HEAT TRANSFER AND PRESSURE DROP EVALUATION OF DIFFERENT TRIANGULAR BAFFLE PLACEMENT ANGLES IN CROSS-CORRUGATED TRIANGULAR CHANNELS

by

Koray KARABULUT*

Electric and Energy Department, Sivas Vocational High School, Sivas Cumhuriyet University, Sivas, Turkey

Original scientific paper
https://doi.org/10.2298/TSCI190813466K

Plate heat exchangers have a widespread usage and the simplest parallel plate channel structures. Cross-corrugated ducts are basic channel geometries used in the plate heat exchangers. In this study, the increasing of heat transfer from the cross-corrugated triangular ducts by inserting triangular baffles with different placement angles into the channel upper side and pressure drop have been numerically investigated. Numerical calculations have been carried out to solve Navier-Stokes and energy equations by employing k-ε turbulence model as 3-D and steady with ANSYS-FLUENT program. While inlet temperature of the air used as working fluid is 293 K, constant surface temperature values of the the lower corrugated channel walls are 373 K. The height of the baffle and apex angle of the corrugated duct have been taken constant as 0.5 H and 60°, respectively. Investigated Reynolds number range is 1000-6000 while the baffle placement angles are 30°, 45°, 60°, and 90°. Numerical results of this study are within 3.53% deviation with experimental study existed in literature. The obtained results have been presented as mean Nusselt number temperature and pressure variations of the fluid for each baffle angle. The temperature and velocity vector contour distributions have been also assessed for different Reynolds numbers and baffle angles. The value of the Nu_m *for the corrugated channel with 60° baffle angle is 8.2% higher than that of the 90° for the Re = 4000. Besides, for Re = 1000 the value of the pressure drop is 39% lower in the channel with 60° baffle angle than that of 90°.*

Key words: *cross-corrugated triangular channel, baffle, angle, heat transfer*

Introduction

Plate heat exchangers offer the highest technology, efficiency and ease of use in a wide range of applications from special construction technology to the automotive sector and general industrial uses from food processing to chemical and pharmaceutical applications. However, cross-corrugated ducts are extensively employed in the plate heat exchangers. Generally, shape of cross-triangular or cross-sinusoidal is used as a corrugation profile. In the plate heat exchangers, many corrugated plates are connected to each other to form flow channels. These flow ducts are constituted by giving an angle to adjacent plates and so, hot and cold fluids are segregated each other by means of these plates. Thus, the cross-corrugated channel whose fundamental geometrical features are angle of inclination, apex angle, corruga-

* Authors' e-mail: kkarabulut@cumhuriyet.edu.tr

tion height and pitch is obtained. The corrugation profile of the cross-corrugated channel has vortex and secondary flow production ability in corrugation troughs, which augments heat transfer between adjacent flow ducts. This increment substantially depends on the channel geometric properties.

Many numerical and experimental works were carried out to investigate heat transfer performance of the cross-corrugated channel. In one of these researches, Focke and Knibbe [1] performed experimental and numerical flow and heat transfer study for a crossed-corrugated geometry represented compact heat exchangers. They analyzed the effects of corrugation angle, geometry and Reynolds number. In the range researched, the average Nusselt number was found to increase approximately as $Re^{2/3}$ for all geometries and also the typical effects of the corrugation angle and the pitch-to-height ratio on Nusselt number were evaluated. Han *et al.* [2] were simulated temperature, pressure, and velocity fields of the widely used chevron corrugated-plate heat exchanger as 3-D. They determined that the temperature gradient increased gradually and got the maximum value in the central of the flow and then it became smaller again. When the simulated results were compared with the experimental values, they found that the outside temperature trends were consistent with those of pressure drop and also experimental values were similar to those of simulated values. Liu and Niu [3] numerically searched the effects of channel sizes and apex angles on the heat transfer, pressure drop and thermohydraulic performance in the corrugated duct. They found that the apex angle substantially affected the heat transfer and pressure drop in the cross-sectional corrugated channel. In an another study, Sharif *et al.* [4] investigated the effect of the apex angle on the thermal and hydraulic properties of the triangular cross-corrugated heat exchangers for a Reynolds number range of 310-2064. They observed that pressure drop and heat transfer coefficient enhance with increasing of the apex angle and the highest turbulence intensity, friction factor and Colburn j factor were achieved around the apex angles of 90°-100°. Hall *et al.* [5] obtained a correlation for the coefficient of mass transfer by implementing experiments on a triangular cross-corrugated heat exchanger having 90° apex angle at the Reynolds number range of 200-2300. They found that in the studied Reynolds number range the flow regime was not laminar. Guo-Yan *et al.* [6] carried out a research both experimentally and numerically for a sinusoidal cross-corrugated heat exchanger and the Reynolds numbers between 84 and 1168. They determined that laminar flow with steady-state was a logical selection for this Reynolds number range. However, it was found to be around 15% deviation between the experimental and numerical results for Colburn j, and friction factor.

