Numerical Analysis for Flow and Heat Transfer in New Shapes of Corrugated Channel

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Abstract

Present study numerically investigates the flow field and heat transfer in two dimensional new shapes of corrugated wall channels. Simulations are performed for fully developed flow conditions. The governing continuity, momentum and energy equations are numerically solved using finite volume method based on SIMPLE technique. The investigation covers Reynolds number in the rage of 200-1400. The effects of Reynolds number and varying (LR/LL) ratio from 0.28 to 3.6 are studied for keeping wavy wall wavelength (L_w) value and distance between triangle in corrugated channel (d) fixed for all shapes.

The results show that heat transfer enhancement increases by using corrugated channel and also, heat transfer increases and vortices are decreases with decreasing the value (LR/LL).

Keywords: Corrugated Channel, CFD, Flow Characteristics, Heat Transfer

الخلاصة

تم في هذا البحث دراسة الجريان وانتقال الحرارة ثنائي البعد لمجرى ذو أشكال جديدة من التموج تحت ظروف الجريان كامل التطور ، تم حل المعادلات الحاكمة (الاستمرارية، الزخم والطاقة) عدديا باستخدام طريقة الحجوم المحددة (FVM) بالاعتماد على تقنية SIMPLE ، الدراسة تمت ضمن مدى لرقم رينولدز (200-1400)، في هذا البحث تم دراسة تأثير رقم رينولدز والنسبة (LR/LL) لمدى (0.28-3.6) بثبوت طول موجة الجدار (L_w) والمسافة بين التعرجات (d) لكل الاشكال،

اوضحت النتائج ان انتقال الحرارة يتحسن باستخدام المجرى المتموج كذلك فان انتقال الحرارة يزداد وتقل الدوامات بتقليل النسبة (LR/LL)،

الكلمات المفتاحية: القناة المتموجة، خصائص الجريان، انتقال الحرارة

Nomenclature

Symbol	Description	Unit
A	Wavy amplitude	m
D	Diffusion term coefficient	
d	Distance between triangle in corrugated channel	m
h_{ij}	Local heat transfer coefficient	W/(m2.K)
J	Jacobean of transformation	
H	Channel height	m
K	Thermal conductivity	W/(m.K)
L	The length of the channel	m
L_w	wavelength of the wavy wall	m
LR	Right length of wave	m
LL	Left length of wave	m
Nu	Nusselt number	
P	Source term in grid generation system of equation	
P_{ij}	Local static pressure	N/m^2
Pr	Prandtl number	
q	Heat flux per unit area	W/m^2
Q	Source term in grid generation system of equation	
T	Temperature	K
U	Velocity in x-direction	m/s
U, V	Contravariant velocities	
V	Velocity in y- direction	m/s
x, y	Cartesian coordinates	
P_0	Total pressure	N/m^2
2-D	Two dimensions	

FVM	Finite volume method	
N-S	Naveir-Stoke equation	
ξ, η	General coordinates	
Φ	General variable	
Ρ	Density	kg/m^3
Γ	Diffusivity	

1-Introduction

At present, heat transfer enhancement in heat exchanger devices has gained great popularity because of the importance of these devices in numerous engineering applications,due to the recirculation regions near the wall caused by wavy wall improves mixing of the fluid, which leads to the improvement in heat transfer .In addition,corrugated walls affect heat transfer enhancement by breaking and destabilizing the thermal boundary layer. Scientists from different countries dealt with the issue of heat transfer in flat and curved surfaces. [Goldstein *et. al.*,1977] were the first to study the local heat and mass transfer characteristics of wavy wall channels. A channel consisting of two corrugated waves was used for laminar, transitional, and low Reynolds number turbulent flow regimes. It was concluded in this study that corrugated walls were moderately effective on heat transfer rate in the laminar flow regime. However, heat transfer rate increased by a factor of three compared with the smooth duct when the flow was in the low-Reynolds-number turbulent regime.[Niceno *et al.*,2001] numerically studied two-dimensional steady and time-dependent fluid flow and heat transfer in sinusoidal and arc-shaped channels. For both geometrical configurations, corrugated walls had a negligible effect on heat transfer enhancement in comparison with plate channel in steady flow regimes at lower Reynolds number values. Moreover, they both have higher pressure drop compare to that of parallel-plate channel under fully developed flow conditions.

[Gradeck *et al.*, 2005] studied effect of wavy channel on heat transfer where wavy surfaces operate as turbulence promoters to increase the local heat transfer.[Islamoglu *et. al.*, 2004] numerically studied the effect of channel height on the heat transfer characteristics in a heat exchanger with corrugated channels.They demonstrated that the Nusselt number increases with an increment in the channel height in the same manner as friction factor.[Naphon *et al.*,2008] numerically studied the heat transfer and pressure drop characteristics in lower-side corrugated channels under constant heat flux. The obtained results showed that the corrugated surface has significant effect on the enhancement of heat transfer. Therefore, using corrugated plates is a proper method to increase the thermal performance and higher compactness of the heat exchangers.

