FLOW CHARACTERISTICS AND HEAT TRANSFER FOR IN-PHASE TRAPEZOID AIR CHANNELS

R. Y. SAKR*, H. E. FAWAZ**, A. A. LASHINE*

*Faculty of Engineering, Shoubra, Benha University
**Mechanical Engineering Department, National Research Centre

ABSTRACT

This work is concerned with the numerical investigations of heat transfer enhancement and friction factor penalty inside corrugated-plate air channels. The investigation was evaluated for in-phase trapezoidal corrugation shapes. The influence of the flow rate represented by Reynolds number (100-1000) and cutting edge ratio (0-0.6) on the flow characteristics and heat transfer were investigated. The flow characteristics were depicted through the presentation of stream function contours, local wall shear stress, air velocity profiles and isothermal fanning friction factor. Whereas the heat transfer characteristics were determined through isotherms contours, air temperature profiles coefficient, local and average Nusselt number.

Lateral vortices or re-circulation cells developed in the corrugated-wall troughs as the axial flow get separated downstream of the wall-corrogation peaks and re-attaches upstream of the subsequent wall peak, and their strength and coverage was a function of Reynolds number and cutting edge ratio. By cutting the sharp edged corners for trapezoidal channel, the large lateral vortex of the triangular channel was gradually damped resulting in lower recirculation area, lower recirculation velocities and lower velocity gradients which damping the high peaks of the shear stress observed in the triangular case resulting in lower isothermal friction factor. However, the mixing effect produced by the lateral vortex decreased with its damping, which led to less uniform temperature profile resulting in lower sharper wall temperature gradients with thicker boundary layer thickness, so lower values of the Nusselt number was obtained.

New correlations were developed based on the present results to evaluate Nusselt number and friction factor as a function of Reynolds number, corrugation aspect ratio, relative spacing ratio and cutting edge ratio for different corrugation shapes. The predicted results are consistent with the numerical data and lie within ±10% deviation.
KEYWORDS: trapezoidal channel, heat exchanger, forced convection, corrugated channel

1. INTRODUCTION:

The optimal design of heat exchangers is required to increase energy savings. The way to improve the global performances of plate heat exchangers is to find efficient heat transfer surfaces which do not induce much pressure loss. Using corrugated plates is a suitable method to increase the compactness which is needed for automotive, space and aeronautic devices. Besides, the wavy geometries are known to enhance the heat transfer by breaking and destabilizing the thermal boundary layer. So, corrugated surfaces serve as turbulence promoters to increase the local heat and mass transfer. Several studies on heat transfer enhancement using such passages have been reported.

