, $A_C$ is the area of the corrugated wall, and $\Delta T_{LMTD}$ is defined as:

$$\Delta T_{LMTD} = \frac{\left(T_{S,Ave} - T_{a,Ave,in}\right) - \left(T_{s,Ave} - T_{a,Ave,out}\right)}{\ln\left(\dfrac{T_{s,Ave} - T_{a,Ave,in}}{T_{s,Ave} - T_{a,Ave,out}}\right)} \tag{9}$$

The flow velocity at channel entrance and the convective heat transfer enhancement ratio E are respectively defined as:

$$u_{in} = \frac{\mu Re}{\rho D_h} \tag{10}$$

$$E = \frac{Nu_{corr}}{Nu_{stra}} \tag{11}$$

in which $Nu_{corr}$ is the mean Nusselt number on the corrugated wall and $Nu_{stra}$ is the mean Nusselt number on the same wall but without the corrugation.

## 4) RESULTS AND DISCUSSION

In order to validate our numerical simulations, the results obtained for the average temperature and the mean Nusselt number on a corrugated wall with a wavy angle of 20° and a constant heat flux of $580W/m^2$ have been compared with the Naphon's experimental results [12] respectively in Figs.3 and 4. The regime of air flow is laminar. These comparisons show a good agreement.

In Figs.5 and 6, the variation of mean Nusselt number on a corrugated wall with four various wavy angles of 0°(straight wall),20°,40°, and 60°in a channel with four various heights of

1cm, 1.5cm, 3cm, and 6cm versus the Reynolds number in the laminar and turbulent regimes of flow has been presented. In all cases, the heat flux on the corrugated wall is constant and equals 800W/m$^2$. Fig.5 shows that in the laminar regime, the corrugated wall in a channel has a greater Nusselt number in comparison with the straight wall in the same channel. The main reason for this is its considerable role in increasing the surface for the contact between the wall and fluid. Moreover, the corrugated wall positively disturbs the flow and thins the boundary layers. The formation of vortexes also enhances the convective heat transfer. In the turbulent regime, again this convective heat transfer enhancement is observed in Fig.6. It is found that in the laminar regime, the highest mean Nusselt number on the corrugated wall is achieved for the wavy angle of 40$^{\circ}$ at Reynolds number of 800 but in the turbulent regime, it is achieved for the wavy angle of 60$^{\circ}$at the Reynolds number of 8000. It is seen in both Figs.5 and 6 that the mean Nusselt number on the corrugated wall increases with the Reynolds number in both the laminar and turbulent regimes. Moreover, it can be concluded from Figs.5 and 6 that in both the laminar and turbulent regimes, the channel with a greater height has a more capacity for the heat transfer enhancement.

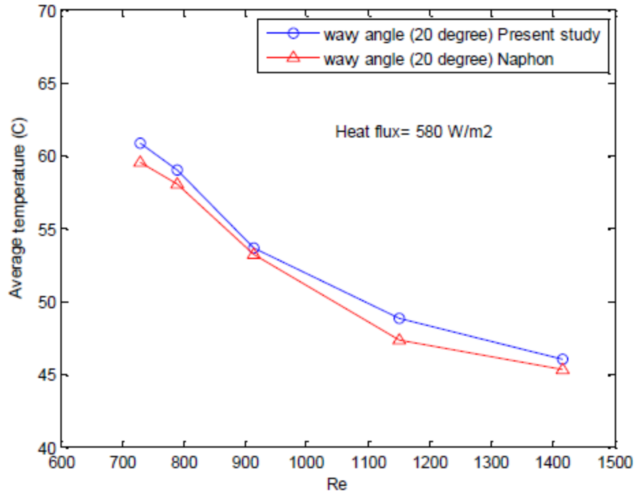


Fig.3. The average temperature on a corrugated wall versus the Reynolds number in the laminar regime of flow

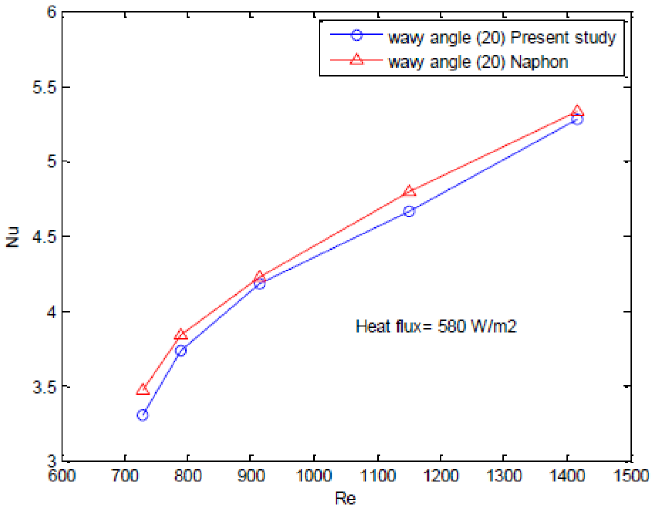


Fig.4. The mean Nusselt number on a corrugated wall versus the Reynolds number in the laminar regime of flow