Baffles which are extensively used in shell and tube heat exchangers can influentially enhance the heat transfer area. Besides, they have properties in varying the mainstream direction, enhancing flow disturbance and heat transfer. However, there are very few studies on baffle caused to increase heat transfer by composing the turbulent flow in the channel at the low Reynolds number. Handoyo *et al.* [7] numerically researched the heat transfer and pressure drop characteristics of delta-shaped obstacles for V-corrugated channels by taking into considering interval between obstacles. Determined ratio of the interval to height of the delta-shaped obstacles was varied range from 0.5 to 2. They found that when the ratio of the interval to height was 0.5, Nusselt number and friction factor increased 3.46 and 19.9 times, respectively. Leung and Robert [8] examined thermal characteristic of air-flow with turbulence regime which had Reynolds number range of 5000-20000 for three heat exchangers having isosceles triangular corrugations with 40°, 60°, and 90° apex angles. Zhang [9] numerically researched the heat transfer and flow features in a triangular cross-corrugated heat exchanger having apex angle of 60° by using k-ω turbulence model with Reynolds number range of 100-

6000. They suggested two correlations related to Nusselt number depending on boundary conditions of constant heat flux and temperature in their work. Liang *et al.* [10] carried out a numerical study to analyze the effect of baffle on flow and temperature distribution in the cross-corrugated triangular channels having six different baffle installations by using a k-ω turbulence model. They obtained as a result that the Nusselt number of the duct with baffle for the case of setup 6 was 1.5-1.6 times higher than that of the channel without baffle case.

As previously discussed, the many of the studies in the examined literature focus on the effects of different baffle setups, baffle heights and intervals and apex angles of the corrugation troughs on the heat transfer and flow characteristics in the cross-corrugated channels. Unlike the mentioned studies, in this work, the influences of the placement angles of the triangular baffles used as suitable to the cross-corrugated triangular channel have been investigated on the heat transfer, flow structure and pressure variation. Numerical investigation has been carried out by solving the steady, 3-D energy and Navier-Stokes equations using ANSYS-FLUENT software program with the k-ε turbulence model. In the corrugated channel used in this study has been designated by considering the works in the literature. While the inlet temperature, T_i, to the channel having equilaterals of the air used as working fluid is 293 K, wall surface temperature values, T_s, that are applied as constant in the the lower corrugated channels are 373 K. The Reynolds number and the baffle placement angles, θ, to the top section of the corrugated channel have been considered as variable parameters. The study has been carried out using a constant 60° apex angle, which has the best heat transfer property according to datas obtained in the literature. Investigated Reynolds number range is 1000-6000 while the baffle placement angles are 30°, 45°, 60°, and 90°. Also, the height of the baffle has been taken constant as half of the top channel height (H_b = 0.5 H). The numerical results of the presented work have been compared with the experimental and numerical results of the studies in the literature and it has been seen that the results are agreement with each other. The obtained results have been presented as temperature and pressure variations of the fluid along the middle section of the top corrugated channel and changes of the fluid outlet temperature and the mean Nusselt number for each baffle angle and analyzed by comparing with each other and case of the without baffle. The temperature and velocity vector contour distributions of the fluid along the cross-corrugated triangular channel have been evaluated for the different angles of the triangular baffles and the Reynolds number value of 1000.

Numerical method

The solution of the flow and heat transfer through the cross-corrugated channels having different baffle placement angles was performed by the solution of PDE derived from the equations of time-averaged mass, momentum and energy conservation for turbulent flow under the steady-state conditions where there was not body forces and velocities and mass fractions were divided into a mean and fluctuating value ($u_j = \bar{u}_j + u'_j$ and $T = \bar{T} + T'$) as described below [11].

– Continuity equation

$$\frac{\partial(\bar{u}_i)}{\partial x_i} = 0 \tag{1}$$

– Momentum equation

$$\frac{\partial}{\partial x_j}(\bar{u}_i \bar{u}_j) = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \nu \frac{\partial}{\partial x_j}\frac{\partial \bar{u}_i}{\partial x_j} - \frac{\partial \overline{u'_i u'_j}}{\partial x_j} \tag{2}$$

– Energy equation

$$\frac{\partial}{\partial x_j}(\bar{u}_j \bar{T}) = \left(\frac{\nu}{\mathrm{Pr}} + \frac{1}{\rho} \frac{\mu_t}{\mathrm{Pr}_t} \right) \frac{\partial}{\partial x_j} \frac{\partial \bar{T}}{\partial x_j} \tag{3}$$

where ρ is the density of the fluid, p – the pressure, u_i – the velocity component, μ – the dynamic viscosity, ν – the kinematic viscosity, and Pr – the Prandtl number.