Goldstein and Sparrow [1] were the first to report local and average convection heat transfer coefficients in a corrugated passage using the naphthalene technique. Their passage included two different corrugations with an angle of 30°. Comparison of their results with parallel plate passage data showed an enhancement in the average convection heat transfer by a factor of 3 in turbulent flow regimes. O'Brian and Sparrow [2] obtained an empirical correlation of the average Nusselt number for turbulent flow in corrugated channels with sharp edged corrugation peaks. They reported that friction factor was almost independent of Reynolds number. The promoted flow separation accompanied with sharp edged corrugations however, causes large pressure drops. Sparrow and Comb [3] reported that the increase of the channel height resulted in a valuable increase in Nusselt number but the friction factor substantially increased. Islamoglu and Parmaksizoglu [4] investigated the heat transfer coefficient and friction factor for air flowing in a corrugated channel of two different spacing and single corrugation angle of 20°. It is reported that over a flow range of 1200 < Re< 4000, a substantial increase in both fully developed Nusselt number and friction factor was obtained due to changing of channel spacing from 5-mm to 10-mm, however the flow area goodness factor slightly decreased. Ali and Ramadhyan [5] demonstrated the heat transfer and the flow visualizations in triangular corrugated channel with water as a working fluid for two different channel spacing. They concluded that the performance of corrugated channel with larger spacing is much better than that of smaller spacing. Comparing their results with that
of parallel plate channel, they pointed out that the corrugated channel had much higher Nusselt number. Naphon [6] presented the heat transfer characteristics and pressure drop in the channel with V corrugated upper and lower plates heated under constant heat flux conditions. Corrugated plates with three different corrugation angles are tested. It is reported that the Nusselt number increases with increasing the wavy angle. Lin, et. al. [7], derived dimensionless correlations to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger using the Buckingham theorem. Hamza and Hanaoka [8] carried out an experimental study of the effects of flow Reynolds number, corrugation angles, inlet air bulk temperatures and incident radiation fluxes on laminar flow forced-convection heat transfer for air flowing in a channel having a V-corrugated upper plate heated by radiation heat flux. Metwally and Manglik [9] numerically investigated laminar periodically developed forced convection in sinusoidal corrugated-plate channels, with uniform wall temperature. Numerical solutions are obtained using the control volume finite-difference method for a wide range of viscous liquids (Pr = 5, 35, and 150). Islamoglu [10] determined experimentally the convection heat transfer characteristics for the rounding of the protruding peaks of a corrugated converging–diverging channel. Results were carried out for a Prandtl number of 0.7 in the Reynolds number ranged from 2000 to 5000 for a corrugation angle of 30°. Asako and Faghri [11] developed a finite volume methodology to predict fully developed heat transfer coefficient, friction factor, and streamlines for laminar flow in a corrugated duct for Prandtl number having values of 0.7, 4 and 8, corrugation angle(15°,30° and 45°), and a number of dimensionless height to pitch ratios. Asako, et. al. [12] determined numerically the heat transfer and pressure drop responses of a corrugated duct with rounded corners. Computations were carried out for a Prandtl number of 0.7, in the Reynolds number ranged from 100 to 1000, for three assigned corrugation angles, and for four values of aspect ratios. Rounding of the corners resulted in a decrease of friction factor and Nusselt number.

Many studies on the general performance of the corrugated plates can be found in the literature but there is still a lack of knowledge on the local structure of the flow and on its links to the heat transfer efficiency. The present work deals with the study of the local temperature distribution and its link with the local flow structure for the trapezoidal corrugated-plate channels. The global heat transfer performances will also be considered.
2. PROBLEM FORMULATION AND NUMERICAL SOLUTION

Typical in-phase trapezoidal wavy ducts are depicted in Fig.1. and their geometric attributes are described by the waviness amplitude $A$, pitch $P$, inter-plate spacing $S$ and height of the cutting edge for trapezoidal shape $L$. Their dimensionless representation is given by the corrugation aspect ratio $\gamma = (2A / P)$, spacing ratio $\varepsilon = (S/2A)$ and cutting edge ratio for trapezoidal shape $\delta = (L/A)$.

Because the plate spacing $S$ is much smaller than the plate width, the flow passages can essentially be modeled as two-dimensional wavy-parallel-plate channels. For the trapezoidal in-phase wavy ducts, constant property, periodically fully developed, laminar incompressible air flow ($Pr = 0.7$), steady state, constant heat flux boundary conditions, with negligible axial conduction ($Pe >> 1$) and viscous dissipation ($Br << 1$). Thus for the two-dimensional convection, the governing equations for mass, momentum, and energy conservations can be expressed as follows:

Mass conservation equation:
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)
\]

Momentum conservation equation:
\[
u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + v \frac{\partial^2 u}{\partial x^2} + v \frac{\partial^2 u}{\partial y^2} \quad (2.a)
\]
\[
u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{\partial p}{\partial y} + v \frac{\partial^2 v}{\partial x^2} + v \frac{\partial^2 v}{\partial y^2} \quad (2.b)
\]

Energy conservation equation:
\[
u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (3)
\]

For the present problem, the no slip, uniform heat flux boundary conditions can be stated as:

The governing equations (1-3) were solved numerically using FLUENT-V6.2 code. This code employs a control-volume based discretization technique along with a pressure-correction algorithm.

