Journal University of Kerbala , Vol. 16 No.2 Scientific . 2018

Numerical study on forced convection heat transfer and pressure drop for different configurations of corrugated channels

دراسه عدديه لانتقال الحراره بالحمل القسري وهبوط الظغط لقنوات مستطيلة المقطع بأشكال مختلفه من السطوح المتعرجه

Abdalrazzaq K. Abbas
Lecturer - Dep. of Mech. Engineering
College of Engineering, Kerbala University, Kerbala, Iraq
Field of Work: Heat Transfer and Fluid Flow

ABSTRACT

A numerical study to investigate the effects of different corrugated channel configurations on the flow and thermal fields are performed with Reynolds number from 6000 to 20000 and heat flux of 6000 W/m^2. The simulation adopts FLUENT- based control volume to solve the turbulent flow field and the associated thermal behaviour of the different configurations of corrugation. Seven types, in three groups, of trapezoidal corrugated channels; i.e., one face outward (1FOC), one face inward (1FIC), inward on one face and outward on opposite face (2FIOC). Moreover, two faces inward (2FIC), two faces outward (2FOC), outward and inward are aligned on both surfaces (IAO), and outward and inward are staggered on both surfaces (IOSIO), were examined and compared with smooth one. Simulated results displayed that the 2FIC provides maximum friction factor and heat transfer rate than the others. The performance evaluation criterion (PEC) values decrease with increasing Reynolds number, and the highest was predicted for 1FOC and 2FOC in turbulent flow. PEC comparative evaluation of the influence of flow regime on corrugated channel configurations showed that the (PEC) in laminar regime ($600 \leq Re \leq 2000$) increases and the 2FOC seem to have better values. However, laminar regime offered higher value than the turbulent flow. Computed results are validated as they have an acceptable agreement with the findings of previous works as well as with the existing well-established correlations.

Keywords: Corrugated channel configurations, Turbulent flow, Laminar flow, Outward and inward corrugations, Performance evaluation criterion.

الخلاصه

اجريت دراسه عدديه لفحص تأثير تفيير هيئة وتصميم القنوات متعرجة السطوح ذات المقطع المستطيل على معدل انتقال الحراره والجريان ضمن مدى لعدد رينولدز من (6000) الى (20000) وبتسليط فيض حراري ثابت وبمقدار ($6000\ W/m^2$) .تمت الدراسه باستخدام برنامج فلونت (Fluent-CFD-6.3) والذي يعتمد على تقنية الحجم المحدد لحل المعادلات الحاكمه الخاصه بالجريان الاضطرابي والمتعلقه بانتقال الحراره لمختلف انواع القنوات المتعرجه. اجريت الدراسه لسبعة انواع بثلاث مجاميع لقنوات ذات تعرج على شكل شبه منحرف وهي قناه ذات تعرج لسطح واحد باتجاه الداخل(1FIC)،سطح واحد متعرج والى الخارج (1FOC)،سطح باتجاه الداخل والاخر باتجاه الخارج (2FIOC)،التعرج لسطحين باتجاه الداخل (2FIC)،التعرج لسطحين باتجاه الخارج(2FOC)،تعرج لسطحين الى الداخل متعاقب مع تعرج للسطحين الى الخارج(IAO)،تعرج الى الخارج يقابل تعرج الى الداخل ويتكرر على طول القناة (IOSIO).وقد تمت مقارنة النتائج مع نتائج لقناة اعتياديه (Smooth Channel).النتائج اظهرت بان أعلى معامل احتكاك وانتقال حراره كان للقناة (1FOC) واقلها للقناة(IAO).لغرض تقييم ثأثير نوع الجريان على أداء القنوات بأعتماد تقييم (PEC) ،فقد أظهرت النتائج ان معامل الاداء يقل بزياده سرعة الماء للجريان الاظطرابي وقد حقق النوع (1FOC) ويليه (2FOC)اعلى قيم بينما كان معامل الاداء يزداد بزيادة سرعة المائع لرقم رينولدز من (600) الى(2000) للجريان الطباقي لحدود معينه من السرعه.تم التحقق من دقة النتائج بمقارنتها مع نتائج العلاقات التجريبيه لهذا الخصوص وكان التطابق مقبول جدا.

1. INTRODUCTION

Heat exchangers are used in various applications throughout the different industrial fields. Therefore, they are bulleted to reach high heat transfer and small volume at lower cost.Employing coarsening walls is an appropriate technique to provide higher compactness and increase the performance. Corrugated channels represent one of the coarsening walls methods in both types: inward and outward corrugations were considered in present paper [1, 2]. Many studies have examined the characteristics of heat transfer, and fluids flow in a one corrugation direction (inward or outward) in different geometrical parameters, and some of them tested the using of corrugated channels surface with upper and lower with a specific configuration. Studies on characteristics of single or double corrugation direction with one or two side coursed channels are summarized chronologically as follows:

The experimental work of two sides corrugated channel with an in-line arrangement laminar flow and staggered arrangement turbulent flow are performed by Naphon [3, 4]. The results displayed that the corrugated wall has a significant influence on the increment of pressure drop and heat transfer due to the occurrence of recirculation flow regions. Naphon and Kornkumjayrit [5] studied the numerical work on the flow developments and heat transfer along the channel with the single surface corrugated plate under constant heat flux condition. The influences of rib and grooved in a single wall corrugated rectangular channel on a forced convection heat transfer were described experimentally by Eiamsa-ard and Promvonge [6]. Results showed that the duct with rectangular rib and triangular groove type provides maximum friction factor and heat transfer rate. Mohammed et al. [7] investigated the results of variant corrugation parameters numerically through inward roughened tubes. The results revealed that the Nusselt number increased with the increase of height, the width of roughness and Reynolds number also with the reduction of corrugations pitch. Mohammed et al. [8] studied the influences of corrugation parameters of inward wavy roughened channel height, wavy tilt angle and channel height. The results indicated that the using of the corrugated channel is an efficient method to increase the thermal performance.