[Ramgadia *et al.*,2012] depicted that the critical Reynolds number of unsteadiness is dependent on the geometrical parameters and the geometry with Hmin/Hmax (minimum-to-maximum distance between plates) ratio equal to 0.2 produced the highest Nusselt number and the best thermal performance factor. [Deylamiet *et al.*, 2013] studied influence of rib-height to channel-height ratio (e/H) on the heat transfer, friction factor and thermal enhancement characteristics in order to achieve higher efficiency for the corrugated channel figure(1).

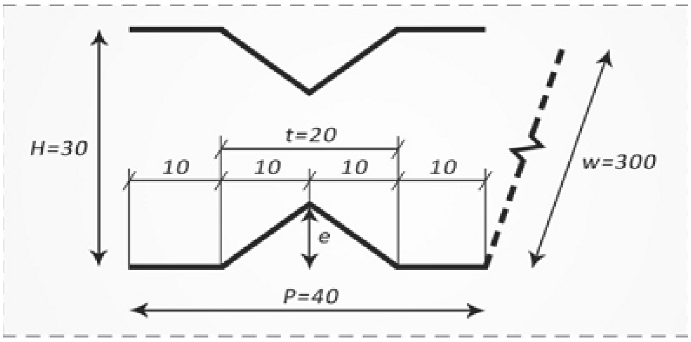


Figure (1) schematic view of the numerical domain for [Deylami *et al*., 2013]

[Cernecky *et al*.,2013] studied heat transfer intensification on shaped heat exchange surfaces using regulating elements.In this conditions the regulating elements can increase the values of local heat transfer coefficients along shaped heat exchange surfaces figure (2)

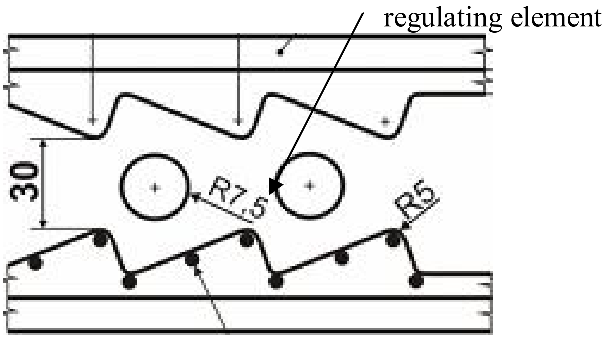


figure (2) Geometric arrangement of shaped heat exchange with regulating elements [Cernecky, 2013]

[Veli Ozbolat *et al*., 2013] studied numerically the flow field and heat transfer of water in two dimensional sinusoidal and rectangular corrugated wall channels.The investigation covered Reynolds number in the rage of 100-1000.The effects of the distance between upper and lower corrugated walls are studied by varying H_{min}/H_{max} ratio from 0.3 to 0.5 while keeping wave length and wave amplitude values fixed for both geometries figure (3).It was found that heat transfer enhancement increases by usage of corrugated horizontal walls in an appropriate Reynolds number regime and channel height.

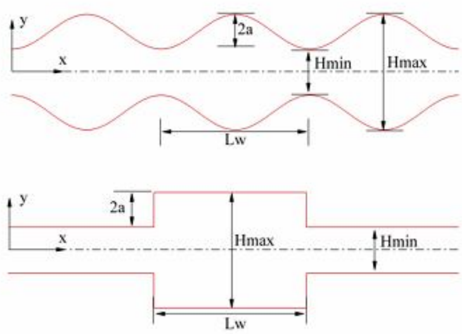


Figure (3) Schematic diagram of the channel [Veli Ozbolat *et al*., 2013]

[**Ağra** *et al*.,2011] investigated the pressure drop and heat transfer characteristics for five different tubes in turbulent flow conditions. The results showed that the corrugated tubes have higher heat transfer coefficients than the smooth tube, but lower heat transfer coefficients than the helically finned tubes.

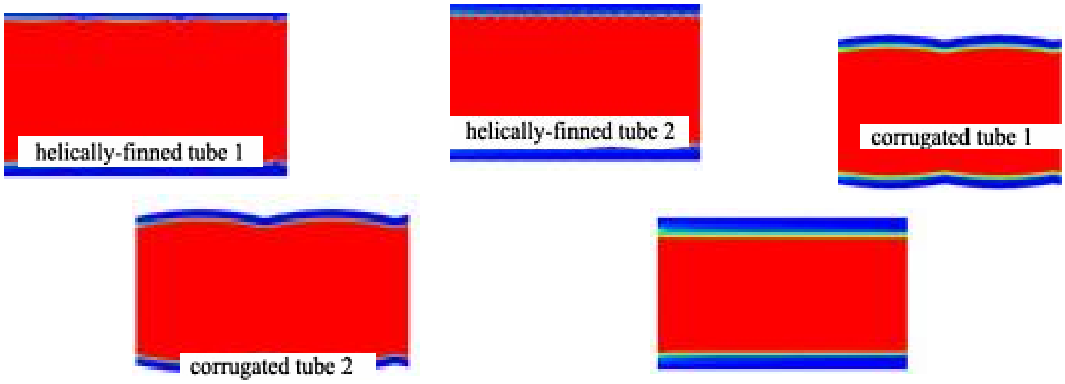


Figure (4) Schematic diagram for investigated tubes [Ağra *et al*., 2011]