The hydraulic diameter of the duct is defined as:
\[
D_h = \frac{2BS}{B + S} \quad (4)
\]

For two dimensional problem ($B >> S$) $D_h = 2S$ \quad (5)
Fig. 1 Geometrical parameters of the physical model (a) triangular shape, (b) trapezoidal shape.
The flow Reynolds number is defined as:

\[ \text{Re} = \frac{\rho v D_h}{\mu} = \frac{2m^*}{\mu} \]  

(6)

Considering that the variation of the mean temperature of the fluid between the inlet and the outlet of the cycle is linear. The local mean temperature of the fluid is given by:

\[ T_b(x) = \frac{T_{b,o} - T_{b,i}}{P}.x + T_{b,i} \]  

(7)

The local heat transfer coefficient is given by the following relation:

\[ h(x) = \frac{q}{(T_w(x) - T_b(x))} \]  

(8)

The local Nusselt number can finally be estimated from the local temperatures as follows.

\[ Nu(x) = \frac{h(x)D_h}{k} \]  

(9)

The average Nusselt number was obtained by numerical integration of local Nusselt number over one pitch (periodic module) using the trapezoidal rule.

The fanning friction factor is given by the following equation:

\[ f = \frac{-dp}{dx} \frac{D_h}{0.5 \rho u_m^2} \]  

(10)

3. RESULTS AND DISCUSSION

3.1 Fluid Flow Characteristics

The effects of the variations in cutting edge ratio (\( \delta \)) and Reynolds number (Re) on the dynamic behavior of the flow field are shown in Fig.2. Streamlines for flows with Re = 100, 300, 500, 700 and 1000 in five different wall cutting edge ratios \( \delta = 0.12, 0.24, 0.36, 0.48 \) and 0.6 are graphed along with the triangular case (\( \delta = 0 \)). Lateral vortices or re-circulation cells developed in the corrugated-wall troughs as the axial flow get separated downstream of the wall-corrugation peaks and re-attaches upstream of the subsequent wall peak, and their strength and coverage was a function of Reynolds number and cutting edge ratio. As the cutting edge ratio increase, the large lateral vortex of the triangular case (\( \delta = 0 \)) was gradually damped and lower recirculation area was obtained. Also, the flow reattachment point moves further
Fig. 2 Streamlines distributions for air flows inside trapezoidalcorrugated-plate channels at different cutting edge ratios and Reynolds numbers but constant corrugation aspect ratio ($\gamma = 0.25$) and relative spacing ratio ($\varepsilon = 1$).
Fig. 2 Streamlines distributions for air flows inside trapezoidal corrugated-plate channels at different cutting edge ratios and Reynolds numbers but constant corrugation aspect ratio ($\gamma = 0.25$) and relative spacing ratio ($\varepsilon = 1$).
upstream on the oblique face of the wavy wall. At low Reynolds number or with large cutting edge ratio, the surface geometry has no effect on the flow structure and the fluid moves through the channel by simply “adopting” its shape, and the viscous forces dominate to produce undisturbed streamline flows and no swirl structure developed.

The effects of the variations in cutting edge ratio (δ) and Reynolds number (Re) on the local shear stress distributions are shown in Fig.3 along with the shear stress profile for triangular passage with the same aspect ratio (γ = 0.25) and the flat duct profile.

- As the cutting edge ratio increase, the variation of the local shear stress was decreased and the shear stress profile was shifted from the triangular profile toward the flat duct profile (constant value).

- The high peak of the shear stress for the triangular duct was reduced for trapezoidal duct as a result of cutting the sharp edged corner of the triangular duct which reduce the sudden impact of the air flow to the wall, and hence reduce the high velocity gradient at this point. As the cutting edge ratio increase this high peak decrease.

At Re = 300, γ = 0.25 the peak point values of the shear stresses equal to 0.046, 0.033, 0.03, 0.0275, 0.026, 0.025 Pa for triangular, trapezoidal with cutting edge ratios of 0.12, 0.24, 0.36, 0.48 and 0.6 respectively. While the corresponding values of the shear stresses at Re = 500 equal to 0.11, 0.089, 0.078, 0.07, 0.06, 0.055 Pa.

- The position of the peak point (X/P) for the triangular case equal to 0.25, but for the trapezoidal duct, this position was shifted gradually to the left as the cutting edge ratio increase, this is due to the left shifting of the sharp edged corner with increasing the cutting edge ratio.

