

**ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES**

PhD THESIS

Veli ÖZBOLAT

**FLOW CHARACTERISTICS AND HEAT TRANSFER ENHANCEMENT OF
SINUSOIDAL CORRUGATED CHANNELS**

DEPARTMENT OF MECHANICAL ENGINEERING

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ABSTRACT

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In this study, flow characteristics and heat transfer enhancement in various sinusoidal corrugated channels were investigated. The channel height and phase shift effects on the flow structures and thermal performances were examined for Reynolds numbers in the range of $2500 \leq Re \leq 7500$. The geometrical configurations of this study cover three different channel heights ($H_{min}/H_{max}=0.36, 0.54, 0.72$) and phase shift angles ($\phi=0^\circ, 90^\circ, 180^\circ$). Firstly, flow characteristics in corrugated channels were investigated both experimentally and numerically. Experiments were conducted using the Particle Image Velocimetry (PIV) technique to compare with the results obtained from the numerical simulations. Afterwards, the hydrodynamic structures and thermal performance of the sinusoidal corrugated channels were studied. Velocity distribution, turbulence intensity, local and averaged Nusselt numbers, Nu and friction factor, f were calculated for various configurations. Finally, thermal performance factor (TPF) were also calculated to determine the optimum channel configuration and operating condition.

Key Words: Corrugated channel, heat transfer enhancement, phase shift, PIV

ÖZ

DOKTORA TEZİ

**SİNÜS EĞRİSİ ŞEKLİNDEKİ OLUKLU KANALLARDA AKIŞ
KARAKTERİSTİĞİ VE ISI TRANSFERİ İYİLEŞTİRMESİ**

Veli ÖZBOLAT

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Bu çalışmada, sinüs eğrisi şeklindeki oluklu çeşitli kanalların içindeki akış karakteristiği ve ısı transferi gelişimi incelenmiştir. Kanal yüksekliği ve faz açısının akış yapısı ve ısı performansına etkileri Reynolds sayısı, $2500 \leq Re \leq 7500$ aralığında incelenmiştir. Bu çalışmanın geometrik yapılandırılması; üç farklı kanal yüksekliğini ($H_{min}/H_{max}=0.36, 0.54, 0.72$) ve üç farklı faz açısını ($\varphi=0^\circ, 90^\circ, 180^\circ$) kapsamaktadır. İlk olarak, oluklu kanallardaki akış karakteristiği hem deneysel hem de sayısal olarak incelenmiştir. Parçacık görüntülemeli hız ölçüm (PIV) tekniği ile elde edilen deneysel sonuçların, sayısal benzeşim yöntemi ile elde edilen sonuçlarla kıyaslaması yapılmıştır. Sonrasında, sinüs eğrisi şeklindeki oluklu kanalların hidrodinamik yapıları ve ısı performansları çalışılmıştır. Hız dağılımı, türbülans düzeyi, yerel ve ortalama Nusselt sayısı, Nu ve sürtünme katsayısı, f çeşitli geometrik yapılar için hesaplanmıştır. Nihai olarak, en uygun kanal geometrik parametrelerini ve çalışma şartını belirlemek amacıyla ısı performans faktörü (TPF) de hesaplanmıştır.

Anahtar Kelimeler: Faz açısı, ısı transferi gelişimi, oluklu kanal, PIV

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LIST OF ABBREVIATIONS

x	:	Streamwise direction
y	:	Transverse direction
u	:	Velocity in x direction
v	:	Velocity in y direction
U_{∞}	:	Free-stream velocity
μ	:	Dynamic viscosity
ν	:	Kinematic viscosity
u'	:	Streamwise velocity fluctuation
v'	:	Streamwise velocity fluctuation
$\langle u \rangle$	:	Time-averaged velocity component in x direction
$\langle V \rangle$	:	Time-averaged velocity fields
$\langle \Psi \rangle$	:	Time-averaged streamline
$\langle \omega \rangle$	:	Time-averaged vorticity
$u'u'$	:	Streamwise Reynolds normal stress
$v'v'$	:	Transverse Reynolds normal stress
$\langle u'v' \rangle$	:	Time-averaged Reynolds shear stress correlations
$\langle \text{TKE} \rangle$	:	Time-averaged turbulent kinetic energy
u_{rms}	:	RMS value of the streamwise velocity fluctuation
v_{rms}	:	RMS value of the vertical velocity fluctuation
Re	:	Reynolds number
Nu	:	Nusselt number
Nu_0	:	Nusselt number in parallel plate channel
$\overline{Nu}$	:	Averaged Nusselt number
S_t	:	Stokes number
Pr	:	Prandtl number
Ra	:	Rayleigh number
f	:	Friction factor
f_0	:	Friction factor in parallel plate channel
H	:	Height of the channel

$H_{\max}$	:	Maximum height of the corrugated channel
$H_{\min}$	:	Minimum height of the corrugated channel
a	:	Amplitude of the sinusoidal wavy wall
L	:	Length of the channel
L_w	:	Length of the sinusoidal wave
ϕ	:	Phase angle
TPF	:	Thermal performance factor
TI	:	Turbulence Intensity
t	:	Time
p	:	Pressure
Δp	:	Pressure difference
ρ	:	Density
T	:	Temperature
T_m	:	Bulk temperature
T_w	:	Wall temperature
T_i	:	Inlet temperature
k	:	Thermal conductivity
h	:	Heat transfer coefficient
q''	:	Heat flux
D_h	:	Hydraulic diameter
G_k	:	Kinetic energy generation
ε	:	Turbulent kinetic energy dissipation rate
C_p	:	Pressure coefficient

1. INTRODUCTION

In recent years, energy became more important to all human activities such as manufacturing, heating, cooling, lighting, transportation, food processing, etc. An increase of energy consumption also related to life standards. In addition, there is a strong relationship between energy consumption and economic growth. During worldwide economic crisis between 2008 and 2009, most of the countries had to decline energy demands. Since reducing the energy demands is a temporary solution, developed countries created some policies on energy saving and energy efficiency with the aim of energy conservation both in industrial and local businesses.

Management of heat recovery in industrial applications has become substantially important as we endeavor to improve the efficient use of energy. Therefore, control of heat transfer for its enhancement and reduction heat losses in these systems is an important topic from the energy saving perspective. A process of heat transfer becomes very important whenever energy needs to be generated, exchanged, or transformed. In a wide variety of circumstances, heat transfer principle is to increase the temperature by generating heat, however, sometimes the goal is to remove heat and decrease the temperature. Heat transfer can be observed everywhere in practical engineering. A process of heat transfer can be used in food industry, climate control, thermal management of electronic devices and systems, automotive engineering, chemical engineering, nuclear engineering, materials processing. It is well known that some energy devices such as condenser, evaporator, and heat exchanger can be used in heat transfer applications.

In recent years, an increase in cost of energy generation and it has obligated the manufacturers to develop more efficient energy systems and new methods to generate energy. Therefore, engineers are obligated to try to improve more efficient heat exchangers by optimizing the geometry of heat exchangers and using fluids with higher thermal performance.

1.1. Heat Transfer Enhancement Techniques

There are three different control techniques for heat transfer enhancement: these are passive, active, and compound techniques which are commonly used in thermal systems such as thermal power plants, heating, cooling and evaporators systems, air conditioning equipment, and radiators of automobiles, etc. The active method needs some external power input from an external source for the enhancement of heat transfer and has not shown much potential because of the design complexity. In addition, it is difficult to provide external power in many heat transfer augmentation applications. Some examples of active methods can be counted as induced pulsation by cams and reciprocating plungers, the use of a magnetic field etc. Passive techniques do not require external power directly. For passive techniques, heat transfer enhancement can be achieved by modifying surface geometries, roughness of materials, fluid properties or attaching objects to increase surface area and turbulence. The common objective of these methods is to disturb boundary layer on the surface, and replace it from core fluid, which creates new boundary layer with higher temperature gradient. Relevant examples of this method are twisted tapes, wire coils, ribs, vortex generators, louvers, offset-fins, etc. Compound method is the combination of the passive and active methods, such as rough surface with a twisted tape swirl flow device, rough surface with twisted tapes, or rough surface with fluid vibration.

1.1.1. Passive Flow Control Techniques

Passive methods does not need input power externally, but additional power requirements are supplied from the power available in the system which may causes fluid pressure drop. Surface or geometry of the flow channel are modified by including additional parts or any other inserts. Passive control techniques are advantageous compared with active control techniques because it is inconvenient to employ active control methods under compact geometry. Moreover, active control methods do require external power which causes more operational costs and

problems associated with vibration or acoustic noise (Webb, 1987). Thus, passive control methods, which employs special geometries in order to enhance heat transfer are usually preferred by designers. Although there are numerous passive methods to enhance heat transfer, the followings are the most commonly used in different industrial applications (Liu and Sakr, 2013):

- **Treated surfaces:** A fine scale changes either continuous or discontinuous are made to the heat transfer surfaces with coating. Very small roughness can be obtained to increase single-phase heat transfer, and they are used mainly for condensing and boiling applications.
- **Rough surfaces:** One of the passive techniques in order to improve heat transfer enhancement is modifying the heated surface characteristics by increasing the roughness of the surface without increasing the heat transfer surface area. Rough surfaces reduce the thermal boundary layer thickness and make the turbulence transition earlier.
- **Extended surfaces:** Heat transfer enhancement can be achieved by using modified fin surfaces which increase the heat transfer surface area as well as improve the heat transfer coefficients by disturbing the flow field.
- **Displaced enhancement devices:** These devices are preliminary used for force convection. They increase heat transfer by displacing the fluid from the heated or cooled surface with bulk fluid to the core flow.

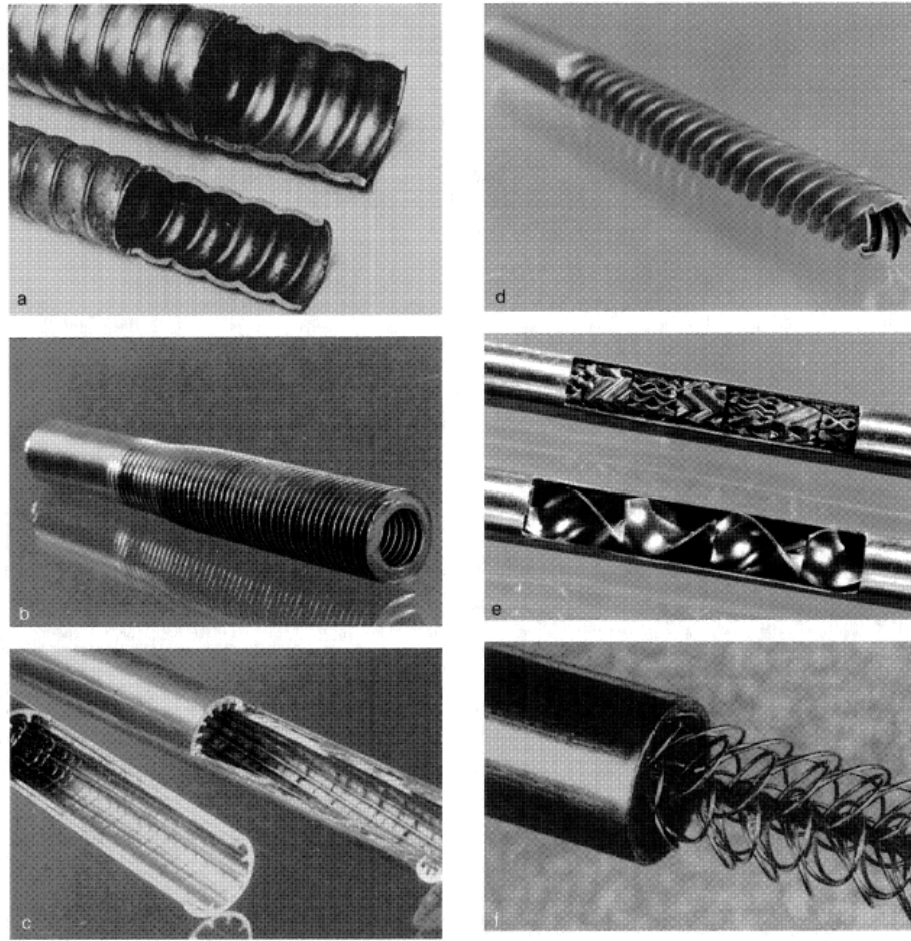


Figure 1.1 Enhanced tubes for augmentation of single-phase heat transfer. (a) Corrugated or spirally indented tube with internal protuberances. (b) Integral external fins. (c) Integral internal fins. (d) Deep spirally fluted tube. (e) Static mixer insert. (f) Wire-wound insert. (Kreith, F.; Boehm, R.F.; et. al. "Heat and Mass Transfer" Mechanical Engineering Handbook, CRC Press LLC, 1999)

- **Swirl flow devices:** These devices which can be used for both single phase and two-phase flows heat exchangers generate swirl flow or secondary recirculation on the axial flow in a channel. Two examples of these devices are twisted tapes and tube inserts.
- **Coiled tubes:** These tubes are classified as one of the passive heat transfer enhancement techniques and are widely used since they have compact structures. As fluid flows through curved tubes, centrifugal force is generated due to the curvature of the tubes. A secondary flow and vortices produced by centrifugal force improve heat transfer rate. Spiral and helical coils are two

common types of coiled tubes and they are used air conditioning and refrigeration systems, heat recovery processes, chemical reactors, etc.

- **Surface tension devices:** These devices improve the boiling and condensing surface by wicking and grooving the surfaces.
- **Fluid Additives:** The addition of small particles containing a phase change material (PCM) may improve the heat transfer enhancement since presence of the PCM changes heat capacity of the fluid.

1.1.2. Active Flow Control Techniques

Active flow control methods need some external power input to the system for the heat transfer augmentation. These control techniques has not shown much potential because of the complexity and power input required to the system. The input to the system can be form of electricity, power, external pump, or radio frequency (RF) signals.

- **Mechanical Aids:** Such instruments stir the fluid by mechanical means or by rotating the surface. These include rotating tube heat exchangers and scrapped surface heat and mass exchangers.
- **Vibration:** Vibration in the fluid or surface can be used as an active control of heat transfer enhancement techniques. They are used preliminary for single phase flow. Fluid vibration techniques are most practical vibration types for heat transfer enhancement.
- **Electrostatic fields:** Electric or magnetic fields or combination of these two forms AC or DC sources can be applied.
- **Injection:** This technique can be used by injecting the same or different fluid into the bulk fluid.
- **Suction:** In this method, fluid can be withdrawn through a porous heated surface in single-phase flow or vapor can be removed through a porous heated surface in nucleate or film boiling.

- **Jet impingement:** It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.

1.1.3. Compound Flow Control Techniques

One of the above methods can be used simultaneously in order to achieve further improvement in heat transfer enhancement. Since this control technique needs complex design, it has limited applications.

1.2. Classification of Heat Exchangers

Heat exchangers are devices built for efficient heat transfer between two or more fluids at different temperature. Heat exchangers are widely used in many applications, such as power plants, chemical and food industries, refrigerators and air-conditioners, waste heat recovery system, space heating, manufacturing industry, electronics, petroleum refineries, etc. Kakaç et al. (2002) classified heat exchangers according to following main criteria:

- Recuperators/regenerators
- Transfer processes: direct and indirect contact
- Construction geometry: tubes, plates and, extended surfaces
- Heat transfer mechanism: single phase and two phase
- Flow arrangement: parallel flows, counter flows, and cross flows

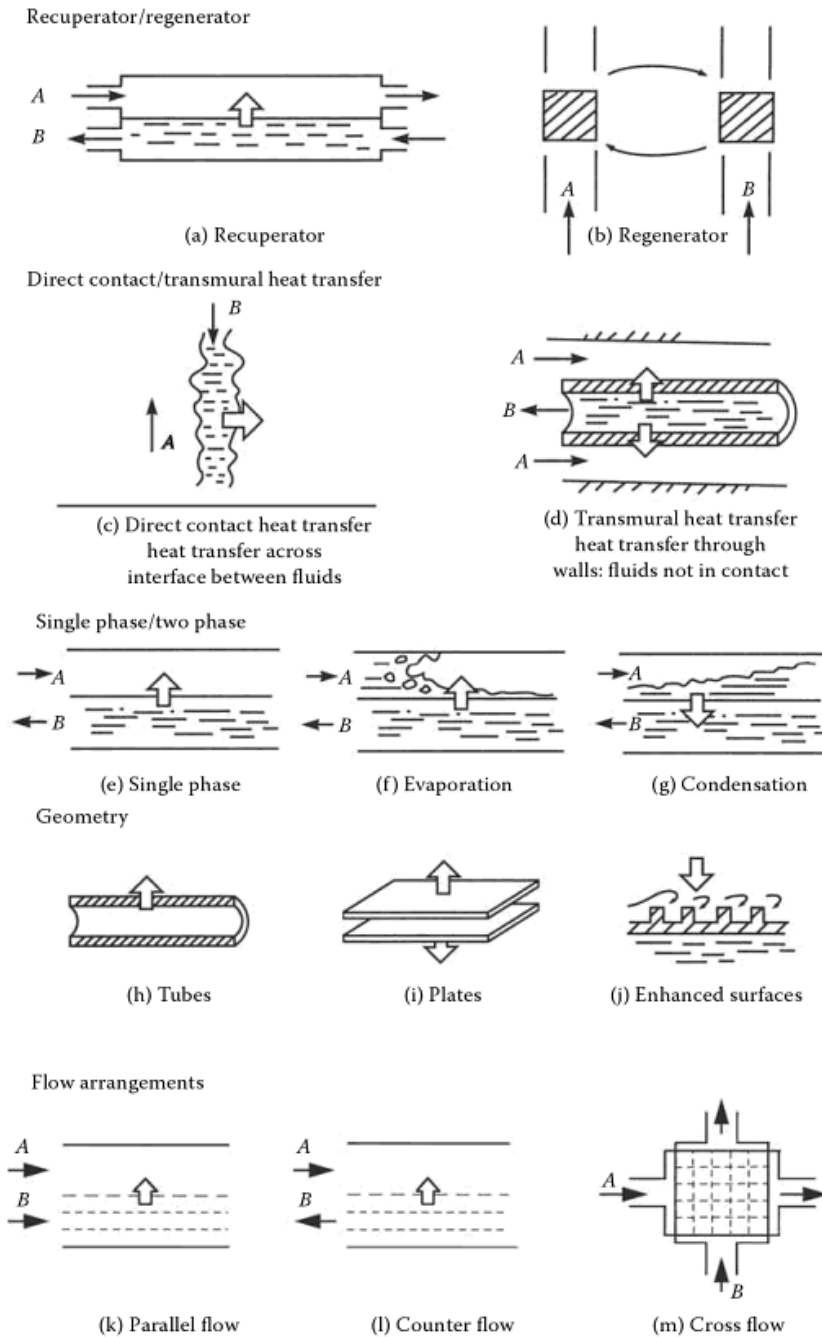


Figure. 1.2. Heat Exchangers Classification Criteria. (From Hewitt, G. F., Shires, G. L., and Bott, T. R., Process Heat Transfer, CRC Press, Boca Raton, FL, 1994.)

A plate heat exchanger (PHE) is a type of heat exchanger using metal plates to heat transfer between fluids. A plate heat exchanger consists of a series of thin corrugated plates which produce an extremely large surface area. As compared to other types, plate heat exchanger is more compact due to the large heat transfer area

provided by the plates. Plate heat exchangers are used due to the advantages over alternative shell and tube type heat exchangers such as efficiency, size, economy, energy savings, and ease of cleaning.

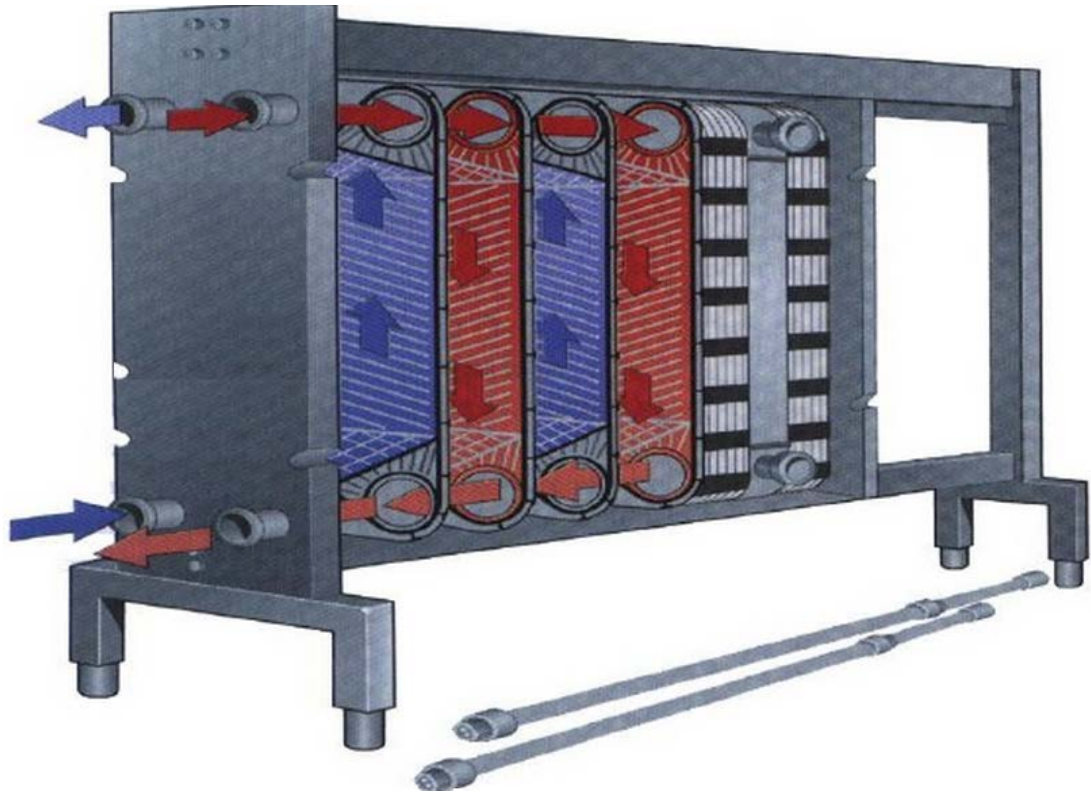


Figure 1.3. Plate Heat Exchanger

1.3. Heat Transfer Enhancement of Heat Exchangers

The need for improved efficiency in heat exchangers has recently gained great popularity due to the increased demand for built and operate heat exchanger devices less expensive. Efficiency of heat exchangers can be increased by reducing the size and improving heat transfer enhancement. The common objective of these methods is to disturb boundary layer on the surface, and replace it from core fluid, which creates new boundary layer with higher temperature gradient. Relevant examples of this method are vortex generators, louvers, offset-fins, etc.

1.3.1. Corrugated Walls

A simple and effective way to increase heat transfer enhancement of heat exchangers is to use optimum surface design which do not bring considerable pressure loss. Corrugated channels are used in a wide array of heat transfer applications because they are easy to fabricate, increase compactness and can be used to increase the heat transfer rate if implemented in an appropriate Reynolds number regime. Wavy channels do not have significant effects if operated in steady regime but they are very effective on heat transfer enhancement when the flow is unsteady since interactions between boundary layer and core fluid are improved as observed experimentally and numerically (Nishimura (1984), Nishimura et al. (1990), and Wang and Vanka (1995)). Wavy walls in heat exchangers increase mixing of the fluid, which leads enhancement in heat transfer. Furthermore, wavy walls are effective on heat transfer enhancement by breaking and destabilizing the thermal boundary layer. Therefore, wavy surfaces operate as turbulence promoters to increase the local heat and mass transfer.