Figs.7 and 8 represent the variation of convective heat transfer enhancement ratio, E, respectively in the laminar and turbulent regimes. In the laminar regime, this ratio increases with the Reynolds number and the highest E is achieved for the wavy angle of 40$^{\circ}$at Reynolds number of 800. But in the turbulent regime and for the channel with the height of 1cm, as the Reynolds number increases, E decreases. The highest E is achieved for the wavy angle of 60$^{\circ}$ at Reynolds number of 1600. For the channel with the height of 1.5cm, the highest E is achieved for the wavy angle of 60$^{\circ}$ at Reynolds number of 4000. For the channel with height of 3 and 6cm, E increases with the Reynolds number and the highest E is achieved for the wavy angle of 60$^{\circ}$ at Reynolds number of 8000. It is found in Figs.7 and 8 that the channel with the height of 1cm in both the laminar and turbulent regimes has the greatest convective heat transfer enhancement ratio.

The pressure drop in a channel is an important parameter. Figs.9 and 10 represent the effect of channel geometry on the mean pressure drop dP/dx in laminar and turbulent regimes, respectively. It is seen that in both regimes, the mean pressure drop increases with the Reynolds number. The highest mean pressure drop is for wavy angle of 60$^{\circ}$ at Reynolds number of 800 in the laminar regime. The greatest mean pressure drop in the turbulent regime is for the wavy angle of 60$^{\circ}$ at Reynolds number of 8000. In both regimes, the channel with the height of 6cm has the minimum mean pressure drop.

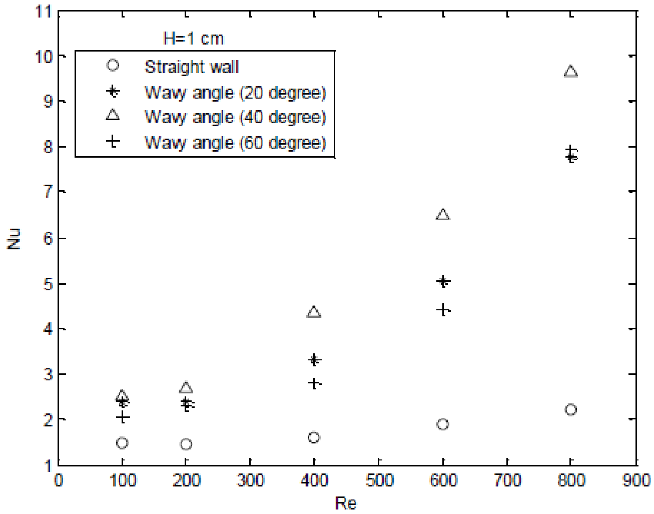


Fig.5a. The mean Nusselt number on the corrugated wall versus the Reynolds number in the laminar regime of flow for H=1cm

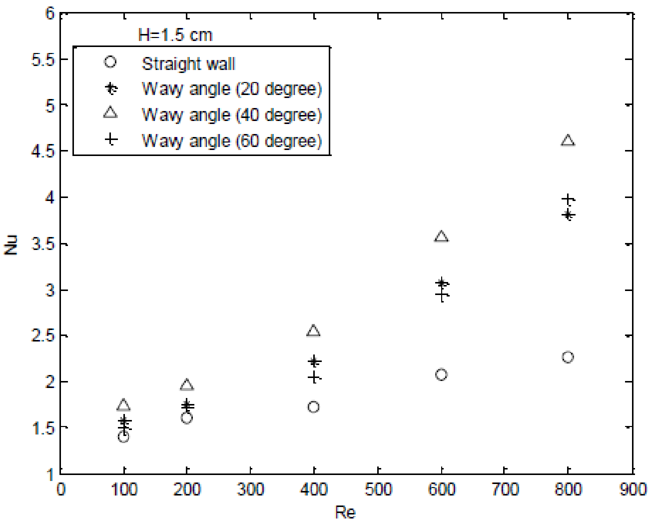


Fig.5b. The mean Nusselt number on the corrugated wall versus the Reynolds number in the laminar regime of flow for H=1.5cm