The positions of the peak points at γ = 0.25 for cutting edge ratios of 0.12, 0.24, 0.36, 0.48 and 0.6 equal to 0.22, 0.19, 0.16, 0.13, 0.1 respectively for all values of flow rates.

- Also another peak was appeared as a result of the other sharp corner of the trapezoidal profile, but the value of the shear stress at this edge was lower than that at the peak edge.

- As the cutting edge ratio increase, the distance between the two points of minimum shear stress values that encapsulate the lateral swirl decreased as the separation point
Fig. 3 Local wall shear stress distributions at different values of cutting edge ratio inside trapezoidal corrugated-plate channels at (a) Re = 300 and (b) Re = 500.
moves downstream and the reattachment point moves upstream. So, the recirculation zone was damped with increasing cutting edge ratios as mentioned in Fig. 2.

The effects of the variations in cutting edge ratio (δ) and Reynolds number (Re) on the velocity profile across the duct cross section at X/P = 0.25, which represents the peak-and-valley planes, are displayed in Fig. 4 along with the velocity profile for triangular passage with the same aspect ratio (γ = 0.25) and the flat duct profile. As the cutting edge ratio increase, the velocity profile was gradually shifted from the triangular velocity profile to the flat duct velocity profile, and this is illustrated through the following points:

1- The velocity magnitude was decreased as the cutting edge ratio increase, and hence the wall velocity gradient decrease.

Max velocity magnitude for cutting edge ratios of 0.12, 0.24, 0.36, 0.48 and 0.6 equal to 0.54, 0.52, 0.5, 0.47, 0.45 m/s respectively which lies between the triangle value (0.56 m/s), and the flat duct value (0.42 m/s) at Re=300, while at Re=500 the corresponding values equal to 1.03, 0.95, 0.87, 0.82 and 0.77 which lies between the triangle value (1.05) and the flat duct value (0.7).

2- The positions of the peak points (X/P) were shifted toward the duct center line as the cutting edge ratio increase.

The positions of the peak points for cutting edge ratios of 0.12, 0.24, 0.36, 0.48 and 0.6 are equal to 0.25, 0.29, 0.33, 0.36 and 0.41, which lies between 0.5 and 0.23 for triangle and flat duct at Re = 300, and equal to 0.23, 0.27, 0.31, 0.36, 0.38 which lies between 0.5, 0.21 for triangle and flat duct at Re=500.

3- The recirculation velocity decreases with increasing cutting edge ratio.

Max recirculation velocities for cutting edge ratios of 0.12, 0.24, 0.36, 0.48 and 0.6 are equal to 0.0096, 0.00406, 0, 0, 0, which lies between 0, 0.014 for flat, triangle duct at Re=300, while the corresponding values at Re=500 equal to 0.09, 0.065, 0.04, 0.02, 0, which lies between 0, 0.16 for flat, triangle duct.

4- The reattachment point position (X/P) was shifted toward the wall as the cutting ratio increase, this is due to the damping of the swirl with increasing cutting edge ratio.

The reattachment point position (X/P) for cutting edge ratios of values 0.12, 0.24, 0.36, 0.48 and 0.6 is equal to 0.86, 0.92, 0, 0, 0 which lies between 0 and 0.81 for flat and triangular duct respectively, while the corresponding values at Re=500
Fig. 4 Axial velocity profiles at X/P = 0.25 at different values of cutting edge ratio inside trapezoidal corrugated-plate channels at (a) Re = 300, and (b) Re = 500
equal to 0.72, 0.78, 0.84, 0.9, which lies between 0, 0.7 for flat and triangular duct respectively.

The variation in the isothermal fanning friction factor with Reynolds number for different cutting edge ratios (δ=0.12, 0.24, 0.36, 0.48 and 0.6) is presented in Fig.5. Compared with the performance of both the triangular passage with the same aspect ratio (γ=0.25) and the flat duct in fully developed laminar flow. As a result of damping the lateral vortex with increasing cutting edge ratio, lower velocity gradients were occurred in the acceleration zone, which damping the high peaks of the shear stress result in lower shear stress values and hence the isothermal friction factor was reduced with increasing the cutting edge ratio.

