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Research Paper

Numerical Study on Heat Transfer and Pressure Drop in a Mini-Channel with Corrugated Walls

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Abstract. This study presents the numerical results relative to the development of heat transfer and pressure drop inside a corrugated channel, under constant heat flux conditions applied to the walls; the working fluid is air. The test section is a channel with two plates having trapezoidal-shaped corrugations with V-folds. The corrugated plates were placed inside a 12.5 m high channel and tested for three different inclination angles, i.e. 20°, 40° and 60°. The model was simulated for a heat flux of 0.58 kW /m^2, while the Reynolds numbers were considered within the interval ranging from 600 to 1400. The standard turbulent model (k-ε) was employed to simulate the flow and heat transfer developments within the channel. In addition, the governing equations were solved using the finite volume method in a structured uniform grid arrangement. Moreover, the effects of the geometric parameters on heat transfer and flow evolution were discussed as well. It is also worth noting that the corrugated surface had a significant impact on the enhancement of heat transfer and pressure drop due to breakage and destabilization occurring in the thermal boundary layer.

Keywords: Forced flow; Laminar; Turbulent; Numerical; Heat exchanger.

1. Introduction

In general, heat exchangers are used in industrial engineering applications such as heaters, oil coolers, air conditioner condensers and air conditioner evaporator units, petrochemical industry, power plants and chemical processing. A lot of effort has been made to develop more efficient and more compact heat exchangers using various heat transfer enhancement techniques in order to produce cheaper heat exchange equipment. The use of corrugated plates is a worthy approach for enhancing the thermal performance of exchangers and providing higher compactness. The breakage and destabilization of the thermal boundary layer occur as a result of the fluid flowing over the wrinkled surfaces. Therefore, corrugated surfaces can be used to promote the onset of turbulence and consequently increase heat transfer.

Sunden and Skoldheden [1-2] conducted an experimental study on heat transfer and pressure drop in corrugated tubes and smooth tubes. In their experiments, they used Reynolds numbers within the range extending from 800 to 5000. These same authors found out that the amount of heat transferred within a corrugated channel was 3.5 times higher than that obtained with a smooth channel. Similarly, Sunden and Trollheden [3] numerically investigated laminar flow heat transferring convectors in a two-dimensional corrugated channel whose section varies periodically, under constant heat flux. Finite difference approximations were used to solve the governing equations. Over the last few years, a large number of researchers have investigated the characteristics of heat transfer and pressure drop in corrugated channels. For example, Mohammed et al. [4] conducted a numerical study on the effects of tilt angles and height of corrugated channels on the heat enhancement rate. They recommended a wavy angle of 60 °, a corrugation depth of 2.5 mm with a channel height equal to 17.5 mm for a better heat transfer rate. Likewise, Elshafei et al. [5-6] carried out an experimental study on the effect of phase difference in V-shaped corrugated channels on the features of heat transfer and pressure drop. It was found that the average heat transfer coefficient was significantly enhanced; however, the pressure drop decreased as a function of the spacing and phase difference. Similarly, an experimental investigation was carried out by Ali and Ramadhyani [7] on heat transfer in the inlet region of the channel for a 20° wave angle and a laminar flow; they suggested that the Nusselt number increased from 140% in the parallel plate channel to 240% in the corrugated channel; likewise, the friction factor went from 130% to 280% for the same case.

Furthermore, a numerical and experimental study was undertaken by Islamoglu and Parmaksizoglu [8] on forced convection heat transfer and pressure drop in a corrugated channel; they noted a large difference between the minimum and maximum local heat transfer coefficients on the corrugated walls. Moreover, these same authors found out that the Nusselt number increased as the channel size augmented, which caused the pressure gradient to decrease. On the other hand, Zimmerer et al. [9] made an

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attempt to investigate the effects of inclination angle, wavelength, amplitude, and fold shape on heat and mass transfer inside the heat exchanger. Similarly, Hamza et al. [10] conducted an experimental study on the effects of the parameters used on the laminar flow forced convection heat transfer in a channel provided with a V-corrugated upper plate. All experiments were carried out for channel inclination angles and Reynolds numbers in the ranges from 0° to 60° and from 750 to 2050, respectively. In the same way, Pehlivan [11] performed an experimental investigation on the effect of sharp and rounded corrugation peaks on heat transfer and pressure drop within a folded channel; his results suggested that increasing the corrugation angle and channel height causes the heat transfer to rise.

A numerical study was conducted by Sukhmeet Singh et al [12] to investigate the heat transfer and pressure drop characteristics in a rough pipe with periodic cross ribs in a solar heater. Four types of transverse ribs have been studied, circular, square, trapezoidal, and saw-tooth. They found that the highest Nusselt number and the lowest friction factor are observed in the roughened pipe with saw-tooth ribs, followed by trapezoidal ribs.

On the other hand, a numerical analysis was performed by S. Rashidi and M. Akbarzadeh [13] for the purpose of studying the thermal-hydraulic performance and entropy generation of a turbulent flow inside a corrugated channel. The numerical simulation was performed with the Reynolds number varying within the interval from 5000 and 50 000. In this analysis, it was decided to consider three different wave amplitude values (a = 0.1, 0.2 and 0.3) and three wavelength values for the corrugated wall (k = 1, 2 and 3). It turned out that the global thermal performance was significantly enhanced through the use of a corrugated channel, where A = 0.1 for all Reynolds numbers. It was therefore found that the total entropy production had a minimum value at Re = 20,000, for all values of wave amplitude and wavelength of the corrugated wall.