[Mohamed *et al*.,2006] performed numerical analysis of laminar forced convection in the entrance region of a wavy-channel figure (5).The results for this study showed that in the channel entrance region, the viscous constraint tangential as well as the local Nusselt number were characterized by a very fast decrease and their amplitudes increase with increasing the wall corrugations and the Reynolds number.

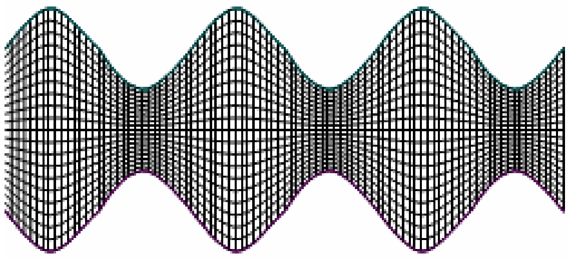


Figure (5) Geometry of wavy channel [Mohamed *et al*., 2006]

The studies presented above provide important insights into the correlation between the heat transfer and the geometry characteristics of the corrugated channel However, the definitions of the channel geometry parameters are generally different in each study. Different geometry parameter configuration can be considered. our research is focused on heat transfer intensification on new shaped heat exchange surfaces as shown in figure (6).

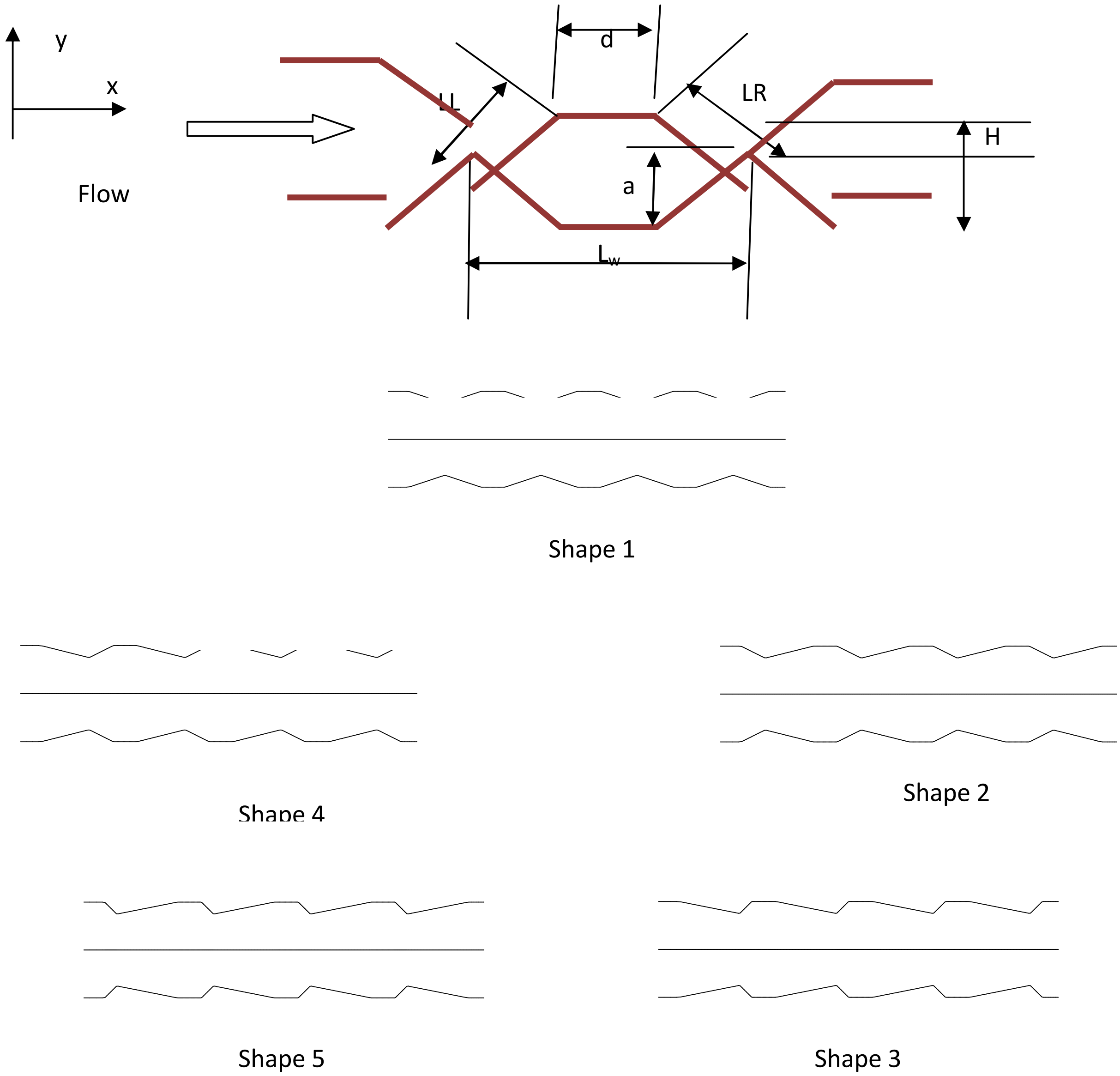


Figure (6) the shapes of new shaped heat channel for present studying
(a, H, L_w and L are constant)

Table (1) the dimensions change of new shapes of corrugated channel

Channel shape	LR/LL
shape 1	1
Shape2	0.54
Shape3	0.28
Shape4	1.85
Shape5	3.6

2-Grid Generation

Figure (7)show the schematic of two dimensions body fitting grid used for present computation, this grid obtained by solving non homogeneous 2-D Laplace equations [Thompson *et al*., 1977].

$$\left.\begin{aligned}\xi_{xx}+\xi_{yy}&=P(\xi,\eta)\\ \eta_{xx}+\eta_{yy}&=Q(\xi,\eta)\end{aligned}\right\} \quad \text{--1}$$

where P and Q used to clustering the grid near the walls to sense the velocity gradient because there is a friction between the wall and the fluid. Equations (1) are transformed to (ξ, η) coordinates by interchanging the roles of dependent variables This yields the following system at equations.

$$\left.\begin{aligned}g_{11}x_{\xi\xi}-g_{21}x_{\xi\eta}+g_{22}x_{\eta\eta}&=-J^2(Px_\xi+Qx_\eta)\\ g_{11}y_{\xi\xi}-g_{21}y_{\xi\eta}+g_{22}y_{\eta\eta}&=-J^2(Py_\xi+Qy_\eta)\end{aligned}\right\} \quad \text{-----------------------------------2}$$