Mirzaei et al. [9] performed a numerical investigation of turbulent convective flow through a half-roughed channel for different ranges of the wave amplitudes. The results illustrated that the recirculating region of flow depends on the wave amplitude. As the wave amplitude increases, the Nusselt number of the turbulent region becomes larger then it keeps on approximately constant. Dizaji et al. [10] experimentally investigated flow and thermal fields in heat exchanger corrugating shell and tube. Results showed that the type of corrugation arrangement has an essential effect on thermal and frictional characteristics. The geometrical parameters of fully developed turbulent flow in outward and inward trapezoidal corrugated channels using nanofluids were analyzed by Abed et al. [11]. It was mentioned that the maximum enhancement improves with increasing of the corrugation height. Mohammed et al. [12] studied numerically three types of internally rib and groove shapes (semi-circular, rectangular, and trapezoidal) of turbulent nanofluid flow corrugated channels. The results revealed that the semi-circular has the highest thermal enhancement influence. Chandra et al. [13] used periodic converging and diverging circular cross-section microchannel to study numerically and analyze the impacts of the corrugation size on the thermal performance for laminar flow. Results displayed that significant enhancement in heat transfer over traditional smooth microchannel with a penalty of pressure drop due to flow separation and recirculation. Lee et al. [14] analyzed seven cross-corrugated shape heat exchanger passages with various surface profiles. They observed that the high heat transfer region growths with the increasing of the corrugations radius and the pressure loss also increases.

Wang et al. [15] investigated the effect of geometrical parameters to achieve the optimal values of the height and pitch for undulated plate heat transfer surfaces. Zhai et al. [16] performed a laminar hydrodynamics water flow experimentally in micro heat sinks with ribs and cavities. They found that the flow turbulence and heat transfer area increase due to the presence of ribs and cavities. A new model of one wall corrugated channel heat exchanger was performed and numerically tested by Schmidmayer et al. [17]. The results revealed that the mass flow rate and heat transfer coefficient enhance with an increase in amplitude with applying of the positive pressure difference.

As described in the literature review, these works have studied channels with a detailed corrugation design and flow regime concerning the pressure drop and heat transfer with corrugated surfaces. The primary aim of this study is to investigate the effects of the various kinds of configuration design of corrugated surfaces on pressure drop and heat transfer characteristics of a corrugated channel with numerically analyzed. Another objective of this paper is to have a better balance among different designed channels by using hydraulic diameter based on the enclosed volume to the surface area instead of the cross-section area to the wetted perimeter. Which is more accurate for different types of the surface, when the cross-section area varies with flow direction. Numerical calculations were performed through the test section region in which the flow of water becomes fully developed in both thermally and hydrodynamically boundary layers. The present results, friction factor and the average Nusselt number are validated by comparing with that of the single-phase fluid empirical correlations for water flow in a smooth channel.

2. MATHEMATICAL MODELING

2.1. Physical Model and Assumptions

The rectangular corrugated channel of seven configurations design of successive trapezoidal corrugation along the upper and lower walls is depicted schematically in two dimensions by Figs. 1 and 2. The computational domain is a flat or corrugated channel with the height (H) of 10 mm, and the length is 100 mm. To ensure a fully-developed flow, the length of the upstream heated smooth channel is 200 mm, while the length of downstream adiabatic channels is 100 mm to avoid adverse pressure and reversed flow through the simulation domain [11].Therefore, the simulation adopts 400 mm of the total length of the channel for each configuration type. The upper and lower walls of upstream and domain are subjected to a constant wall heat flux at a rate of 6000 W/m^2 [8]. The trapezoidal roughness shape is defined by the rib height (e) of 6mm, pitch length (p) of 10 mm, and width (w) of 2 mm were retained constant as presented in Fig. 1.The simulation of this study depends on the following assumptions:

1- The flow through the test section is assumed to be fully developed, Newtonian, steady, and incompressible parallel to the x-axis.
2- The thermophysical properties of water are assumed constant.
3- Because the inward and the outward corrugation between them are periodically distributed, and the channel height is much smaller than the width of the channel. Thus, the geometry and the steady flow in this respect are assumed to be two-dimensional [8].
4- The external lower and upper surfaces of each channel are subjected to a uniform heat flux.

2.2 Governing Equations

The turbulent flow is modelled by partial governing conservation equations which are continuity, momentum and energy equations as indicated by equations 1, 2 and 3, respectively as [7]:

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho\, u_i)=0 \tag{1}$$

Momentum equation:

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

In which ρ, μ, u', u_i, u_j and $\overline{u_i' u_j'}$ represent the fluid density, dynamic viscosity, fluctuated velocity, axial velocity, radial velocity and turbulent shear stress, respectively.

Energy equation:

$$\frac{\partial}{\partial x_i}[u_i(\rho E+P)]=\frac{\partial}{\partial x_j}\left[\left(k+\frac{C_p\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_j}+u_i(\tau_{ij})_{eff}\right] \tag{3}$$

Where the total energy (E) and tensor of deviatoric stress should be calculated as:

$$E=C_pT-P/\rho+u^2/2 \tag{4}$$

$$(\tau_{ij})_{eff}=\mu_{eff}\left(\frac{\partial u_j}{\partial x_i}+\frac{\partial u_i}{\partial x_j}\right)-\frac{2}{3}\mu_{eff}\frac{\partial u_i}{\partial x_j}\delta_{ij} \tag{5}$$

The *k-ε* turbulent model introduced by Launder and Spalding [18] is adopted in the present simulation to express the turbulent stresses and heat flux quantities of the related physical phenomenon [19].

2.3. Boundary Conditions

The set of boundary conditions adopted are:

At inlet: The uniform velocity and temperature boundary condition are used:

$$u_x=u_{in}=\text{constant},\ u_y=0,\ T=T_{in}=300\ K \tag{6}$$

And turbulent inlet intensity is set as [19]:

$$I=u'/u_m\ 100\% \tag{7}$$

At outlet: zero normal gradients for all flow variables (B) except pressure whereas the pressure boundary condition is used [20].

$$\frac{\partial B}{\partial x}=0,\ B=u_x,u_y,T \tag{8}$$

At the upper and lower walls: no-slip condition and constant heat flux are supplied at the upper and lower walls for the inlet fluid and test sections.