R.K. Ajeel et al [14-16] performed a series of numerical investigations, on corrugated channels to study the impact of geometric parameters on the thermo-hydraulic performance of corrugated channel heat exchangers. the simulation results proved that the height / width (H/L) ratio was more efficient than the height / length (H/W) ratio, from the point of view performance.

The forced turbulent flow of the SiO_2-water nanofluid, through different corrugated channels is studied numerically and experimentally by the same authors, R. K. Ajeel et al.[17]. Two corrugated channels namely, the semi-circular corrugated channel and the trapezoidal corrugated channel were studied. The results show that the ripple profile has a significant effect on improving heat transfer. In addition, the use of nanofluids further promotes the improvement of heat transfer compared to the base fluid. The comparison shows a good agreement between the two numerical and experimental results.

H. Ameur [18] carried out a numerical study on a rectangular channel heat exchanger, equipped with corrugated baffles. The influence of the corrugated baffles on the thermohydraulic behavior of the exchanger was discussed.
The results showed that the overall performance factor increased from 1.27 to 1.53 as the ripple angle increased from 0° to 45°. Contrary to the case with the smooth channel without baffles.

In another work, M. Akbarzadeh and S. Rashidi [19] undertook a numerical investigation on the entropy generation and thermohydraulic performance of a corrugated channel with three corrugation profiles, i.e. triangular, sinusoidal, and trapezoidal. The simulations were performed for Reynolds numbers within the range from 400 to 1400 and the results found indicated that the triangular channel offers the highest generation rate of thermal entropy; it is followed by the sinusoidal and trapezoidal channels. In the end, it may be said that a channel with sinusoidal walls is recommended to have high performance and low entropy generations.
A large number of experimental and numerical studies have been conducted on topics related to heat transfer and pressure drop for various corrugated surface configurations. The purpose of this paper is to study the thermo-hydraulics characteristics in corrugated trapezoidal plate channels for different spacing angles θ = 20°, 40°, and 60°, as only a limited number of related studies have been reported in the literature so far.

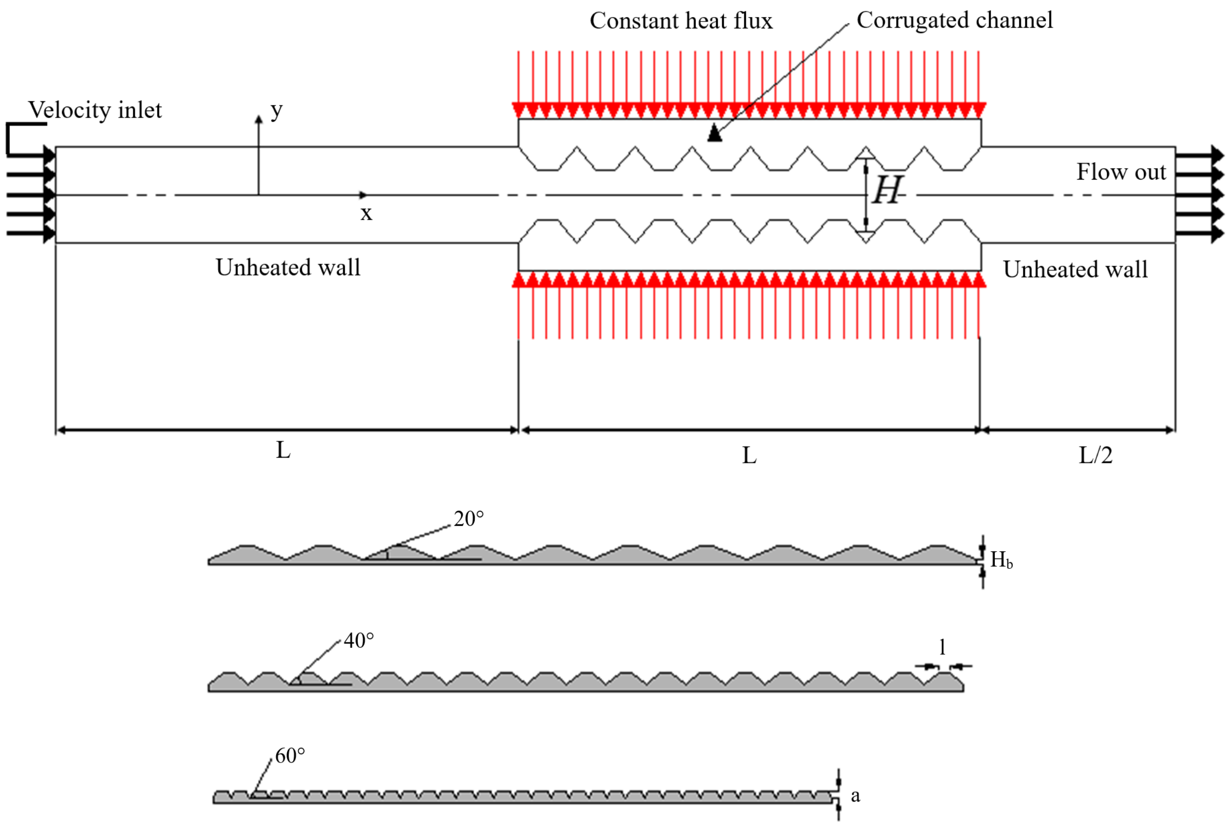

Fig. 1. (a) Geometry of the problem under study, and (b) General diagram of various corrugations.