Figure 1.4. Corrugated surface

1.3.2. Nanofluids

Heat transfer efficiency of heat exchangers can also be improved by increasing the thermal conductivity of working fluid since the thermal conductivity of these fluids play important role on the heat transfer coefficient between the heat transfer medium and the heat transfer surface. Due to poor thermal conductivity, heat exchangers of conventional fluids such as water, air, oil show low performance. Suspension of solids in fluids enhances the effective thermal conductivity of the fluids. Several researchers have investigated to increase the thermal conductivity of fluids by suspending micro or larger-sized solid particles in fluids for more than 100 years since thermal conductivity of solids higher than that of liquids. However, they faced with some drawbacks which make unfeasible to use larger-sized solid particles in practical heat transfer applications. These drawbacks are sedimentation of particles, clogging of channels and erosion in channel walls, and increase in pressure drops. The recent advance in materials technology has made it possible to produce nanometer-sizes particles. Nanometer-sized particles suspended fluids, known as “nanofluids” which was first used by Choi in 1995 of the Argonne National Laboratory, have generated considerable interest for their potential to enhance the heat transfer rate in engineering systems. Since nanoparticles fluidize easily inside the base fluid owing to small size, using nanofluids eliminates the sedimentation, clogging of channels and erosion in channel walls that were common problems for solid-liquid mixtures with larger particles. Heat transfer performance of the fluid can be significantly improved by suspending nanoparticles in cooling or heating fluids. The main reasons of such improvement may be listed as follows (Xuan and Li, 2000):

- The suspended nanoparticles increase the surface area which lead to increase in the heat capacity of the fluid and the stability of the suspensions.
- The suspended nanoparticles increase the effective thermal conductivity of the fluid.

- The interaction and collision among particles, fluid and the flow passage surface are intensified.
- The mixing fluctuation and turbulence of the fluid are intensified.
- The dispersion of nanoparticles flattens the transverse temperature gradient of the fluid.

2. PREVIOUS STUDY

2.1. Experimental Studies of Flow in Corrugated Channels

Goldstein and Sparrow (1977) were probably the first to study the local heat and mass transfer characteristics of wavy wall channels. Two corrugated waves were used for laminar, transitional, and low Reynolds number turbulent flow regimes. They concluded that corrugated wall had a negligible effect on heat transfer rate in the laminar flow regime. However, heat transfer rates were improved by a factor of three over those of conventional straight channel. O'Brien and Sparrow (1982) conducted experiments to investigate forced convection heat transfer coefficients, h and friction factors, f for flow in a corrugated duct. In that study Reynolds numbers, Re ranged from 1500 to 25000 based on the hydraulic diameter, and the Prandtl number, Pr between 4 and 8. The results showed that heat transfer was enhanced by a factor of 2.5, but the friction factor, f of corrugated duct was significantly greater than the friction factor, f of straight duct. Sparrow and Comb (1983) made a similar study for a corrugated channel with an interwall spacing that was approximately 45% greater than the channel used by O'Brien and Sparrow (1982), and then investigated the effects of this variation. They showed that 45 % increase in spacing resulted in 30 % rise in the Nusselt number, Nu of fully-developed, but the friction factor, f increased more than doubled rate. An extensive experimental study analyzing turbulent flow in the entrance region for fully developed flow in periodically converging-diverging tubes was performed by Mendes and Sparrow (1984). They varied geometrical parameters and the Reynolds number, Re . They obtained significant heat transfer enhancement accompanied by large pressure drop for converging-diverging channel over the straight channel. Oyakawa et al. (1989) conducted an experimental study to investigate the effects of the height, H of a wavy sinusoidal passage on fluid flow and heat transfer. They obtained an optimum spacing for fully-developed flow using the chosen wall geometry. Nishimura et al. (1990) investigated flow characteristics and mass transfer in two dimensional symmetrical sinusoidal and arc-shaped wavy-walled channels at Reynolds numbers,

Re ranging from 20 to 300. Since earlier transition to turbulence occurs for arc-shaped wall as compared with the sinusoidal wall, the arc-shaped wall has larger mass transfer rate than the sinusoidal wall. Ali and Ramadhyani (1992) performed experiments in order to analyze the developing flow region of triangular corrugated channels. Experiments were carried out for two different interwall spacing in both laminar and transitional regions. They observed a transitional to unsteady flow around $Re = 500$ for both spacing, accompanied by a sharp increase in Nusselt number, Nu. Synder et al. (1993) conducted an experimental study of thermally fully-developed region of serpentine channel to measure the heat transfer rates and pressure drops. Experiments were carried out for Reynolds numbers, Re ranging from 250 to 10000. The purpose of the choosing the channel design was minimizing the extent of flow separation. They found that the heat transfer in the serpentine channel was as much as 9 times greater than in a straight parallel-plate channel for equal Reynolds number. Saniei and Dini (1993) investigated the effect of height and geometry on local heat transfer and pressure drop in a channel with corrugated walls. In that study the bottom wall of the rectangular channel was corrugated and opposite top wall took one of the three different constructions such as: (i) plane; (ii) corrugations making an in-phase arrangement with the bottom wall and; (iii) corrugations making an out-of-phase arrangement with the bottom wall. Reynolds numbers, Re for these measurements ranged from 22000 to 52000 in the turbulent region for three different aspect ratios. They showed that local heat transfer increases with the higher Reynolds number, Re and higher aspect ratio. Heat transfer results were almost the same for in-phase and out-of-phase arrangement but pressure drop was higher for the out-of-phase arrangement. Nishimura and Kojima (1995) experimentally investigated mass transfer characteristics in a sinusoidal wavy-walled channel for pulsatile flow with different flow parameters. They varied net flow, amplitude and frequency of fluid oscillation. It is stated that combination of fluid oscillation and flow separation results in significant mass transfer rate enhancement under laminar flow conditions. Gschwind et al. (1995) conducted flow visualization and mass transfer experiments with air in sinusoidal wavy duct. They investigated both effects of height of the channel and Reynolds number, Re. In addition to

showing all instabilities, they also provide stability diagrams describing the longitudinal vortices range. Lorenz et al. (1995) investigated heat transfer and pressure drop distributions along the wall inside an asymmetrical grooved channel for thermally developed and periodic turbulent flow for Reynolds numbers in the range of $10^4 < Re < 10^5$. Enhancement in Nusselt number, Nu at the grooved wall is achieved 1.52 to 1.75 times compared to a plane channel. In addition, it is found that thermal periodicity occurs at fifth heated period. Choi et al. (1996) made experiments with the naphthalene sublimation technique to study heat transfer analogy in a two-dimensional serpentine channel. In order to examine the development of the flow near the entrance of the channel, they determined the variation of the regionally averaged transfer coefficient, h the distribution of the local heat transfer coefficient, h and static pressure, p distribution. Heat transfer and pressure drop were investigated experimentally by Wang et al. (1997) for typical wavy fin-and-tube heat exchangers. They tested 18 samples of wavy fin-and-tube heat exchangers with different geometrical parameters. Plots of the Fanning friction factor, f and the Colburn factor, j against Reynolds numbers, Re ranging from 400 to 8000 was presented. Their results showed that tube row has negligible effect on the friction factor, f and the effect of fin pitch on the Colburn factor, j is negligible. Russ and Beer (1997) performed experiments to study mass transfer and flow characteristics in pipe with sinusoidal wavy walls for Reynolds numbers ranging from 400 to 2000. The local mass transfer was measured along the axis of the pipe with the naphthalene sublimation method. In addition, they applied smoke technique for flow visualization experiments. Rush et al. (1999) studied the fluid flow and local heat transfer in sinusoidal wavy passages for laminar and transitional flows. Phase angle, ϕ , wave amplitude, a and distance between plates, H were varied. They showed that instabilities were mostly near the exit of the channel at the low Reynolds number, Re , but these instabilities moved to inlet as the Reynolds number, Re was increased. Kim (2001) performed experiments to investigate flow characteristics in two dimensional wavy channel by PIV. The purpose of the study was to analyze the onset position of the transition and the global mixing. In addition, he produced experimental data for developing and fully developed flow to understand the

relationship between flow characteristics and heat transfer enhancement with low pumping power. Zimparov (2001) conducted experiments to study heat transfer and isothermal frictional pressure drop in corrugated tubes combined with twisted tape inserts for Reynolds numbers, Re ranging between 3000 and 60000. Enhancement in heat transfer coefficients can be achieved with an increase of friction factor, f compared to smooth tube. In another experimental study conducted by Zimparov (2002), the effects of corrugation and twisted tape inserts geometry on heat transfer and pressure drop were investigated. It was concluded that considerable heat transfer enhancement was obtained as the relative peach H/D_i decreases at a certain roughness value of $e/D_i=0.044$. Herman and Kang (2002) investigated the heat transfer enhancement in a grooved channel with curved vanes. Their experimental results reveal that heat transfer and pressure drop are 1.5-3.5 and 3-5 times higher, respectively, compared to the basic grooved channel. Ali and Hanaoka (2002) conducted experiments to investigate the effects of the operating parameters on laminar flow forced convective heat transfer for air flowing in a channel having V-shape corrugated upper wall heated by radiation heat flux. Their results showed that effects of Reynolds number, Re on local Nusselt number, Nu are more significant at the channel entrance region. In addition, the local Nusselt number, Nu increases with increase in tilting angles of the channel and incident radiation fluxes. Zimmerer et al. (2002) discussed the effect of different geometrical parameters of corrugated walls on the local and integral mass transfer. They observed that the flow phenomena caused by the different geometries affected homogeneity, quantity of the local heat and mass transfer and pressure drop. Islamoglu and Parmaksizoglu (2003) investigated the effects of channel height, H on the heat transfer enhancement of heat exchanger with corrugated walls. They measured forced convection heat transfer coefficients, h and friction factor, f in corrugated channel for two different values of channel height 5 and 10 mm while keeping corrugation angle fixed. Their results indicated that both fully developed Nusselt number, Nu and friction factor, f increase with channel height, H . Nishimura et al. (2003) conducted an experimental study on the flow and mass transfer in a sinusoidal wavy tube for steady flow in Reynolds numbers, Re ranging from 50 to 1000. The flow transition from steady to unsteady

occurs when Reynolds numbers, Re are in the range between 160 and 200. In addition, the Reynolds number, Re affects mass transfer and the wall shear stress at minimum cross section of the tube. Vicente et al. (2004) studied laminar and turbulent flow conditions in corrugated tubes experimentally. Their study revealed that corrugated tubes increase heat transfer enhancement up to 30 % at a high Rayleigh number, Ra . On the other hand, friction factor, f values of corrugated tubes are 5 to 30% higher than those of smooth tubes. Gradeck et al. (2005) conducted experiments to study effects of hydrodynamic conditions on the heat transfer enhancement for single phase flow inside corrugated channels. Local temperature distribution at the wall and its relation with general structure of fluid flow was found. They concluded that heat transfer in the corrugated channels is always higher than those in straight channels as expected. Nilpueng and Wongwises (2006) studied flow characteristics and pressure loss of gas-liquid flows in vertical sinusoidal wavy channels with different phase angles between the side walls of 0° , 90° and 180° . Flow structure and pressure drop are strongly affected from the phase shifts. Bubbly, slug, churn, and dispersed bubbly flows are observed and recorded during the experiments. As the phase shifts are increased, bubbly and dispersed bubbly flow regions become larger. In addition, slug flow do not appear when the phase shift is 180° . Heat transfer and friction characteristics of high aspect ratio rectangular duct using rib-grooved artificial roughness was studied by Jaurker et al. (2006). The Reynolds number, Re , relative roughness height, relative roughness pitch, and groove position to pitch ratio were varied during the experiments. The effects of these parameters on heat transfer and friction factor, f were discussed. Their results showed that the heat transfer coefficient, h and friction factor, f for rib-grooved duct is higher than that for the transverse ribs. Rozzi et al. (2007) studied convective heat transfer and pressure loss enhancement in helically corrugated wall tubes for both Newtonian and non-newtonian fluids. In addition, comparison between smooth and helically corrugated tubes were presented. They concluded that helically corrugated tubes are significantly effective in enhancing the overall heat transfer for generalized Reynolds numbers, Re ranging from about 800 to the limit of the transitional flow regime. On the other hand, negligible enhancement is obtained for the low

generalized Reynolds number, Re . They obtained moderate heat transfer enhancement accompanied by high pressure loss in the fully developed regime. A comprehensive experimental study related to heat transfer and pressure drop in the corrugated channel was carried out by Naphon (2007a). Two opposite corrugated plates were arranged as an in phase configuration. They varied corrugated tile angle, heat flux and the Reynolds number to investigate the effects of these parameters on heat transfer enhancement and pressure drop. Since recirculation zones occur, the corrugated surface enhances heat transfer and pressure drop compared to smooth surface. Another study investigating heat transfer and pressure drop in corrugated channel with out of phase arrangement was made Naphon (2007b). Their results revealed that heat transfer enhancement was achieved with corrugated surfaces. On the other hand, the pressure drop also increases. Jin et al. (2007) experimentally investigated the heat transfer enhancement by pulsating flow in a triangular grooved channel for the range of Reynolds numbers, $270 \leq Re \leq 910$ and Strouhal numbers, $0.08 \leq St \leq 0.67$. It was measured that the heat transfer improved up to 350% compared with the steady flow case. The heat transfer enhancement was found to increase as the Reynolds number, Re decreases. It was also found that the optimal Strouhal number increases as the Reynolds number, Re decreases. The PIV results show that the heat transfer enhancement results from the strong fluid mixing caused by the repeating sequence of vortex generation, growth, expansion and ejection from the groove to the main stream by the pulsating flow. The fluid mixing enhancement is maximized when the pulsation period matches with the time duration for the vortex to grow large enough to fill the groove and then to be ejected to the main stream. Lin et al. (2007) obtained dimensionless correlations using experimental data and the Buckingham Pi theorem to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger. According to the dimensionless analysis, Nusselt number, Nu is determined primarily by Re , R/D_h , x/D_h and β . Here R is Radius of curvature, D_h is hydraulic diameter, and β is corrugated angle. Hasanuzzaman (2007) made an experimental study to investigate the natural convective heat transfer and the effect of mass flow rate of cooling water, input heat energy and inlet water temperature on heat transfer in the square enclosure of two V-

corrugated parallel vertical plates. The heat transfer rate increases with increase in mass flow rate of water and the input heat energy, and decrease in inlet water temperature. Tolentino et al. (2008) conducted experiments to visualize the fluid flow in the entrance region of sinusoidal channel. The distance between upper and lower plates, H , phase angle, ϕ and the Reynolds number, Re impacts on the flow characteristics were analyzed. Steady, unsteady, and significantly mixed flows were observed during the experiments. The instabilities of the flow move from the exit region of the channel to entrance region with increasing the Reynolds number. It was also concluded that the flow shows similar characteristics after eight waves. Wen and Ho (2009) observed the fluid flow characteristics and discussed the heat transfer enhancement of air flow in fin-and-tube heat exchanger for three different shapes of fins. Heat transfer enhancement improves for wavy and compound fins compared to plate fin. However, friction factors for wavy and compound fins are also higher than that of plate fin. They conclude that compound fins are the most effective for heat exchangers. Durmus et al. (2009) experimentally studied effects of surface geometries of heat exchangers on heat transfer, friction factor, f and exergy loss under laminar flow condition. With obtained results they presented correlations for heat transfer, friction factor, f and exergy loss. Bilen et al. (2009) conducted an experimental study to investigate the heat transfer and friction characteristics of a fully developed turbulent flow for Reynolds numbers, Re ranging from 10000 to 38000 for circular, trapezoidal and rectangular grooved tubes. Among all cases, the maximum enhancement in heat transfer occurs 63% for circular groove, 58% for trapezoidal groove and 47% for rectangular groove compared to smooth tube at the Reynolds number, $Re=38000$. In addition, optimum value of entropy generation for all shapes was observed at about the Reynolds number, Re of 17000. Elshafei et al. (2010) discussed the effects of channel spacing and phase shift variations on heat transfer and pressure drop. Relatively to parallel plates, the average heat transfer coefficient, h and pressure drop increased by a factor of 2.6 to 3.2 and 1.9 to 2.6, respectively. They concluded that best geometrical configuration of corrugated channel has spacing ratio, $\varepsilon \leq 3.0$ and phase shift, $\phi \leq 90^\circ$ according to the flow area goodness factor. Another study investigating heat transfer and pressure drop was

made by Laohalertdecha and Wongwises (2010). The effects of corrugation pitch on heat transfer and pressure drop were discussed and the results obtained from corrugated tubes were compared with those of smooth tubes. It was concluded that corrugation pitches affect heat transfer coefficient, h and pressure drop increase. Kim et al. (2010) performed an experimental study on corrugated cross-flow air-cooled plate heat exchangers. (PHEs) with single-wave plates and double-wave plates were tested and compared. For double-wave PHEs, they obtained 50% heat transfer enhancement compared to single-wave PHEs. However, double-wave PHEs has 30% pressure drop penalty. Pethkool et al. (2011) studied convective heat transfer enhancement in helically corrugated tubes and discussed the effects of pitch-to-diameter ratio, P/D and rib-height to diameter ratio, e/D on heat transfer augmentation, isothermal friction, f and thermal performance factor (TPF). According to the experimental results, heat transfer rate of corrugated tube is in a range between 1.23 and 2.32 times over the smooth tube, depending on the rib height/pitch ratios, P/D and the Reynolds number, Re . They also found that the average friction factor, f of the corrugated tube increases between 146% and 193% compared to those of the smooth tube. Faizal and Ahmed (2012) experimentally studied the heat transfer in corrugated plate heat exchanger by varying the distance between plates to determine the distance giving the optimum thermal performance. They concluded that increase in flow rate increases pressure drop and the optimum heat transfer is observed for minimum distance of $\Delta X = 6$ mm. Chang and Huang (2012) conducted experiments to analyze the heat transfer, pressure drop and thermal performance factor (TPF) of sinusoidal wall channel with 45° ribs. The Reynolds number, Re was varied from 800 to 9000 during the experiments. Obtained results were compared with those of other passive heat transfer enhancement devices. Pehlivan et al. (2013) performed experiments to study heat transfer rate for corrugated channel with different type sharp corrugation peak fins and a plain surface. The effects of the wavy geometry and channel height on Nusselt number (Nu), convection heat transfer coefficient (h), Colburn factor (j) and enhancement ratio (E) for various Reynolds numbers (Re) were discussed. Their results revealed that heat transfer rate enhances with corrugated angle, θ .

2.2. Numerical Studies of Flow in Corrugated Channels

Asako and Faghri (1987) used a finite volume method to study laminar flow and heat transfer in a corrugated duct. Results were found for laminar flow, for Reynolds numbers, Re ranging from 90 to 1000 under uniform wall temperature. Prandtl numbers, Pr corrugation angle, and dimensionless interwall spacing were also varied. They concluded that pressure drop and friction factors f are higher than the corresponding values for a straight channel. Asako, Nakamura and Faghri (1988) made a numerical study to determine the heat transfer and pressure drop in a corrugated duct with a rounded corners. Reynolds numbers, Re for these computations ranging from 100 to 1000 and the Prandtl number, Pr of 0.7 were used. They showed that friction factor, f and the Nusselt number, Nu in corrugated duct with rounded corners were lower than those in corrugated duct with sharp corners. However, heat transfer enhancement of rounded corners duct is greater than that of straight duct. Another numerical study conducted by Garg and Maji (1988) investigated the heat transfer of sinusoidal wavy channels with phase shift of $\theta=0^\circ$. It is obtained that the local Nusselt number, Nu is proportional to the Reynolds number, Re and changes sinusoidally in the flow direction. Wang and Vanka (1995) performed numerical investigations for periodic wavy passages and found that the flow was steady until a critical Reynolds number, $Re=180$. Above $Re=180$ self-sustained oscillatory was observed. In steady flow regime average Nusselt numbers, $\overline{Nu}$ were slightly larger than those for parallel flat plate however in transitional flow regime heat transfer enhancement rate was about 2.5, and the friction factor, f were about twice that of parallel flat plate. Tanda and Vittori (1996) presented a numerical study for the laminar fully-developed flow and heat transfer in a two-dimensional channel. In that study, effects of the channel geometry, Reynolds and Prandtl numbers, Re and Pr , on the flow field and heat transfer was investigated. They showed that wall-to-wall spacing and wave height had significantly effects on streamlines contours and heat transfer coefficients. Pressure drop of the wavy channel was always greater than that of straight channel for the same flow rate and heat transfer conditions. A comprehensive numerical investigation, performing by

Russ and Beer (1997), discussed the fluid flow and heat transfer in sinusoidally varying diameter pipe for a wide range of Reynolds numbers, Re . It is shown that Nusselt and Sherwood numbers, Nu and Sh , in turbulent flow is proportional to the wall function gradient. Sawyers et al. (1998) studied the effects of three dimensional steady flow characteristics on heat transfer enhancement in sinusoidal corrugated channels employing a combination of analytical and numerical methods. The Reynolds number, Re was varied between 0 and 250 to keep the flow in steady regime. Since recirculation occurs in two-dimensional corrugations, heat transfer enhances with compared to the flat plates. In three dimensional case, the additional heat transfer enhancement occurs because of the adventive transport across the boundary of the recirculation region. However, as the transverse flow becomes stronger, the heat transfer rate decreases. Stone and Vanka (1999) investigated the developing flow and heat transfer in a sinusoidal wavy channel consisting of 14 waves. Their numerical simulations were performed for several Reynolds numbers, Re . The flow is steady in the entire channel at low Reynolds numbers, Re and the flow becomes unsteady as the Reynolds number, Re is increased. Since the mixing of the core and wall fluids increases with unsteadiness, the heat transfer enhancement improves. Fabbri (2000) made a study of heat transfer under laminar flow condition for both smooth and corrugated wall channels. The heat transfer performance of the corrugated wall channel is compared with that of a smooth wall and genetic algorithm program is written in order to maximize the heat transfer by optimizing the corrugated plate profile. Yang (2000) employed Large Eddy Simulation (LES) method to observe the turbulence flows in periodically grooved channel. The relations between the length and depth of a groove and flow characteristics was studied. As the groove length decreases, the recirculation of flow in the groove becomes weaker so heat transfer enhancement and pressure drop decreases. Niceno and Nobile (2001) numerically studied the fluid flow and heat transfer in periodic wavy channels. Sinusoidal and arc-shaped wavy channels were used for a fluid with a Prandtl number of 0.7. They showed that wavy channels have a little or no effects on heat transfer enhancement at lower values of Reynolds numbers, Re . Heat transfer rate increases for both sinusoidal and arc-shaped wavy channels when the flow

becomes unsteady. Ergin et al. (2001) performed numerical simulations for obtaining periodic fully developed flow characteristics in corrugated channel. Their simulations were performed by varying the ratio of the corrugation height to the interval spacing from 0.45 to 2 and corrugation angles of 30° and 45°. In addition, they conducted experiments for pressure drop, velocity, and turbulence measurements. Wang and Chen (2002) studied the effects of wavy wall geometry, the Reynolds number, Re and Prandtl number, Pr on the skin friction, C_f and the Nusselt number, Nu . The study showed that corrugated walls had a considerable effect on heat transfer rate when the amplitude-wavelength ratio was high, particularly at higher Reynolds numbers, Re . Mahmud et al. (2002) numerically investigated separation characteristics of fluid flow inside two parallel wavy plates for steady-laminar flow. The simulations carried out for surface waviness 0-0.3 and aspect ratio 1.5-2.25. They presented effect of the waviness, a/H and aspect ratio, w/H on separation characteristics of fluid flow. The appearance of first separation of fluid flow was identified by using the critical Reynolds number, Re . They showed that the structure of separation bubbles depends strongly on waviness, a/H of the surface and aspect ratio, w/H for a particular Reynolds number, Re . In another numerical study conducted by Mahmud et al. (2003), effects of surface waviness, a/H on flow structure and heat transfer enhancement of laminar steady flow in sinusoidal wavy wall pipe were studied. When the surface waviness, a/H is higher, flow separation occurs earlier which causes larger heat transfer rate and pressure drop. Comini et al. (2003) conducted a numerical study to investigate the effects of aspect ratio on heat transfer and pressure drop in the three-dimensional wavy channels by varying the aspect ratio from one to infinity. They observed that heat transfer enhancement and pressure drop increase with decreasing aspect ratio. The overall performances also improve. Laminar forced convection in two dimensional wavy plate fin cores for Reynolds numbers ranging from 10 to 1000 was simulated by Zhang et al. (2004) to analyze effects of fin waviness and spacing on the flow structure and heat transfer. Their results indicated that overall heat transfer coefficient, h enhances significantly compared to that of straight channel with pressure drop penalty. Metwally and Manglik (2004) performed numerical