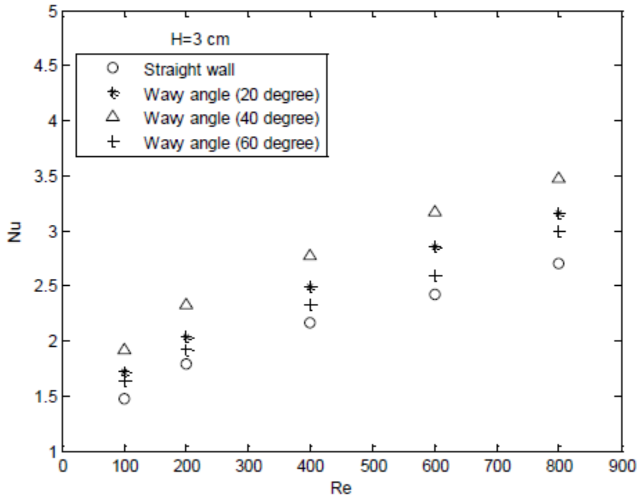


Fig.5c. The mean Nusselt number on the corrugated wall versus the Reynolds number in the laminar regime of flow for H=3cm

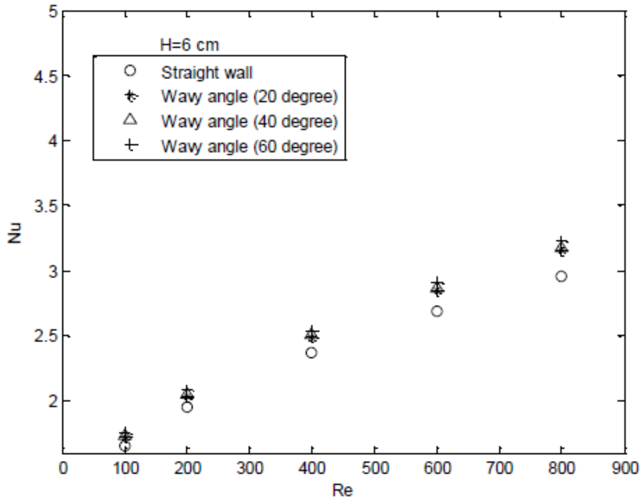


Fig.5d. The mean Nusselt number on the corrugated wall versus the Reynolds number in the laminar regime of flow for H=6cm

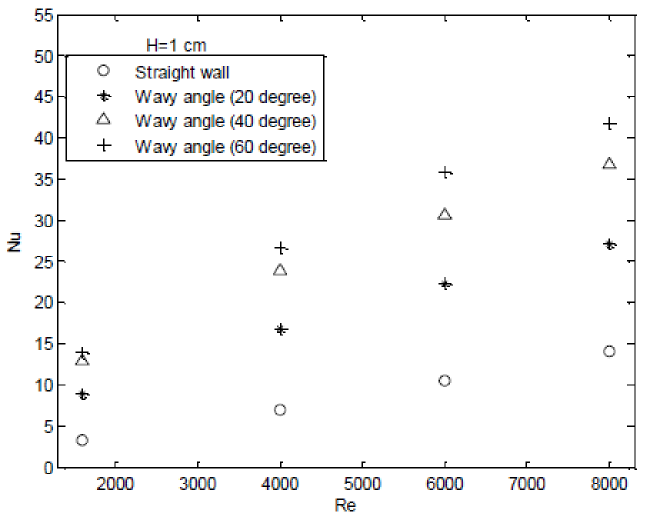


Fig.6a. The mean Nusselt number on the corrugated wall versus the Reynolds number in the turbulent regime of flow for H=1cm

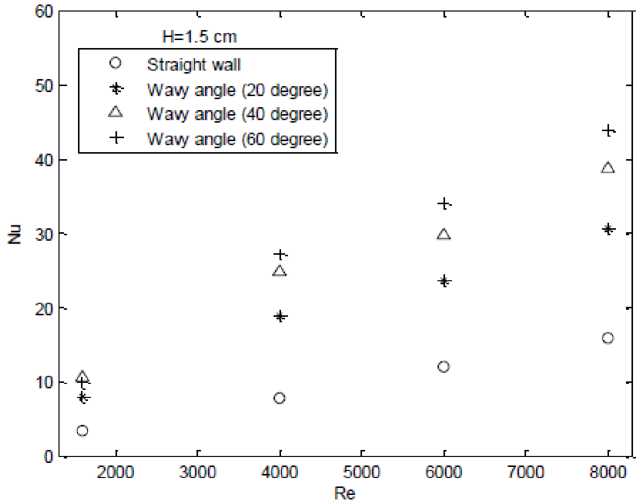


Fig.6b. The mean Nusselt number on the corrugated wall versus the Reynolds number in the turbulent regime of flow for H=1.5cm