Table 1. Geometric dimensions of the studied section. [20]

θ (°)	H (mm)	a (mm)	l (mm)	H_b (mm)	L (mm)	Ripple number
0	12.5	--	--	2.5	323	--
20	12.5	5	5	2.5	323	10
40	12.5	5	5	2.5	323	19
60	12.5	5	5	2.5	323	33

2. Physical model

Figure 1 illustrates the geometry of the problem. It includes a rectangular channel provided with two trapezoidal upper and lower corrugated plates and traversed by a stationary turbulent air flow. A number of hypotheses must be satisfied, namely (i) The physical properties of the fluid are assumed to be constant, (ii) The velocity and flux profiles is uniform, (iii) The applied turbulence model is (k-ε), and (iv) The flow is fully developed at the inlet.

3. Mathematical model

Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

Momentum equation

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

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

Energy equation
For fluid:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

with: $\alpha = k/\rho c_p$

For solid:

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}=0 \tag{5}$$

Furthermore, the standard turbulence model (k-ε) proposed by Launder and Spalding [21] was also applied in this simulation. This model, which may be employed to predict the secondary flow movement [24], consists of two equations, namely the equation for turbulent kinetic energy k and the equation for dissipation rate ε.

The two equations are given below:

Equation for the turbulent kinetic energy k:

$$\frac{\partial}{\partial x_i}[\rho k u_i]=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k-\rho\varepsilon \tag{6}$$

Equation for dissipation rate ε:

$$\frac{\partial}{\partial x_i}[\rho\varepsilon u_i]=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\left(\frac{\varepsilon}{k}\right)G_k+C_{2\varepsilon}\rho\left(\frac{\varepsilon^2}{k}\right)-\rho\varepsilon \tag{7}$$

In the above equations, G_k represents the generation of turbulent kinetic energy due to the mean velocity gradient; σ_k and σ_ε are the effective Prandtl number for turbulent kinetic energy and the dissipation rate, respectively; $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are constants and μ_t is the turbulent viscosity which is expressed as:

$$\mu_t=\left(\frac{\rho C_\mu k^2}{\varepsilon}\right) \tag{8}$$

Patankar [22] suggested that the empirical constants for the turbulent model can be determined by the complete fitting of data for a range of turbulent flows. They are given as:

$$C_\mu = 0.09,\ C_{1\varepsilon} = 1.47,\ C_{2\varepsilon} = 1.47,\ \sigma_k = 1.0,\ \sigma_\varepsilon = 1.3 \tag{9}$$

3.1 Boundary conditions

At the inlet:

$$u = u_{in},\ v = 0,\ T = T_{in} = 300K,\ k = k_{in},\ \varepsilon = \varepsilon_{in} \tag{10}$$

The turbulent kinetic energy k_{in} and turbulent dissipation ε_{in} in the inlet section of the channel are estimated in terms of the turbulent intensity I as follows:

$$k_{in} = \frac{3}{2}(u_{in}I)^2 , \quad \varepsilon_{in} = C_{\mu}^{\frac{3}{4}} \frac{k^{\frac{3}{2}}}{L} \tag{11}$$

$$I = 0.16\,\mathrm{Re}^{-1/8} \tag{12}$$

At the outlet:

$$\frac{\partial u}{\partial x} = 0 , \frac{\partial v}{\partial x} = 0 , \frac{\partial T}{\partial x} = 0 , \frac{\partial k}{\partial x} = 0 , \frac{\partial \varepsilon}{\partial x} = 0 \tag{13}$$

At the wall we have $q_{wall} = 580 \ w/m^2$.

Symmetric condition: x=0

$$\frac{\partial T}{\partial y} = \frac{\partial u}{\partial y} = \frac{\partial v}{\partial y} = 0 \tag{14}$$

3.2. Characteristic parameters

The present numerical simulation is aimed at investigating the impact that the channel angle may have on heat transfer characteristics and pressure drop for an entirely developed flow inside narrow corrugated channels. The average velocity at the inlet of the channel was calculated in terms of the Reynolds number and the hydraulic channel diameter; it is defined as:

$$u_{ave} = \frac{\mathrm{Re}\,\nu}{D_h} \tag{15}$$

The hydraulic diameter is defined as:

$$D_h = \frac{4S}{P} = 2H_{ave} \tag{16}$$

Dimensional analysis of the system shows the following dimensionless magnitudes:

Local friction coefficient

The local friction coefficient at the wall C_f is given by:

$$C_f = \frac{\tau_w}{\frac{1}{2}\rho u_{ave}^2} \tag{17}$$

Average friction coefficient

$$f = \frac{(\Delta p/L)D_h}{\frac{1}{2}\rho u_{ave}^2} \tag{18}$$

The pressure drop within the corrugated channel may be calculated as:

$$\Delta p = p_{ave,in} - p_{ave,out} \tag{19}$$

An approximation of the local heat transfer coefficient $h(x)$ can be obtained by:

$$h(x) = \frac{Q_w(x)}{T_w(x) - T_b(x)} \tag{20}$$

where Q_w is the heat flux density at the wall, T_w and T_b represent the average temperature of the wall and bulk flow temperature.