simulations to analyze the effects of corrugation aspect ratio, Reynolds number, Re , and Prandtl number, Pr on heat transfer augmentation and flow structure in sinusoidal corrugated-plate channels. Corrugations create transverse vortices in trough regions and recirculation increases with aspect ratio and the Reynolds number, Re . Heat transfer increases with less pressure drop penalty. Three-dimensional simulations were performed by Savino et al. (2004) to study heat transfer and pressure drop in corrugated channels that are nominally two-dimensional. Their results show that three-dimensionality has an important impact on overall friction factor, f and Nusselt number, Nu . Hossain and Sadrul Islam (2004) numerically investigated fully developed unsteady fluid flow and heat transfer in two-dimensional sinusoidal wavy channels. The effects of variations of height, H , amplitude, a and wavelength, L_w were studied. Their results showed that friction factor, f and heat transfer increases with increase in amplitude, a and decrease in height, H . On the other hand, variation of wavelength, L_w has minimal effect. Fabri and Rossi (2005) analyzed heat transfer in the entrance region of channel composed of a flat and a corrugated profile walls. Simulations were performed for various Reynolds numbers, Re and corrugated profile amplitude, a and their results were compared with heat transfer coefficient, h in the entrance region of a flat plate. Bahaidarah et al. (2005) numerically studied two dimensional steady developing flow and heat transfer in sinusoidal and arc-shaped wavy channels and compared with the results of flow in parallel-plate channel. They showed that these two configurations have a little heat transfer enhancement compared to parallel-plate channel at the low Reynolds number, Re . However, heat transfer enhancement ratios, Nu/Nu_0 increase up to 80% at the higher Reynolds number. Xie et al. (2007) conducted numerical study on flow characteristics and heat transfer of the periodically fully developed flow in two-dimensional wavy channel. Effects of geometrical parameters and the Reynolds number, Re on flow and heat transfer were discussed. According to simulation results, increase in Reynolds numbers, Re increases overall Nusselt number, Nu and friction factors, f . They concluded that wavy surface strongly enhances heat transfer. Bahaidarah (2007) presented a numerical study for two dimensional steady developing flow and heat transfer in

sinusoidal and arc-shaped wavy channels with staggered walls and compared the results of flow in the corresponding channel with nonstaggered walls. Rosaguti et al. (2007) numerically studied the heat transfer enhancement in sinusoidal channels with circular and semi-circular cross-sections for the low Reynolds number, Re . In that study effect of the Reynolds number, Re and amplitude to half wavelength ratio on heat transfer enhancement and pressure drop was investigated. Eiamsa-ard and Promvonge (2008) simulated turbulent forced convection in a two-dimensional channel with periodic transverse grooves on the lower channel wall for Reynolds numbers, Re ranging from 6000 to 18000. It is found that the grooved channel increases heat transfer up to 158% over the smooth channel Guzman et al. (2009) numerically investigated the enhancement characteristics of heat transfer due to flow bifurcations in asymmetric wavy wall channels in transition scenarios from laminar to transitional time-dependent flow regimes. Two-dimensional direct numerical simulations (DNS), of the time-dependent, incompressible continuity, Navier Stokes, and energy equations were carried out for three aspect ratios and two Prandtl numbers. They concluded that significant heat transfer enhancements were obtained when the asymmetric wavy channel was operated in the appropriate transitional Reynolds number, Re range. Naphon (2009) investigated the heat transfer and flow distributions in the wavy plate channels with various geometry configurations. In that study he considered the effects of geometry configurations and arrangement of wavy plates, air flow rates on the temperature and flow developments under constant heat flux conditions. It was seen from this study that the sharp edge of the wavy plate has significant effect on the flow structure and heat transfer enhancement. Bahaidarah (2009) numerically studied the two-dimensional steady developing fluid flow and heat transfer through a periodic, corrugated triangular channel with and without horizontal pitch for a fluid with the Prandtl number, Pr of 0.7. They solved the governing equations using finite volume method. In that study, they reported average Nusselt number, Nu and normalized pressure drop. Zhang and Che (2011) performed three-dimensional simulations to analyze the effects of Reynolds number, Re and corrugation profile on the heat transfer and pressure drop of the flow in cross-corrugated plates. Thermalhydraulic performance of the cross-corrugated plates with

sinusoidal, trapezoidal, rectangular, isosceles triangular, and elliptic corrugations were compared. The Nusselt number, Nu and friction factor, f of the trapezoidal channel are 1-4 times higher than those of elliptic channel. Yin et al. (2012) numerically investigated the effect of phase shift, θ on flow characteristics and heat transfer in corrugated wall channels. The computations was performed on uniform wall temperature for air while phase shift and Reynolds Number, Re vary. They showed that corrugated wall channels have a significant effect on heat transfer enhancement and phase shift, θ is more effective in higher Re region than lower Re region. Sui et al. (2012) performed numerical simulations of fully developed flow and heat transfer in rectangular cross section wavy channels. Their results showed that corrugation enhances heat transfer and pressure drop due to the better mixing of the fluid. Also it is calculated that heat transfer enhancement is much higher than pressure drop penalty of wavy channels. Since wavy channels improve efficiency and compactness, they are suggested to be used in heat transfer devices. Desrues et al. (2012) investigated the pressure drop and heat transfer enhancement of ribbed channel for several Reynolds numbers. They found that heat transfer enhances only when Re is greater than a critical value, while pressure drop increases monotonically with Re . Rahimi-Esbo et al. (2012) simulated two-dimensional flow in sinusoidal converging jet in turbulence flow regime. They presented influence of the Reynolds number, Re , channel aspect ratio, and amplitude of wave of the channel on heat transfer enhancement. Their results showed that the average Nusselt number, Nu increases with increase in the Reynolds number, Re , aspect ratio and amplitude, a . Ramgadia and Saha (2012) conducted a numerical study on wave amplitude, a and the Reynolds number, Re effects on heat transfer characteristics. It was concluded that $H_{min}/H_{max}=0.2$ exhibited maximum thermal performance at a critical Reynolds number, Re of 600. In addition, thermal performance factor decreased with an increase in the Reynolds number when the flow was steady; however, opposite scheme was observed when the flow was unsteady. Ramgadia and Saha (2013) simulated two-dimensional fully developed flow in the sinusoidal wavy channel to investigate the effects of streamwise domain length and the Reynolds number on flow characteristics and heat transfer enhancement. The Reynolds number was varied

in the range of 25-1000. They obtained that streamwise domain length does not influence the fluid flow and heat transfer. A little heat transfer rate enhancement was observed for steady flow. On the other hand, heat transfer rate increases significantly due to the improvement in mixing between core and near wall fluids. Ramgadia and Saha (2013) presented also three-dimensional numerical study for analyzing fluid flow and heat transfer in the wavy wall channel using the Large Eddy Simulation (LES). They varied Reynolds numbers, Re between 2053 and 10000 while keeping geometrical parameters constant. Numerical results showed that the Reynolds number, Re affects normalized heat transfer ratio. However, the effects of Reynolds number, Re on friction factor, f can be negligible. Mohammed et.al (2013) investigated the effects of geometrical parameters of corrugated tilt angles, wavy heights and channel heights on flow and heat transfer. In that study, optimum geometrical parameters are found and they are very effective on heat transfer enhancement. In addition, using wavy channel improves compactness and thermal performance of heat exchangers. Errico and Stalio (2014) simulated turbulent forced convection in a channel with upper flat wall and lower sinusoidal wavy wall. They arranged three Prandtl number, $Pr=0.025, 0.20, 0.71$ while keeping the Reynolds number, Re fixed at 18900. Flow and heat transfer in half corrugated channel was also studied by Mirzaei et al. (2014). They simulated the flow using Large Eddy Simulation (LES) model at the Reynolds number, $Re=10000$ for wave amplitudes between 0 and 0.15. Their results revealed that increasing wave amplitude, a increases the Nusselt number, Nu until a specific value. However, the Nusselt number, Nu remains constant with further increase in wavy amplitude, a . 0.1 was found to be optimum value of wave amplitude, a by considering heat transfer enhancement and pressure drop.

2.3. Both Experimental and Numerical Studies of Flow in Corrugated Channels

Flow and heat transfer enhancement in crossed corrugated passages were investigated experimentally by Stasiek et al. (1996) and numerically by Ciofalo et al. (1996). They analyzed the impacts of corrugation angle, θ , geometry and the

Reynolds number, Re on heat transfer enhancement to define optimum design parameter. It is found that the average Nusselt number, $\overline{Nu}$ increases with the Reynolds number, Re . On the other hand, the friction coefficient, f decreases as the Reynolds number, Re increases. Although the corrugation angle, θ affects both Nusselt number and friction coefficient, f it has stronger impact on the friction coefficient, f compared with the Nusselt number, Nu . Nishimura and Matsune (1996) carried out experiments to analyze the mass transfer enhancement of laminar pulsatile flow in sinusoidal wavy channels with phase angle of 180° and 0° . They also visualized the dynamics of vortices both experimentally and numerically. Mass transfer enhancement factor of 0° phase angle channel was found to be greater than that of the channel with phase angle, ϕ of 180° due to the specific fluid mixing caused by vortices dynamics. In another study of Nishimura and Matsune (1998), numerical simulations of flow structure in sinusoidal wavy channels with phase angle of 180° and 0° for pulsatile flow was performed. They also visualized the flow to confirm the validity of the numerical simulations. They found that the flow structure strongly depends channel geometry. Islamoglu and Parmaksizoglu (2004) numerically and experimentally studied the convective heat transfer and pressure drop in corrugated heat exchanger channels. In that study all the walls were in phase arrangement and the corrugation angle were 20° . In experimental and numerical study conducted by Kwon et al. (2008), effects of corrugation angle and the Reynolds number, Re on two dimensional flow structures and heat/mass transfer characteristics in rectangular corrugated channel were studied. Flow transitions to turbulent is observed earlier as the corrugation angle decreases. In addition, maximum performance factor was calculated as 1.8 at the Reynolds number, Re of 1000 and corrugation angle, θ of 100° . Han et al. (2010) studied flow and heat transfer enhancement in chevron corrugated-plate heat exchanger. Three dimensional temperature, pressure, and velocity fields obtained from numerical simulations were compared with experimental results. It is observed that the temperature gradient in the inlet and outlet sections are very small. Arsenyeva et al. (2011) investigated new corrugation pattern for the Plate Heat Exchangers (PHEs) as condenser. A channel with variable corrugation along the plate length was used in experiments. Confirmed

mathematical model was used to calculate parameters and condenser performance. Their principle for designing plate condenser with variable cross section area can reduce pressure loss up to 40% while maintaining high heat transfer enhancement.

2.4. Experimental and Numerical Studies of Nanofluid Flow in Corrugated Channels

Heidary and Kermani (2010) numerically studied heat transfer enhancement and fluid flow in a sinusoidal wall channel with copper-water nano-fluid. Effects of nano-fluid volume fraction, the Reynolds number, Re , wave amplitude, a on local and average Nusselt numbers, Nu and $\overline{Nu}$ and skin friction coefficient, C_f were investigated. Results showed integrating wavy wall and nanoparticle within base fluid can increase heat transfer by 50%. Ahmed et al. (2011) studied heat transfer enhancement and pressure drop due to flow of copper-water nano-fluid through corrugated channels. In that study, numerical simulations were generated for Reynolds numbers, Re ranging from 100 to 1000 with a nanoparticle volume fraction of 0-0.05. Their findings showed that increase in the Reynolds number, Re and nanoparticle volume fraction increase heat transfer enhancement with a slight increase in pressure drop. In another study conducted by Ahmed et al. (2012), effects of a wide array of parameters including Reynolds number, Re , nanoparticle volume fraction, wavelength, L_w and amplitude, a of the sinusoidal wavy channel were studied and their impact on the heat transfer enhancement, local and average Nusselt number, Nu and $\overline{Nu}$ and local skin-friction coefficient, C_f were determined. Significant raise in Nusselt number, Nu and slight increase in friction coefficient, f were observed as the nanoparticle volume fraction increased. As the amplitude of the wavy wall increases, Nusselt number, Nu and the friction coefficient, f increase. Their study also elicited that the Reynolds number, Re , nanoparticle volume fraction and wavy wall amplitude, a greatly inspire heat transfer enhancement in contrast to wavelength, L_w demonstrating negligible effect. Pandey and Nema (2012) conducted experiments to investigate the heat transfer enhancement and pressure loss of nanofluid flow in corrugated plate heat exchanger. They also obtained empirical correlations for mean

Nusselt number, $\overline{Nu}$ and friction factor, f using experimental data. Ahmed et al. (2013) also investigated the laminar forced convective heat transfer in trapezoidal corrugated channel using nanofluid. For various Reynolds number and nanoparticle volume fractions, the influences of wave length, L_w and amplitude, a of corrugated channel on pressure loss and heat transfer enhancement were analyzed. It is seen that increase in nanoparticle volume fraction and the amplitude, a of corrugated channel enhances Nusselt number, Nu with pressure drop penalty. Moreover, to become wavelength, L_w shorter increases the average Nusselt number, $\overline{Nu}$ and decreases pressure loss. Lastly, Ahmed et al. (2014) studied phase shift, ϕ impacts on thermalhydraulic performance of the sinusoidal wavy channel. Four different phase shifts, ϕ of 0° , 45° , 90° and 180° were considered in their simulations. Reynolds numbers, Re ranging from 100 to 800 and nanoparticle volume fraction between 0 and 5% were considered. It is concluded that phase shift, ϕ of 0° has optimum thermalhydraulic performance over the ranges of Reynolds number, Re and nanoparticles volume fractions. Aliabadi (2014) studied the heat transfer and friction factor, f of nanofluid in two dimensional sinusoidal corrugated channel for different geometrical parameters such as channel height, H , channel length, L , wave length, L_w , wave amplitude, a , and phase shift, ϕ . Reynolds number, Re and nanoparticle volume fraction effects were also analyzed. Nusselt number, Re and Darcy friction factor, f were considered as performance factor. Channel height, H and wave amplitude, a has strongest effects on Nusselt number, Nu and friction factor, f , respectively. Using nanofluids in sinusoidal corrugated channel enhances Nusselt number, Nu considerably. On the other hand, almost the same pressure loss values were calculated for nanofluid and water flow. Akdag et al. (2014) simulated heat transfer enhancement of two dimensional laminar pulsating nanofluid flow in sinusoidal wavy channel. They found that considerable heat transfer enhancement can be achieved with increasing nanoparticle volume fraction and the amplitude, a of pulsatile flow. Abed et al. (2015) studied flow characteristics and heat transfer augmentation of nanofluid in trapezoidal corrugated channel. Influences of geometrical parameters such as wavy amplitude, a , longitudinal pitch, P on the performance of the trapezoidal corrugated channel were analyzed for designing

optimum plate geometry. They also examine the effects of Reynolds number, Re , nanofluid type, and nanoparticle volume fraction and particle diameter, d . It is concluded that using nanofluids in corrugated channel brings additional enhancement in heat transfer that makes heat exchangers more compact. Abed et al. (2015) also investigated the heat transfer enhancement and pressure drop characteristics of nanofluid flow in channel with V-shaped wavy lower plate. Numerical simulations were performed for various type of nanofluid with various volume fractions in the range of 0-4% over a range of Reynolds numbers, Re 8000-20000. The average Nusselt number, $\overline{Nu}$ increases with adding nanoparticle to the base fluid. An increase in volumetric concentration improves heat transfer enhancement with pressure loss penalty. In addition, the average Nusselt number, $\overline{Nu}$ increases as the Reynolds number, Re increases.

3. MATERIAL and METHOD

3.1. Experimental Study

3.1.1. Particle Image Velocimetry (PIV)

Particle Image Velocimetry (PIV) is a non-intrusive laser optical measurement technique for research and industry to investigate the flow. In this technique, the flow field is seeded with some tracer particles and two rapid images of investigation region are obtained with charged coupled device (CCD) camera while periodically illuminating that region by a high power laser light source. A velocity vector map can be calculated in the flow field since the displacement of tracer particles and the time interval between image captures is known. The PIV technique has advantages over traditional techniques using measurement probes. PIV do not require any measurement probes in the flow medium affecting the overall flow. In addition, PIV gives information about entire flow field.

Adrian (1988) introduced the theory of PIV in the late 1980s. Keane et al. (1990 and 1991) were the first to study experimental implementations. At that time, a single photographic frame was multiply exposed and analyzed using an auto-correlation technique because of hardware limitations. Improvements in recording speed could allow to capture images on separate frames for analysis by cross-correlation (Kean et al. 1992). Particle images could directly recorded with applying the digital camera technology in PIV technique (Willert et al. 1991), at the expense of reduced resolution, resulting digital PIV development (Westerweel, 1997). Accuracy and speed of PIV analysis have been increased with improvement in cameras, lasers, optics and software.

3.1.1.1. Principles of PIV Technique

The PIV technique mainly calculate the flow velocity by measuring the distance travelled by particles in the flow field within a known time period. It is assumed that tracer particles move with flow velocity. These special particles are named as seeding particles. Seeding particles are chosen depending on the flow

nature. Investigated plane is illuminated by pulsating light sheets and the movement of the particles is recorded with high performance digital cameras. Therefore, the movement of seeding particle can be detected by the CCD camera when the laser light sheets illuminate the flow field. Particle positions can be seen as light specks on a dark background on each camera frame. Particle position at the instant of laser light pulse number 1 is recorded on camera frame 1 while particle position at the instant of pulse number 2 is recorded on the frame 2 by synchronizing the camera and laser light sheet.

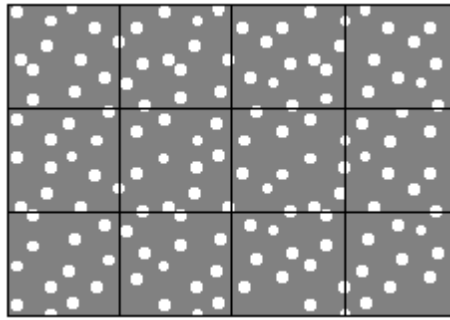


Figure 3.1. Interrogation areas

The digital PIV records are divided into small square subareas called “interrogation areas”. An average particle displacement vector is obtained by correlation of images from the first and second pulse of the light sheet for each interrogation areas. It is assumed that all particles within one interrogation area move homogeneously between two pulses. The displacement vectors can be converted into a raw velocity vectors by dividing the known time interval between two images.

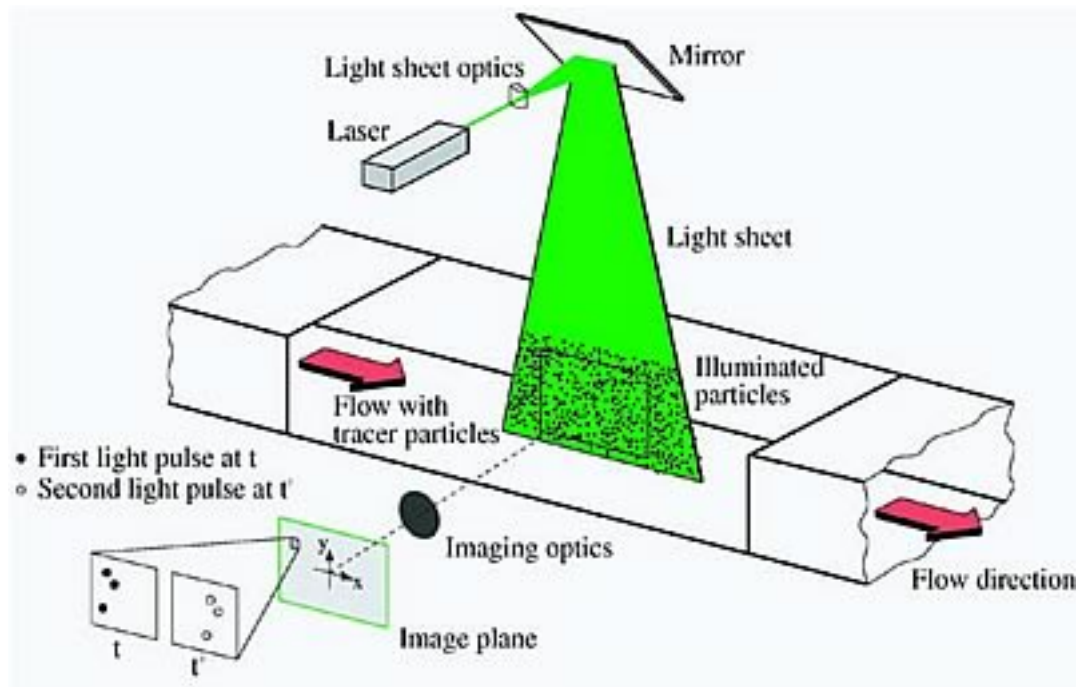


Figure 3.2. Illustration of seeding, illuminating and capturing of the image process in PIV

Figure 3.3 shows the main components of the PIV system consisting a CCD (Charge Coupled Device) digital camera, a laser system, a synchronizer, a frame grabber and a computer.

The PIV technique mainly contains two stages; image acquisition and image evaluation as shown in Figure 3.4.

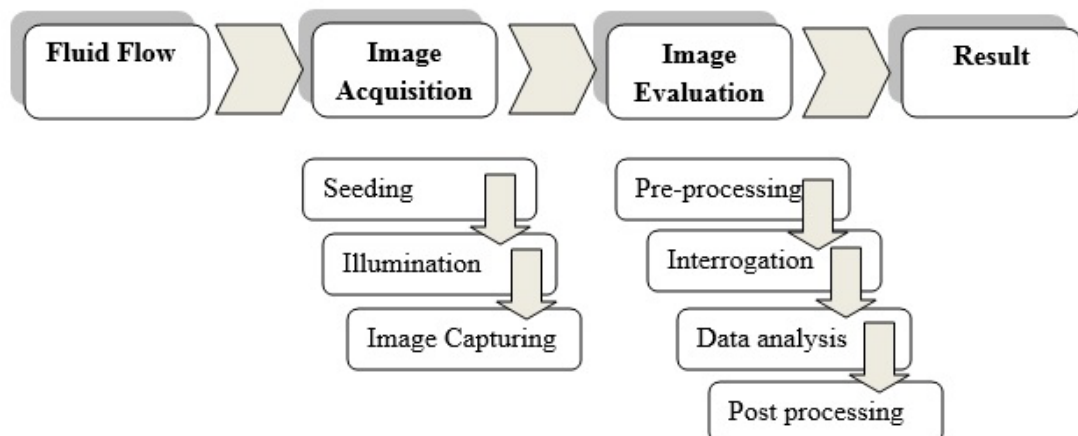


Figure 3.3. Flowchart of the PIV system

3.1.1.2. Image Acquisition

3.1.1.2.(1). Seeding

Seeding particles are added to the flow to track the movement of the particle. The type, size, number, distribution and tracking ability of the particles used in the measurements are very important in the PIV technique. In addition, density and size of the particle, and density and viscosity of the fluid affect the buoyancy and inertia. Although achieving the exact neutral buoyancy is difficult, the particles must be suspended throughout the experiments. Therefore, seeding particles should be carefully selected in order to avoid the discrepancies between the particle and the flow.

The particles' ability to track the flow field is determined by the Stokes number and limiting Stokes number depends on the flow geometry and existence of considerable accelerations or decelerations.

Stokes number is defined as the ratio of the stopping distance of a particle to a characteristic dimension of the obstacle;

$$S_t = \frac{\tau_p \cdot U_\infty}{d_{particle}}$$

where; τ_p is the response time of the particle

U_∞ is the fluid velocity away from the obstacle

$d_{particle}$ is the diameter of the seeding particle

The response time of a particle is;

$$\tau_p = \frac{\rho_{particle} \cdot d_{particle}^2}{18\rho_{fluid} \cdot \nu_{fluid}}$$

where; $\rho_{particle}$ is the particle density

ρ_{fluid} is the fluid density

ν_{fluid} is kinematic viscosity of the fluid

For, $S_t \gg 1$, particles will continue in a straight line as the fluid turns around the obstacle. However, for $S_t \ll 1$, particles will follow the fluid streamlines closely. For $S_t \ll 1$, tracing accuracy errors are below 1%.