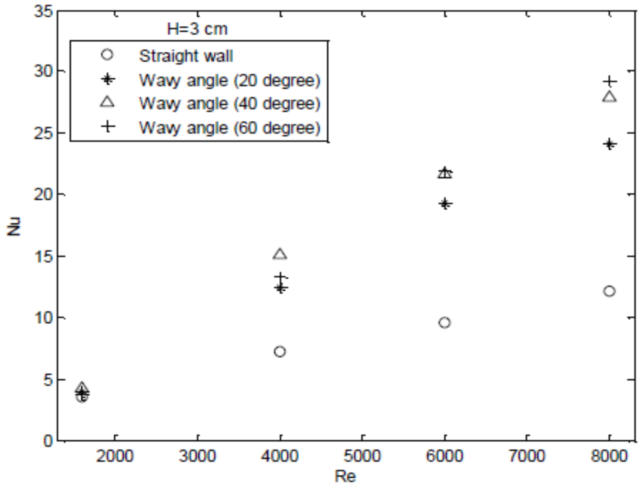


Fig.6c. The mean Nusselt number on the corrugated wall versus the Reynolds number in the turbulent regime of flow for H=3cm

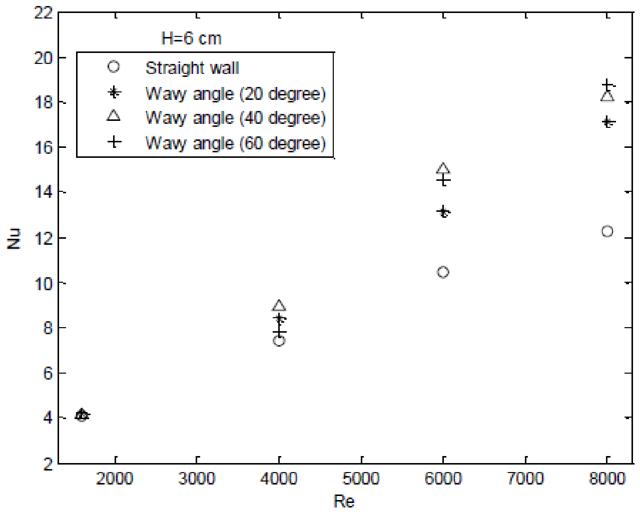


Fig.6d. The mean Nusselt number on the corrugated wall versus the Reynolds number in the turbulent regime of flow for H=6cm

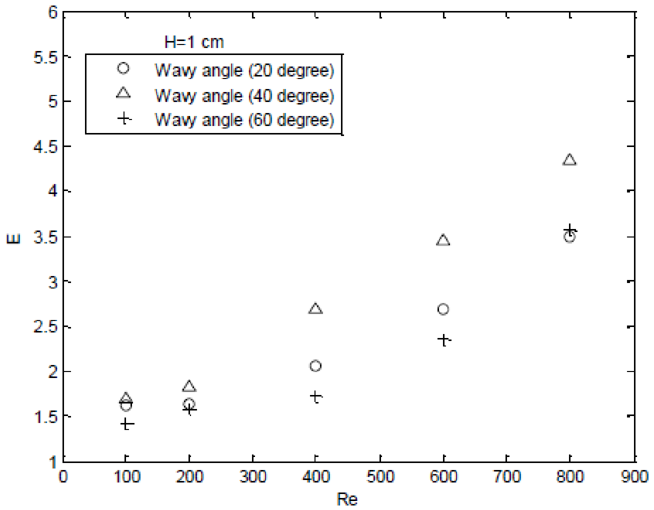


Fig.7a. The convective heat transfer enhancement ratio, E versus the Reynolds number in the laminar regime of flow for H=1cm