Average heat transfer coefficient

$$h = \frac{1}{L}\int_0^L h(x)dx \tag{21}$$

Local Nusselt number

$$h(x) = \frac{D_h}{k} h(x) \tag{22}$$

Average Nusselt number

$$Nu = \frac{1}{L}\int_0^L Nu(x)dx \tag{23}$$

As for the Performance evaluation criterion (PEC), it is given as:

$$PEC = \frac{(Nu_{ave} / Nu_{ave,s})}{(f / f_s)^{1/3}} \tag{24}$$

4. Numerical solution

The governing equations are a set of equations that govern the phenomenon of forced convection in a channel with velocity/pressure coupling, based on the finite volume method, SIMPLE algorithm of Patankar [22] is used to treat the problem of coupling velocity/pressure. A second order scheme (Upwind) and a structured uniform grid system are used to discretize the main governing equations, as shown in Fig. 2. In order to evaluate the accuracy of these calculations, the independence of the mesh is analyzed by adopting different grid distributions of 25 000, 65 000, 80 000 and 95 000 for *Re* = 1100. The test of independence of the mesh showed that the systems mesh of 80 000 nods guarantee a satisfactory solution. This is verified by the axial velocity profile in the corrugated channel with a mesh size greater than 80,000 (for example 95,000) in the 1% indicated in Fig. 3.
At the inlet, the fluid with an intensity turbulent [24], the inlet temperature, Tin, enters the studied section at the velocity of uin. The velocity boundary condition is applied to the input section while the pressure boundary condition is used at the output section. The commercial code ANSYS/Fluent has been used as a numerical solver. The numerical computation is completed if the residual added on all the computation nodes satisfies the criterion (10^{-5}).

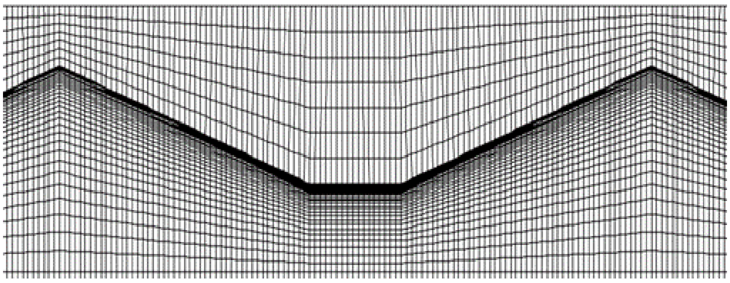

Fig. 2. Grid distribution for structured mesh inside the computational domain.

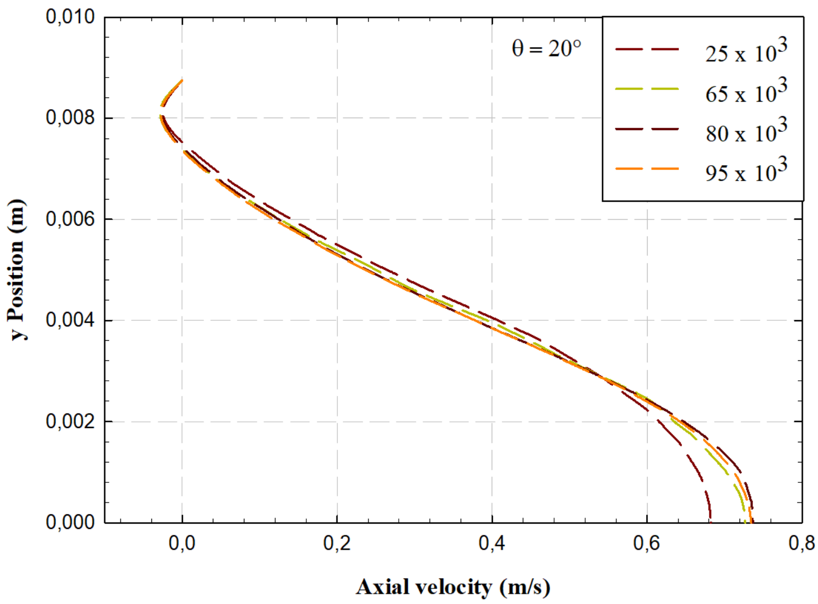

Fig. 3. Axial velocity profile for different number of elements at *Re* = 1100, and x = *L*/2, for inclination angle θ = 20°.

5. Validation of simulation results

For the purpose of validating the numerical results of the present study, the predicted outcomes for the average Nusselt number were compared with the previous experimental and numerical ones of Naphon [23], [24], as shown in Figs. 4 and 5.
Figure 4 illustrates a comparison between the average Nusselt numbers of the present numerical study and the values previously obtained by Naphon [23]. Likewise, Fig. 5 displays a comparison between the average Nusselt number of air in the corrugated channel and the numerical result attained by Naphon [24]. One may distinctly note from these two figures that the results of the present study are consistent with those reported in previous articles.

6. Results and discussion

The evolution of the average Nusselt number as a function of Reynolds number of air for various corrugated corners is depicted in Fig. 6. It is easy to observe that the Nusselt number rises as the Reynolds number of air increases, which may be attributed to the fact that the Nusselt number depends on the heat transfer rate. The average Nusselt number attains greater values for all corrugated corners as compared to a smooth channel. This same Fig. 6 shows the effect of a corrugated angle on the Nusselt number. Note that for larger corrugated angles, the Nusselt number is more important as compared to that obtained at smaller angles. Therefore, when the corrugation angle increases, i.e. the spacing decreases, there is a higher fluid recirculation with a larger eddy flow, which results in a greater intensity. Therefore, the number of Nusselt should grow as the corrugated angle goes up.