$$u_x=0,\ u_y=0,\ q=q_{wa} \tag{9}$$

2.3 Calculation of Thermal and Hydrodynamic Parameters

The dimensionless parameters considered in the current investigation are the Nusselt numbers, the Reynolds number, friction factor, and performance evaluation criteria [7, 14]. The variation of the local Nusselt number along the lower and upper walls could be defined as:

$$Nu(x)=\frac{h(x)D_h}{kt} \tag{10}$$

The local convection heat transfer coefficient h(x) is representing as:

$$h(x)=\frac{q}{T_{wa}(x)-T_b(x)} \tag{11}$$

The local and average Nusselt numbers are calculated as:

$$Nu(x)=\frac{q\ D_h}{kt\ (T_{wa}(x)-T_b(x))} \tag{12}$$

The Reynolds number can be defined as:

$$\mathrm{Re} = \frac{\rho\, u_m\, D_h}{\mu} \tag{13}$$

For the corrugated surface, the cross flow area varies with passage length, so for this situation an alternative definition of the hydraulic diameter D_h is determined as [15]:

$$D_h = 4V/A \tag{14}$$

Where V is the enclosed volume, and A is the wetted surface area. The Darcy friction factor over the entire walls is determined as follows [21]:

$$f = 2\,\Delta P\, D_h / L\, \rho\, u_m^2 \tag{15}$$

Where ΔP is the pressure difference between inlet and outlet which is calculated by:

$$\Delta P = P_{av,\,in} - P_{av,\,out} \tag{16}$$

The enhancement of heat transfer using corrugated channels goes together with a penalty of pressure drop which increases the pumping power. The performance evaluation criteria index (PEC) is employed to study the fluid-dynamic and thermal performance of channels for seven various configurations. The PEC is defined using the results of friction factors and Nusselt numbers as follows [7]:

$$\mathrm{PEC} = \frac{Nu/Nu_s}{(f/f_s)^{1/3}} \tag{17}$$

2.4. Computational Procedure

The thermal characteristics and flow field for all geometries are simulated by implementing finite volume based FLUENT software to solve the equations and the boundary conditions of continuity, momentum and energy. GMBIT software was adopted to configure the geometries and meshing. The model of standard *k–ε* turbulence, RNG model, the SIMPLEC algorithm, and first-order differencing upwind scheme were employed to resolve the pressure-velocity coupling system [8, 21]. The convergence criterion solutions for all dependent variables is realized when the residual results are lower than 10^{-5}, where the steady computation is completed if $\sum_{ij}\left|M_{ij}^{n} - M_{ij}^{n-1}\right| / \left|M_{ij}^{n}\right| \le 10^{-5}$. Here M stands for either velocity or temperature components, and (n) represents the iteration step. The non-uniform structured grid with an appropriate number of control volumes is generated with suitable expansion ratio to obtain fine meshing near the wall.

2.5. Grid Test

The 2D channel with trapezoidal corrugation shape (2FIC) was used to evaluate the numerical procedure accuracy. Eight different arrangements of mesh were tested to assess the number of elements required in this study by adopting the variation of the Nusselt number with the grid number as shown in Fig. 3.By comparing the results regarding the average Nusselt number rate, the sixth grid layout with approximately 88,000 nodes was adopted to get an adequate compromise between the result accuracy and the computational time, an additional increase in cells number shows less than 3% relative error in the average Nusselt number rate.

2.6. Code Validation

The code validation was performed based on the boundary conditions and channel geometry in the previous results of Mohammed et al. [8]. They carried out Numerical study through a 2D corrugated channel with outward, and inward roughness arrangement over heat flux and Reynolds number in the range of 0.4 – 6 KW/m^2 and 8000 –20,000, respectively. The average Nusselt number results of the present simulation are relatively close to that of Mohammed et al. [8] with maximum deviation for Nu of 6% as shown in Fig. 4.

3. RESULTS AND DISCUSSION

Simulations of forced convection fluid flow and heat transfer through one smooth and seven types of the corrugated channel using water were carried out to study the effects on the thermal and flow fields as well as on the performance evaluation criteria. Flow velocity, friction factor and the Nusselt number values are calculated based on the hydraulic diameter and the inlet cold water temperature was kept constant at 27°C. At the upstream outlet section, the temperature and flow fields are fully developed where the entrance length >10 D_h [22]. Therefore, we can assume fully developed turbulent flow through the middle part of total geometry.

3.1. Validation of smooth channel

In the following validation, the average Nusselt number and friction factor found from the present work for the water flow through the smooth channel (Fig.2 case A) are calculated and compared with the results of empirical correlations of Petukhov, Eq. (18), and Filonenko, Eq. (19) [18].

$$\mathrm{Nu} = \frac{\mathrm{Pr}\,\mathrm{Re}\,(f/8)}{1.07+12.7\,(f/8)^{1/2}\,(\mathrm{Pr}^{2/3}-1)} \qquad (18)$$

$$f = (1.84\log_{10}\mathrm{Re} - 1.64)^{-2} \qquad (19)$$

The comparison of the results calculated using these two equations and their deviations with the current numerical values are presented in Figs. 5(a) and (b). The comparison displays a good agreement between the predicted present results and the previous correlations for fully developed turbulent and water flow. Therefore, this solution procedure with its boundary conditions can be employed to study the current work through corrugated channels.

3.2. Heat Transfer Investigations (Nusselt number)

Nusselt number results with Reynolds number for the smooth and corrugated channels are presented in Fig. 6. As illustrated in the figure, the increased flow rate has a more significant effect on the Nusselt number of the corrugated channels in compare with the smooth channel. As Reynolds number increases, the mixing of core stream and recirculating flows coming from the regions between corrugations bring additional heat from the heating wall and can efficiently increase the total heat transfer rate. Also, adapting equal channel length of corrugated and smooth configurations tends to increase the surface area of the corrugated in comparison with the smooth channel. This is one of the corrugated channels advantages when they are used in the same size of heat exchangers. The highest values of the Nusselt number were obtained in the channel with two faces corrugation (2FIC) followed by IAO and IOSIO types at the same Reynolds number. The two types of two face corrugation channels (2FIOC, 2FOC) and channels with one face (1FIC, 1FOC) take the next place, while the lowest is the smooth channel. The slope of this enhances for channel made of two faces outward corrugated walls is less than the others. Therefore, the using of various types of corrugated channel configurations can create different amounts of thermal performance and frictional losses due to generate a particular motion of flow in the longitudinal and vertical directions.

The effect of different configuration type on enhancement in heat transfer ($\mathrm{Nu}/\mathrm{Nu}_s$) is presented in Fig.7. It is clear that the ratio decreases as the Reynolds number increase. However, the slope of this drop depends on configuration type. The reduction slope for the channel (2FOC) is more than the other corrugated channels. The ($\mathrm{Nu}/\mathrm{Nu}_s$) of the 2FIC kind is most significant, and that of 1FOC is apparently the smallest. Aligned type IAO has a slight advantage in heat transfer enhancement ($\mathrm{Nu}/\mathrm{Nu}_s$) compared to staggered case IOSIO, and both of them have much higher enhancement than configuration 2FIOC. From the above results, it can be suggested that the corrugated channels are not reasonable to save energy at higher values of Reynolds numbers [6, 24].