Where

$g_{11}=x_\eta^2+y_\eta^2$

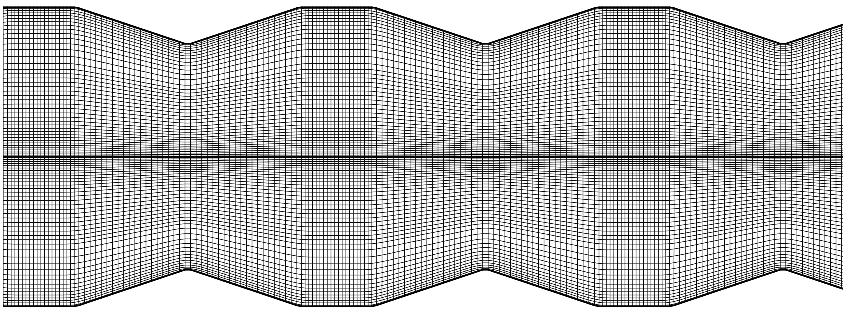
$g_{21}=2(x_\xi x_\eta+y_\xi y_\eta)$

$g_{22}=x_\xi^2+y_\xi^2$

$J=x_\xi y_\eta-x_\eta y_\xi$

Because of the complex geometry of the domain the (2-D) N-S, continuity and energy equations must be transported to general coordinates (ξ, η).

Figure (7) grid generated



3-Governing Equations

The conserve form of all fluid flow transport equations include equation of scalar quantity(T) can be written in general form for the dependent variable (Φ):-

$$\frac{\partial(\rho u\phi)}{\partial x}+\frac{\partial(\rho v\phi)}{\partial y}=\frac{\partial}{\partial x}\left[\Gamma_\phi(\frac{\partial\phi}{\partial x})\right]+\frac{\partial}{\partial y}\left[\Gamma_\phi(\frac{\partial\phi}{\partial y})\right]+S_\phi \quad \text{----------------------------3}$$

The two terms on the left hand side of the general equation (3) are convective terms while the first two terms on the right hand side are diffusive terms and the last term is the source term. The source term include both the source of (Φ) and any other terms that cannot be found in any of convective or diffusive terms.

Equation (3) serves as a starting point for the computation procedure in (FVM). By setting the dependent variable (Φ) and (Γ_Φ) according to the table (1). The term S_Φ in equation (3) is the source term of quantity Φ. When Φ stands for velocity components, S_Φ contains the appropriate pressure gradient term as well as terms arising from coordinate curvature. When Φ stands for temperature, S_Φ contains the terms arising from coordinate curvature. By applying partial differential chain rule to equation(3) and using table (1) the (mass, momentum and energy equations) in general coordinate with conterevariant velocity components are obtained. The procedure for transporting this equation to general equation can be found in [Ferziger *et al.*, 1999].