Density value of the seeded particle should be nearly that of the fluid. These particles should be large enough to scatter sufficient light for the camera to be deflected accurately. On the other hand, they should be small enough to track the flow. For water flow, fluorescent, polystyrene, polyamide, silver-coated particles or hollow glass spheres with diameter range of 5 μm to 100 μm are used. The number of the particles added to the fluid is another parameter that affects accuracy of the measurements. It is important for obtaining a good signal peak in the cross-correlation. As a rule of thumb, 10 to 25 particles per interrogation area can be seen. (Karakuş, 2007)

3.1.1.2.(2). Illumination

Continuous wave lasers, pulsed lasers, laser diodes and Xenon flash lamps are potential light sources which can be used for illumination. Any of these sources can be used to illuminate the flow field in the PIV technique. Illumination source should have sufficient energy density to capture successful images of tracer particle in the flow. Lasers are mostly used to illuminate the flow field since they can produce a pulsed, collimated and monochromatic light beam which can easily be formed into a thin light sheet by a cylindrical lens.

Solid-state frequency-doubled Nd:YAG (Neodymium-Doped Yttrium Aluminum Garnet) lasers is the most common device used in PIV. They have high output energy (25 to 400 mJ/pulse), short pulse duration (6 to 10 ns) and emit lights with a wavelength of 532 nm. The nominal pulse repetition rate of Nd:YAG lasers is 10 to 15 Hz (Gürlek, 2008).

The illumination method used in the PIV system should satisfy following criteria:

- Power of the light which determines the intensity of scattered light from the seeding particles should be high enough so that images of particles can be recorded on the camera used in the PIV system.
- Time interval between the illumination pulses should be short enough to avoid particles with an out-of-plane velocity component and long enough to obtain sufficient resolution images of the tracer particle's movement.
- The duration of the light pulse should be short enough so that the particle does not move significantly during its exposure in order to avoid to blurring of the images.

3.1.1.2.(3). Image Capturing

CCD (Charge Coupled Device) camera records the images of the light scattered by the tracer particles with the aid of double pulsed laser sheet. The initial and final positions of the tracer particles are captured by the camera so that the displacement vector can be obtained.

Full-frame interline transfer progressive scan CCD camera, also called a cross-correlation CCD camera is used to achieve two single exposed images with a time separation of the order of microseconds. The main idea is that the image exposed by the first laser pulse is transferred very rapidly to light-hidden areas on the CCD-chip. Each pixel has its own storage site in immediate vicinity of the light sensitive pixel area. After the second exposure, both images are transferred to the computer. In a second, the camera can capture only few double-images because of the limited data transfer capacity. Therefore, in this technique, the temporal resolution quality of the flow is low.

The laser pulse and camera must be triggered with the correct time and sequence to capture the flow images accurately so the camera and the laser pulse should be synchronized.

3.1.1.2.(4). Image Evaluation

The image evaluation in the Particle Image Velocimetry technique consists of pre-processing, interrogation, data analysis and post processing. The PIV technique mainly determines the distances traveled by particles within a time between two laser pulses. Correlation, and particle tracking are the common methods to determine this distance. Knowing the time interval between the captured images enables a velocity vector to be calculated from the displacement vector. The correlation technique can be used for a single frame multiple exposed (auto-correlation) or multiple frames singly exposed (cross-correlation). The choice of the technique depends primarily on the available equipment used for the particle image data recording and the seed particle concentration. The recorded image frame is divided into small sub-regions, each containing particle images. A velocity vector map is generated by processing the image over a regular grid of small sub-regions.

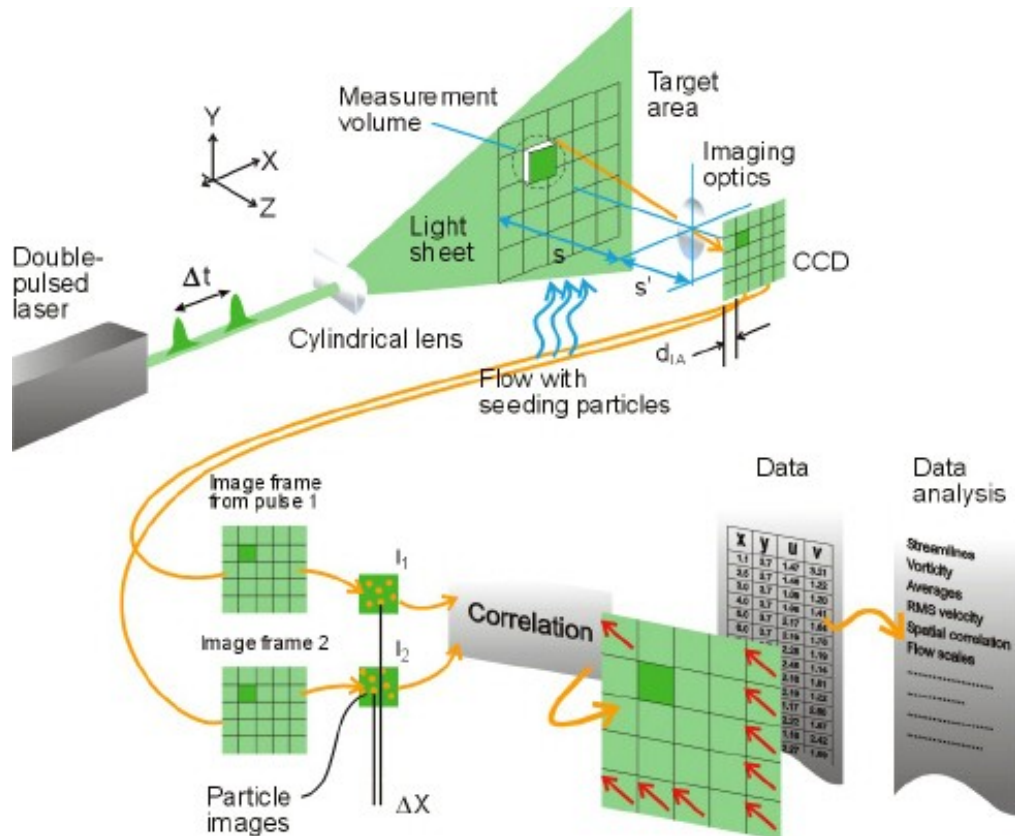


Figure 3.4. Typical 2D PIV setup

Both auto and cross-correlation operations are implemented using Fast Fourier Transforms (FFT). The differences between these correlation methods are the image window areas for the first and second images. In the auto-correlation, same frame is used for both first and second images. Auto-correlation cannot correlate the first and the second image with respect to initial and final positions since only one camera recorded. Instead both initial and final particle positions are recorded on the same camera frame, and then the recorded image is correlated with spatially shifted versions of itself. Since the recorded image is correlated with itself, auto-correlation produces a large central peak in the correlation plane corresponding to zero displacement. This happens since seeding particles are correlated with themselves, and thus it does not represent physical zero velocity. In cross-correlation, two single exposure image frames must be recorded. Similar to auto-correlation, the cross-correlation operation divides the image frames into small sub-regions. However, now a sub-region from image 1 (recorded at the first laser pulse) is cross-correlated with a sub-region from image 2 (recorded at the second laser pulse). The direction of the displacement is determined exactly since the images from exposures 1 and 2 are recorded separately (Gürlek, 2008).

The basic principle of Adaptive correlation is an iterative procedure: From an initial guessed offset value, an offset is introduced from the first window (the interrogation area in the image frame from laser pulse one) to the second window. The obtained vector is validated and is used as a new estimate for the window offset. This time, a new run with a smaller interrogation area is made. Capturing the images that left the interrogation area between the two light pulses is the main advantage of using the shifted window. Loss of these particle images is known as the "in-plane dropout", which reduces signal strength and, as a result, the number of successful vectors that can be obtained. Capturing more particle images for each vector permits the interrogation area to be refined. It allows to obtain an adequate number of successful vectors without increasing the seeding density in the flow.

The post-processing of the raw velocity field includes vector validation, removal of spurious vectors, replacement of the removed vectors, and data smoothing and filtering. The vectors are validated using CLEANVEC, a program

written by Ron Adrian's group in Theoretical and Applied Mechanics at the University of Illinois, after the velocity field is calculated. The software CLEANVEC contains four statistical filters designed for incorrect vectors removal:

- Absolute range filter
- RMS tolerance filter
- Magnitude difference filter
- Quality filter

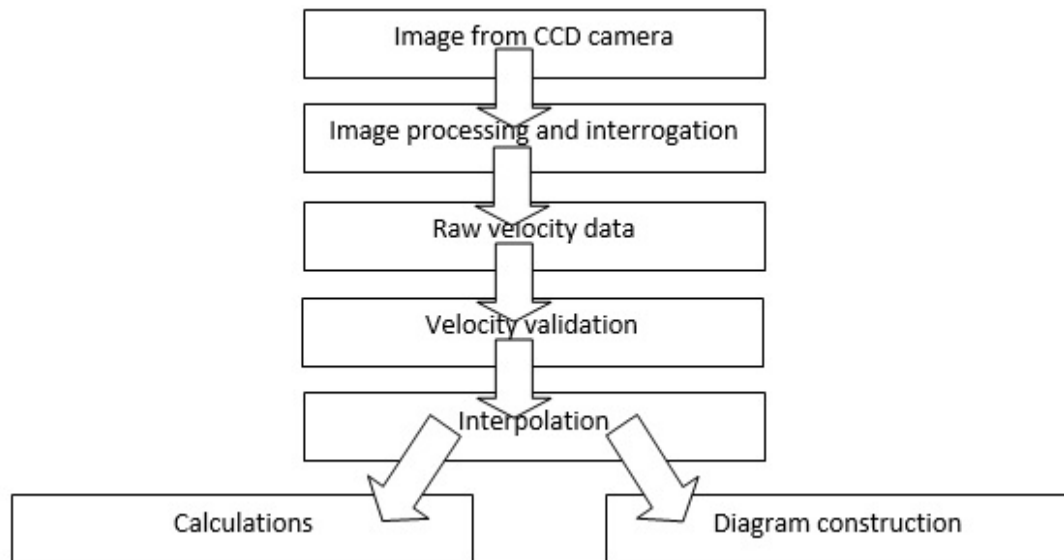


Figure 3.5. General procedure for image processing

Absolute range filter, RMS tolerance filter and Magnitude difference filter are used for purposes of eliminating incorrect vectors. A bilinear least-square fit technique is used to fill in the areas of missing vectors and scale the image field to the actual flow field after the process of removing incorrect vectors. The interpolated and scaled velocity field is also smoothed with a smoothing parameter of 1.3 by Gaussian weighted averaging technique based on the work of Landreth and Adrian (1989). The vorticity is calculated by circulation method. Constant vorticity contours are calculated using a spline fit technique with a tension factor of 0.1 for smoothing process.

3.1.2. Water Channel System

The experiments were conducted in a closed-loop water channel located in the Fluid Mechanics Laboratory at Çukurova University. The water channel has a dimensions of 750 mm height, 1000 mm width and 8000 mm length. Bottom and side walls of the working section of the channel were made of 15 mm thick transparent Plexiglas for optical access. The flow was driven by a 15 kW centrifugal pump having a variable speed controller. The maximum flow speed in the water channel is 250 mm/s. The flow passed through a settling reservoir, a honeycomb and a 2:1 contraction before entering the test chamber. The mean velocity was uniform and average turbulent intensity was less than 0.5% in the empty test section.

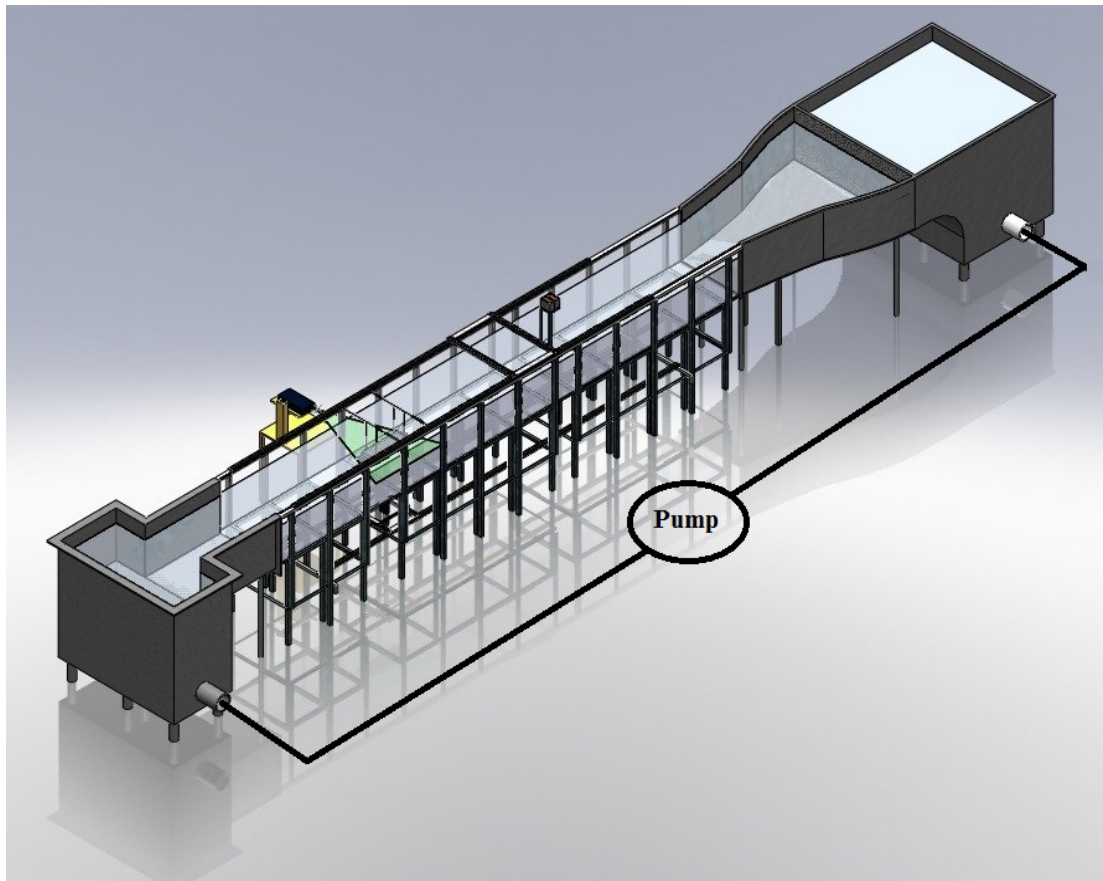


Figure 3.6. Schematic representation of water channel system

3.1.2.1. PIV Instrumentation

The velocity fields were measured employing a Dantec PIV system. A schematic diagram of the PIV setup in the test section of the water channel is shown in figure 3.6. The flow field illumination was provided by two Nd: Yag pulsed laser sources of a wavelength of 532 nm, each with a maximum output energy of 120 mJ. The thickness of the laser sheet was approximately 2 mm. The time interval between pulses was 1.2 ms. The flow was seeded with silver coated spherical particles 12 μm in diameter. A 1024x1024 pixel resolution CCD camera, equipped with a 60 mm focal length lens, was used to record the movement of the particle. The Dantec flow map processor was used to synchronize the CCD camera and the Nd: Yag lasers. 32x32 pixel rectangular effective interrogation windows with an overlap of 50% provide 3844 velocity vectors. Thousands images were taken at a rate of 15 Hz to determine the time-averaged mean flow structure. Adrian (1991) reported the principles of the PIV technique in detail. It is known that the rate of uncertainty for the measurement of velocity using the PIV technique can be affected by the size, nonuniform distribution of the particles, particle overlap and the interrogation window size, as well as electronic noises. An overlap of 50% was taken into account to satisfy the Nyquist criterion during the data process. During the performance of the experiment, false velocity vectors may happen in the image field induced by the distortions of laser sheet and shadows. These false vectors were replaced by using the local median-filter and bilinear interpolation techniques. The PIV technique used in the present study was similar to the system used by Sahin et al. (2007) and Gurlek et al. (2012), who stated that the uncertainty rate in the measured data of instantaneous velocity was less than $\pm 2\%$.

3.1.3. Experimental Model

Experimental model was manufactured from plexiglass material using CNC machine. Experiments were carried out for three different phase angles and channel heights and they were adjusted by moving the upper wall. The channel height was

adjusted by moving the upper wall up and down. On the other hand, phase shift angle was adjusted by moving the upper wall left and right. Figure 3.7 shows the dimensions of the experimental models with phase shifts, $\phi = 180^\circ$, 90° and 0° . For all models, the wavy wall geometry was kept constant. The wave length, L_w and wave amplitude, a of the sinusoidal corrugated wall were taken 28 mm and 3.5 mm, respectively.

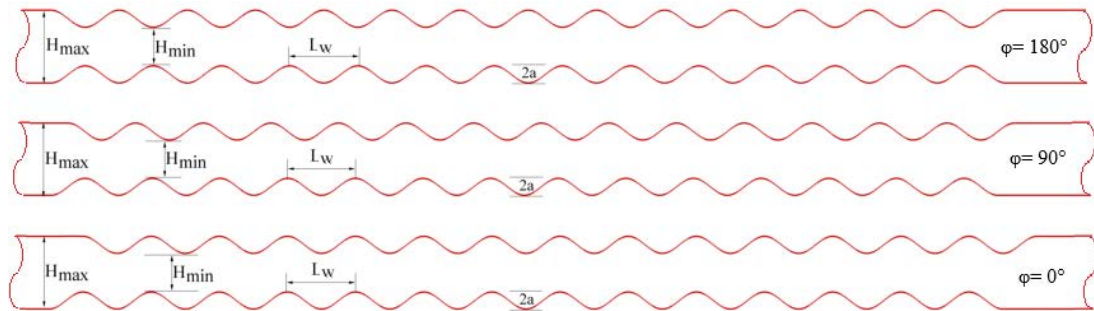


Figure 3.7. Dimensions of the experimental models

Experimental models with three different phase shifts ($\phi = 180^\circ$, 90° and 0°) were presented in figure 3.8. The width of the channel (180 mm) was very large compared to the height of the channel as seen in figure 3.8.



Figure 3.8. Experimental models used in this study

The position of the test model in the water channel is shown in figure 3.9. The model was placed on a 700 mm long and 300 mm height horizontal flat plate raised from the floor of the working section. The depth of the water in the channel was 480 mm.



Figure 3.9. The view of immersed model in the water channel

3.2. Numerical Procedure

3.2.1. Definition of Physical Domain

Numerical simulation of fully developed flow and heat transfer through a sinusoidal wavy channel consisting of thirteen wavy sections has been considered. The adiabatic smooth sections before and after the wavy sections were added to the geometry in order to obtain appropriate boundary conditions. The wavy sections of the geometry consist of two isothermally heated sinusoidal wavy walls with wavelength of (L_w) and amplitude of (a) as seen in figure 3.10.

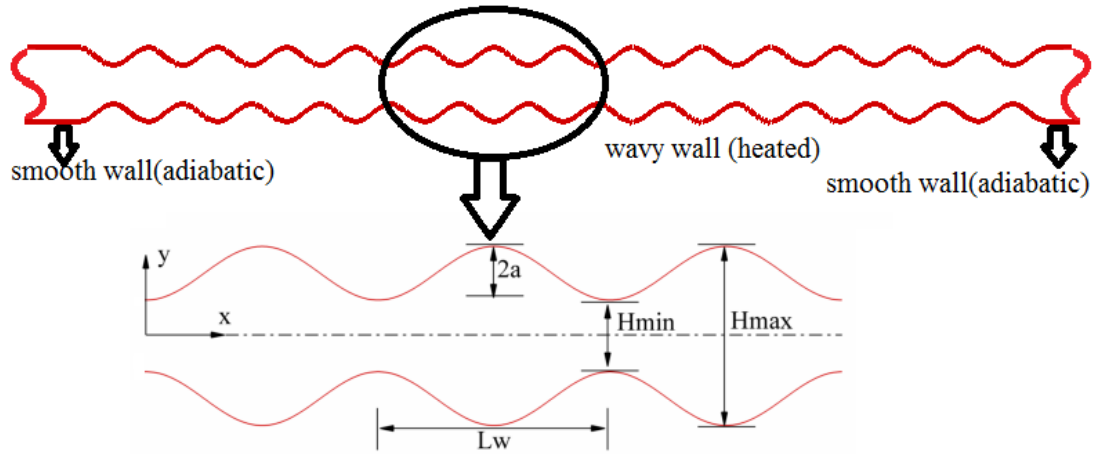


Figure 3.10. Schematic of the 180° phase shift physical model

The wavy surface function was defined by $y = \frac{H_{\min}}{2} + a \sin \frac{2\pi x}{L_w}$. Effect of geometry, generated by varying $H_{\min}/H_{\max}$ ratio and phase angle while keeping wavy surface geometry fixed, on fluid flow and heat transfer characteristics has been investigated in detailed. Three different $H_{\min}/H_{\max}$ ratios and phase angles were considered. The wave length and wave amplitude of the wavy wall are 28 mm and 3.5 mm, respectively. For the present study, the following geometric parameters are applied; $H_{\min}/H_{\max}$ of 0.36, 0.54 and 0.72 and the phase angle of 180°, 90° and 0°.

3.2.2. Governing Equations

The main governing equations can be expressed as follows:

Continuity Equation:

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

Momentum Equation:

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \frac{\partial^2 u}{\partial x^2} + \nu \frac{\partial^2 u}{\partial y^2}$$

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \frac{\partial^2 v}{\partial x^2} + \nu \frac{\partial^2 v}{\partial y^2}$$

Energy Equation:

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

Where $\alpha = \frac{k}{\rho \cdot C_p}$

3.2.3. The k-ε Turbulence Model

The k-ε turbulence model is the one of the most commonly used turbulence model although it has well-known shortcomings. The k-ε turbulence model is a two-equation model, which means, two additional transport equations are used to represent turbulent flow. The first transported variable is turbulent kinetic energy, k and the second transported variable is turbulent dissipation, ϵ . The k-ε turbulence model uses gradient diffusion hypothesis to relate the Reynolds stresses, $\overline{\rho u'_i u'_j}$ to the mean velocity gradients and the turbulent viscosity, μ_t . The turbulent viscosity is modeled as the product of a turbulent velocity and turbulent length scale. The transport variable turbulent kinetic energy determines the turbulent velocity scale whereas the turbulent length scale is obtained from both turbulent kinetic energy and turbulent dissipation rate. In the derivation of k-ε model, it was assumed that the flow is fully turbulent, and the effects of molecular viscosity are negligible. Therefore, the standard k-ε turbulence model is only valid for fully turbulent flows.

In the present numerical study, the k-ε turbulence model proposed by Launder and Spalding (1974) is used to solve the numerical problem due to the robustness, economy, and acceptable accuracy for turbulent flow and heat transfer simulations. The turbulent kinetic energy (k) and the rate of turbulent kinetic energy dissipation (ϵ) are defined, respectively, as:

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

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

Where the rate of turbulent kinetic energy generation is

$$G_k = -\rho \overline{u'_i u'_j} \frac{\partial u_j}{\partial x_i}$$

The turbulent viscosity, μ_t is computed by combining k and ε as follows:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$$

The following constants can be obtained by comprehensive data fitting for a wide range of turbulent flow. (Launder and Spalding (1974))

$$C_{\varepsilon 1} = 1.44, \quad C_{\varepsilon 2} = 1.92, \quad C_\mu = 0.09, \quad \sigma_k = 1.0, \quad \text{and} \quad \sigma_\varepsilon = 1.3$$

3.2.4. Boundary Conditions

A no slip boundary condition is imposed at the walls, where both velocity components are set to zero. A uniform temperature of the input fluid is taken less than constant wavy wall temperature while the flat walls are thermally insulated. A fully developed velocity profile is assigned at the inlet in order to shorten the straight section of the channel which reduces the computational time. The flow is assumed to be Newtonian, turbulent, two-dimensional and incompressible.

$$u_{in} = 6\bar{u} \left[0.25 - \left(\frac{y}{H} \right)^2 \right] \quad \text{and} \quad T = T_{in}$$

On the walls of the channel

$$u = v = 0 \quad \text{and} \quad T = T_w \text{ at sinusoidal wavy walls}$$

$$T = T_{in} \text{ at smooth walls}$$

On the axis of the channel

$$\frac{\partial \bar{u}}{\partial y} = \frac{\partial \bar{v}}{\partial y} = \frac{\partial \bar{T}}{\partial y}$$

To represent the results and characterize fluid flow and heat transfer in a wavy channel, the following parameters are presented.