3.3. Friction Factors Characteristics

The pressure drop of different corrugated channel configurations was numerically measured to determine friction factor (f) over the tested range of Reynolds numbers. The smooth and corrugated channels friction factors are presented in Fig. 8. The results show that the friction factor decreases with increasing of Reynolds number. The sharp trapezoidal edge presences in flow area cause substantial flow recirculation and separation which creates an extra pressure drop. Further, in the diverging section, the losses slowly increase, and fluctuate wildly due to the stagnation region grow at the exit of the diverging zone. As seen from Fig. 8, the use of IAO, IOSIO and 2FIC corrugated channels increases the friction factor about 43-60 times while with one face, 2FOC and 2FIOC channel configurations, the friction factor increases about 1.5-12 times only in comparison with the smooth channel. The channel with IAO configuration has the most significant friction factor, while the channel with 1FOC design in the same condition of Reynolds number has the lowest friction factor.

Comparison of the channel configurations regarding the ratio of friction factor in corrugated channels to that in smooth one (f/f_s) is presented in Fig. 9. In general, enhancement of heat transfer leads to increase in pressure drop and as result high (f/f_s) value, therefore with an increase of the Reynolds number, f/f_s values tend to increase for the seven channel configurations and lies between 2 and 90 values. From this figure, the increase in the ratio of friction factors of case IAO is most substantial and that of case 1FOC is lowest. This ratio of cases 2FIC and IOSIO are a comparatively large while of all one face configurations, 2FOCand 2FIOC are low.

3.4. Performance Evaluation Criteria.

The overall performance comparison regarding performance evaluation criteria (PEC) is presented as a function of Reynolds number in Fig. 10. The values of PEC for all configurations tend to decrease as Reynolds number increase. It means that, as the velocity increases the friction factor grows much faster than the Nusselt number and as result the value of (f/f_s) becomes higher than the value of (Nu/Nu_s). In general, the PEC for IAO, IOSIO, 2FIC, 2FIOC and 1FIC geometry configurations are relatively low while for 1FOC followed by 2FOC geometries is relatively high. The outer corrugation profile in both one and two sides leads to a reduction of frontal area requirement. This suggests that enhancement in heat transfer is more efficient at the low range of Reynolds number while at the high Reynolds number the resistance to fluid flow increases and gradually becomes dominant.

3.5. Effect of the Reynolds number

A sample of the results showing the velocity contours is illustrated in Figs. 11 and 12 along corrugated flow channels (1FOC and 2FOC) at Re of 2000 and 20000, respectively. The results of these figures show the hydrodynamic flow in the tested domains. The separation and recirculation flow show an increase with the increase of fluid velocity due to the presence of corrugation when the mainstream becomes perpendicular to the core stream towards the walls where the velocity becomes slow and reaches zero value. At Reynolds number of 20000, flow vortices increase near the corrugated wall, then friction factor shows a sudden increase in the flow field. Therefore the mixing between the near-surface region and main flow enhances, so the temperature decreases at channel walls, and as a result, the rate of heat transfer between the walls and fluid is improved.

The PEC results for Reynolds number in the range of 600 to 2000 in case of laminar flow and of 6000 to 20000 for turbulent type are presented in Fig.13.The PEC profiles show that the 2FOC configuration at the lower range of Reynolds number is the most efficient one with values greater than unity and vary between 1.48 and 1.16, and the highest value is provided at Reynolds number of 1600. For 1FOC and 2FOC of laminar flow, the PEC values increase rapidly when the Reynolds number is less than 1600 and then decrease for all Reynolds number values. The values of PEC were reduced for the range of turbulent flow and reach the minimum value at highest Reynolds number. The reasons behind that, the heat transfer enhancement at the low Reynolds number is more dominant, but as Reynolds number increases more than 1600 for all cases, the flow resistance increases gradually and becomes more dominant. This is because the corrugations of the channel have a disturbance effect on the core flow. So the improvement of heat transfer or resistance becomes dominant, depending on the Reynolds number value. Therefore, the flow regime has a vital influence on improving the PEC of the corrugated channel.

4. CONCLUSION

The influences of different corrugated channel configurations on the Nusselt number, friction factor, and performance evaluation criterion were presented numerically over Reynolds number ranges with water was used as working fluid. In the present study, the primary objective is to compare various configurations of outward and inward corrugated channels with detailed investigation through a comprehensive performance analysis. The findings can be summarized as follows:

- It is measured that the addition of corrugation to smooth channel is advantageous and can significantly enhance the flow and thermal fields with a flow resistance penalty.
- The using of outward and inward trapezoidal corrugated channel is a strong function of configuration type and Reynolds number: the higher the Reynolds number, the higher the Nusselt number and friction factor in the ranges of 25-130% and 150 -5000% respectively.
- The performance evaluation criterion decreased with the increasing of Reynolds number in the range of 6000 to 20000 for all configuration types, both 1FOC and 2FOC achieve the highest overall enhancement, and it is increased sharply in case of laminar flow, and it has the maximum value at optimal Reynolds number of 1600 with 2FOC type.

The finding of the present study can be a helpful guiding principle for designers and in the manufacturing of the compact plate heat exchangers units.

5.REFERENCES

[1] N. Afgan and E. U. Schlunder, Heat Exchanger: Design and Theory Sourcebook. Washington D.C.: McGraw- Hill/Scripta, 1974.

[2] R.L. Webb, N.H. Kim, Principles of Enhanced Heat Transfer, Second Edition, Taylor & Francis, Boca Raton, FL, 2005.

[3] P. Naphon, Laminar convective heat transfer and pressure drop in the corrugated channels, International Communications in Heat and Mass Transfer 34 (2007) 62–71.

[4] P. Naphon, Heat transfer characteristics and pressure drop in the channel with V corrugated upper and lower plates, Energy Conversion and Management 48 (2007) 1516–1524.

[5] P. Naphon, K. Kornkumjayrit, Numerical analysis on the fluid flow and heat transfer in the channel with a V-shaped wavy lower plate, International Communications in Heat and Mass Transfer 35 (2008) 839–843.

[6] S. Eiamsa-ard, P. Promvonge, Thermal characteristics of turbulent rib–grooved channel flows, International Communications in Heat and Mass Transfer 36 (2009) 705–711.

[7] H.A. Mohammed, A. K. Abbas, J.M. Sheriff, Influence of geometrical parameters and forced convective heat transfer in transversely corrugated circular tubes, International Communications in Heat and Mass Transfer 44 (2013) 116–126.