Table (2) definition of terms of the general transport equation

Equation	Φ	Γ_Φ
Continuity	1	0
x-momentum	U	μ
y-momentum	V	μ
Energy	T	μ/pr

4-Boundary conditions

The (FVM)required that the boundary flux, either be known or expressed in term of known quantities on interior nodal value.The boundaries of our solution include solid wall, inlet and outlet planes.

4-1 Inlet boundary

Usually at inlet boundary, all quantities have to be prescribed as following:-

$$\left.\begin{aligned}(u)_{inlet}&=u_{in}\\(v)_{inlet}&=v_{in}\\(p)_{inlet}&=p_{in}\\(T)_{inlet}&=T_{in}\end{aligned}\right\} \text{---}4$$

4-2 Outlet boundary

Always all values of dependent variables are unknown at exit boundary. It is often adequate to set outlet boundary values equal to intermediate up stream neighbor. This is equivalent to saying the first derivative of property Φ in the direction normal to the boundary is equal to zero.

$$\frac{\partial u}{\partial x}=0,\frac{\partial v}{\partial x}=0,\frac{\partial p}{\partial x}=0,\frac{\partial T}{\partial x}=0 \text{ -------------------------------------}5$$

4-3 Wall boundary

The velocity on the wall equal to zero

u=v=0

The first derivative of pressure in the direction is normal to the wall equal to zero

$$\frac{\partial p}{\partial y}=0 \text{---}6$$

For constant heat flux on bottom plane

$$T_w=q*\frac{y}{k}+T_{ij} \text{ ---}7$$

The aim of present numerical study is to know logically the effect of wavy channel on flow and local heat transfer by calculation Nusselt number (Nu).

Heat transfer calculations

In this study calculate heat transfer Nusselt number(Nu) from heated wall at constant heat flux:-

$$q = -k\frac{\partial T}{\partial y} \quad \text{--} 8$$

$$Nu = \frac{h_{ij} * d}{k} \quad \text{---} 9$$

Where

$$h_{ij} = \frac{k}{\Delta T} * \frac{\partial T}{\partial y} \quad \text{---} 10$$

$\Delta T=(T_{ij}-T_b)$ -- 11

$T_b=\sum T_{ij}/n$ --- 12

n is the number of points in y-direction

5-Results and discussion

Figure (8) shows the streamline for five geometrical models of corrugated channels at a Reynolds number of(1200).The streamline pattern for all shapes exhibits existence of the recirculation region in each wave of the channel due to separation of fluid. It can be seen from the shape(1) that the vortices are nearly equals in all waves from the beginning to end of the channel. In the case of shapes (2 and 3) the vortices are smaller in the waves and it is disappearing at the end of the channel because of the gradually enlargement in each corrugation. Shapes (4 and5) shows that the vortices become bigger in all of the waves due to the sudden enlargement in each corrugation.

Figure(9)shows the effect of Reynolds number on the vortices in shapes (3 and 5). The vortices are increased with increasing (Re) specially in the shape (5) because of increasing momentum of fluid flow that lead to increases the separation.

Figure (10) shows the variation of local Nusselt number along the lower wall of flat channel at (Re=500) for present study and reference [Veli Ozbolat *et al.*, 2013]. The data are in good agreement between them.

Figure (11) shows the variation of the local Nusselt number along the lower walls of flat channel and all shapes of corrugated channel at Re=600.The local Nusselt number decreases in diverging section due to the vortices and increasing in converging section of corrugated channel due to sweeps the heat by the flow since as flow moves, the velocity decreases and consequently heat transfer increased.

Shapes (4 and 5) have minimum heat transfer due to large vortices in the channels, the vortices decreases and heat transfer increased at the shapes (2 and 3). All shapes of corrugated channel have higher Nusselt number than the flat channel.

Figure (12) shows the variation of Nusselt number along the channels for the shapes (1) with different Reynolds number. Heat transfer increases when increases Reynolds number due to high sweeping of heat from the wall.

Figure (13) shows the effect of Reynolds number on average Nusselt number for shape (1) of corrugated channel, where heat transfer increased with increasing the velocity of fluid. This procedure is the same for all shapes.

6-Conclusion

The effect of using new shapes of corrugated channels on heat transfer and flow were numerically investigated, The major results of this study can be summarized as follows.

1- Heat transfers enhancing when using corrugated channel.
2- The governing continuity, momentum and energy equations are numerically solved using finite volume method based on SIMPLE technique

3- Heat transfers enhancing when using corrugated channel at narrowing region and then decreasing at the diverging region.
4- Heat transfer increasing with increased Reynolds number.
5- Shapes (2and 3) are the best of enhance heat transfer due to decreasing vortices.
6- New shapes of corrugated channel lead to appearing vortices.
7- Decreasing (LR/LL) of corrugated channel lead to disappearing vortices.

7-References

Niceno, B.E. Nobile,2001,"Numerical analysis of fluid flow and heat transfer in periodic wavy channels", *International Journal of Heat and Fluid Flow*, 22, 156-167.