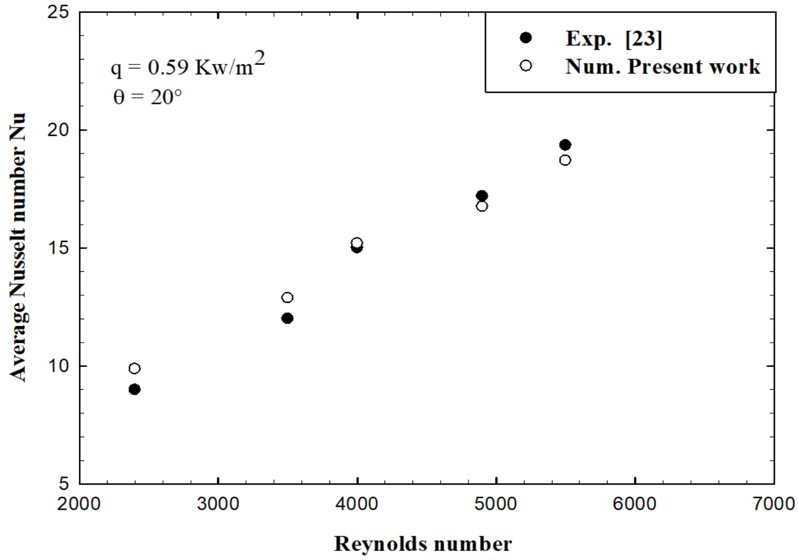

Fig. 4. Comparison between the average Nusselt numbers of our simulations and Naphon experimental results [23].

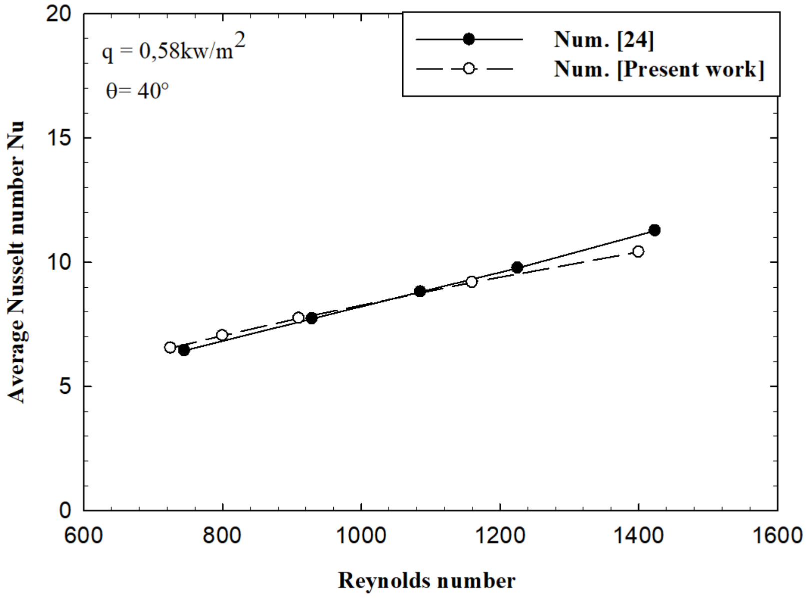

Fig. 5. Comparison between the average Nusselt numbers of our simulation and Naphon's numerical results [24].

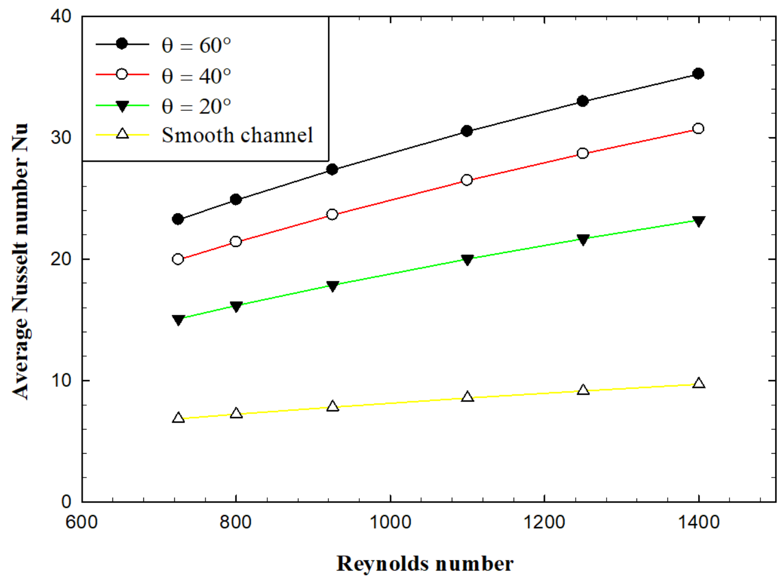

Fig. 6. Average Nusselt number for different inclination angles.