[8] H.A. Mohammed, A. M. Abed, M.A. Wahid, The effects of geometrical parameters of a corrugated channel within out-of-phase arrangement, International

Communications in Heat and Mass Transfer 40 (2013) 47–57.

[9] M. Mirzaei, A. Sohankar, L. Davidson, F. Innings, Large Eddy Simulation of the flow and heat transfer in a half-corrugated channel with various wave amplitudes, International Journal of Heat and Mass Transfer 76 (2014) 432–446.

[10] H. S. Dizaji, S. Jafarmadar, F.Mobadersani, Experimental studies on heat transfer and pressure drop characteristics for new arrangements of corrugated tubes in a double pipe heat exchanger, International Journal of Thermal Sciences 96 (2015) 211e220.

[11] A. M. Abed, M.A. Alghoul, K. Sopian, H.A. Mohammed, H. S. Majdi, A. N. Al-Shamani, Design characteristics of corrugated trapezoidal plate heat exchangers using nanofluids, Chemical Engineering and Processing 87 (2015) 88–103.

[12] A.S. Navaei, H.A.Mohammed, K.M. Munisamy, H. Yarmandc, S. Gharehkhani, Heat transfer enhancement of turbulent nanofluid flow over various types of internally corrugated channels, Powder Technology 286 (2015) 332–341.

[13] A. K, Chandraa, K. Kishora, P. K. Mishrab, Md. S. Alama, Numerical simulation of heat transfer enhancement in periodic converging-diverging microchannel, Procedia Engineering 127 (2015) 95 – 101.

[14] J. M. Lee, P. W. Kwan, C, M, Son, M. Y. Ha, Characterizations of the aerothermal performance of novel cross-corrugated plate heat exchangers for advanced cycle aero-engines, International Journal of Heat and Mass Transfer 85 (2015) 166–180.

[15] L. Wang, L. Deng, C. Ji, E. Liang, C. Wangb, D. Che, Multi-objective optimisation of geometrical parameters of corrugated-undulated heat transfer surfaces, Applied Energy 174 (2016) 25–36.

[16] Y. Zhai, G. Xia, Z. Chen, Z. Li, Micro-PIV study of flow and the formation of a vortex in micro heat sinks with cavities and ribs, International Journal of Heat and Mass Transfer 98 (2016) 380–389.

[17] K. Schmidmayer, P. Kumar , P. Lavieille, M. Miscevic, F. Topin, Thermo-hydraulic characterization of a self-pumping corrugated wall heat exchanger, Energy 128 (2017) 713e728.

[18] Launder BE, Spalding DB, Mathematical Models of Turbulence, Academic Press, New York, 1972.

[19] Fluent 6 User's Guide, Lebanon, NH, Fluent Inc., 2006.

[20] H. Mohammed, A.M. Abed, M.Wahid, The effects of geometrical parameters of a corrugated channel within the out-of-phase arrangement, Int. Commun. Heat Mass Transfer 40 (2013) 47–57.

[21] V. Ozceyhan, S. Gunes, O. Buyukalaca, N. Altuntop, Heat transfer enhancement in a tube using circular cross-sectional rings separated from wall, Applied Energy, vol. 85, 2008, pp 988-1001.

[22] F. P. Incropera, D. P. DeWitt, T. L. Bergman, and A. S. Lavine, Fundamentals of Heat and Mass Transfer, 6th ed. (Wiley, 2007).

[23] B.E. Launder, D.B. Spalding, The numerical computation of turbulent flows, Computer Methods in Applied Mechanics and Engineering 3 (1974) 269–289.

[24] H. Han, B. Li, W. Shao. Effect of flow direction for flow and heat transfer characteristics in outward convex asymmetrical corrugated tubes. International Journal of Heat and Mass Transfer 92 (2016) 1236–1251.

NOMENCLATURE

A	Surface Area, (m^2)
C_p	Specific Heat (kJ/kg · °C)
D_h	Hydraulic Diameter, (m)
e	Height of Roughness, (m)
F	Friction Factor,($f = 2\ \Delta P\ D_h / L\ \rho\ u_m^2$)
H	Heat Transfer Coefficient,($W/m^2.K$)
H	Smooth Channel Height, (m)
I	Turbulent Intensity
K	Kelvin
kt	Thermal Conductivity, (W/m. K)
k	Turbulent Kinetic Energy,(J/kg)
Nu	Nusselt Number, ($Nu = h\ D_h/kt$)
P	Pressure, (N/m^2)
pe	Corrugation Pitch, (m)
ΔP	Pressure Drop,(Pa)
PEC	Performance Evaluation Criteria,($PEC = Nu/Nu_s/(f/f_s)^{1/3}$)
q	Heat Flux, (W/m^2)
Re	Reynolds Number, ($Re = \rho\ u_m\ D_h/\mu$)
T	Temperature, (K)
u	Component of Flow Velocity,(m/s)
u'	Local Flow Velocity (Fluctuated Velocity), (m/s)
V	Volume, (m^3)
w	Width of Corrugated,(m)
x	Horizontal Coordinate, mm
y	Vertical Coordinate, mm
Greek Symbols	
ρ	Fluid Density, (kg/m^3)
μ	Dynamic Viscosity, $N.s/m^2$)
ε	Turbulent Dissipation Rate, m^2/s^3)
τ	Shear Stress, (kg/m^2)
Subscripts	
av	Average
s	Smooth Channel
Wa	Wall
in	Inlet
out	Outlet
m	Mean
t	Turbulent
i, j	Components
b	Bulk
eff	Effective

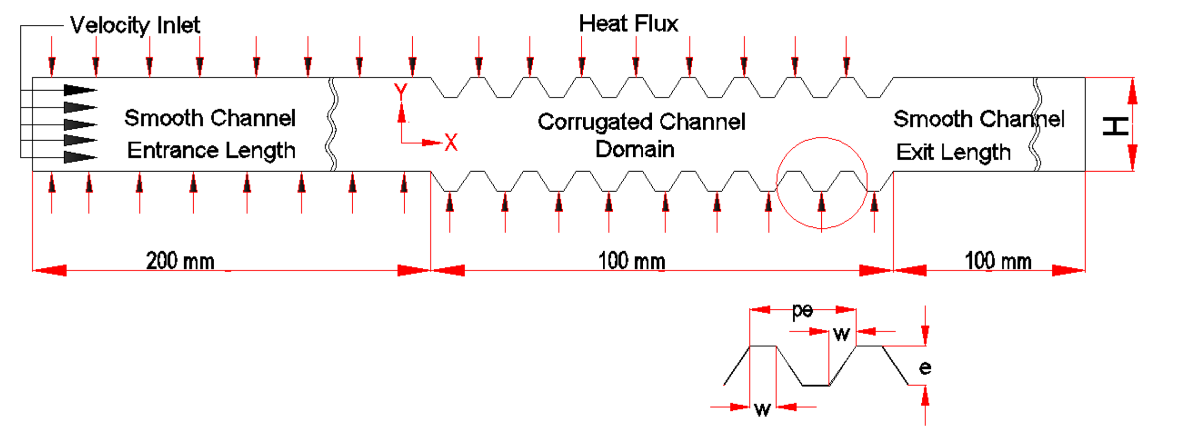


Fig. 1: Schematic diagram of the studied test section.