Ferziger, J. and Peric, M.,1999," Computational method for fluid dynamic "2-thedition, Springer, Berlin

Deylami, H.M. , Amanifard*, N., Sanaei, M., Kouhikamal, **R.**,2013, "Numerical Investigation of HeatTransfer and Pressure Drop in a Corrugated Channel" *International Journal of Engineering TRANSACTIONS* A: Basics Vol.26, No.7, 771-780 .

Islamoglu, Y. and Parmaksizoglu, C,2004,"Numerical investigation of convective heat transfer and pressure drop in a corrugated heat exchanger channel", *Applied Thermal Engineering*, Vol. 24, No. 1, 141-147.

Goldstein, J.L. , Sparrow, **E.M. ,**1977,"Heat/mass transfer characteristics for flow in a corrugated wall channel" *ASME* Journal of Heat Transfer, 99, 187-195.

Cernecky, J, Koniar, J, Brodnianska, Z, 2013,"The effect of regulating elements on convective heat transfer along shaped heat exchange surfaces"*Chemical and Process Engineering*, 34 (1), 5-16.

Gradeck, M., Hoareau, B., Lebouche, M.,2005,"Local analysis of heat transfer inside corrugated channel", *International Journal of Heat and Mass Transfer*, 48, 1909-1915.

Mohamed, N, Khedidjaand, B, and Belkacem, Z,2006," Numerical Analysis of Flow and Heat Transfer in the Entrance Region of a Wavy Channel with Thermally Asymmetric Walls" *International Journal of Applied Engineering Research ISSN* 0973-4562 Vol. 1 No. 2, pp. 273-293.

Naphon, P. and Kornkumjayrit, K.,2008,"Numerical analysis on the fluid flow and heat transfer in the channel with v-shaped wavy lower plate*", International Communications in Heat and Mass Transfer,* Vol. 35, No. 7, 839-843.

Ağra, Ö, Demir, H, Atayılmaz, S,Ö, Kantaş, F, Dalkılıç, A, S, 2011,"Numerical investigation of heat transfer and pressure drop in enhanced tubes" *International Communications in Heat and Mass Transfer* 38, 1384–1391.

Ramgadia, A. G. and Saha, A. K., 2012,"Fully developed flow and heat transfer characteristics in a wavy passage: Effect of amplitude of waviness and Reynolds number", *International journal of Heat and Mass Transfer,* Vol. 55, No. 9, 2494-2509.

Thompson, J. F. Thomas, F. C. and Mastin,1977,"TOMACA-A code for numerical generation of boundary-fitted curvilinear coordinate system on field containing any number or arbitrary 2-D bodies" *J. of computational physics,*vol.24, pp 274-303.

Veli Ozbolat, Nehir Tokgoz, Besir Sahin, 2013," Flow Characteristics and Heat Transfer Enhancement in 2D Corrugated Channels" *International Journal of Mechanical Science and Engineering* Vol:7 No:10.