These averaged equations include the Reynolds stress term $\overline{u'_i} \overline{u'_j}$ which shows the effect of the turbulence on the mean flow field. The Reynolds stress terms can be explained:

$$-\rho \bar{u}_i \bar{u}_j = \mu_\mathrm{t} \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \frac{2}{3} k \rho \delta_{ij} \tag{4}$$

where δ_ij indicates the Kronecker delta function. It is unity for $i = j$ and zero otherwise. The turbulence viscosity, μ_t, is forthrightly related to the turbulence kinetic energy, k, and energy dissipation rate, ε:

$$\mu_\mathrm{t} = 0.09 \rho \frac{k^2}{\varepsilon} \text{ and } k = \frac{1}{2}\left(\overline{u}_i'^2 + \overline{u}_j'^2 \right) \tag{5}$$

The hydraulic diameter of the duct is defined:

$$D_\mathrm{h} = \frac{4A_\mathrm{c}}{P} = \frac{4\left(\frac{WH}{2} \right)}{3W} = \frac{2}{3} H \tag{6}$$

where A_c and P are the fluid inlet cross-sectional area and perimeter length of the fluid inlet, respectively.

The Reynolds number, is calculated by:

$$\mathrm{Re} = \frac{u_\mathrm{m} D_\mathrm{h}}{\nu} \tag{7}$$

where u_m is the mean velocity of the fluid in the duct entry.

Nusselt number is considered to be ratio of conduction heat transfer rate of convection heat transfer rate:

$$-k \left(\frac{\partial T}{\partial n} \right)_\mathrm{s} = h \Delta T_\mathrm{lm} \text{ and } \mathrm{Nu} = \frac{hL}{k} \tag{8}$$

where h is the local convective heat transfer coefficient on the surface, n – the perpendicular direction to wall surface of the duct bottom corrugation, and Nu – the local Nusselt number.

Logarithmic temperature difference between throughout wall of the corrugated channel and fluid is calculated by:

$$\Delta T_\mathrm{lm} = \frac{(T_\mathrm{i} - T_\mathrm{o})}{\ln\left(\frac{T_\mathrm{w} - T_\mathrm{i}}{T_\mathrm{w} - T_\mathrm{o}} \right)} \tag{9}$$

where T_i, T_o, and T_w are the fluid inlet and outlet temperatures and the wall surface temperature in the bottom section of the cross-corrugated duct, respectively.

Mean heat transfer coefficient, h_m, and mean Nusselt number, Nu_m:

$$h_m = \frac{1}{L}\int_0^L h\mathrm{d}x \quad \text{and} \quad \mathrm{Nu}_m = \frac{h_m L}{k} \tag{10}$$

Pressure drop can be calculated with:

$$\Delta p = \frac{1}{2} f \rho \frac{L}{D_h} u_m^2 \tag{11}$$

where Δp is the pressure drop of the cross-corrugated channel between the inlet and outlet, f – the friction factor, and L – the channel length.

Dimensionless variable is defined:

$$z^* = \frac{z}{L} \tag{12}$$

where the z points out local duct length.

Geometric model

Perspective view of the corrugated channel with baffle used in the calculations and duct view including boundary conditions and the corrugated channel sizes are indicated in figs. 1(a) and 1(b), respectively. While the length, L, and width, W, of the corrugated channel are 70.71 mm and 8.165 mm, respectively, the height of the channel, H, and distance between two baffles, d, are 7.071 mm and 8.165 mm, respectively. Also, the used triangular baffle height, H_b, is 3.5355 mm (H_b = 0.5 H). However, the inlet velocity range of the fluid is 2.1436 m/s (Re = 1000) – 12.8616 m/s (Re = 6000). Depending on the duct sizes employed in literature, there are ten triangular baffles and corrugated troughs in the channel lower section. Besides, the constant wall surface temperature (T_s = 373 K) is valid as a boundary condition on the lower surfaces of the corrugated duct when the inlet temperature of the air is 293 K. However, this study has been performed under the following assumptions: the flow domain for the corrugated duct is 3-D, steady and turbulent, the used working fluid is incompressible air, the thermal properties of the fluid are constant, and there is not heat generation for the fluid and the baffle surfaces.