3.2 Heat Transfer Characteristics

The effect of cutting edge ratio on the local convective heat transfer in airflows between uniform heat flux plates is illustrated by the isotherms plots in Fig. 6. As the cutting edge ratio increase, the boundary layer thickness increase and the flow field exhibits regions of less thermal mixing and lower sharper wall temperature gradients occur. So, a gradually reduction occur in the convection heat transfer as the cutting edge ratio increase as a result of damping the lateral vortex with increasing cutting edge ratio.

In order to illustrate the reduction in the mixing effect produced by the cutting edge ratio, the normalized temperature profile across the duct cross section is illustrated in Fig.7 at X/P having a value of 0.25, which represents the peak-and-valley plane. The normalized temperature profiles with respect to its mean value are illustrated for δ = 0.12, 0.24, 0.36, 0.48 and 0.6 at a fixed Re and corrugation aspect ratio.

It is shown from this figure that, for flat duct, the temperature profile is symmetric around the duct center, and the point of lowest temperature lying on the duct center. As the cutting edge ratio increase, the temperature profile was gradually shifted from the triangular profile to the flat duct profile.

As the cutting edge ratio increase, the mixing effect produced by the lateral vortex decrease, and the deviation of the temperature profile around its average value increases, which means less uniform temperature profile.

For trapezoidal passage with Re=700 and γ =0.25, the lowest point having values of 0.91, 0.87, 0.84, 0.82, 0.81 for cutting edge ratio of values δ=0.12, 0.24, 36, 0.48 and
Fig. 5 Effect of Reynolds number on isothermal fanning friction factor inside trapezoidal corrugated-plate channels at different values of cutting edge ratio.
Fig. 6 Isotherms maps for air flows inside trapezoidal corrugated-plate channels at different cutting edge ratios and Reynolds numbers but constant corrugation aspect ratio ($\gamma = 0.25$) and relative spacing ratio ($\varepsilon = 1$).
\[
\delta = 0.36 \\
\delta = 0.48 \\
\delta = 0.6 \\
\]

Re = 100  \\
Re = 100  \\
Re = 100  \\

Re = 300  \\
Re = 300  \\
Re = 300  \\

Re = 500  \\
Re = 500  \\
Re = 500  \\

Re = 700  \\
Re = 700  \\
Re = 700  \\

Re = 1000  \\
Re = 1000  \\
Re = 1000  \\

Fig. 6 Isotherms maps for air flows inside trapezoidal corrugated plate channels at different cutting edge ratios and Reynolds numbers but constant corrugation aspect ratio (\( \gamma = 0.25 \)) and relative spacing ratio (\( \varepsilon = 1 \)).
Fig. 7 Normalized temperature profiles at $X/P = 0.25$ at different values of cutting edge ratio inside trapezoidal corrugated-plate channels at (a) $Re = 700$ and, (b) $Re = 1000$. 

(a) $Re = 700$
- $\gamma = 0.25$
- $X / P = 0.25$
- $\varepsilon = 1$

(b) $Re = 1000$
- $\gamma = 0.25$
- $X / P = 0.25$
- $\varepsilon = 1$
0.6 respectively. Which falls between the triangular value (0.93) and the flat duct value (0.74) while the corresponding values for Re=1000 and \( \gamma = 0.25 \) equal to 0.93, 0.9, 0.89, 0.85, 0.83 Which falls between the triangular value (0.95) and the flat duct value (0.73).

As the cutting edge ratio increase, the temperature profile was shifted toward the duct centerline. For trapezoidal passage with Re=700 and \( \gamma = 0.25 \), the positions of the lowest points(Y/S) are 0.27, 0.33, 0.34, 0.36 and 0.41 for cutting edge ratio of values \( \delta = 0.12, 0.24, 0.36, 0.48 \) and 0.6 respectively which falls between the triangular value (0.25) and the flat duct value (0.5). While the corresponding values for Re=1000 and \( \gamma = 0.25 \) equal to 0.25, 0.31, 0.34, 0.36, 0.39 Which falls between the triangular value (0.23) and the flat duct value (0.5).