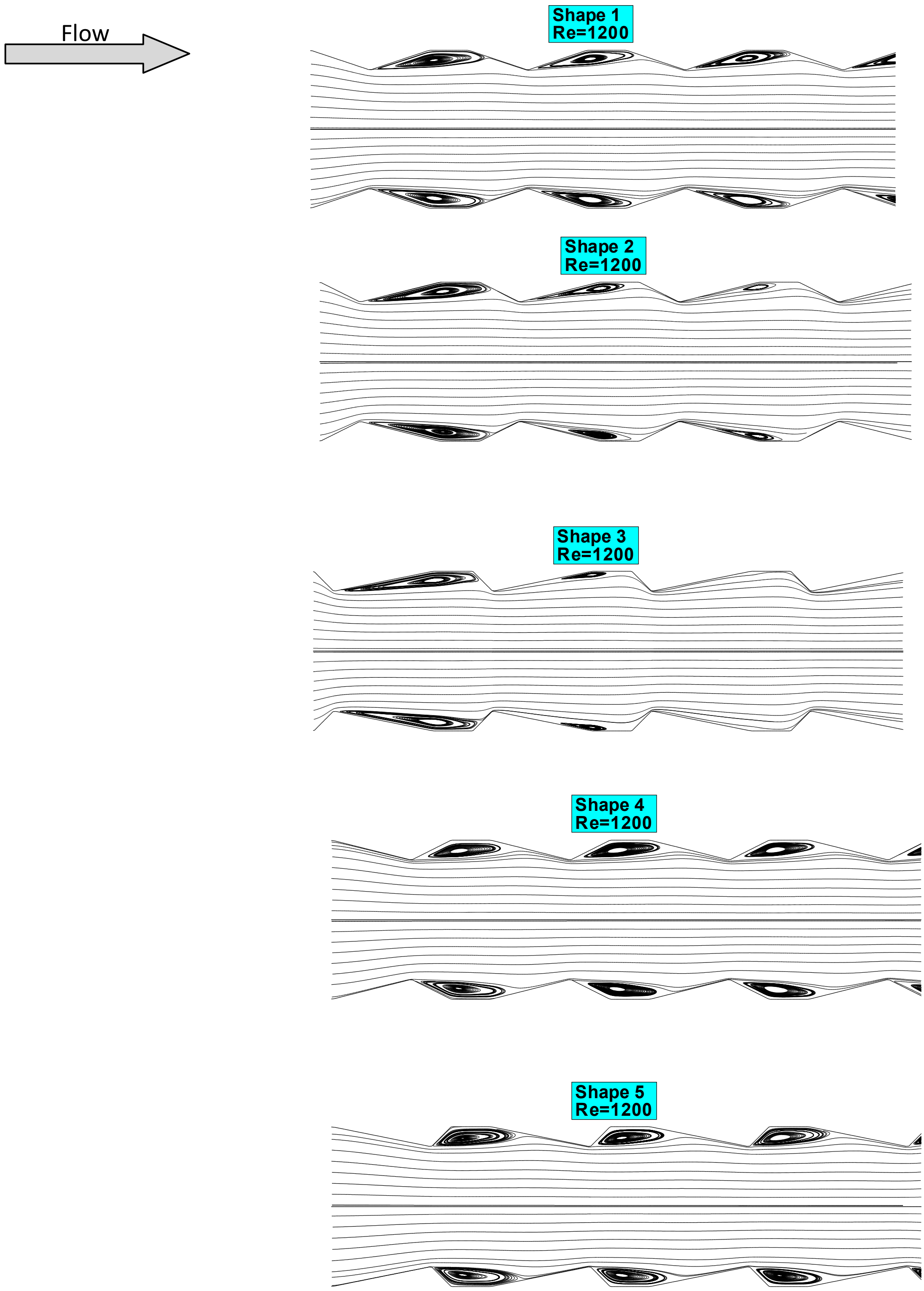
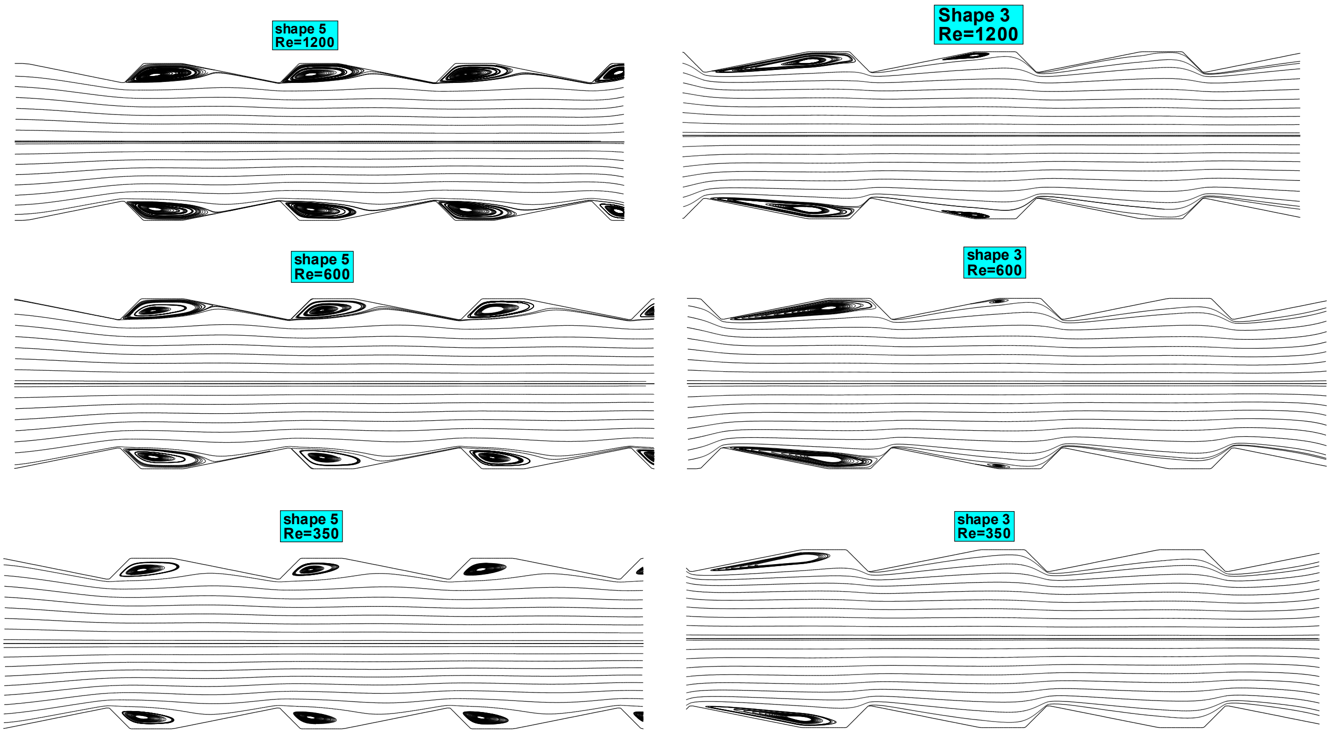