Reynolds number, Re is defines as:

$$Re = \frac{\rho \cdot \bar{u} \cdot D_h}{\mu}$$

Heat transfer coefficient, h along the wavy surface can be calculated using the following equation:

$$h(x) = \frac{q''(x)}{T_w - T_m}$$

Where T_m is bulk temperature which can be determined as:

$$T_m = \frac{\int_0^{H/2} u \cdot T \cdot dA}{\int_0^{H/2} u \cdot dA}$$

The local Nusselt number, Nu is defined as:

$$Nu_x = \frac{h_x \cdot D_h}{k}$$

The average Nusselt number along the corrugated wavy wall is calculated by integrating the local Nusselt number, $\overline{Nu}$ over the wavy section.

$$\overline{Nu} = \frac{1}{L} \int_0^L Nu \cdot dx$$

The friction factor, f for the flow in corrugated channel is computed as:

$$f = \Delta p \frac{2D_h}{(x_e - x_s)(\rho \bar{u}^2)}$$

Using corrugated wall instead of smooth one enhances heat transfer inherently comes with additional pressure loss caused by higher friction at the walls due to the flow separation. Since most of the time heat transfer and friction factor, f increase simultaneously, the thermal performance factor (TPF) is used to analyze the overall

performance of the geometric modifications using the Nusselt number, Nu and friction factor, f information.

$$TPF = \frac{Nu/Nu_o}{\left(f/f_o\right)^{1/3}}$$

Where Nu_o and f_o are Nusselt number, Nu and friction factor, f for fully developed flow in a parallel-plate channel subject to constant wall temperature, respectively.

3.2.5. Numerical Computation

The governing equations of flow have been discretized by a finite volume method and pressure-velocity coupling system has been resolved by using the SIMPLE algorithm. Non-staggered grids are used. As can be seen in figure 3.11 that grids are finer near the boundaries while coarser at the core region since higher gradients of variable values are expected near the walls. The numerical computation is ended if the sum of the absolute residual satisfy the convergence criterion of 10^{-6} for all parameters in computational domain. A constant time step of 0.001 was used for all calculations.

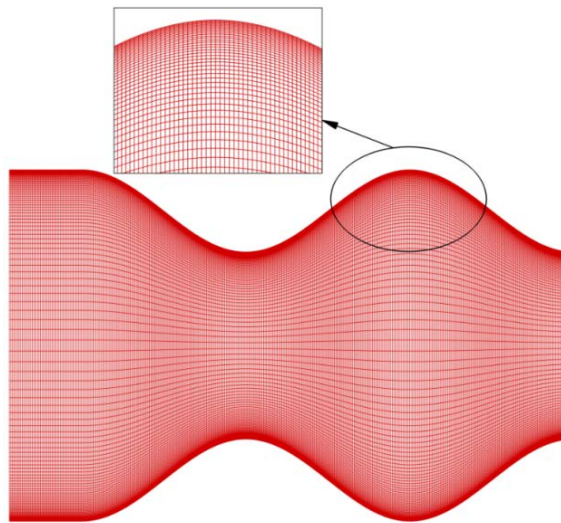


Figure 3.11. A typical non-uniform and non-orthogonal grid in the computational domain

3.2.6. Grid Testing and Code Validation

In the present numerical investigation non-staggered grids which are finer near the walls are used. The accuracy of the numerical results depends on the grid resolution. In order to estimate the accuracy of the numerical computation, the domain for a channel with a phase shift of 180° and $H_{\min}/H_{\max}$ ratio of 0.54 was tested. Three different mesh combinations (139×65 , 160×80 , 228×115) for one wave were used to test the number of elements required. Figure 3.12 shows the axial velocity profiles at the middle of the third wave for $Re=2500$. Since the deviation between the medium coarse and fine grid is very small, the results of the fine grid (228×115) are presented in this study.

The present results were also compared with the experimental results for the fluid flow characteristics in section 4. The comparisons between numerical and experimental results show good agreement.

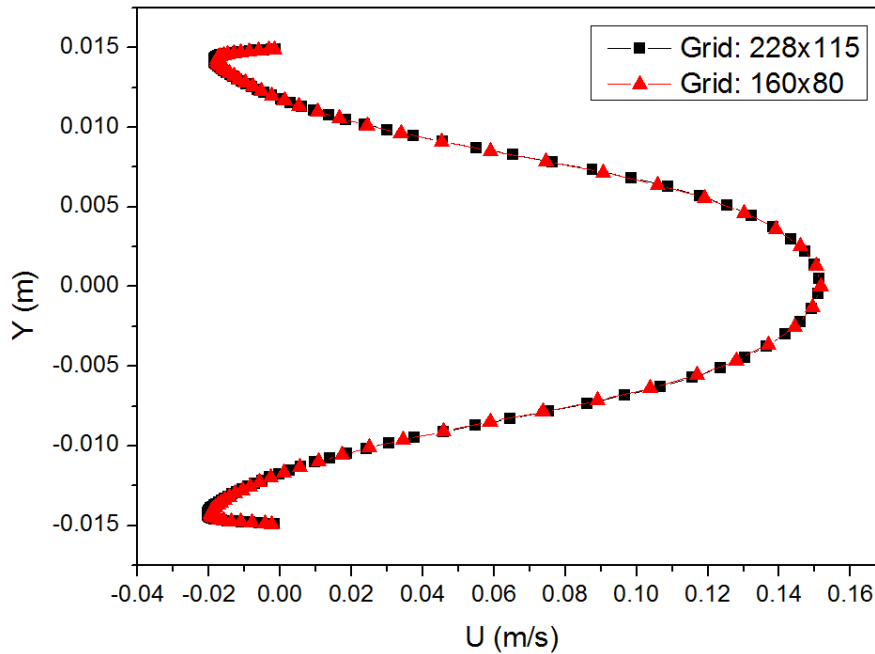


Figure 3.12. Velocity profiles at the widest cross section of 3rd wave at $Re=2500$

4. RESULTS AND DISCUSSIONS

In this study, hydrodynamics and thermal characteristics of incompressible flow through channel formed by lower and upper plates with sinusoidal corrugations are investigated. The test section consists of two subsections. In the first subsection, the experimental results and numerical predictions of the flow behaviors through channel are compared. In the second subsection, heat transfer enhancement was investigated numerically using commercial software ANSYS FLUENT. For both subsections, three different channel ratios, $H_{\min}/H_{\max} = (0.36, 0.54 \text{ and } 0.72)$ and phase shift angles, ϕ , ($0^\circ, 90^\circ \text{ and } 180^\circ$) were considered while keeping wavy surface geometry fixed.

4.1. Experimental and Numerical Studies of Fully Developed Flow in Corrugated Channels

The experiments were performed for investigation of flow field in corrugated channel using the Particle Image Velocimetry (PIV) technique. Since it is difficult to take a detailed image of thirteen wave long channel in one view, the channel was divided into six field of views. In this section, experimental and numerical results of these six field of views are presented in different figures from the top surface to the bottom surface of channel walls. Experimental results are compared with the numerical predictions. It is worth to mention that it is substantially important to know the flow characteristics in detail for designing optimum geometry of the corrugated channels. Since it is possible to observe the turbulent unsteady flow and provide quantitative numerical data on the spatial characteristics of instantaneous velocity vectors in flow fields, the PIV measuring system is used to determine instantaneous and time-averaged turbulence flow statistics in the sinusoidal corrugated channels in order to investigate the flow structures at a given range of Reynolds numbers, Re .

In this section, contours of time-averaged streamwise velocity component, $\langle u \rangle$, contours of vorticity, $\langle \omega \rangle$, streamline patterns, $\langle \Psi \rangle$, Reynolds stress correlations, $\langle u'v' \rangle$ and turbulence kinetic energy, $\langle TKE \rangle$ correlations are presented.

The channel is divided into six fields of view in order to see the flow field in more detail. Time-averaged flow data are determined using flow data of 1000 PIV images.

4.1.1. Velocity Distribution

In this section, experimentally obtained and numerically predicted time-averaged streamwise velocity components, $\langle u \rangle$ are presented, and compared with each other for straight channel and corrugated channels for three different phase shift angles such as, 180 degree, 90 degree and 0 degree, respectively.

Since 1000 images were taken during the one set of the PIV experiment, the time-averaged experimental parameters are calculated by taking an average of 1000 instantaneous images. The time-averaged streamwise velocity component, $\langle u \rangle$ is calculated as follows:

$$\langle u \rangle = \frac{1}{N} \sum_{n=1}^N u_n(x, y)$$

Where N (1000) is the number of the instantaneous images taken during the experiments for one set.

4.1.1.1. Velocity Contours of Straight Channel

When the fluid entering a channel in a uniform velocity, the fluid particles in the layer in contact with the surface of the channel stop due to no slip condition. The contact layer with the channel surface resists the motion of the adjacent layers due to the fluid viscosity. Therefore, velocity of the fluid decreases gradually. Since the fluid layers near the channel surface reduces velocities, the velocity of middle layers of the fluid in the channel increases which develops a velocity gradient across the cross section of the channel.

The boundary layer starts growing at the entrance of the channel and the flow becomes fully developed when the boundary layers from the walls meet the axis of the channel. The region between the entrance and the point where the boundary layer

covers entire channel is called as the hydrodynamic entrance region. The distance from the entrance at which hydrodynamically fully developed condition is achieved is termed as the hydrodynamic entry length (L_e). The entry length is much shorter in turbulent flow as compared to laminar flow.

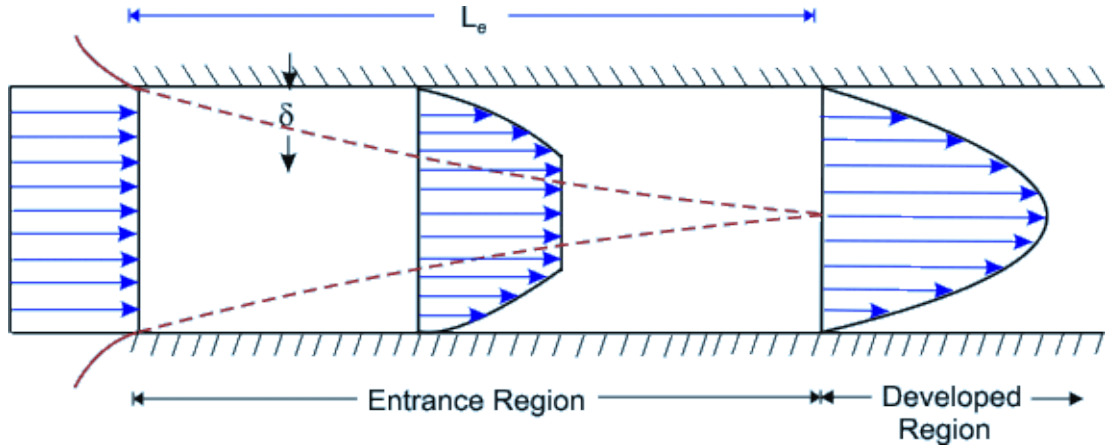


Figure 4.1. Development of fluid flow at the entrance region of a channel

In fully developed flow region, the velocity profile is constant along the specified flow direction. The fluid velocity is zero at the surfaces due to the no slip condition and becomes maximum at the center. Since it is desired to have the flow hydrodynamically fully developed before entering the corrugated section. Namely, the entry length was considered to be enough for having hydrodynamically fully developed flow. In most of the practical engineering applications, entrance effects become insignificant beyond a pipe length of 10 times the diameter. In these experiments, the entry length was taken 630 mm.

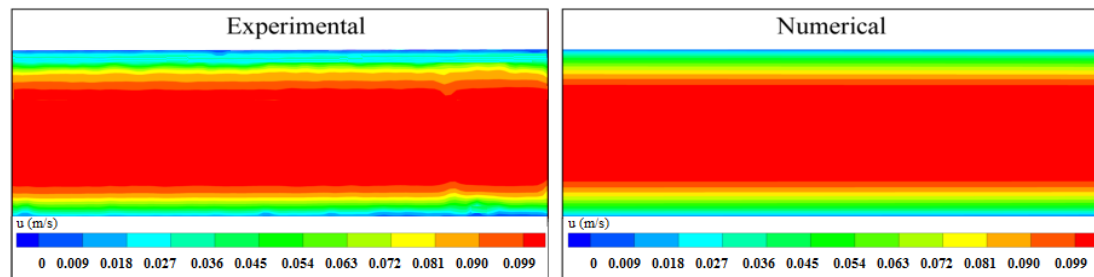


Figure 4.2. Time-averaged streamwise velocity component, $\langle u \rangle$ contours at $Re=2500$ for straight channel

The fully developed flow in straight channel can be seen in figure 4.2. The left hand side column shows the experimental results while the right hand side column presents the numerical results. Both experimental and numerical results are obtained for $Re=3750$. The height of the straight channel is equal to the maximum height of the corrugated channel where $H_{min}/H_{max}=0.54$. The incremental value of the velocity contours is 0.009 m/s and the maximum value of the velocity is 0.099 m/s. It is observed that the flow exhibits symmetrical flow structures. The velocity is higher at the center of channel and lower near the channel walls as expected.

4.1.1.2. Velocity Contours of Wavy Channel

In general, the fundamental geometrical properties of the present gradual diverging channel are similar to the two dimensional diffuser. An immediate consequence of this channel divergency is that in incompressible flow the average velocity at largest cross section of divergent part of the channel is lower than that at inlet which obeys continuity principle. This two-dimensional gradual divergency possess additional fundamental flow characteristics since one-dimensional considerations are not enough to discuss flow behaviors. Two-dimensional considerations have to be taken into account in order to define the dissipation of mechanical energy and other flow characteristics. It is worth to mention that the rate of dissipation of mechanical energy with in the divergent part of channel is not excessive as compared to the sudden enlargement geometry.

During the experiments, the channel was divided into six fields of view to get more detailed images. These six field of views are presented in figure 4.3.

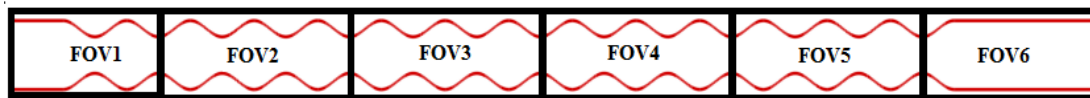


Figure 4.3. Field of views (FOVs) for channel with phase angle, ϕ of 180 degree

Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=2500$ and $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree are

presented in figures 4.4 and figure 4.5, respectively. For both figures, the left hand side columns present the velocity, $\langle u \rangle$ contours obtained from the experimental data. On the other hand, right hand side columns show the numerically predicted results. The maximum and incremental values of the time averaged velocity, $\langle u \rangle$ contours for $Re=2500$ are 0.126 m/s and 0.014 m/s, respectively. On the other hand, an increment of the time-averaged velocity, $\langle u \rangle$ contours is 0.021 m/s and velocity reaches their maximum value at 0.189 m/s for $Re=3750$. Since the upper and lower walls of the channel are symmetrical with respect to the centerline of the channel in the case of $\phi=180^\circ$, the flow is also symmetrical. It is clearly observed that the velocity contours of experimental and numerical results are similar for both Reynolds numbers.

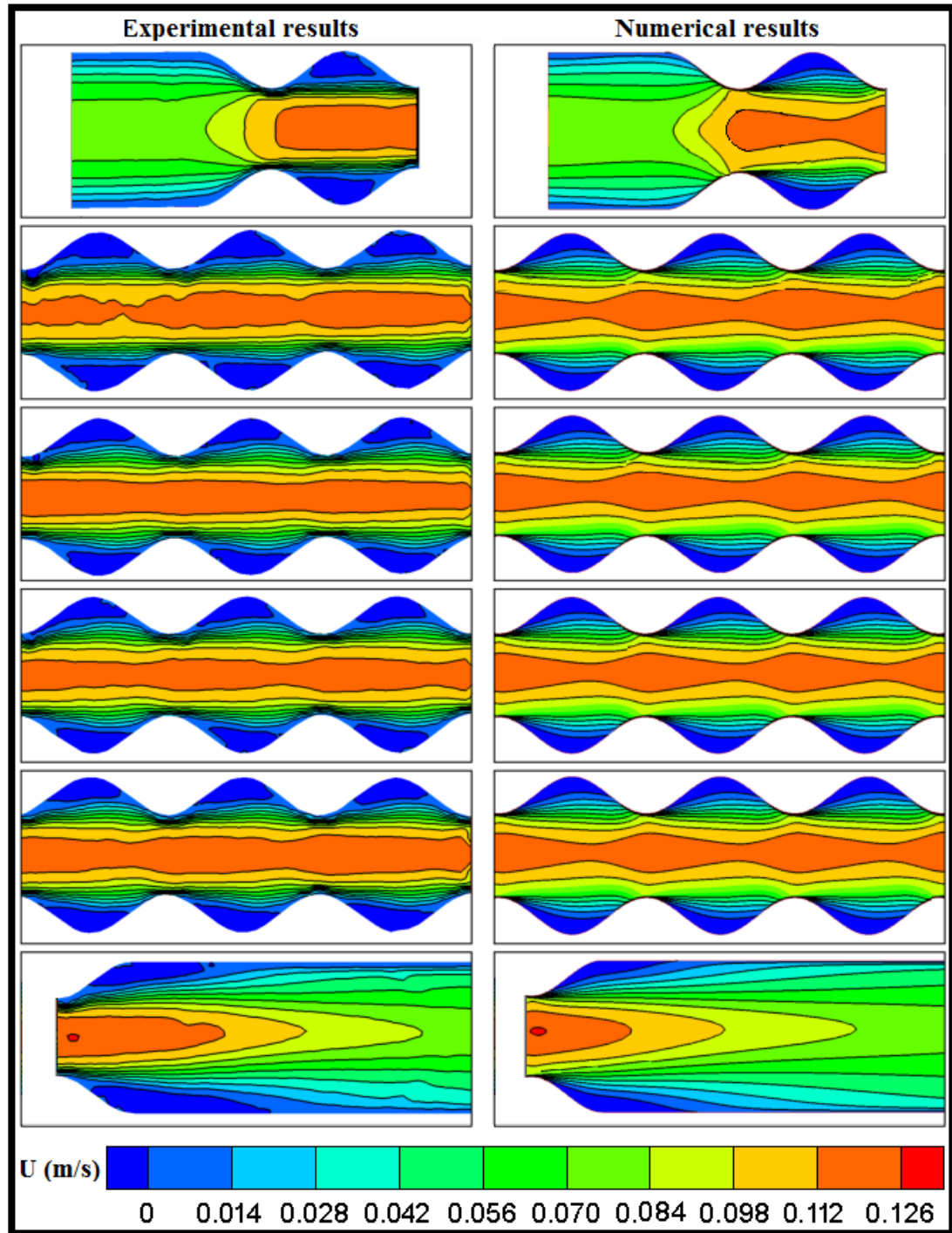


Figure 4.4. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=2500$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree

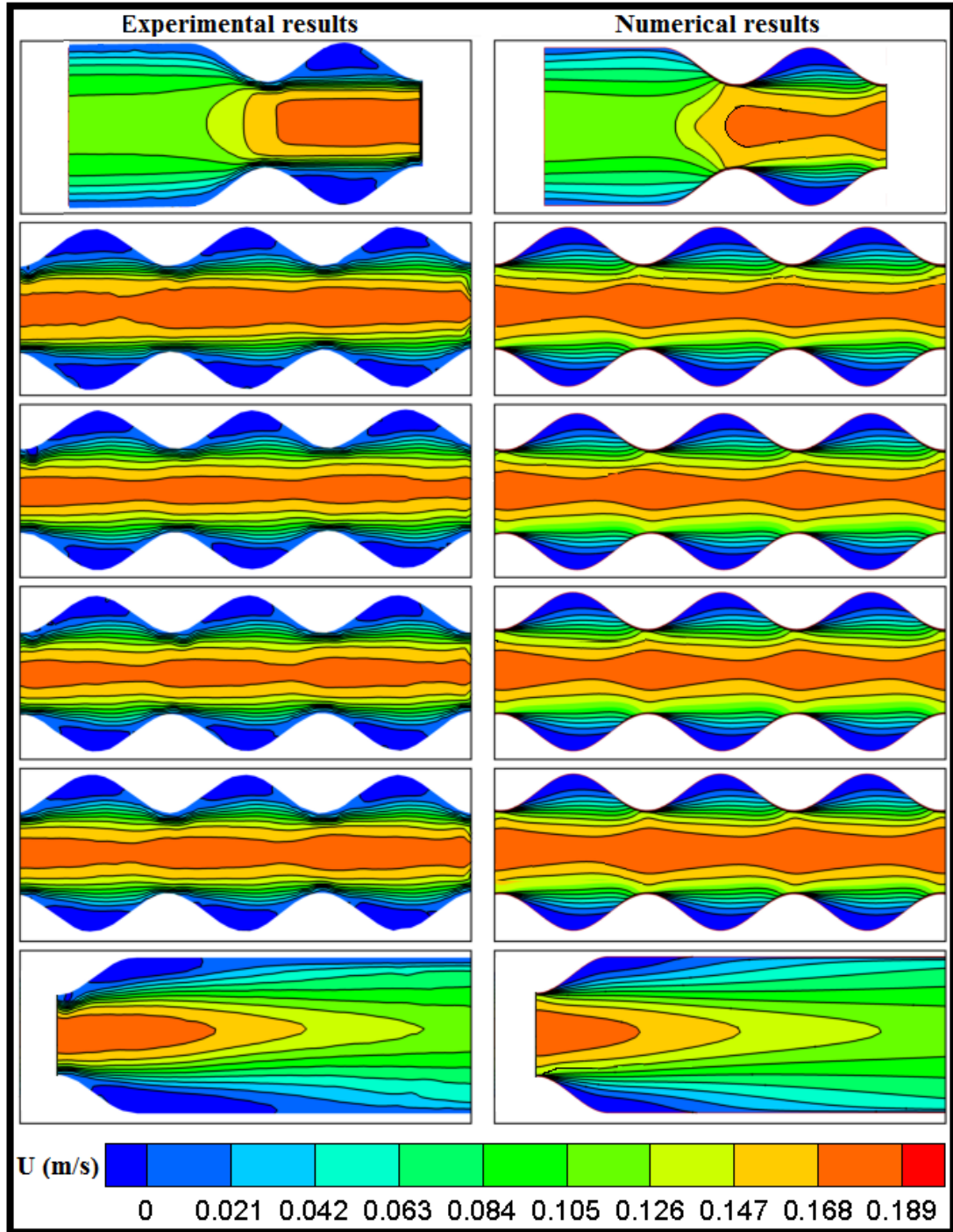


Figure 4.5. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=3750$, for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 180 degree

Time-averaged velocity of the sinusoidal wavy channel with a phase angle, ϕ of 180 degree is higher than that of the straight channel. The momentum enhances in sinusoidal corrugated channels since the wavy surface geometry manipulates the

direction of the fluid. Therefore, the wavy surfaces at channel operate as turbulence promoters when compared to the straight channel case. Since the sinusoidal wavy channel with a phase angle, ϕ of 180 degree has the converging and diverging sections, the time-averaged streamwise velocities, $\langle u \rangle$ vary along the channel. As it can be seen from figures 4.4 and 4.5, the time-average streamwise velocity, $\langle u \rangle$ decreases in the diverging sections and increases in the converging sections of the wavy channel. The main flow passing through the straight section of the channel is deflected into the corrugated confined area where the large scale vortices are created. These vortices split into a number of small scale eddies just downstream of the cavity corner, and these eddies disperse in wakes and shear layers. Turbulent eddies also cause energy dissipation in the form of heat which is usually stated as pressure losses. Using corrugations increases turbulence intensity resulting in heat transfer rate enhancement. Since friction losses are also higher in the corrugated channels, there would be a parameter called thermal performance factor to determine the overall efficiency. Thermal performance factor, TPF and overall efficiency will be discussed later in this chapter.