The effects of the variations in cutting edge ratio (\( \delta \)) on the local Nusselt number distribution is displayed in Fig. 8 along with that for both triangular passage with the same aspect ratio (\( \gamma = 0.25 \)) and the flat duct.

- As the cutting edge ratio increase the fluctuation of the Nu number profile was decreased and the profile gets gradually shifted from the triangular profile towards the flat duct profile (constant).

- The high peak of the local Nusselt number profile for the triangular passage was gradually reduced for trapezoidal duct as the cutting edge ratio increase as a result of cutting the sharp edged corner of the triangular duct which reduce the wall velocity gradient, and hence the heat transfer rate was reduced too.

- The high peak of the Nusselt number profile was shifted to the left for trapezoidal duct as cutting edge ratio increase. This is due to the left shifting of the sharp edged corner with increasing the cutting edge ratio.

- Another peak was appeared on each Nusselt profile as a result of the other sharp corner of the trapezoidal profile, but the value of the shear stress at this edge was lower than that at the first edge.

Comparing the results of the shear stress profile (Fig. 3) and local Nusselt number profile, we noticed that as the cutting edge ratio increase, the fluctuation of both shear stress and Nu profile was decreased. And a considerable reduction of the peak point value were occurred for both profiles. So, the cutting edge ratio produce the similar effect for both shear stress and Nusselt number, which is agreed with the Reynolds analogy. Although the trapezoidal shape provide a considerable decreasing in the
shear stress and hence in the average friction factor, but this useful benefit is accomplished with a considerable reduction in the Nusselt number. The variation in the average Nusselt number with Reynolds number for different cutting edge ratios ($\delta=0.12, 0.24, 0.36, 0.48$ and $0.6$) is presented in Fig. 9. Compared with the performance of both the triangular passage with the same aspect ratio ($\gamma=0.25$) and the flat duct in fully developed laminar flow. As the cutting edge ratio increase, the lateral vortex of the triangular passage was gradually damped and lower recirculation area was obtained. So, the flow was less mixed resulted in lower sharper wall temperature gradients with thicker boundary layer thickness, So lower values of the average Nusselt number was observed as the cutting edge ratio increase.

From the previous numerical results, Nusselt number and friction factor are correlated as a function of $Re$, $\gamma$, $\varepsilon$ and $\delta$ in the triangular and trapezoidal passage

For sharp edged passage (triangular and trapezoidal):

\[ Nu = 18 Re^{0.27} \gamma^{1.28} \varepsilon^{-0.44} (1 + \delta)^{0.98} \]  
\[ f = 255 Re^{0.21} \gamma^{2.68} \varepsilon^{-1.05} (1 + \delta)^{-2.5} \]  

Where:

\[ 100 \leq Re \leq 1000 \]

$\delta = 0$ for triangular passage  \hspace{1cm}  $0.12 \leq \delta \leq 0.6$ for trapezoidal passage

The comparison between the present results with the results developed from Eqs.11 and 12 for Nusselt number and friction factor are shown in Figs. 10 and 11 respectively. It can be clearly seen from these figures that the values developed from the previous equations are consistent with the numerical and experimental data and lie within $\pm 10\%$.

The developed model is validated by comparing the obtained numerical results from the present model with the available published results. The validation of the model has been applied for both flat parallel plate duct and triangular duct with an angle of $30^\circ$ as shown in Figs 12 and 13. Fig.12. shows the comparison between the predicted friction factor for flat duct and the well known formula for flat duct ($f = 96 / Re$). Also, for triangular duct with an angle of $30^\circ$, the predicted friction factor are compared with that obtained by Asako [11]. Fig.23. shows the deviation of the predicted Nusselt number for flat duct from the constant value of 7.54. (The value for
Fig. 8 Local Nusselt number distributions at different values of cutting edge ratio inside trapezoidal corrugated-plate channels at Re = 300 and γ = 0.25.