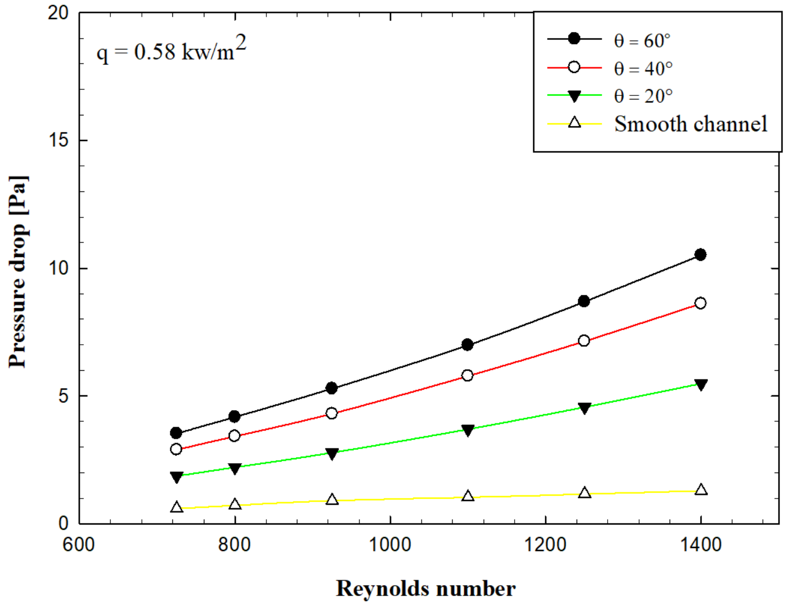

Fig. 7. Pressure drop for various inclination angles.

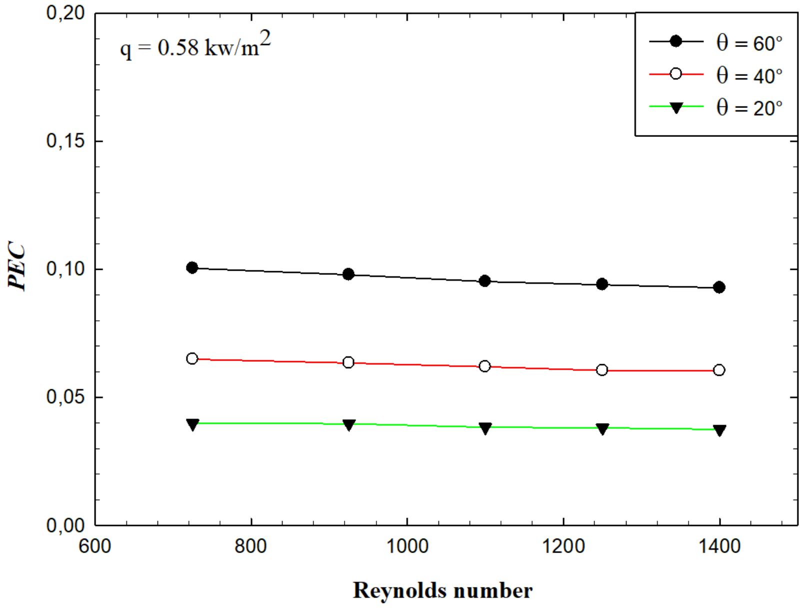

Fig. 8. Thermal–hydraulic performance factor vs. Reynolds number, for various angles.

The variation of the pressure drop as a function of the Reynolds number, for different inclination angles, is illustrated in Fig.7. This figure clearly suggests that the pressure drop keeps rising as the Reynolds number augments, for all inclination angles, which may be assigned to the turbulence effect and recirculation zones, which in turn leads to increased pressure drop. Therefore, it is possible to say that the inclination angle has a significant effect on the pressure drop which increases as the inclination angle rises. The heat transfer enhancement is always accompanied by a pressure drop increase.

The performance evaluation criteria (PECs) were used to assess the thermal-hydraulic performance, as illustrated in Fig. 8. It is noted that the highest wavy angle gives the best performance. One may readily observe that the performance over the entire range of Reynolds numbers is more or less constant for all wavy angles, while it starts decreasing beyond the wavy angle of 60°, as depicted in fig. 7.

Figure 9 shows the temperature contours and stream function for a channel with height H = 12.5 mm, and for different inclination angles. It can clearly be noted that the corrugated wall has a high impact on the temperature distribution and flow structure inside the channel.

It is therefore possible to conclude from the above figures that further increasing the channel angle θ leads to the appearance of swirling flows which continue to grow in a large part of the flow field, particularly in the narrow passage, due to the vortices and flow field along the corrugated wall. As a result, a higher temperature gradient appears near the corrugated wall. Fig. 9 distinctly indicates that the heat transfer rate between the wall and the fluid increases.

It is also worth noting that the onset of the swirling flow and its development encourage the mixing of the cold fluid from the core with the hot fluid from the region near the boundary layer. Furthermore, the corrugated geometry, with low wavy angles, has a slight effect on the flow across the channel. However, higher wavy angle values, i.e. smaller pitches, induce fluid recirculation or swirl flows in the corrugated troughs, which can therefore lead to higher momentum transfer.