The recirculation zones increase in size and shift to the right closer to the next wave by increasing the Reynolds number. This happens due to the higher flow rate pushing the recirculation flow boundaries further downstream.

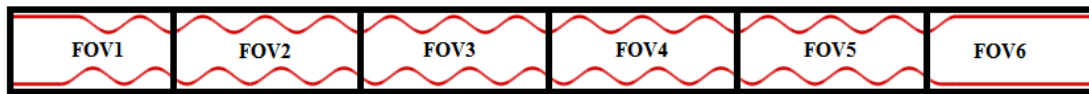


Figure 4.6. Field of views (FOVs) for channel with phase angle, ϕ of 90 degree

Figure 4.6 shows the six field of views for channel with the phase angle, ϕ of 90 degree. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 90 degree are presented in figure 4.7. The left hand side column presents the streamwise velocity, $\langle u \rangle$ distributions obtained from the experimental data. On the other hand, right hand side columns show the numerical results.

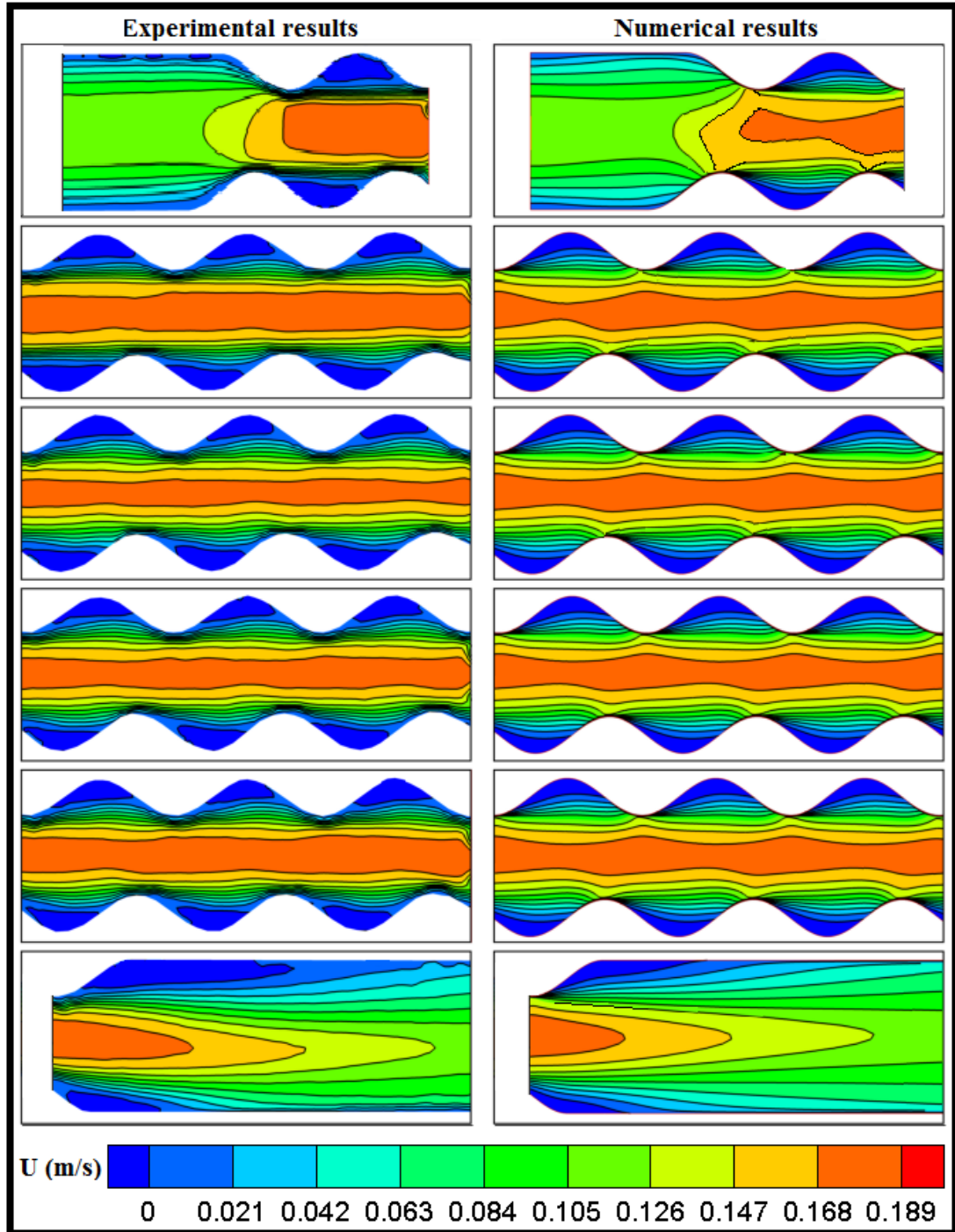


Figure 4.7. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 90 degree

The maximum and incremental values of the time-averaged velocity, $\langle u \rangle$ contours for $Re=3750$ are 0.189 m/s and 0.014 m/s, respectively. Asymmetrical time-averaged streamwise velocity, $\langle u \rangle$ distributions can be observed both from the

experimental and numerical results of sinusoidal wavy channel with phase angle, ϕ of 90 degree because the channel is not symmetrical either. Time-averaged velocity contours distributions of numerical results showing good agreement with the experimental result. As it can be seen from the figure 4.7, negative velocity, $\langle u \rangle$ contours occurred in furrows of the channel due to the lack of fresh fluid moving towards the wake region.

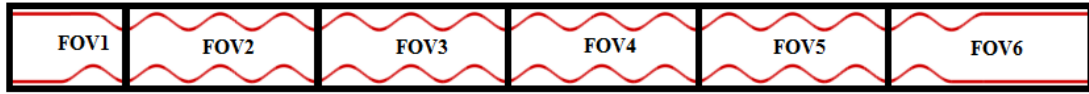


Figure 4.8. Field of views (FOV) for channel with phase angle, ϕ of 0 degree

Figure 4.8 shows the six field of views for channel with the phase angle, ϕ of 0 degree. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 0 degree are presented in figure 4.7. The maximum and incremental values of the time-averaged streamwise velocity, $\langle u \rangle$ contours for $Re=3750$ are 0.189 m/s and 0.014 m/s, respectively.

The position of the reattachment point of the free shear layer changes largely due to the flow mixing occurred by large oscillations in the core flow. When the reattachment point moves far enough upstream, the core flow impinging on the wall injects free-stream fluid into the separation bubble, and this injection is accompanied by an ejection of fluid from the separation bubble into the core flow. This exchange of fluid results in macroscopic mixing. Since the core flow is forced to follow the waves of the surrounding geometry in the channel with phase angle, ϕ of 0° , a higher rate of interactions occur with recirculation. Hence $\phi=0^\circ$ is the case where significant mixing appear first among all shift angles, ϕ investigated in this study.

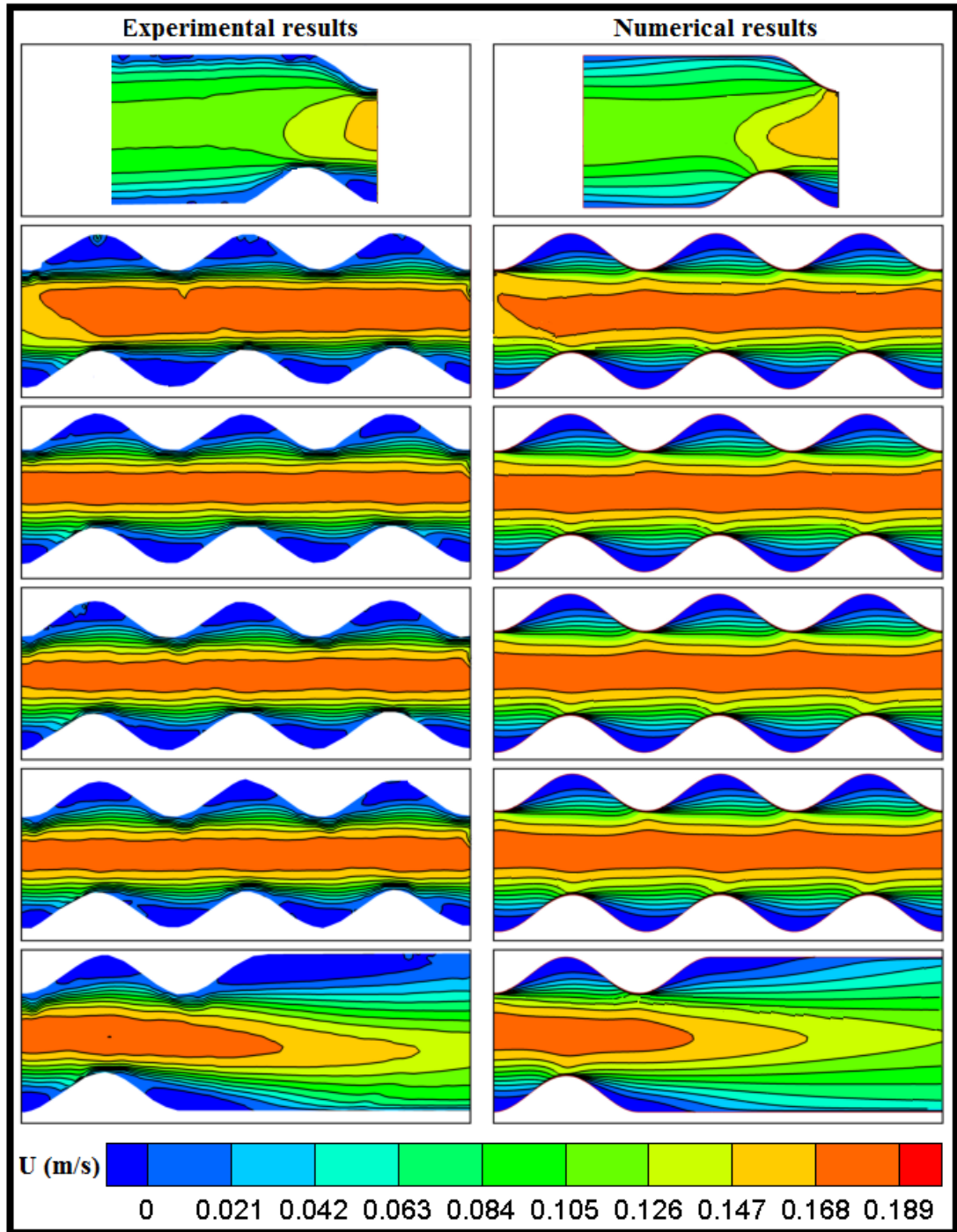


Figure 4.9. Time-averaged streamwise velocity, $\langle u \rangle$ contours at $Re=3750$, for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 0 degree

4.1.1.3. Time-Averaged Velocity Profiles of Wavy Channels

The experimental time-averaged streamwise normalized velocity, $\langle u/U_{ave} \rangle$ profiles at Reynolds numbers 2500 and 3750 for channel with $H_{min}/H_{max}=0.54$ and the phase angle, ϕ of 180 degree are presented in figure 4.11. On the other hand, figure 4.12 shows the comparison of the experimental and numerical velocity profiles at the Reynolds number, $Re= 3750$ for channel with the phase angle, ϕ of 180 degree. Here, time-averaged velocity components, $\langle u \rangle$ are normalized with the average velocities, U_{ave} of the cross sections. Figures 4.11 and 4.12 present the velocity profiles of channel with phase angle of 180° at three sections of 3rd, 6th, 9th, and 12th waves. Figure 4.10 shows the three cross sections for 3rd, 6th, 9th, and 12th waves of the sinusoidal wave channel with phase angle of 180° . The y dimensions are normalized with the length of the sections. The lower and upper walls at sections are assumed to be 0 and 1, respectively. The velocity, $\langle u/U_{ave} \rangle$ profiles at sections 1, 2, and 3 are shown in figures from top to bottom.

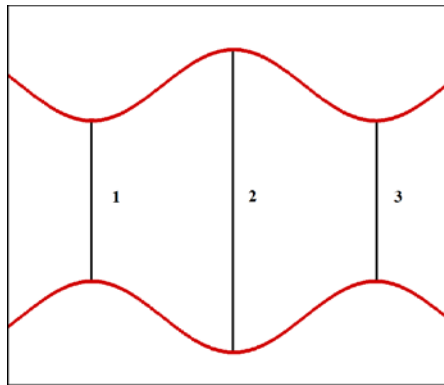


Figure 4.10. Three cross-sections for channel with the phase angle, ϕ of 180 degree

Negative velocities at the upper and lower regions of the waves are observed at sections 2 due to the secondary flow structures. The velocity profiles in the boundary layer at sections 2 represent reverse flow which shows separated flow. Boundary layer detaches from the surface into a broader wake.

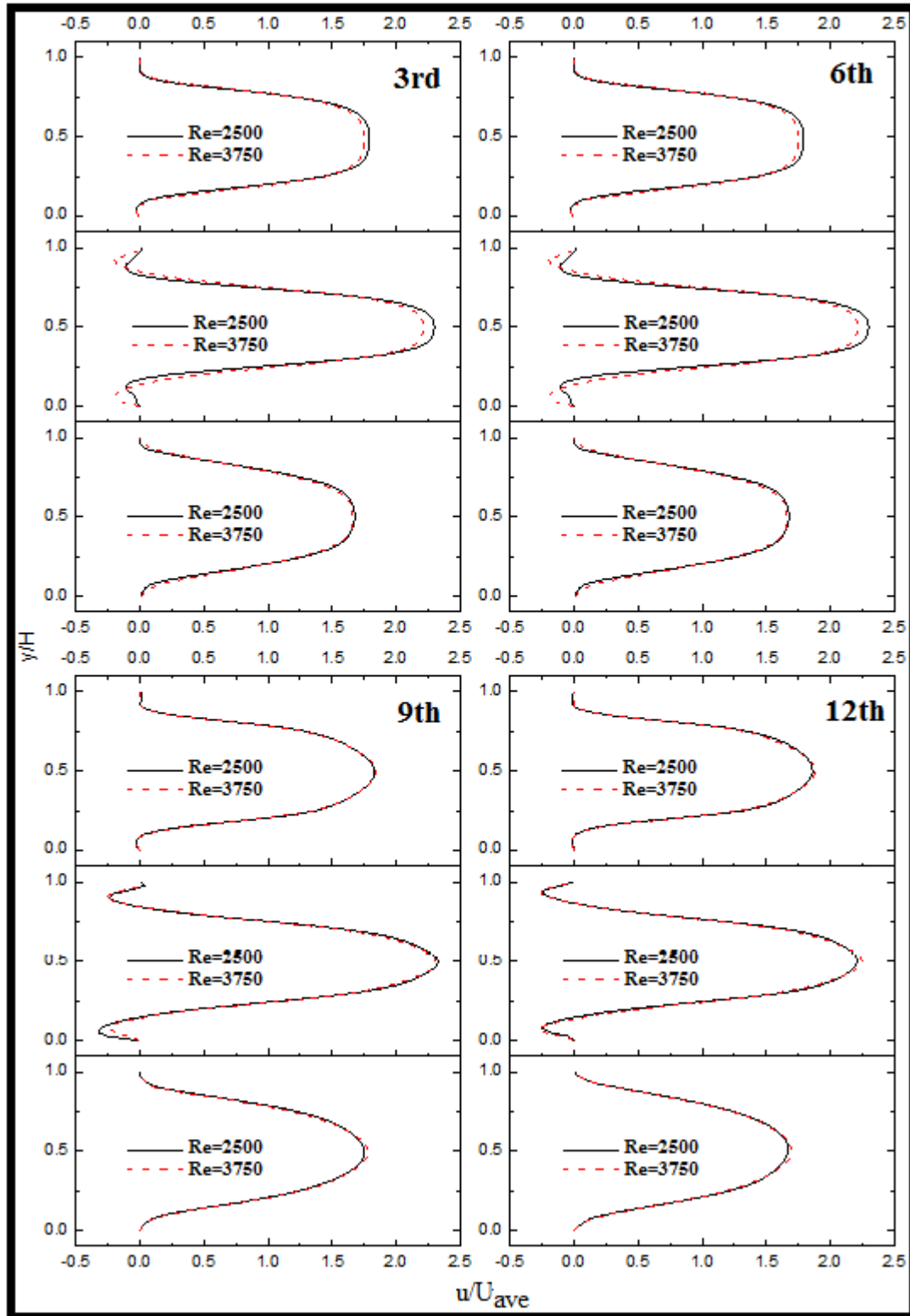


Figure 4.11. Comparisons of time-averaged normalized streamwise velocities, $\langle u/U_{ave} \rangle$ at cross-sections of cavities, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree

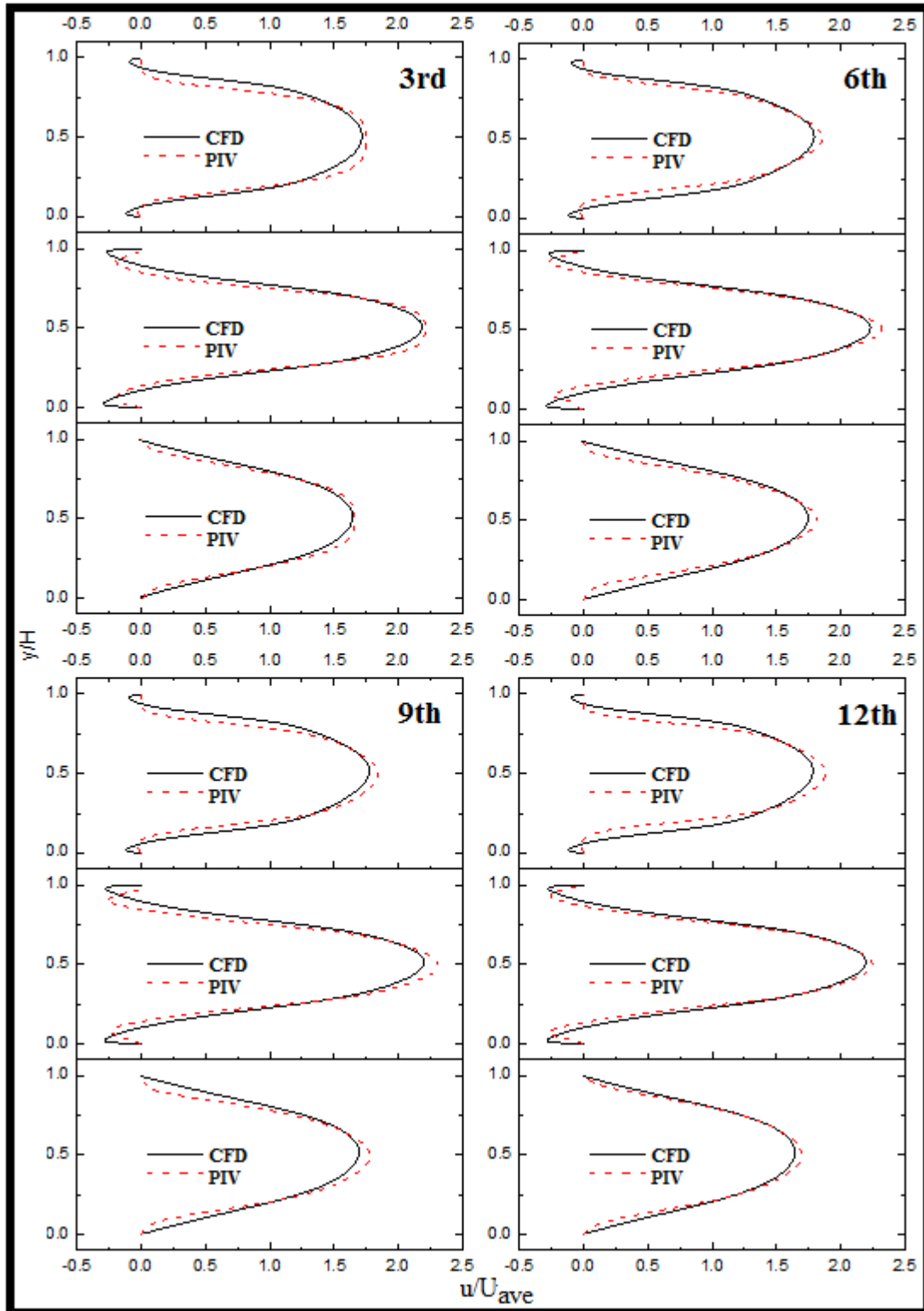


Figure 4.12. Comparisons of time-averaged normalized streamwise velocities, $\langle u/U_{ave} \rangle$ at cross-sections of cavities, at $Re=3750$, channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree

When the boundary layer portion closest to the sinusoidal wall reverses in flow direction, boundary layer separation occurs because of pressure recovery. Namely, separation occurs due to an adverse pressure gradient encountered as the flow expands, causing an extended region growing of separated flow. Recirculating flow is developed in between line of zero velocity and wall surface as seen in figure 4.13. Reattachment point is the point where the dividing streamline attaches to the wall again. The flow finally achieves an equilibrium state and has no reverse flow as the fluid moves further downstream region. Figure 4.13 represents generally the flow separation over a curved surface.

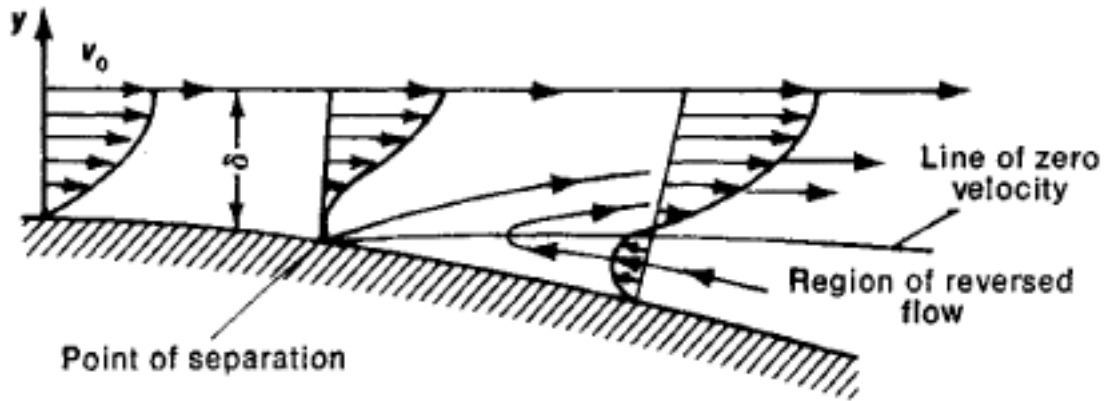


Figure 4.13. Separation of flow over a curved surface

The experimental time-averaged streamwise normalized velocity profiles at Reynolds numbers 2500 and 3750 for channel with the phase angle, ϕ of 90 degree are presented in figure 4.16. Figure 4.14 shows the three cross sections for 3rd, 6th, 9th, and 12th waves of the sinusoidal wave channel with phase angle of 90°. The lengths of the section 1 and 3 are equal, but the length of the section 2 is not. The velocity profiles at sections 1, 2, and 3 are shown in figure 4.16 from top to bottom for four waves.