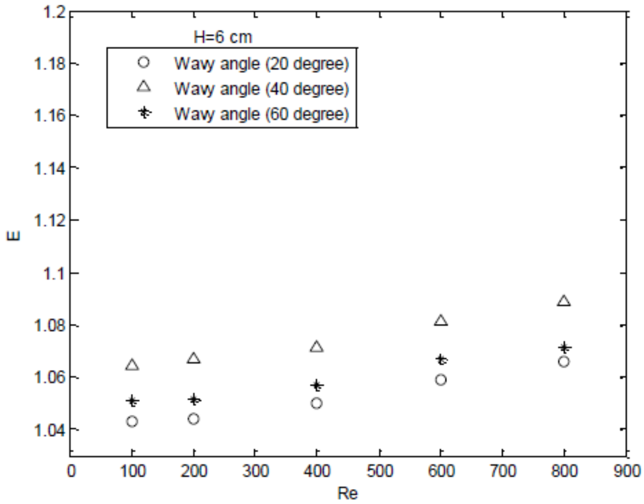


Fig.7d. The convective heat transfer enhancement ratio, E versus the Reynolds number in the laminar regime of flow for H=6cm

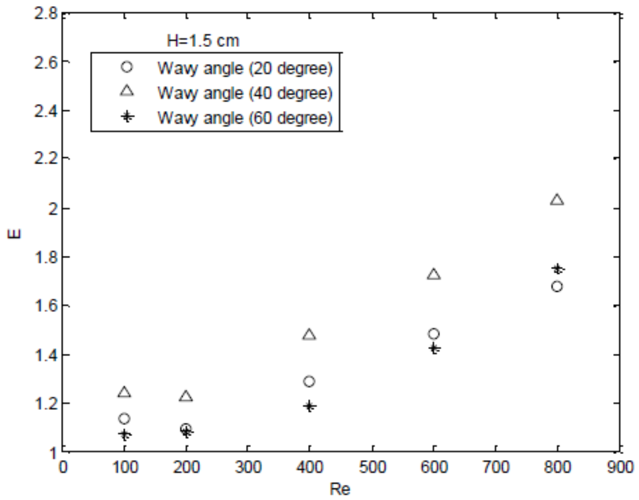


Fig.7b. The convective heat transfer enhancement ratio, E versus the Reynolds number in the laminar regime of flow for H=1.5cm

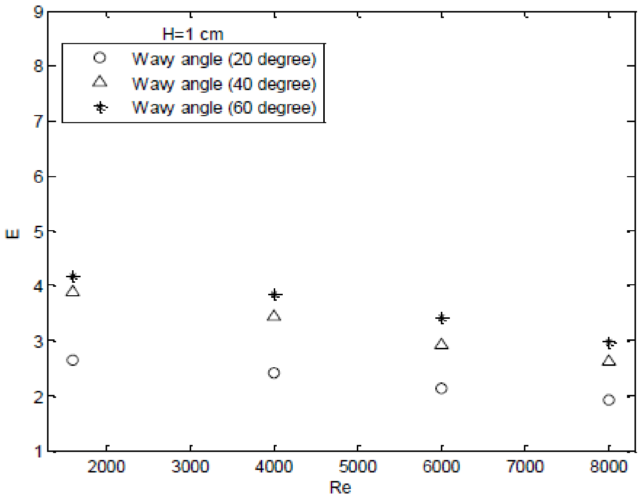


Fig.8a. The convective heat transfer enhancement ratio, E versus the Reynolds number in the turbulent regime of flow for H=1cm

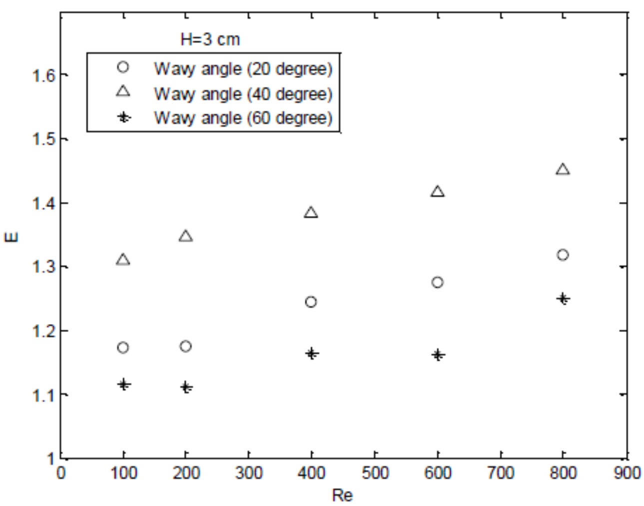


Fig.7c. The convective heat transfer enhancement ratio, E versus the Reynolds number in the laminar regime of flow for H=3cm