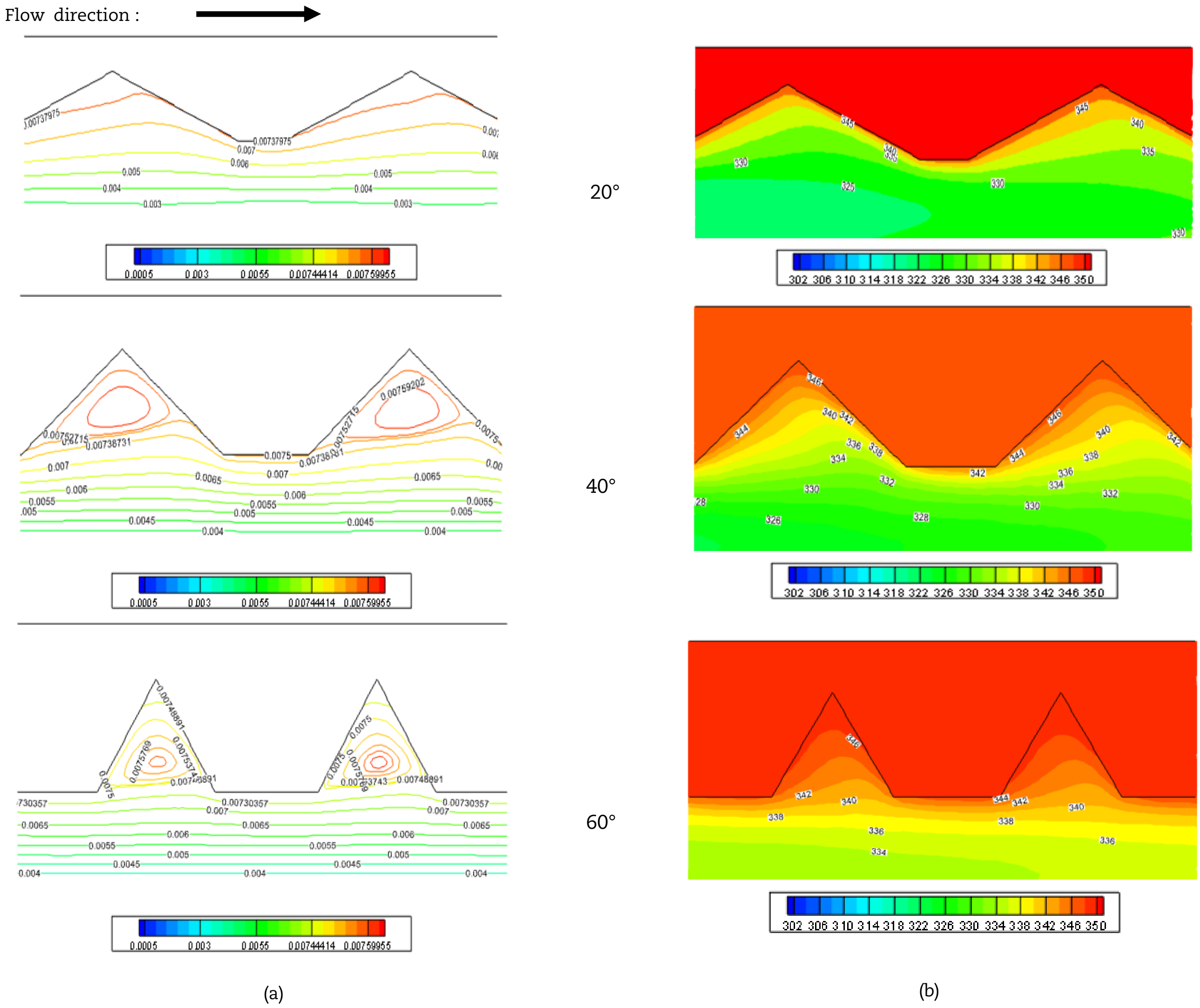

Fig. 9. Variation of (a) stream function, (b) temperature contour, at heat flux = 580 w/m², for θ = 20°, 40° and 60°, with Re = 1100.

7. Conclusion

The present study aimed at presenting the features of heat transfer and pressure drop within a channel with trapezoidal corrugations. Consequently, some interesting conclusions were drawn as:

- The heat transfer and pressure drop can be significantly enhanced by the corrugated surface as compared to the smooth surface.
- The average Nusselt number increases as the wavy channel's inclination angle rises.
- The average Nusselt number for the corrugated channel can be 4.5 times higher than that of a smooth channel.
- The breaking and destabilization of the thermal boundary layer occurs as the fluid flows near the corrugated surfaces.
- Corrugated plates can help to increase the thermal performance rate; it also allows for higher compactness of the heat exchanger.
- Considering the performance factor, it may be stated that a corrugated channel with a wavy angle of 60° gives the best outcomes with regard to energy saving, as compared to other configurations.

Author Contributions

The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed and approved the final version of the manuscript.

Nomenclature

Symbols		*Greek symbols*	
a	Wave amplitude, [mm]	α	Thermal diffusivity [m²/s],
Cp	specific heat, [kJ/kg. K]	ρ	density, [kg/m⁻³]
D_h	hydraulic diameter, [mm]	k	thermal conductivity of the fluid, [W/m.K]
f	friction factor	θ	wavy angle
h	heat transfer coefficient, [kW/m².C°]	μ	dynamic viscosity of the fluid, [kg/m.s]
H	height of channel, [mm]	ε	dissipation kinetic energy, [m².s³]
H_b	height of base [mm]	τ	wall shear stress

K	turbulent kinetic energy, [m^2/s^2]
Nu	Nusselt number
P	wet perimeter, [m]
q	heat flux, [kW/m^2]
Re	Reynolds number
S	surface, [m^2]
Pr	Prandt number
PEC	performance evaluation criteria
P	pressure, [pa]
T	temperature, [K]
u	Fluid velocity in x coordinate, [m/s]
v	Fluid velocity in y coordinate, [m/s]
ν	kinematic viscosity of the fluid, [m^2/s]
ϕ	Stream wise gradients, u, v, k, T, ε