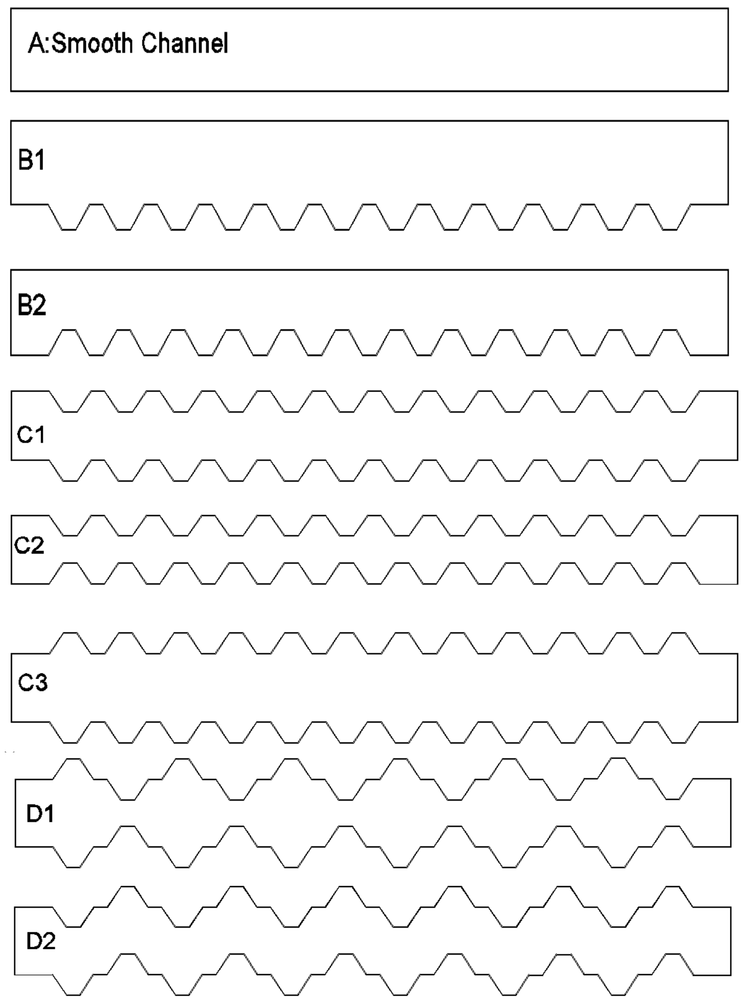


Fig. 2: Investigated region of eight different channel configurations: (1) Case A: smooth channel ; (2) Case B1: one side outward corrugations (1FOC) ;(3) Case B2: one side inward corrugations (1FIC);(4) Case C1: inward and outward corrugations (2FIOC) ;(5) Case C2: inward corrugations (2FIC) ;(6) Case C3: outward corrugations (2FOC) ;(7) Case D1: outward and inward are aligned on both surfaces (IAO), (8) Case D2: inward and outward staggered with outward and inward corrugations (IOSIO).

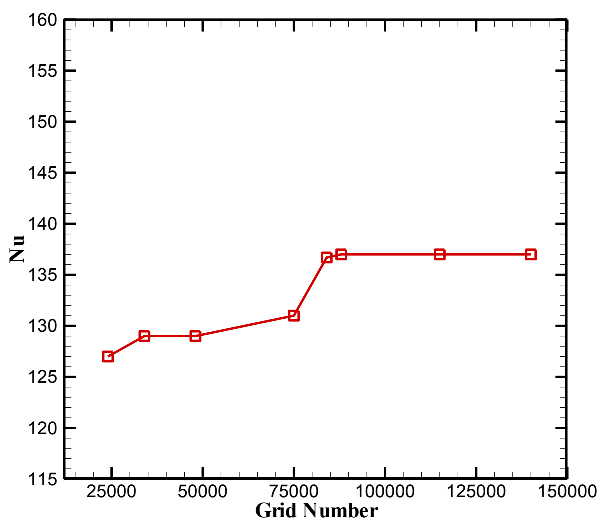


Figure 3: Grid-independence test.