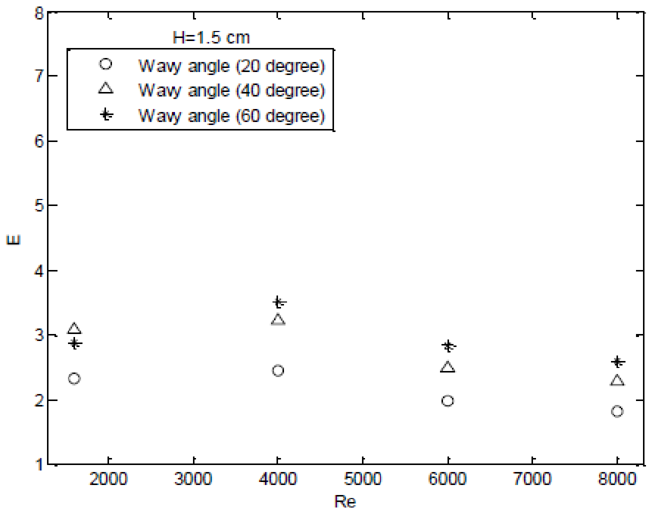


Fig.8b. The convective heat transfer enhancement ratio, E versus the Reynolds number in the turbulent regime of flow for H=1.5cm

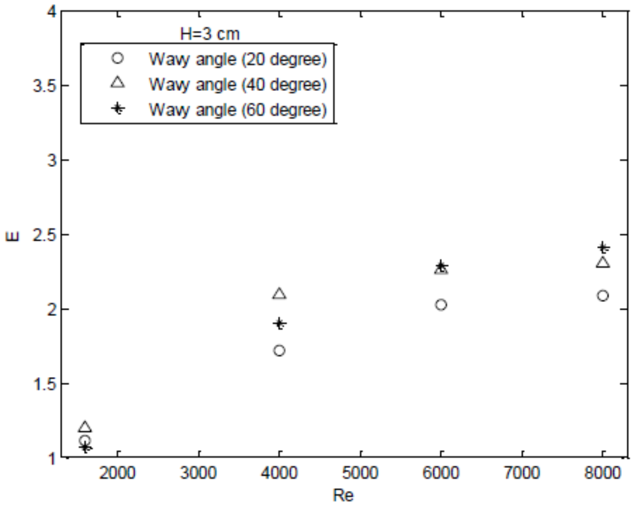


Fig.8c. The convective heat transfer enhancement ratio, E versus the Reynolds number in the turbulent regime of flow for H=3cm

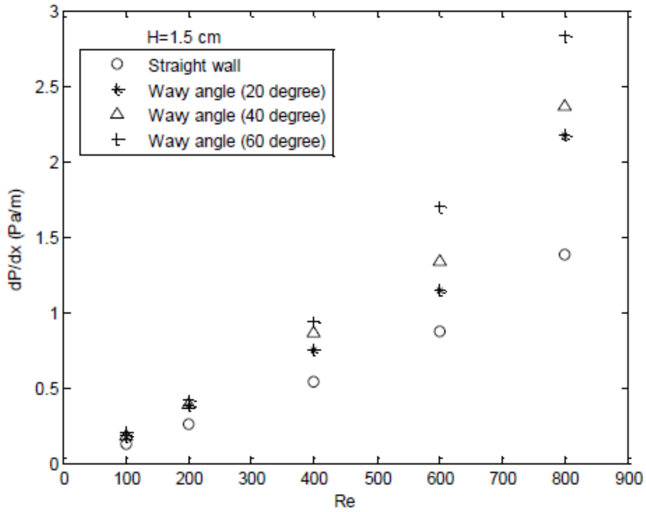


Fig.9b. The mean pressure drop dP/dx along the channel versus the Reynolds number in the laminar regime of flow for H=1.5cm

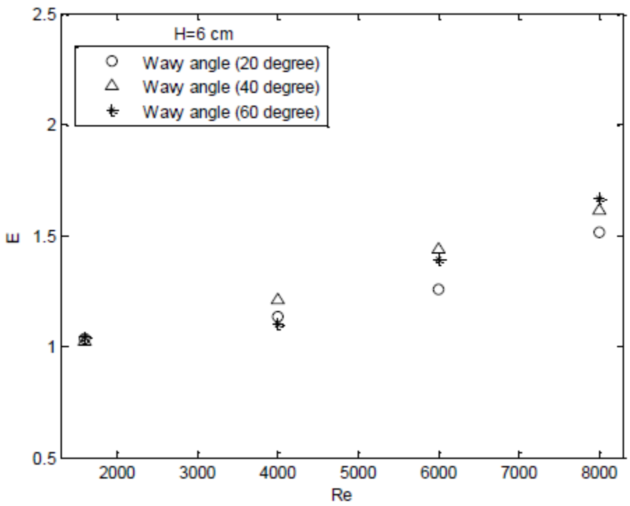


Fig.8d. The convective heat transfer enhancement ratio, E versus the Reynolds number in the turbulent regime of flow for H=6cm