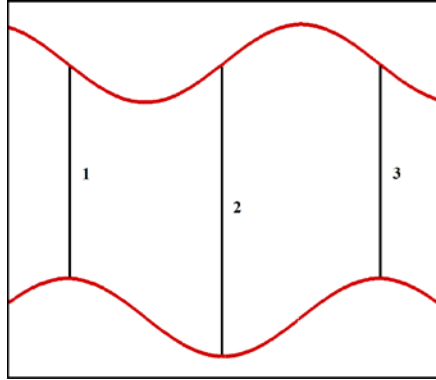


Figure 4.14. Three cross-sections for channel with the phase angle, ϕ of 90 degree

In figure 4.16, negative velocity values are seen at the upper regions of the sections 1 and 3 for all waves. On the other hand, negative velocities are taken place at both upper and lower regions of section 2 due to the reversed flow occurred in cavities at the upper and lower portions of the section 2. Although the upper and lower parts of section 2 have a negative velocity values, the magnitude of the negative values at lower regions are higher since the channel is not symmetrical when the channel axis is taken as a reference line and the distances between the cavity walls and the core region are higher at the bottoms. In lower cavities of the channel with 90° phase angle, separation is developed covering the region between sections 1 and 2 and negative velocity values are seen until reattachment point of the flow which occurs between sections 2 and 3. On the other hand, since the upper parts of sections 1 and 3 are in the reversed flow region, negative velocity values are seen at the cross sections 1 and 3. Since bottom part of the section 2 far away from the separation point, the magnitude of negative velocity values at the bottom of the section 2 is larger than that of the upper parts.

Experimentally measured time-averaged streamwise normalized velocity profiles at Reynolds numbers 2500 and 3750 through the channel with phase angle, ϕ of 0 degree are presented in figure 4.17. Figure 4.15 shows the three cross sections for 3rd, 6th, 9th, and 12th waves of the sinusoidal wave channel with phase angle of 0° . The lengths of sections 1, 2, and 3 are equal. The velocity profiles at sections 1, 2, and 3 are shown in figure 4.17 for four waves.

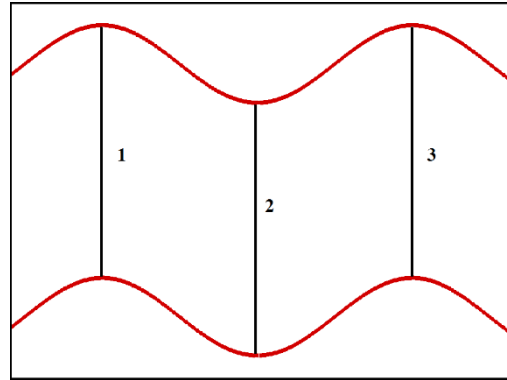


Figure 4.15. Three cross-sections for channel with the phase angle, ϕ of 0 degree

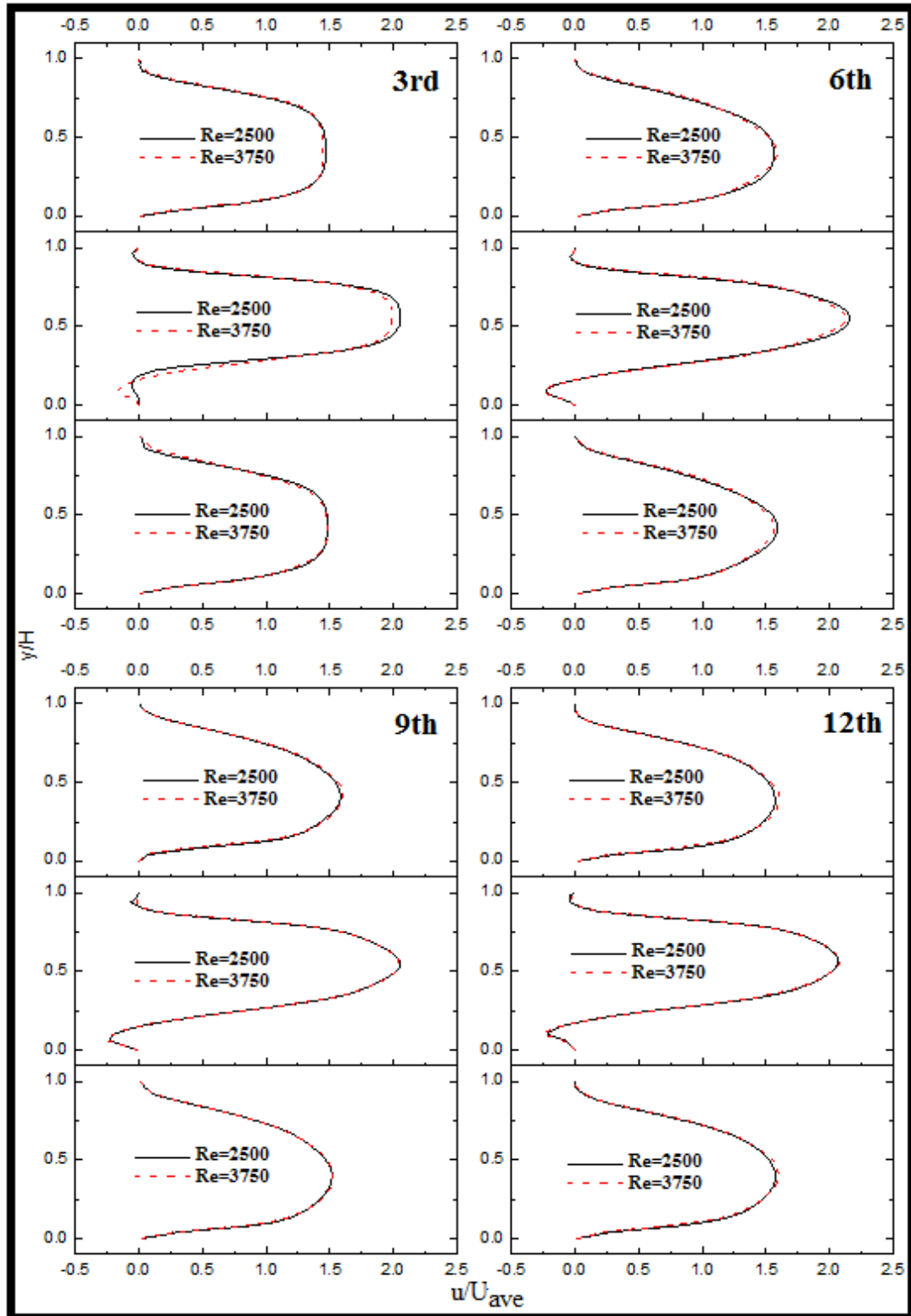


Figure 4.16. Comparisons of time-averaged streamwise velocity components, $\langle u \rangle$ at cross-sections of cavities, at $Re=3750$, channel $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 90 degree

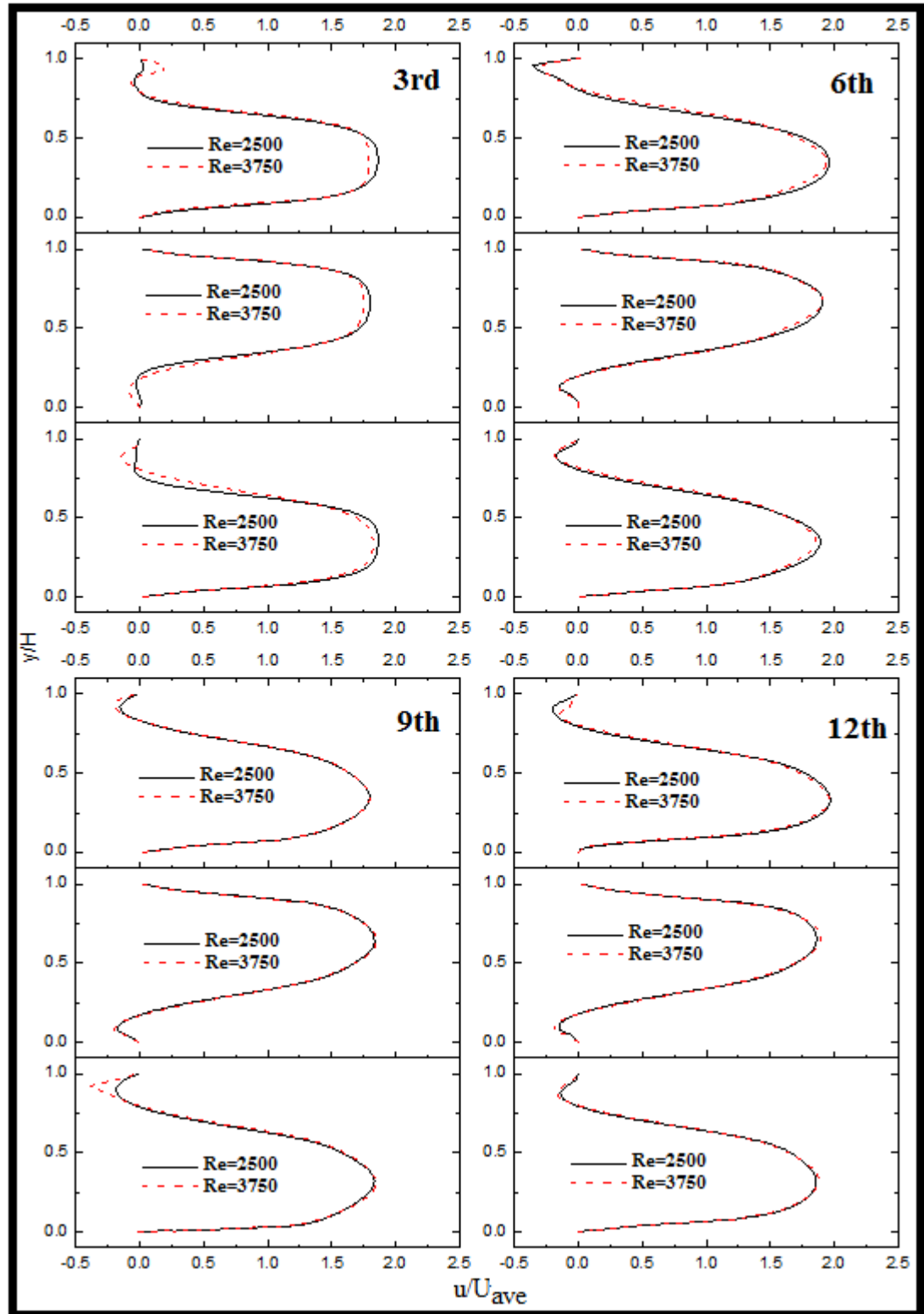


Figure 4.17. Comparisons of time-averaged streamwise velocity components, $\langle u \rangle$ at cross-sections of cavities, at $Re=3750$, with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 0 degree

As it can be seen in figure 4.17, the time-averaged normalized velocity profiles are similar for both Reynolds numbers, Re . In addition, similar velocity profiles are taken place at sections 1 and 3 since lengths and locations of the sections 1 and 3 in cavities are the same. The negative velocity values are seen in the upper regions of sections 1 and 3 for all waves. On the other hand, negative velocities are taken place in the lower regions of section 2 due to the reversed flow in cavities. The velocity profiles of section 2 and sections 1, 3 are almost symmetrical about the horizontal line passing at the point where velocity magnitude is maximum because the upper wall is shifted half wave forward in the flow direction with respect to the 180° case. In lower cavities of the channel with 0° phase angle, separation occurs between sections 1 and 2 and the negative values are seen until reattachment of the flow which occurs between sections 2 and 3. On the other hand, the upper parts of sections 1 and 3 are in the region of reversed flow so negative velocity values are seen. Since the flow reattached before it arrives the top of waves, the negative velocities do not occur at the higher parts of the section 2 and the lower parts of sections 1 and 3.

4.1.2. Streamline Topologies of Corrugated Channels

Streamlines can be described as a curve that is tangential to the velocity at every point in the flow at a given instant so they show the moving direction of the fluid particles at any point in time. The equation of the streamlines can be obtained according to this definition.

$$\frac{du}{u} = \frac{dv}{v} = \frac{dw}{w}$$

where u , v , and w are the velocity components in x , y , and z directions, respectively.

In steady flow, the position of the streamlines does not change. On the other hand, the pattern of the streamlines change from instant to instant in unsteady flow cases since the velocity vector changes with time. Fluid cannot cross a streamline

since they are moving in the same direction. In addition, streamlines cannot cross each other since there is no possibility to have two different velocities at the same point. Therefore, any fluid particle starts on a streamline must have to stay on the same streamline throughout the fluid.

In this section, the patterns of time-averaged streamlines, $\langle \Psi \rangle$ contours of experimental results and numerical predictions results are presented, and compared with each other. The patterns of time-averaged streamlines, $\langle \Psi \rangle$ contours are given for sinusoidal corrugated channel with different phase shift such as 180 degree, 90 degree and 0 degree. Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=2500$ and $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree are presented in figures 4.18 and 4.19, respectively. For both figures, the first columns present the streamline topologies, $\langle \Psi \rangle$ obtained from the experimental data whereas second columns show the numerical predictions. As mentioned in previous sections, the channel was divided into six field of views to get more detailed images. These field of views for channel with 180° phase angle are shown in figure 4.3. Field of views are presented consecutively starting from the first row until the last row. The flow is symmetrical since the plates are symmetrical with respect to the centerline of the channel, as in the case of $\phi=180^\circ$. Similar flow patterns were observed both experimentally and numerically. The streamline plot shows almost perfectly straight core flow and single trapped vortices having a counterclockwise rotation in the upper sinusoidal waves and clockwise rotation in the lower sinusoidal waves. These flow pattern in a wavy channel are expected for steady flow. Therefore, the cool fluid passing straight through the center of the channel does not significantly mix with the warmer fluid near the walls of the channel. Since the warmer fluid becomes isolated in the large recirculating vortices filling the upper and lower furrows of each waves, the heat transfer does not enhance considerably in steady flow cases. As the Reynold number increases, the recirculation magnitude gets smaller and fresh fluid into the core region moves close to the wave furrows so that heat transfer enhances.

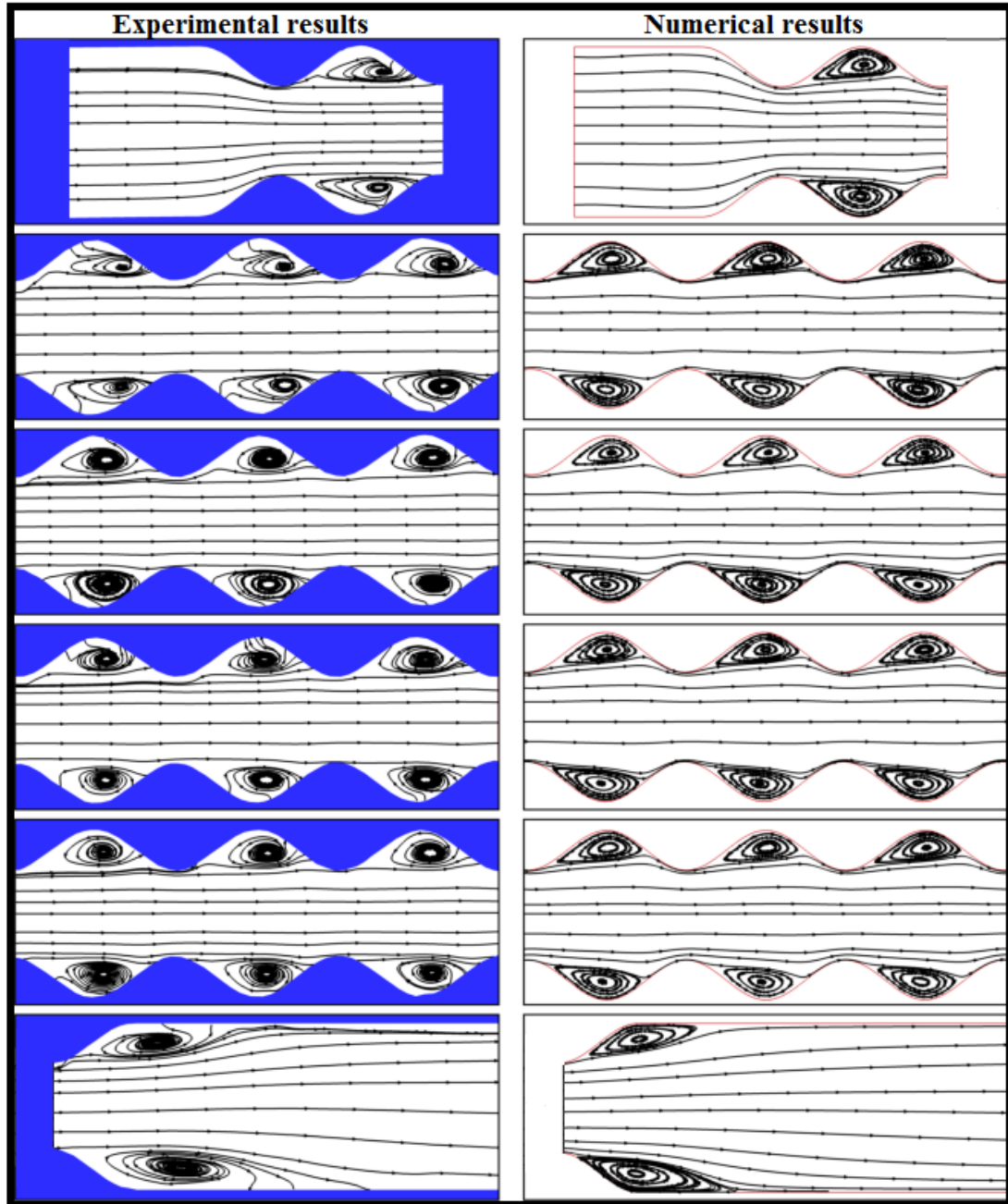


Figure 4.18. Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=2500$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree

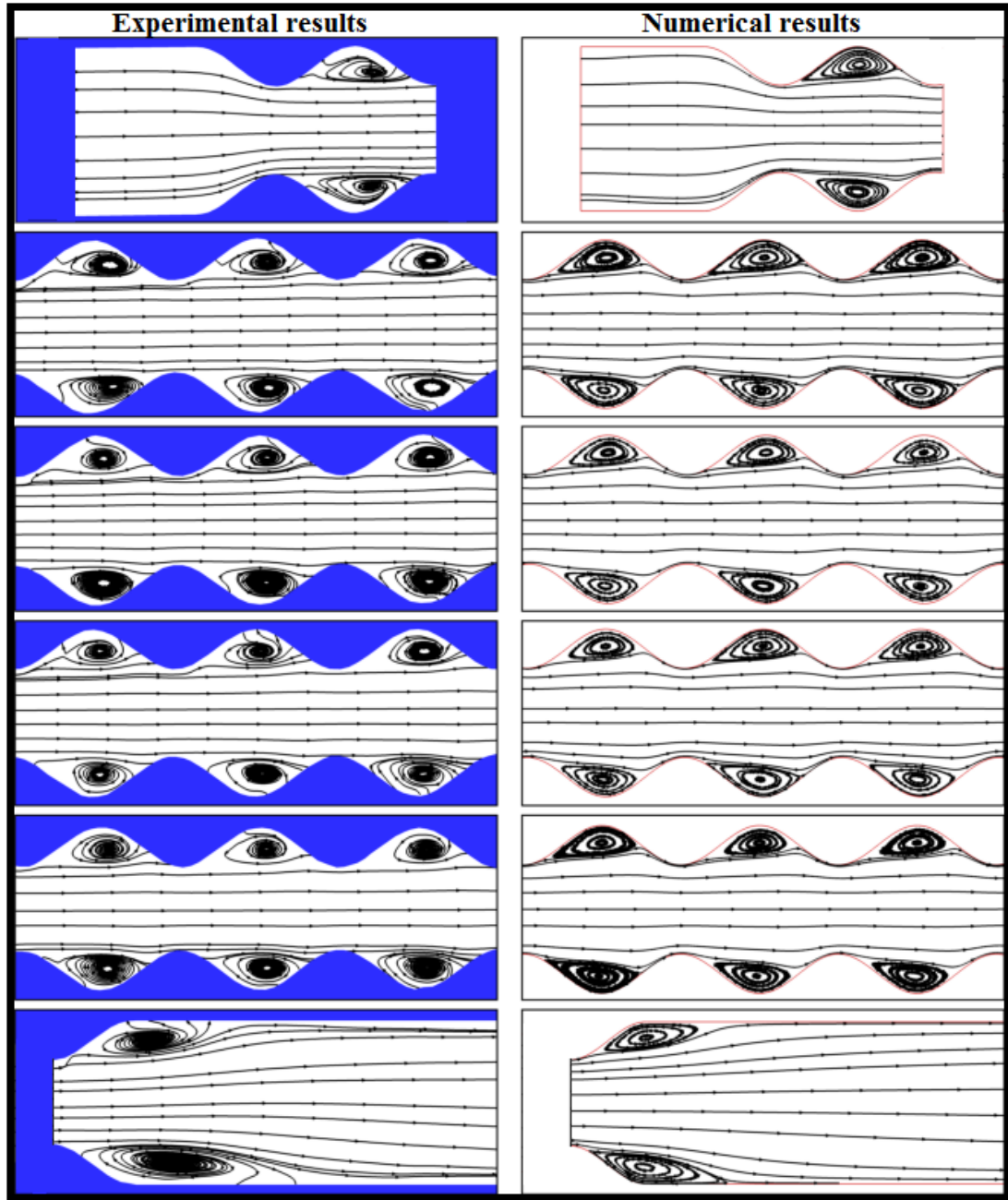


Figure 4.19. Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 180 degree

Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=3750$, through the channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 90 degree are presented in figure 4.20. The first columns of images present the streamline topologies, $\langle \Psi \rangle$ obtained from the experimental data whereas second columns of images show the numerical predictions. Unlike the 180° case, asymmetric flow pattern is observed when the

phase angle is 90° since the plates are not symmetrical with respect to the centerline. Similar flow patterns were observed for experimental and numerical results. Different flow patterns are seen when the phase angle changes. The directions of the recirculation in the upper and lower cavities are counterclockwise and clockwise, respectively. The locations of reattachment points changes with shifting the upper wall and mixing between the core flow and recirculation flow enhances.

Time-averaged streamline topologies, $\langle \Psi \rangle$ for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 0 degree at $Re=3750$ are presented in figure 4.21. It can be seen from the streamline figures for the phase angles, ϕ of 180° , 90° , and 0° , increase in phase angle becomes recirculation region smaller and more separation bubble regions formed in the adjacent outlet. The sinusoidal wall geometry on the core flow behavior becomes less effective due to the decrease in the distortion of streamlines. Hence, a rate of decrease in fluid mixing between the near-wall flow and main flow resulting in large changes in the location of the reattachment point of the free shear layers. The occurrence of separation is associated with large rates of dissipation of mechanical energy. But, the aim of divergent and convergent channel geometry is to stimulate fluid mixing process in regions between wall and central axis.

As mentioned in the previous sections, geometry of the heat exchangers are significantly affects the heat transfer enhancement. Flow characteristics in corrugated channels of heat exchangers are necessary for optimizing design parameters. The shape and dimensions of the corrugated walls, the distance between the upper and lower walls, and Reynolds number are main parameters to optimize the geometry and operation conditions. The flow in the sinusoidal corrugated channel has higher heat transfer rate compare to the straight channel since the wavy walls break the thermal boundary layer. In addition, the wavy walls operate as a turbulence promoters, increase the heat transfer area, and improve the mixing of the core fluid and near-wall fluid. Thus, understanding the flow patterns of corrugated channel flow is important to find optimum geometry design and operation conditions for best results.