Subscripts

ave	average
b	base
eff	effective
in	inlet
s	smooth
t	turbulent
O	outlet
w	wall

References

[1] Sunden, B., Skoldheden, T., Heat transfer and pressure drop in a new type of corrugated channels, *International Communication of Heat and Mass Transfer*, 12, 1985, 559-566.
[2] Sunden, B., Trollheden, S., Periodic laminar flow and heat transfer in a corrugated channel, *International Communication on Heat and Mass Transfer*, 16, 1989, 215-225.
[3] Mohammed, H.A., Abed, A.M., and Wahid, M.A., The effects of geometrical parameters of a corrugated channel with in out-of phase arrangement, *International Communication of Heat and Mass Transfer*, 40, 2013, 47-57.
[4] Elshafei, E.A.M., Awad, M., and Ali, A.G., Heat transfer and pressure loss in narrow channels with corrugated walls, *Second International Conference on Thermal Issues in Emerging Technologies THETA 2,* Cairo, Egypt, 2008, 279-290.
[5] Elshafei, E.A.M., Awad, M., and Ali, A.G., Heat transfer and pressure drop in corrugated channels, *Energy*, 35, 2010, 101-110.
[6] Ali, M., Ramadhyani, S., Experiments on convective heat transfer in corrugated channels, *Experimental Heat Transfer*, 5, 1992, 175-193.
[7] Islamoglu, Y., Parmaksizoglu, C., The effect of channel height on the enhanced heat transfer characteristics in a corrugated heat exchanger channel, *Applied Thermal Engineering*, 23, 2003, 979-987.
[8] Gut, J., Pinto, J.M., Modeling of plate heat exchangers with generalized configurations, *International Journal of Heat and Mass Transfer*, 46, 2003, 2571-2585.
[9] Zimmerer, C., Gschwind, P., Gaiser, G., and Kottke, V., Comparison of heat and mass transfer in different heat exchanger geometries with corrugated walls, *Experimental Thermal Fluid Science*, 26, 2002, 269-273.
[10] Hamza, A., Ali, HH., Experimental study on laminar flow forced-convection in a channel with upper v-corrugated plate heated by radiation, *International Journal of Heat and Mass Transfer*, 45, 2002, 2107-2117.
[11] Pehlivan, H., Experimental investigation of convection heat transfer in converging–diverging wall channels, *International Journal of Heat and Mass Transfer*, 66, 2013, 128-138.
[12] Singh, S., Singh, B., Hans, V.S., Gill, R.S., CFD (computational fluid dynamics) investigation on Nusselt number and friction factor of solar air heater duct roughened with non-uniform cross-section transverse rib, *Energy*, 84, 2015, 509-517.
[13] Rashidi, S., Akbarzadeh, M., Thermal-hydraulic and entropy generation analysis for turbulent flow inside a corrugated channel, *International Journal of Heat and Mass Transfer*, 109, 2017, 812-823.
[14] Ajeel, R.K., Salim, W.I., Hasnan, K., Design characteristics of symmetrical semicircle-corrugated channel on heat transfer enhancement with nanofluid, *International journal of Mechanical Sciences*, 151, 2019, 236–250.
[15] Ajeel, R.K., Salim, W.I., Hasnan, K., Heat transfer enhancement in semicircle corrugated channel: effect of geometrical parameters and nanofluid, *Journal of Advanced Research in Fluid Mechanics and Thermal Sciences*, 53, 2019, 82–94.
[16] Ajeel, R.K., Salim, W.I., Hasnan, K., Influences of geometrical parameters on the heat transfer characteristics through symmetry trapezoidal-corrugated channel using SiO2-water nanofluid, *International Communications in Heat and Mass Transfer*, 101, 2019, 1-9.
[17] Ajeel, R.K., Salim, W.I., Hasnan, K., Turbulent convective heat transfer of silica oxide nanofluid through corrugated channels: An experimental and numerical study, *International Journal of Heat and Mass Transfer*, 145, 2019, 118806.
[18] Ameur, H., Effect of Corrugated Baffles on the Flow and Thermal Fields in a Channel Heat Exchanger, *Journal of Applied and Computational Mechanics*, 6, 2020, 209-218.
[19] Akbarzadeh, M., Rashidi, S., and Esfahan, J.A., Influences of corrugation profiles on entropy generation, heat transfer, pressure drop, and performance in a wavy channel, *Applied Thermal Engineering*, 116, 2017, 278-291.
[20] Sakr, M., Convective heat transfer and pressure drop in V-corrugated channel with different phase shifts, *Heat and Mass Transfer*, 51, 2015, 129–141.
[21] Launder, B.E., Spalding, D.B., The numerical computation of turbulent flow, *Computer Methods in Applied Mechanics and Engineering*, 3, 1974, 269-289.
[22] Patankar, S.V., *Numerical heat transfer and fluid flow*, McGraw-Hill, New York, 1980.
[23] Naphon, P., Heat transfer characteristics and pressure drop in channel with V-corrugated upper and lower plates, *Energy Conversion and Management*, 48, 2007, 1516-1524.
[24] Naphon, P., Effect of corrugated plates in an in-phase arrangement on the heat transfer and flow developments, *International Journal of Heat and Mass Transfer*, 51, 2008, 3963-3971.

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