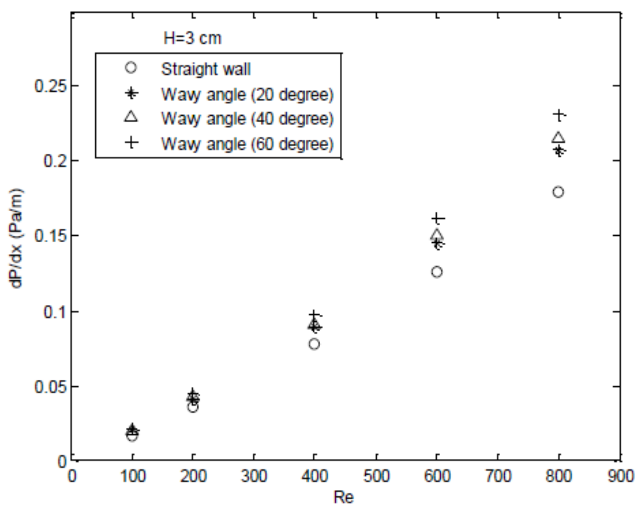


Fig.9c. The mean pressure drop dP/dx along the channel versus the Reynolds number in the laminar regime of flow for H=3cm

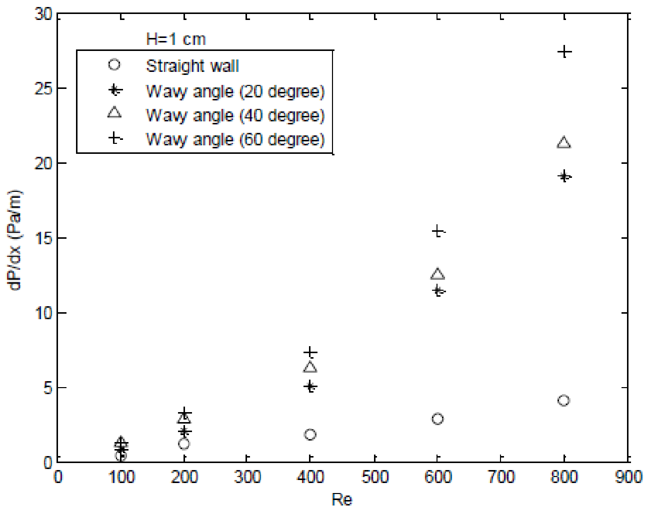


Fig.9a. The mean pressure drop dP/dx along the channel versus the Reynolds number in the laminar regime of flow for H=1cm

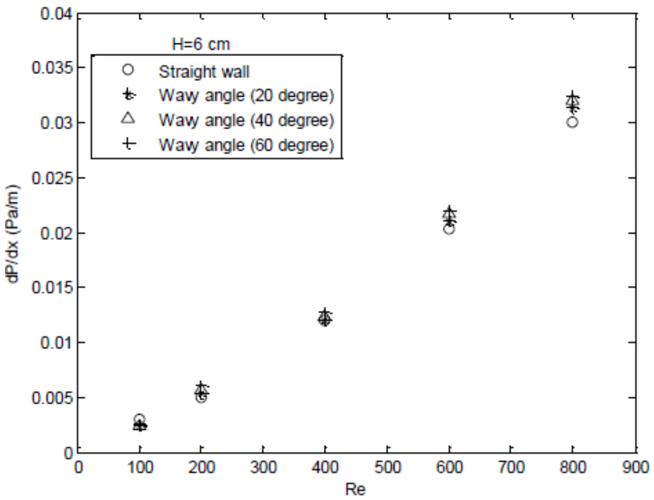


Fig.9d. The mean pressure drop dP/dx along the channel versus the Reynolds number in the laminar regime of flow for H=6cm