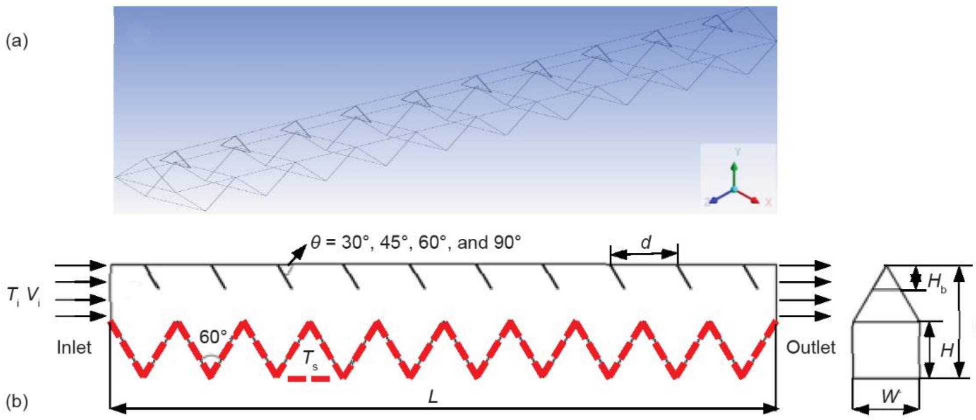

Figure 1. (a) Perspective view of the corrugated duct with baffle used in calculations, (b) channel view with boundary conditions and corrugated channel sizes

Assessment of the results

The mean Nusselt number value results of experimental of Scott and Lobato [12] and numerical of Liu and Niu [3], Zhang [9], and Li and Gao [13] are pointed out in figs. 2(a) and 2(b) for 60° and 90° apex angles of the corrugated channels without baffle as comparative with the numerical results of this study, respectively. While fig. 2(a) exhibits the comparisons of numerical data from Zhang [9] carried out by using a low Reynolds number k-ω turbulence model and Li and Gao [13] performed using k-ε turbulence model and this study for the corrugated duct with 60° apex angle and without baffle. Figure 2(b) displays the comparisons of the experimental result of Scott and Lobato [12] and numerical results of the Liu and Niu [3] done by using Reynolds stress model, Li and Gao [13] and this study for the corrugated channel with 90° apex angle and without baffle. The maximum deviation of this study for Nu_m is 4.8% compared with numerical results for 60° apex angle, fig. 2(a). However, the differences between the numerical results of this study and experimental values are within 3.53% for 90° apex angle, fig. 2(b). By employing k-ε turbulence model, the level of agreement between numerical and experimental is considered good enough to provide comparative studies for different baffle angles and therefore is used in the following research.

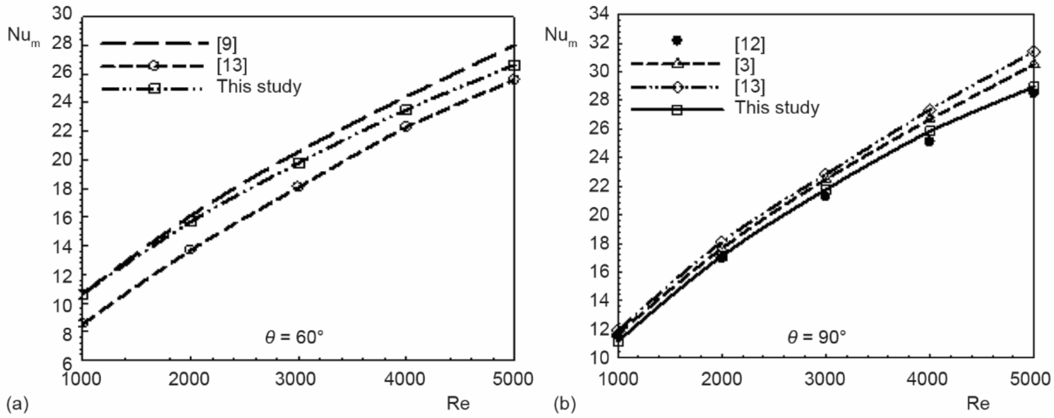

Figure 2. Comparison of the experimental and numerical results with the results of the this numerical study; apex angles of (a) 60° and (b) 90°

To define the effect of grid number on the mean Nusselt number of the corrugated channel surfaces, grid independence tests for the corrugated channel without baffle and the different Reynolds number values of 1000, 2000, and 3000 are displayed in tab. 1. According to obtained results, it is determined that 643673 grid elements are adequate for the corrugated channel without baffle. However, grid elements of 743673 have been employed for the corru-

Table 1. Results of the grid independence test for the Nu_m at the corrugated channel without baffle

Mesh numbers	Nu_m Re = 1000	Nu_m Re = 2000	Nu_m Re = 3000
450571	9.6245	14.6674	18.7645
643673	9.6465	14.6830	18.7880
836774	9.6446	14.6824	18.7572

gated duct with triangular baffle and used mesh has composite structure composed of tetrahedral and quad types as suitable duct type.