Fig. 9 Effect of Reynolds number on average Nusselt number at different values of cutting edge ratio inside trapezoidal corrugated-plate channels.
Fig. 10 Comparison of the present results with the results developed from Eq. 11 for Nusselt number for triangular and trapezoidal corrugated-plate channels.

Fig. 11 Comparison of the present results with the results developed from Eq. 12 for friction factor for triangular and trapezoidal corrugated-plate channels.
Fig. 12. Comparison between the predicted friction factor with those of other authors

Fig. 13. Comparison between the predicted Nusselt number with those of other authors
Fig. 14 Comparison between the stream function distribution for the triangular corrugated-plate channels at $\varepsilon = 1.5$ between (a) the results of Naphon [13] with $\gamma = 0.42$ and $Re = 918$ and (b) The present study with $\gamma = 0.35$ and $Re = 1000$. 
the Nusselt number for the fully developed flat duct is 7.54 and is independent of both Reynolds and Prandtl number). Also, for triangular duct with an angle of 30°, the predicted Nusselt number are compared with that of Asako [11]. As shown from Figs. 12 and 13, a good agreement is achieved between the results of the present numerical model and the available published results.

Fig. 14 shows a comparison between the stream function distribution of the triangular corrugated-plate channels at ε = 1.5 between (a) the results of Naphon [13] with γ = 0.42 and Re = 918 and (b) The present study with γ = 0.35 and Re = 1000 and a very good agreement was observed for both the core flow and the recirculation flow.

4. CONCLUSIONS

For trapezoidal corrugated-plate channels, increasing the cutting edge ratio leads to:

1. The large lateral vortex of the triangular channels is gradually damped accompanying with lower recirculation area.
2. The variation of the local shear stress is decreased and the shear stress profile is shifted from the triangular profile towards the flat duct profile.
3. The velocity profile is gradually shifted from the triangular velocity profile to the flat duct velocity profile.
4. Lower velocity gradients occur in the acceleration zone, resulting in damping the high peaks of the shear stress, lowering shear stress values and hence reducing the isothermal friction factor.
5. The thermal boundary layer thickness increases and the flow field exhibits regions of less thermal mixing and lower sharper wall temperature gradients occur as a result of damping the lateral vortex.
6. The fluctuation of the Nu number profile decreases and the high peak of the local Nusselt number profile for the triangular passage is gradually reduced and the profile is gradually shifted from the triangular profile towards the flat duct profile (constant value).
7. Lower values of the average Nusselt number is observed.

REFERENCES:


[12] Yutaka Asako, Hiroshi Nakamura and Mohammad Faghri, "Heat transfer and pressure drop characteristics in a corrugated duct with rounded corners"


<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>amplitude of sinusoidal corrugation</td>
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<tr>
<td>Br</td>
<td>Brinkman number</td>
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<tr>
<td>B</td>
<td>Width of the corrugated duct</td>
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<td>D&lt;sub&gt;h&lt;/sub&gt;</td>
<td>duct hydraulic diameter, m</td>
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<tr>
<td>F</td>
<td>fanning friction factor</td>
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<td>H</td>
<td>heat transfer coefficient, W/m&lt;sup&gt;2&lt;/sup&gt; K</td>
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<tr>
<td>K</td>
<td>thermal conductivity, W/m·K</td>
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<td>L</td>
<td>height of the cutting edge for trapezoidal shape, m</td>
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<tr>
<td>m&lt;sup&gt;*&lt;/sup&gt;</td>
<td>mass flow rate, kg/s</td>
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<tr>
<td>Nu</td>
<td>Nusselt number</td>
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<td>P</td>
<td>pitch of corrugation, m</td>
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<td>ψ</td>
<td>stream function</td>
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<tr>
<td>μ</td>
<td>dynamic viscosity, kg/m·s</td>
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<td>B</td>
<td>bulk or mixed-mean value</td>
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<tr>
<td>i</td>
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<tr>
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<tr>
<td>n</td>
<td>outward normal vector to the corrugated surface</td>
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<tr>
<td>o</td>
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<td>w</td>
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