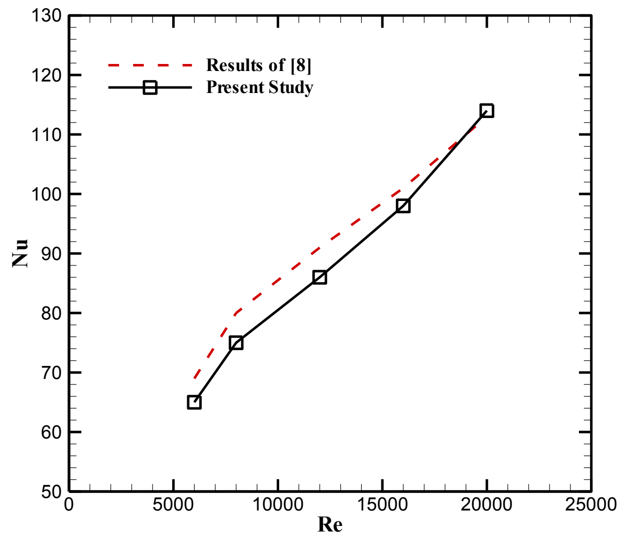
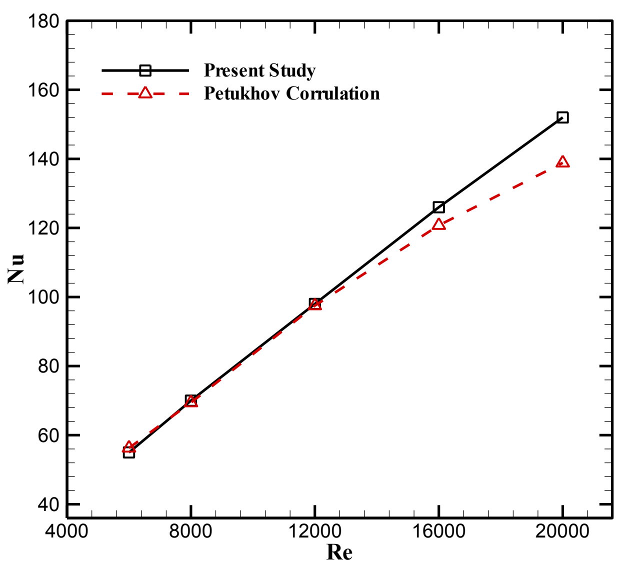
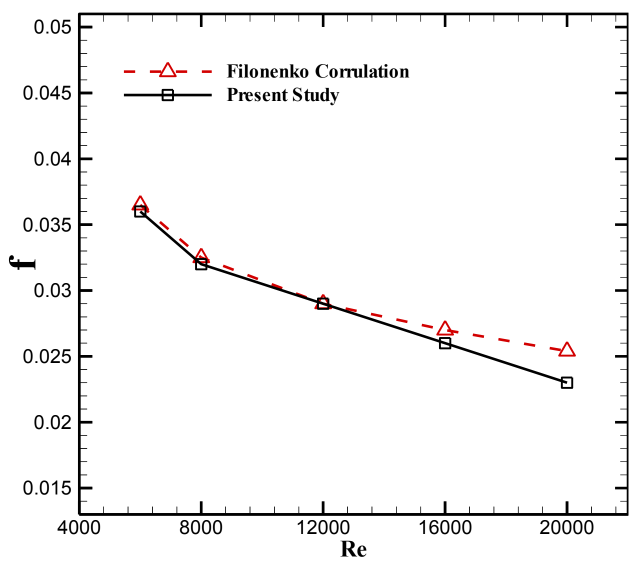


Figure 4: Comparison of the Nusselt number of the present results versus Reynolds number with that of Mohammed et al. [8].



(a)



(b)

Fig. 5: Comparison of the present work results with the results obtained by Mohammed et al. [8]: (a) average Nusselt number, (b) Darcy friction factor.

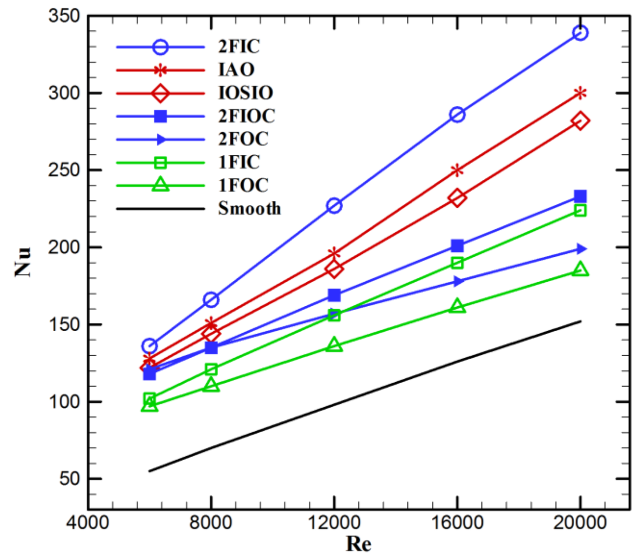


Fig. 6: Nusselt number results for smooth and corrugated channels with different Reynolds number.

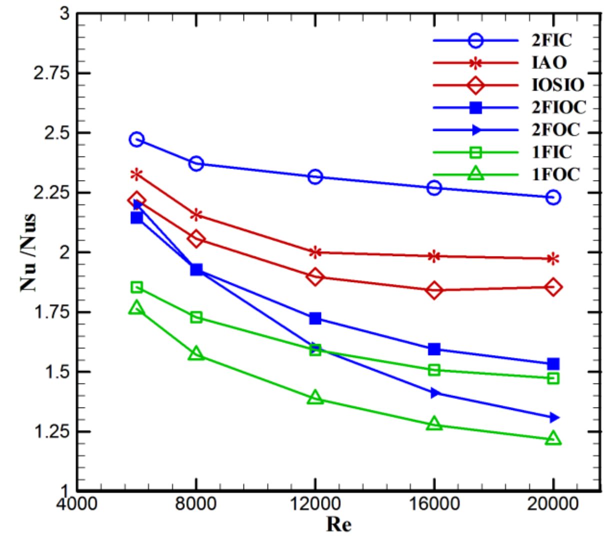


Fig. 7: Comparison of corrugated channel Nusselt number with that of the smooth channel.

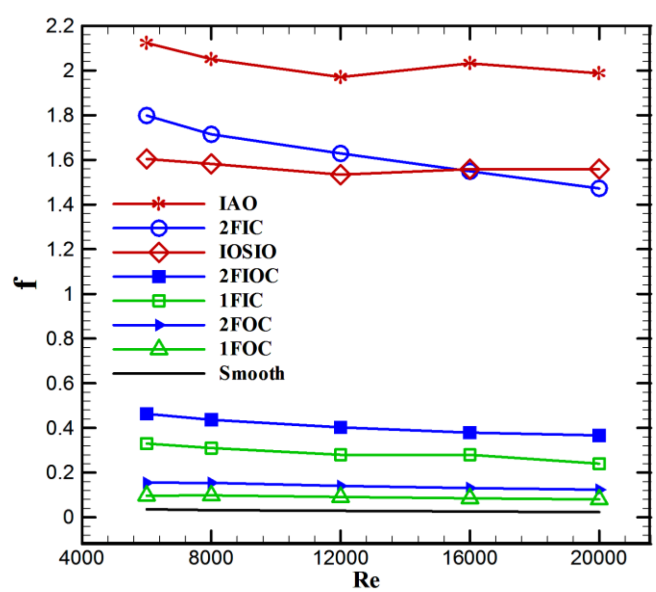


Fig. 8: Friction factors values for smooth and corrugated channels versus Reynolds number.