The distributions of the velocity vector contour for the Reynolds number of 1000 and the first five corrugations are pointed out in fig. 3 for the cross-corrugated triangular channels without baffle and with different triangular baffle placement angles of 30°, 45°, 60°, and 90°, respectively. In the case of without baffle, the fluid flows as parallel flow throughout the top half of the duct and thus the fluid amount flowing in the corrugation troughs of the lower channel is small. However, clockwise vortices occur in the bottom corrugation troughs. When the triangular baffles with 30° location angle are inserted into channel upper side, directions of the vortices are about same as that of the channel without baffle in the lower corrugation troughs. As shown in fig. 3, the vortices begin to occur rear of the baffles. Besides, while the angle of the baffle location is increased from 45° to 90°, densities of the vortices both in rear of the baffles and the lower corrugation troughs augment. For Re = 1000, as can be seen in the cross-corrugated triangular channel with 90° baffle placement angle, the vortex density is higher than that of the 60°. This case can change depending on the Reynolds number that affects the vortex formation. However, the occurred vortices move clockwise in the corrugation of the bottom channel and anticlockwise in the rear of the baffles in the top channel.

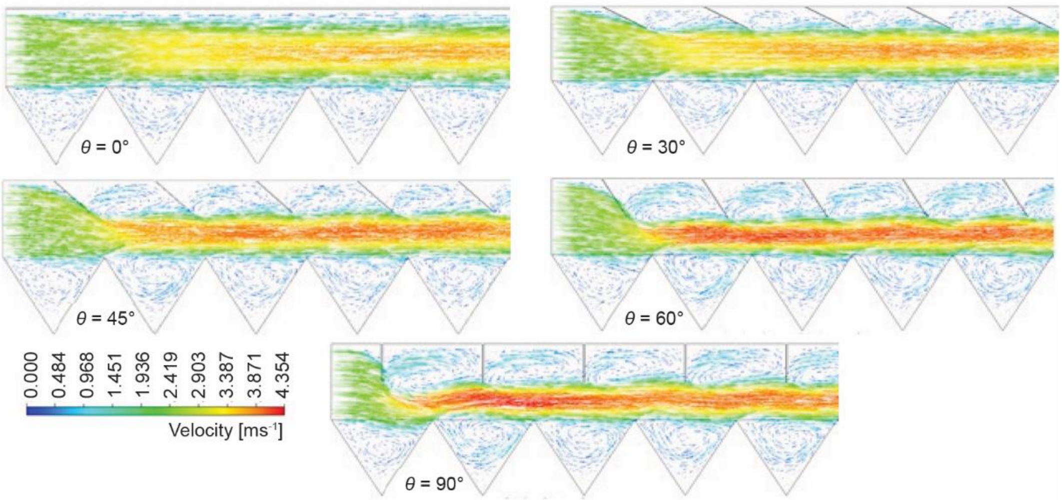

Figure 3. Velocity vector contour distributions of the corrugated triangular channels (Re = 1000)

The temperature contour distributions of the cross-corrugated triangular duct with and without triangular baffles having different placement angles are given in fig. 4. High temperature gradients are obtained close to the trough walls in the corrugation and towards central area of the cross-corrugated triangular channel with different baffle angle according to channel without baffle. This means that the angled baffles direct the flow to troughs better and so the heat transfer improve.

Variations of the temperature along the middle section of the top corrugated channel are indicated in fig. 5 for the different triangular baffle angles of 30°, 45°, 60°, and 90°, respectively. Temperature values in the cross-corrugated triangular channel change according to motion of the fluid in the troughs. For the corrugated channels with baffle, the temperature variations in the channel with triangular baffles whose placement angles are 60° and 90° are

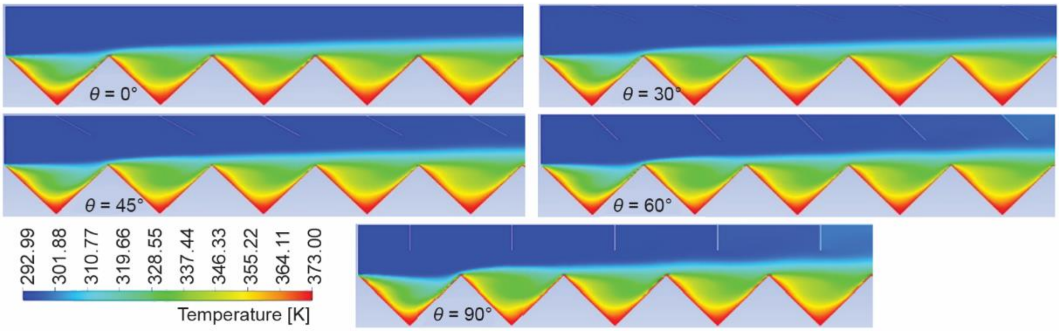

Figure 4. Temperature contour distributions of the corrugated triangular channels (Re = 1000)

higher than that of the 30° and 45° towards outlets of the channels, which demonstrates becoming better heat transfer enhancement from the bottom through walls of the channels with the baffle angles of 60° and 90°. Meanwhile, a flow as jet like occurs due to restriction of the flow in the cross-section of the top corrugated channel by baffles. Besides, it is seen from the graphs that the temperature of the fluid decreases depending on the increasing of the Reynolds number. However, because of larger vortexes and turbulence caused by the baffles with 60° angle for the value of the Re = 6000, the temperature values are higher than of the Re = 4000.