Figure (8) streamline plots at Re=1200 for five different shape of corrugated channels



figure(9)streamline in the shape (3 and 5) for different Reynolds number

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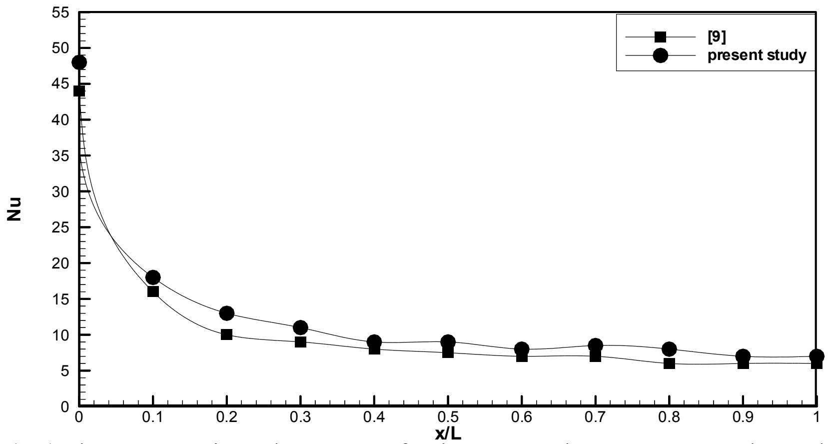

Figure (10) the comparison heat transfer between the present study and reference [Veli Ozbolat *et al.*, 2013] for flat channel

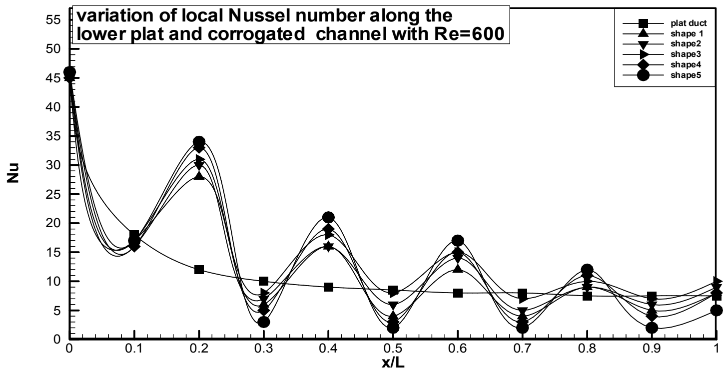

Figure (11) shows the variation of the local Nusselt number along the lower walls of plat and all shapes of corrugated channel at Re=600.

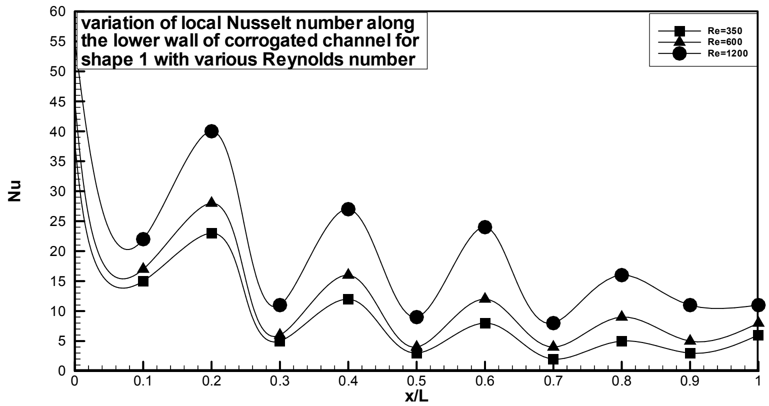

Figure (12) the variation of Nusselt number along the channels for the shapes (1) with different Reynolds number

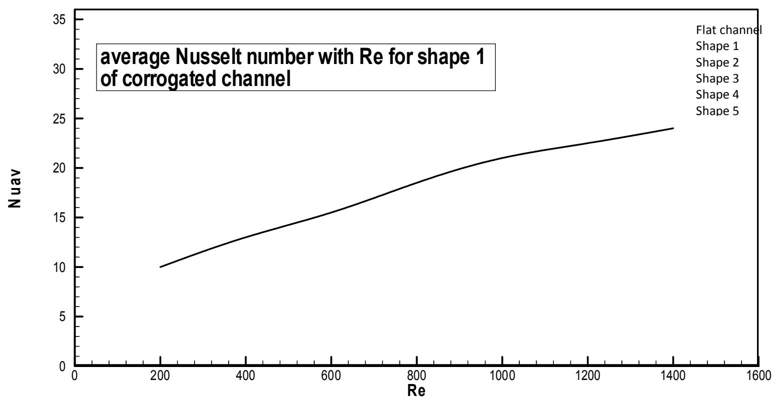

Figure (13) shows the effect of Reynolds number on average Nusselt number for shape 1 of corrugated channel