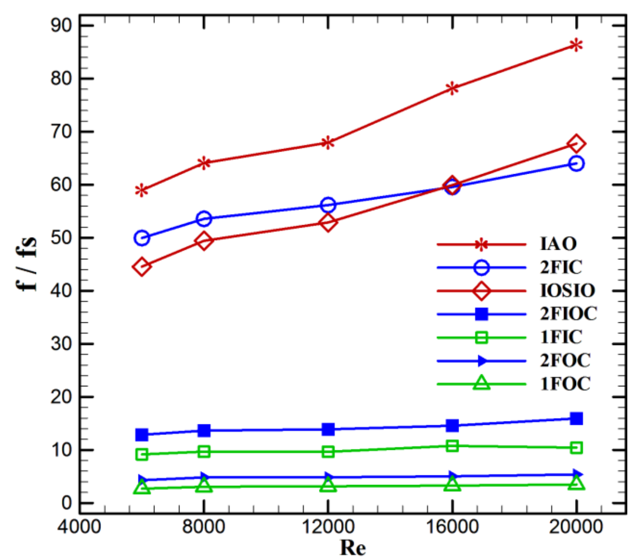


Fig. 9: Comparison of corrugated channel friction factors with the smooth channel friction factors.

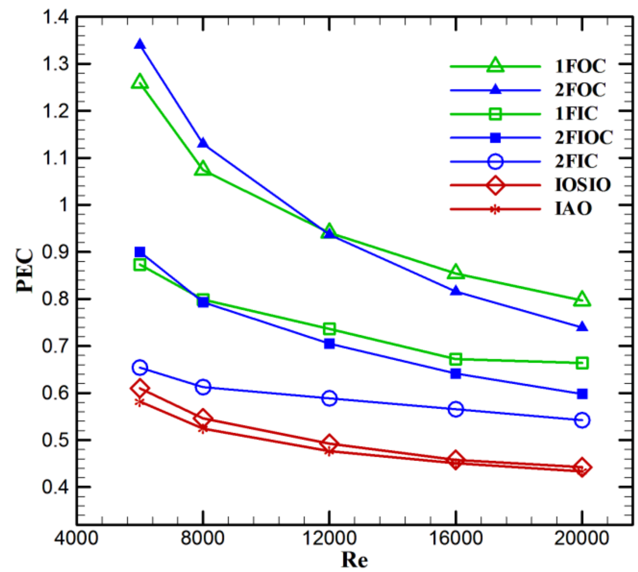


Fig. 10: PEC for channels with different trapezoidal corrugated configurations as function of Reynolds number.

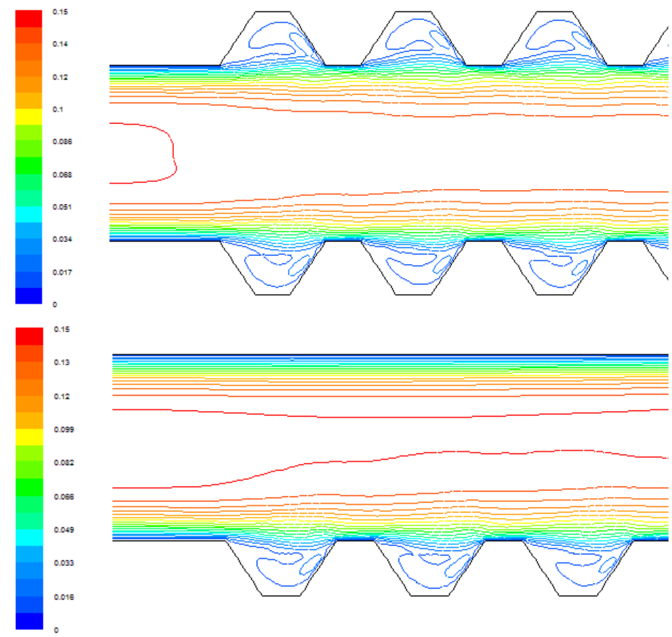


Fig. 11: Velocity contours for 1FOC and 2FOC channels at Re = 2000.

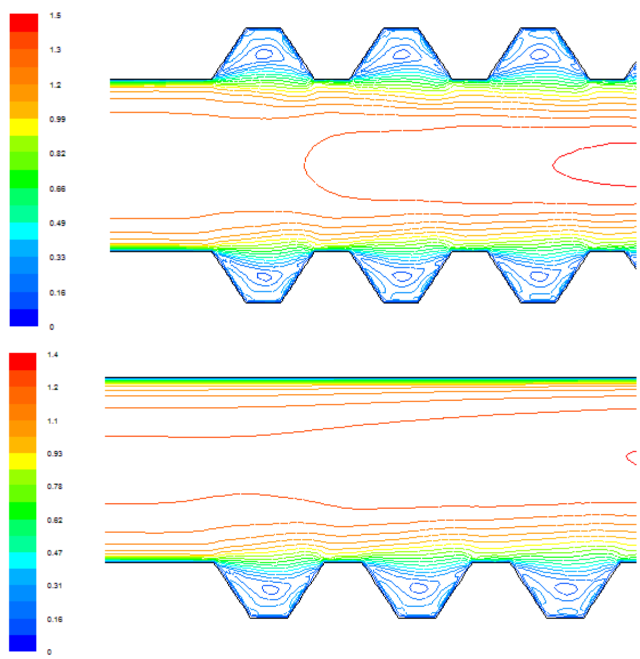


Fig. 12: Velocity contours for 1FOC and 2FOC channels at Re = 20000.

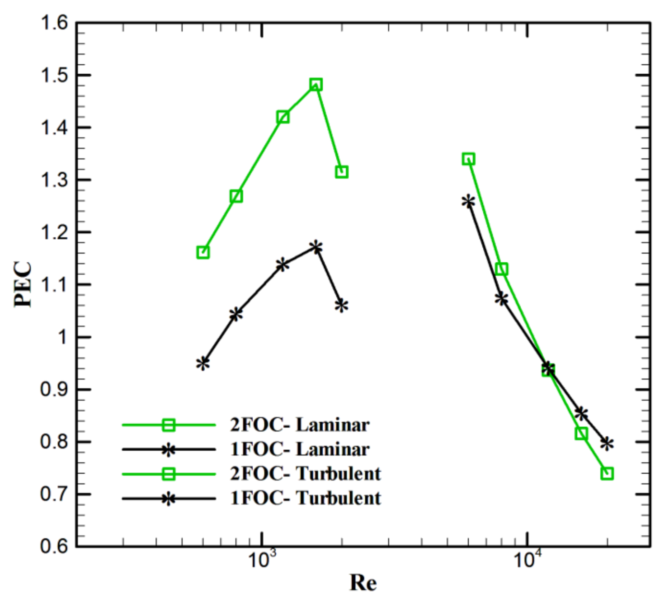


Fig. 13: Variation of PEC with Reynolds number for 1FOC and 2FOC.