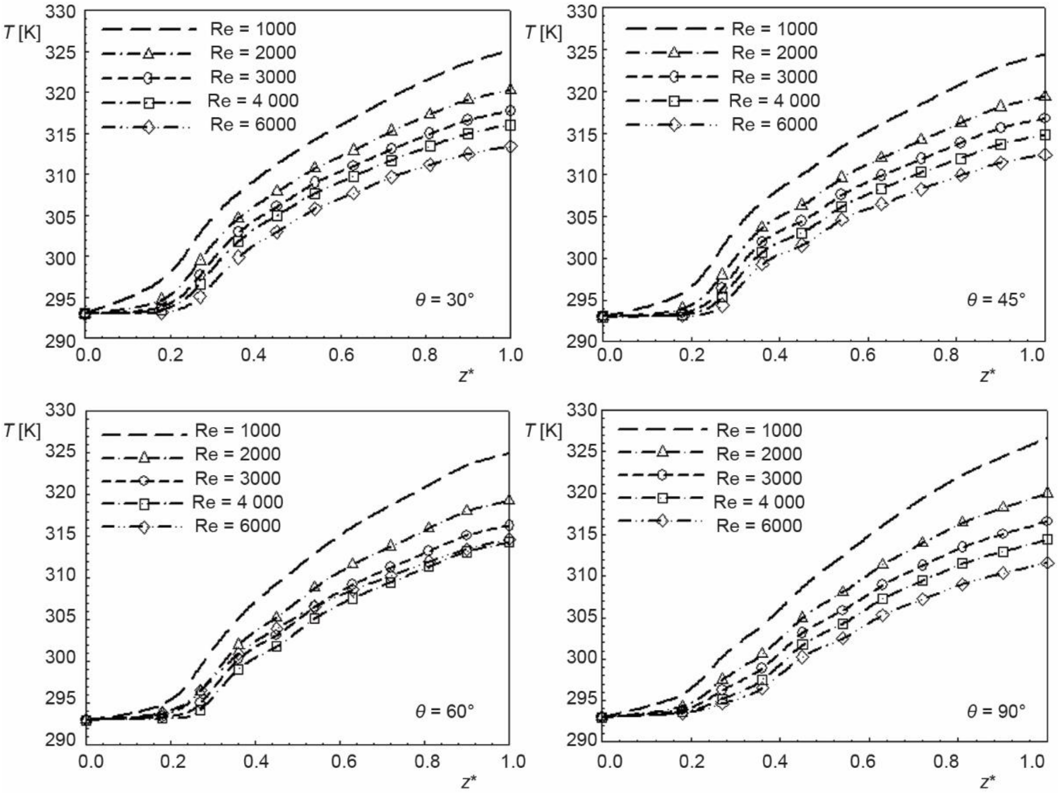

Figure 5. Temperature variations of the corrugated triangular channels with different baffle angles

In fig. 6, pressure variations of the fluid along the cross-corrugated triangular ducts are shown for Reynolds numbers of 1000 and 6000, respectively. While the pressure of the fluid decreases towards outlet of the channel, pressure drop throughout the duct increases with increasing of Reynolds number. However, the highest pressure drop is obtained for the cross-corrugated triangular duct with 90° baffle angle. When the pressure difference between the channel inlet and outlet is 635.26 Pa for the Re = 6000 in the cross-corrugated channel with 90° angle, it is 194.88 Pa for the channel without baffle. Besides, for Re = 1000 the pressure drop value is 39% lower in the corrugated triangular channel with 60° baffle angle than that of 90° angle. In addition to this, fluctuation in the pressure value is seen for Re = 6000 depending on the fluid motion in the corrugation troughs of the duct bottom wall and around the baffles of the channel top wall for 60° baffle angle.

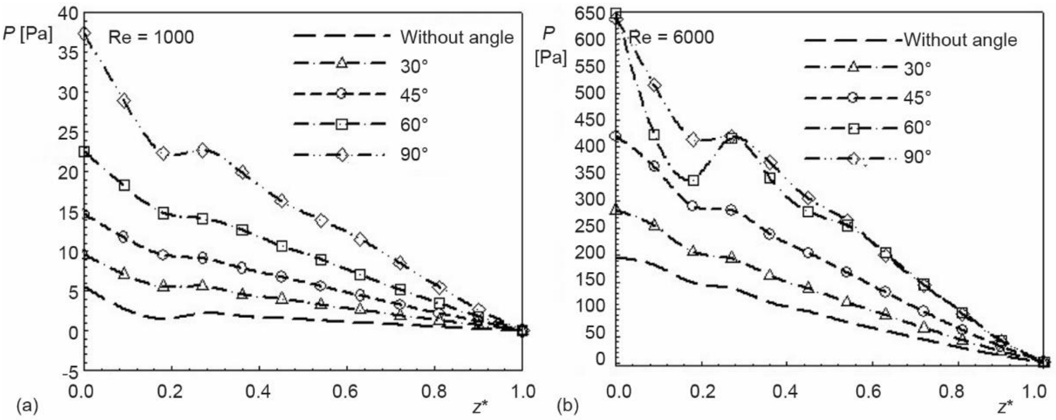

Figure 6. Pressure variations of the fluid in the corrugated triangular channels for the Reynolds numbers of (a) 1000 and (b) 6000