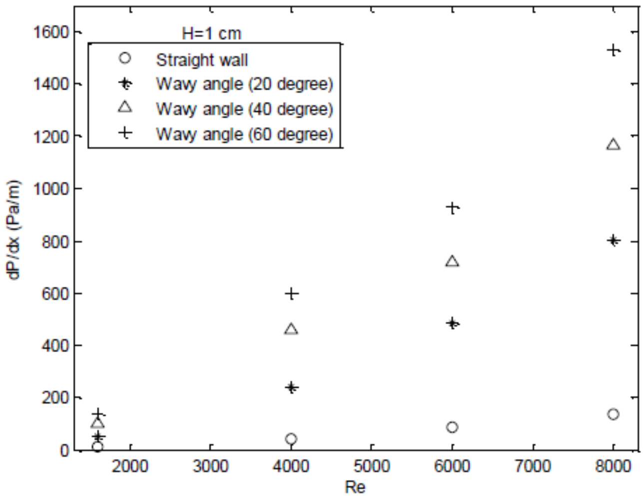


Fig.10a. The mean pressure drop dP/dx along the channel versus the Reynolds number in the turbulent regime of flow for H=1cm

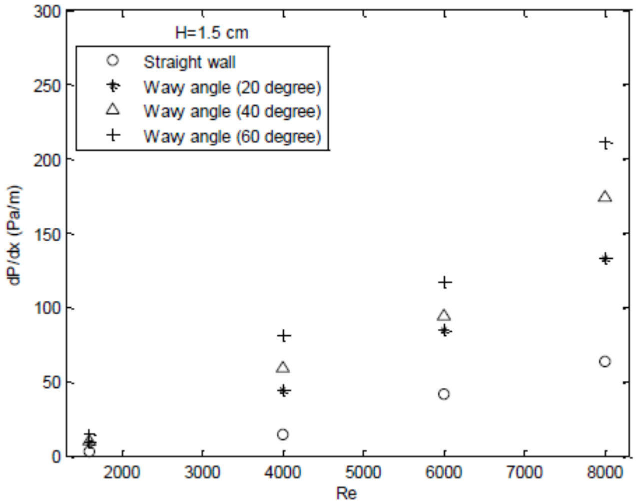


Fig.10b. The mean pressure drop dP/dx along the channel versus the Reynolds number in the turbulent regime of flow for H=1.5cm

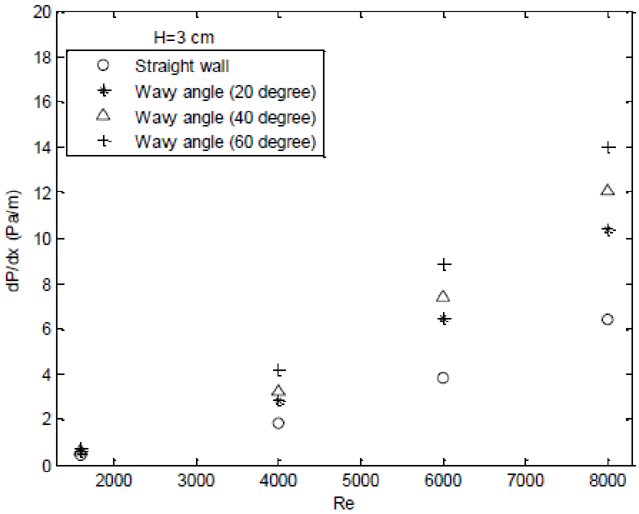


Fig.10c. The mean pressure drop dP/dx along the channel versus the Reynolds number in the turbulent regime of flow for H=3cm

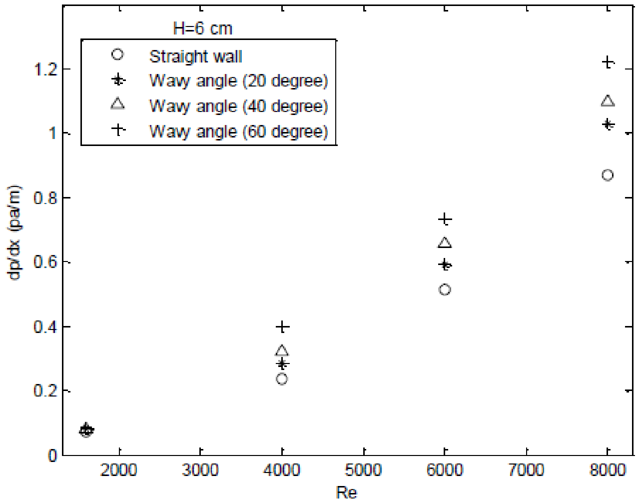


Fig.10d. The mean pressure drop dP/dx along the channel versus the Reynolds number in the turbulent regime of flow for H=6cm

## 5) CONCLUSION

In the present study, the effect of channel geometry and the flow regime on the convective heat transfer and pressure drop along a channel with a corrugated wall was investigated numerically. It was found that the Reynolds number, channel height, and the corrugation angle affect both the convective heat transfer and the pressure drop. Also it was found that the corrugated wall enhances the convective heat transfer.

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