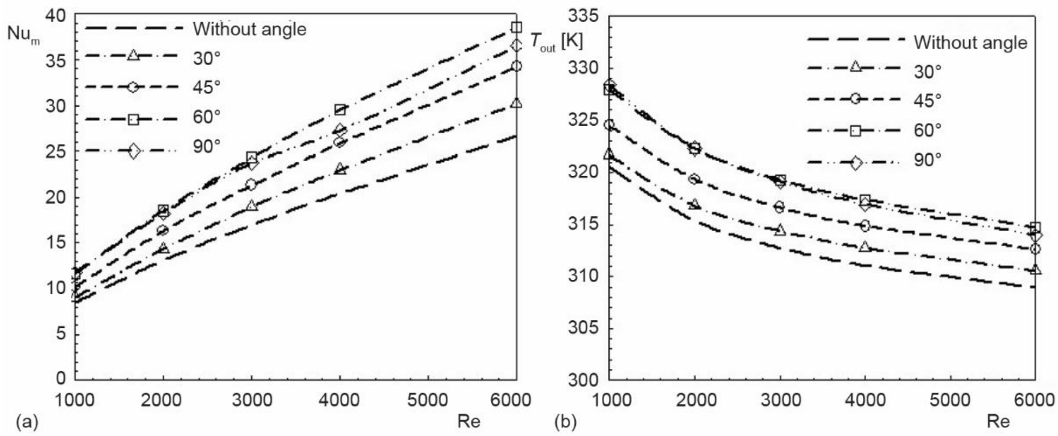

Figure 7. Variations of the fluid in the corrugated triangular duct for (a) Nu_m and (b) T_{out}

The variations of the Nu_m and fluid outlet temperature, T_{out}, are shown in figs. 7(a) and 7(b) *vs.* the Reynolds number in the cross-corrugated triangular ducts. As seen in figs. 7(a) and 7(b), while the Nu_m increases, T_{out} reduces with increasing of the Reynolds number. However, when the Nu_m and T_{out} in the corrugated channel of the 60° baffle placement angle are higher than that of the 90° for the Reynolds number values of 1000, 2000 and 3000 depending on the fluid motions occurred in the corrugation troughs of the duct due to complex geometry with similarity to internally finned tubes and flow pattern caused by the baffles, they decrease for the Re = 4000 and 6000. As a result of this, the value of the Nu_m for the corrugated channel with 60° is 8.2% higher than that of the 90° for the Re = 4000. The increasing of the fluid outlet temperature is attributed to the heat transfer among the fluid and walls of the troughs in the lower section of the corrugated duct, which is more for the cross-corrugated channels with baffles.

Conclusions

In this study, it is aimed to research the heat transfer and flow characteristics of the cross-corrugated triangular ducts by inserting the triangular baffles with different placement angles into the channel upper side. This work has been carried out numerically by solving the steady, 3-D energy and Navier-Stokes equations using ANSYS-FLUENT software program with the k-ε turbulence model. By this numerical investigation, the main conclusions have been obtained as follows.

- In the case of without baffle, the fluid flows as parallel flow throughout the top half of the duct and thus the fluid amount flowing in the corrugation troughs of the lower channel is small. By adding baffles, vortex densities increase in the lower corrugations of the corrugated ducts. However, for Re = 1000, in the cross-corrugated triangular channel with 90° baffle placement angle, the vortex density has been higher than that of the 60°. Besides, the occurred vortices move clockwise in the corrugation of the bottom channel and anticlockwise in the rear of the baffles of the top channel.
- High temperature gradients have been obtained close to the trough walls in the corrugations and towards central area of the cross-corrugated triangular channel with different baffle angle according to duct without baffle.
- The temperature variations of the channel with triangular baffles whose placement angles are 60° and 90° are higher than that of 30° and 45° towards outlet of the channels, which demonstrates becoming better heat transfer enhancement from the bottom trough walls of the channels with baffle angles of 60° and 90°.
- Also, the triangular baffles constitute resistance against to flow in the channel, which results in pressure drop. However, the highest pressure drop has been obtained for the cross-corrugated triangular duct with 90° baffle angle. For Re = 1000 the pressure drop value of the 60° baffle angle is 39% lower than that of 90°. Besides, the value of the Nu_m for the corrugated channel with 60° baffle angle is 8.2% higher than that of the 90° for the Re = 4000.
- As a conclusion, by inserting triangular baffles into the cross-corrugated triangular channels, the heat transfer enhancements have been achieved with relatively negligible pressure drop and the best performances of the heat transfer have been obtained for the 60° and 90° placement angles. In addition to this, it is thought that the results of this study will contribute to mechanical and energy engineers and manufacturers in the design and optimization of the cross-corrugated plate heat exchangers having baffles.

Nomenclature

A_c – inlet cross-sectional area, [m^2]
d – distance between two baffles, [m]
D_h – hydraulic diameter, [m]
f – friction factor, [–]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
H – the height of the channel, [m]

k – turbulence kinetic energy, [m^2s^{-2}]
Nu – Nusselt number, [–]
P – perimeter length of the inlet, [m]
Pr – Prandtl number, ($=\mu c_p/k$), [–]
p – pressure, [Pa]
Re – Reynolds number, ($=u_m D_h/\nu$), [–]
T – temperature, [K]
u_i – velocity component, [ms^{-1}]
$\overline{u_i' u_j'}$ – Reynolds stres term, [m^2s^{-2}]
W – channel width, [m]

Greek symbols

μ – dynamic viscosity, [$kgs^{-1}m^{-1}$]
δ_{ij} – Kronecker delta function, [–]
ε – energy dissipation rate, [$m^2s^{-2}s^{-1}$]
θ – baffle placement angle, [°]
ν – kinematic viscosity, [m^2s^{-1}]
ρ -density, [kgm^{-3}]

Subscript

b – baffle
i – inlet
s – surface
o – outlet

References

[1] Focke, W. W., Knibbe, P. G., Flow Visualization in Parallel-plate Ducts with Corrugated Walls, *Journal of Fluid Mechanics*, *165* (1986), Apr., pp. 73-77
[2] Han, X. H., *et al.*, A Numerical and Experimental Study of Chevron, Corrugated-Plate Heat Exchangers, *International Communications Heat and Mass Transfer*, *37* (2010), 8, pp. 1008-1014
[3] Liu, X. P., Niu, J. L., Effects of Geometrical Parameters on the Thermohydraulic Characteristics of Periodic Cross-Corrugated Channels, *International Journal of Heat and Mass Transfer*, *84* (2015), May, pp. 542-549
[4] Sharif, A., *et al.*, Comparative Performance Assessment of Plate Heat Exchangers with Triangular Corrugation, *Applied Thermal Engineering*, *141* (2018), Aug., pp. 186-199
[5] Hall, D., *et al.*, Determination of Mass Transfer Coefficient of a Cross-Corrugated Membrane Reactor by the Limiting-Current Technique, *International Journal of Heat and Mass Transfer*, *44* (2001), 12, pp. 2201-2207
[6] Guo-Yan, Z., *et al.*, Numerical and Experimental Study on the Heat Transfer and Pressure Drop of Compact Cross-Corrugated Recuperators, *Journal of Heat and Mass Transfer*, *136* (2014), 7, 071801
[7] Handoyo, E. A., *et al.*, Numerical Studies on the Effect of Delta-Shaped Obstacles' Spacing on the Heat Transfer and Pressure Drop in V-Corrugated Channel of Solar Air Heater, *Solar Energy*, *131* (2016), June, pp. 47-60
[8] Leung, C., Probert, S., Forced-Convective Turbulent Flows Through Horizontal Ducts with Isosceles-Triangular Internal Cross-Sections, *Applied Energy*, *57* (1997), 1, pp. 13-24
[9] Zhang, L., Numerical Study of Periodically Fully Developed Flow and Heat Transfer in Cross-Corrugated Triangular Channels in Transitional Flow Regime, *Numerical Heat Transfer Part A: Applications*, *48* (2005), 4, pp. 387-405
[10] Liang, C. H., *et al.*, Numerical Studies on the Effect of Baffle on the Heat Transfer and Flow in Cross-Corrugated Triangular Ducts, *IOP Conference Series: Earth and Environmental Science*, *238* (2019), 1, 012086
[11] *****,** FLUENT User's Guide, Fluent Inc. Lebanon, Netherland, 2003
[12] Scott, K., Lobato, J., Mass Transport In Cross-Corrugated Membranes and the Influence of TiO_2 for Separation Processes, *Industrial&Engineering Chemistry Research, 42* (2003), 22, pp. 5697-5701
[13] Li, Z., Gao, Y., Numerical Study of Turbulent Flow and Heat Transfer in Cross-Corrugated Triangular Ducts with Delta-Shaped Baffles, *International Journal of Heat and Mass Transfer*, *108*, (2017), May, pp. 658-670

Paper submitted: August 13, 2019
Paper revised: December 14, 2019
Paper accepted: December 18, 2019