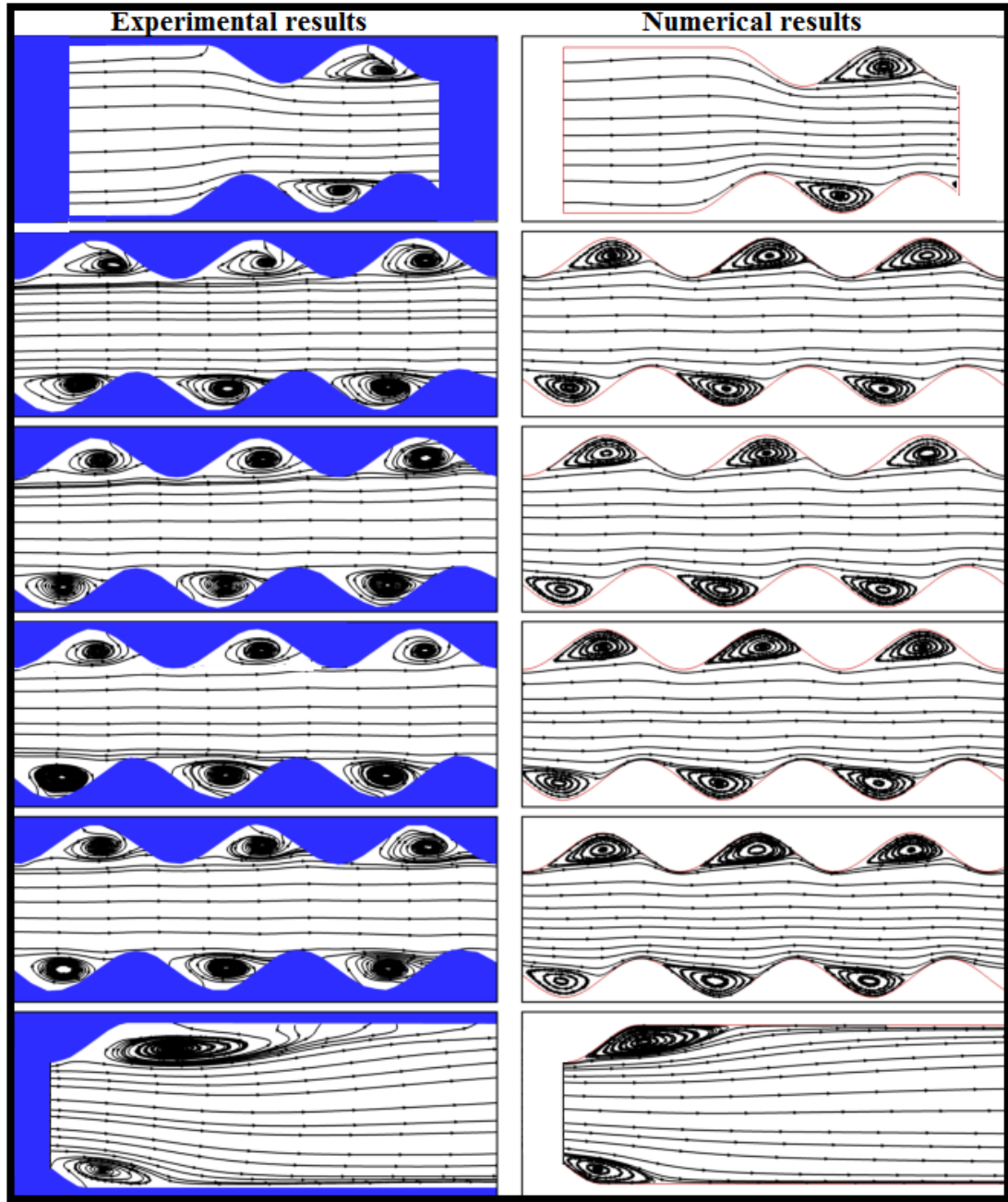


Figure 4.20. Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 90 degree

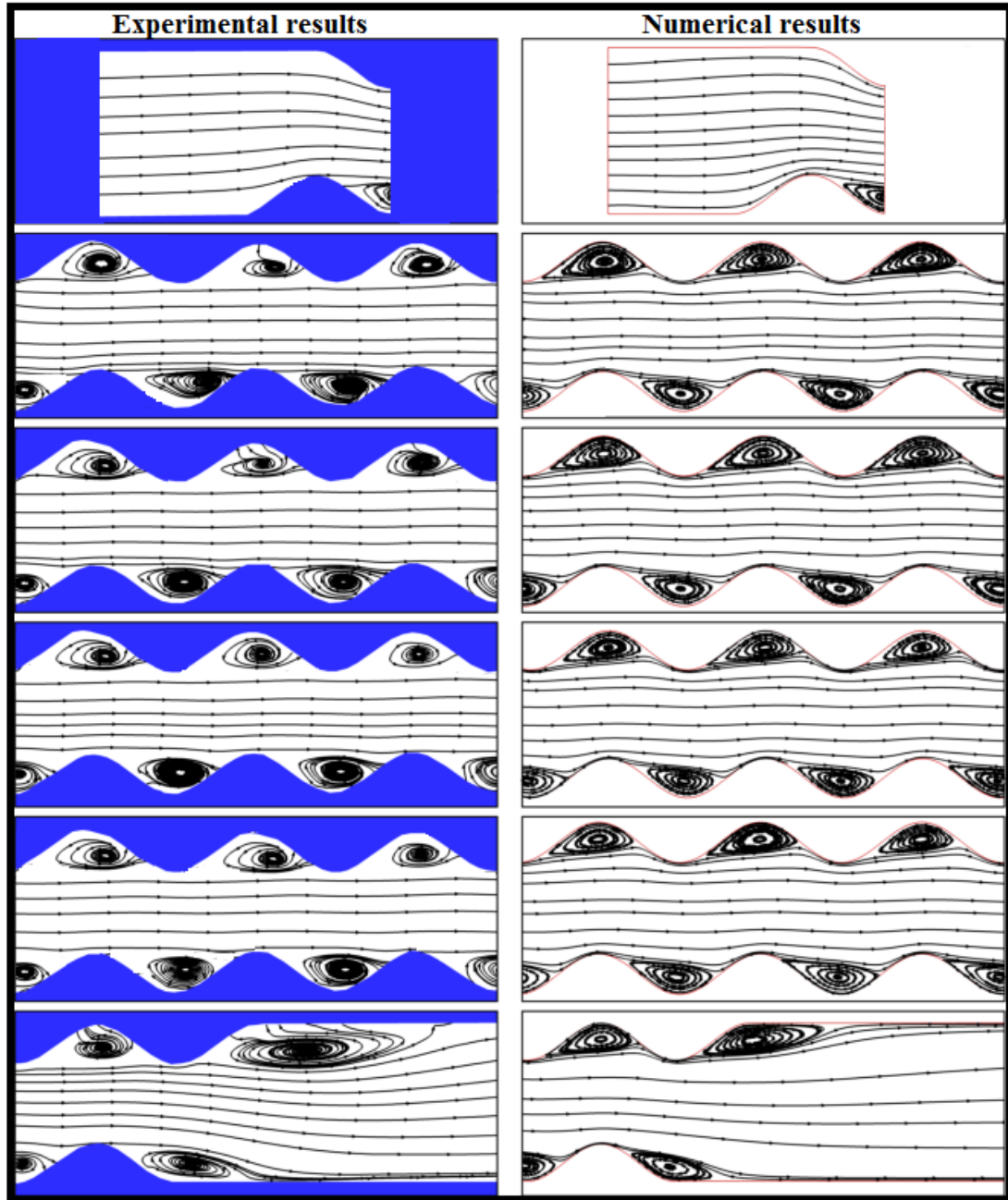


Figure 4.21. Time-averaged streamline topologies, $\langle \Psi \rangle$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 0 degree

4.1.3. Phase Shift and Reynolds Number Effects on Vorticity Contours

The knowledge of flow mechanism in the wake regions and shear layers of the flow become important to define optimum design parameters and operating conditions in order to improve thermal performance of the corrugated channels used in heat exchangers. The dynamics of vortices along shear layers between core and wake regions stimulate the exchanging of mass, momentum and heat transfer.

The time-averaged vorticity, $\langle \omega \rangle$ contours obtained from the results of the Particle Image Velocimetry (PIV) experiments for corrugated channels with three different phase angle are discussed in this section. The vorticity, $\langle \omega \rangle$ contours are presented for Reynolds numbers of 2500 and 3750 for the phase angles, ϕ 180°, 90°, and 0°.

The vorticity, ω , is an important quantity which is defined as the curl of the velocity field as follows:

$$\omega = \nabla \times u$$

The vorticity is a measure of the local swirl or rotation of the fluid. The vorticity counters are positive when it turns counter-clockwise and negative for clockwise rotation.

Since 1000 images were taken during one set of PIV experiment, the time-averaged vorticity, $\langle \omega \rangle$ contours are calculated by taking an average of 1000 instantaneous images. The time-averaged vorticity, $\langle \omega \rangle$ is calculated as follows:

$$\langle \omega \rangle = \frac{1}{N} \sum_{n=1}^N \omega_n(x, y)$$

Where N designates the number of the instantaneous flow data images taken during the experiments.

The time-averaged vorticity, $\langle \omega \rangle$ patterns at Reynolds numbers 2500 and 3750 for channel with $H_{\min}/H_{\max}=0.54$ and the phase angles, ϕ of 180°, 90°, and 0° are presented in figures 4.22, 4.23, and 4.35, respectively. As highlighted in the previous paragraph, vorticity is positive when it rotates counter-clockwise, and it has negative values for clockwise rotation. The positive vorticity contours are drawn with solid line whereas dashed line represents the negative vorticity contours. For all

corrugated channels, the minimum and the incremental values of the vorticity are taken as $\omega_{\min} = \pm 9 \text{ s}^{-1}$ and $\Delta\omega = 9 \text{ s}^{-1}$, respectively.

The first columns of figures 4.22, 4.23, and 4.24 show the time-averaged vorticity, $\langle\omega\rangle$ contours at the Reynolds number 2500, and the second columns of these figures represent the time-averaged vorticity, $\langle\omega\rangle$ contours at the Reynolds number 3750.

Similar time-averaged vorticity, $\langle\omega\rangle$ patterns are seen in figure 4.22 for all cavities. Positive vorticity contours are observed in upper parts of the channel while negative vorticities occur in lower parts. The vorticity layers begin at near wall of the narrowest section of the wave and expand in the flow direction along the shear layer until the next wave. The magnitude of the vorticities become highest near the upper and lower walls at the channel throats because of the interactions between the shear layers and the walls.

The effects of Reynolds number on the time-averaged vorticity, $\langle\omega\rangle$ contours are investigated by comparing the vorticities at Reynolds numbers of 2500 and 3750. The time-averaged vorticity, $\langle\omega\rangle$ contours have similar pattern for both Reynolds numbers. On the other hand, the magnitude of the vorticities are higher for $\text{Re}=3750$. Development of the vortices in the furrows improves the mixing of the wake and core flow resulting in heat transfer enhancement.

For 90° and 0° phase angle cases, the positive and negative vorticity contours lie along the upper and lower walls as occurred in the case of 180° phase angle. The magnitude of the vortices developed near the walls at the wave throats increase with increasing Reynolds number.

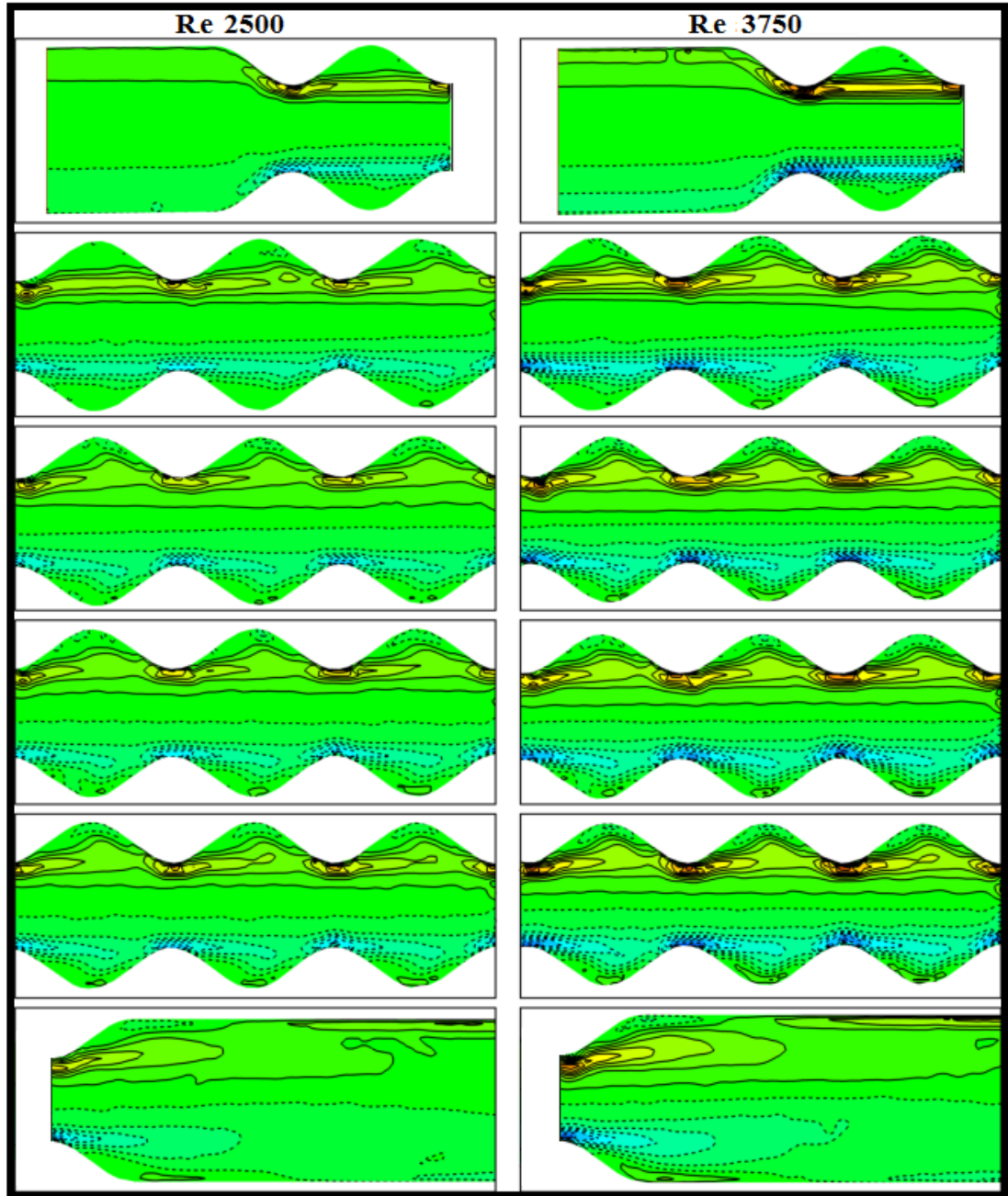


Figure 4.22. Patterns of time-averaged vorticity, $\langle \omega \rangle$ for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 180 degree. Minimum and incremental values of vorticity are $\omega_{\min}=\pm 9 \text{ s}^{-1}$ and $\Delta\omega=9 \text{ s}^{-1}$, respectively

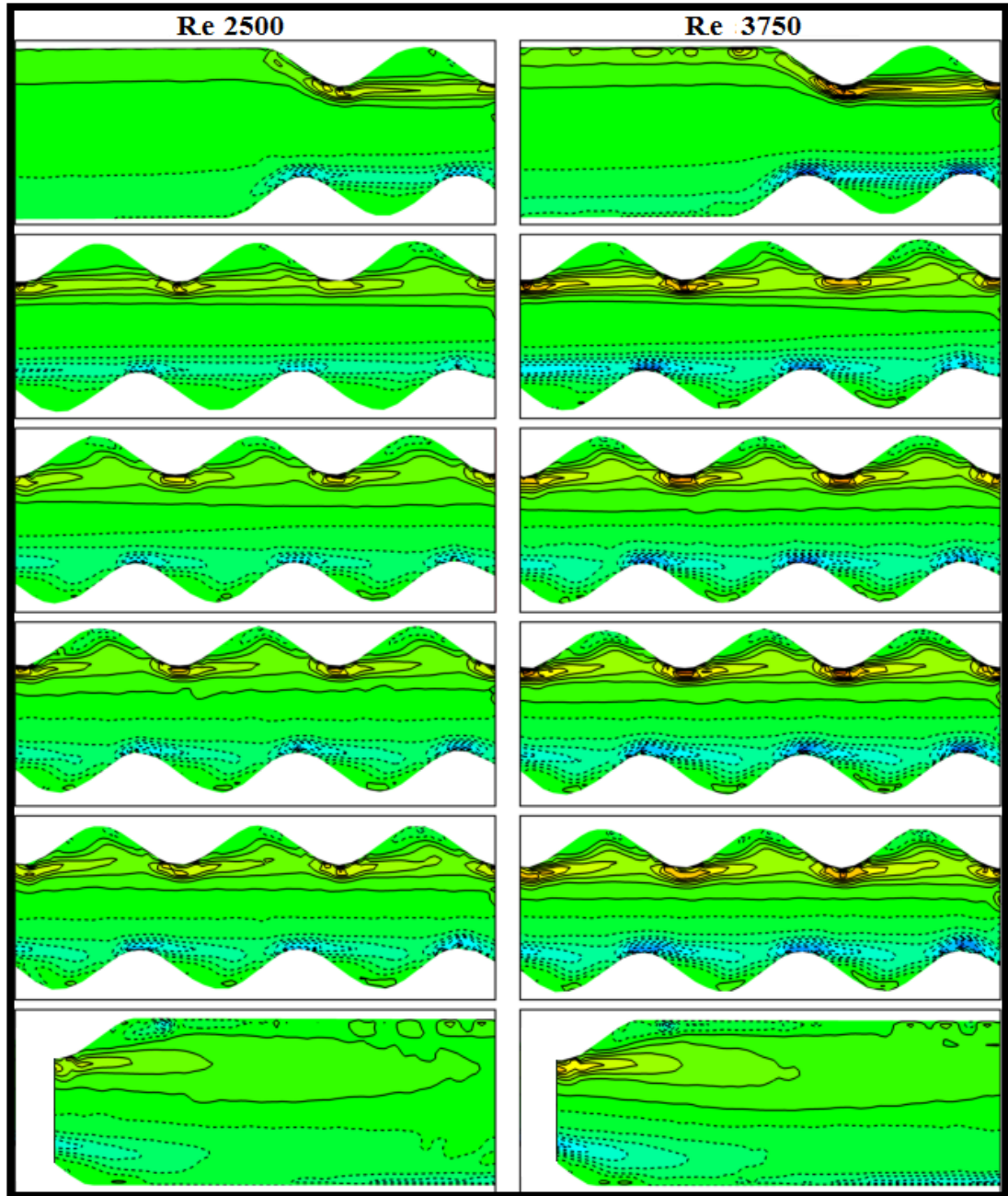


Figure 4.23. Patterns of time-averaged vorticity, $\langle \omega \rangle$ for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 90 degree. Minimum and incremental values of vorticity are $\omega_{\min} = \pm 9 \text{ s}^{-1}$ and $\Delta\omega = 9 \text{ s}^{-1}$, respectively

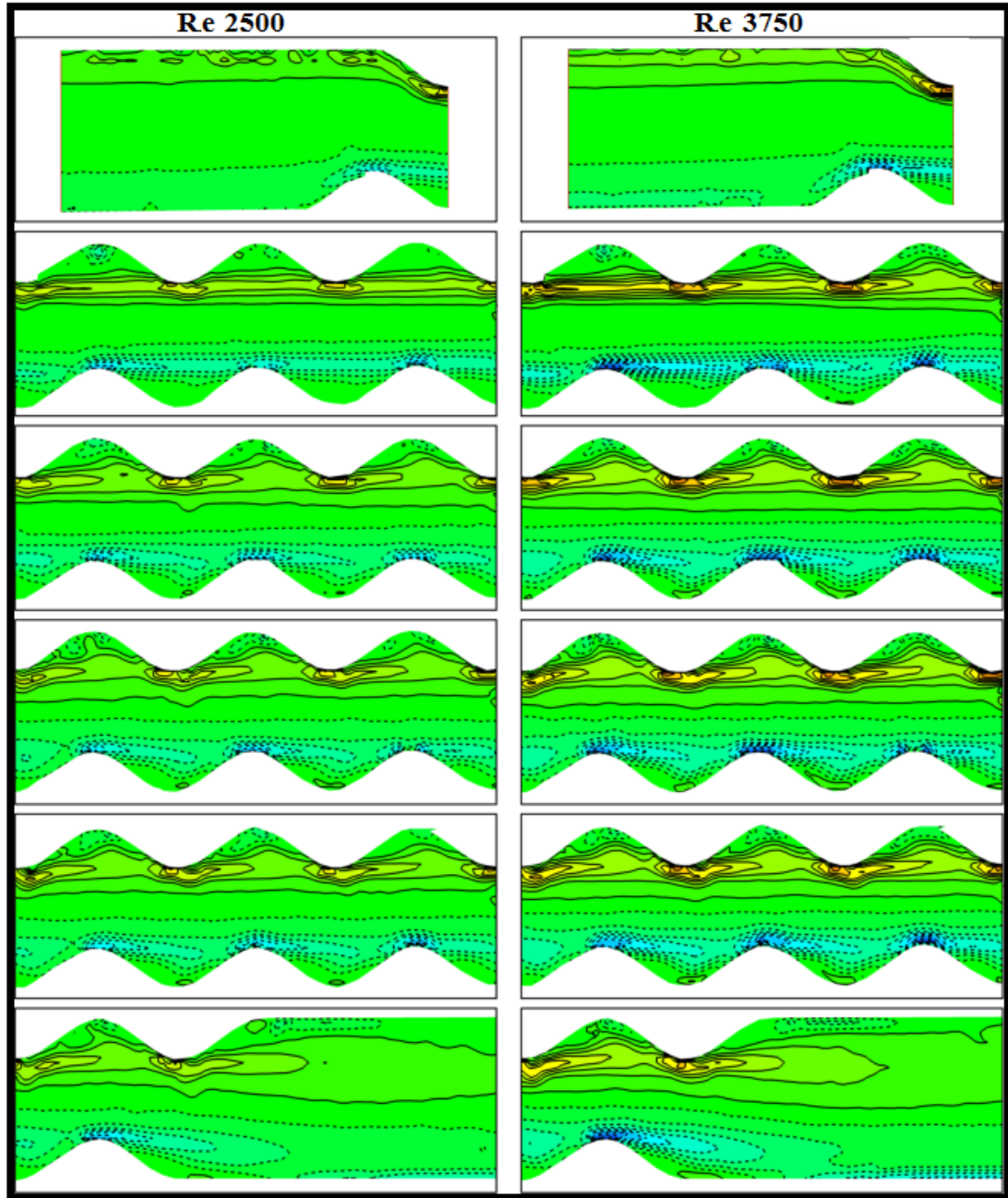


Figure 4.24. Patterns of time-averaged vorticity, $\langle \omega \rangle$ for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 0 degree. Minimum and incremental values of vorticity are $\omega_{\min} = \pm 9 \text{ s}^{-1}$ and $\Delta\omega = 9 \text{ s}^{-1}$, respectively

4.1.4. Reynolds streamwise normal stress, Reynolds transverse normal stress, and Turbulent Kinetic Energy Contours of Corrugated Channels

In this section, Reynolds normal stress, $\langle u'u' \rangle$ in streamwise direction and Reynolds normal stress, $\langle v'v' \rangle$ in transverse direction contours obtained from the results of the Particle Image Velocimetry (PIV) technique are discussed. Distributions of streamwise Reynolds normal stress, $\langle u'u' \rangle$ and transverse Reynolds normal stress, $\langle v'v' \rangle$ contours at Reynolds number 3750 for channels with $H_{\min}/H_{\max} = 0.54$ and the phase angles, ϕ of 180° , 90° , and 0° are presented in figures 4.25, 4.26, and 4.27, respectively. The first columns of the figures 4.25, 4.26, 4.27 show streamwise Reynolds normal stress, $\langle u'u' \rangle$ contours and, the second columns of these figures represent transverse Reynolds normal stress, $\langle v'v' \rangle$ contours.

The values of streamwise Reynolds normal stress, $\langle u'u' \rangle$ and transverse Reynolds normal stress, $\langle v'v' \rangle$ were calculated using the 1000 instantaneous velocity data. The third fluctuating velocity component was estimated by taking the half of the sum of Reynolds streamwise normal stress, $\langle u'u' \rangle$ and Reynolds transverse normal stress, $\langle v'v' \rangle$ (Sheng et al. 2000).

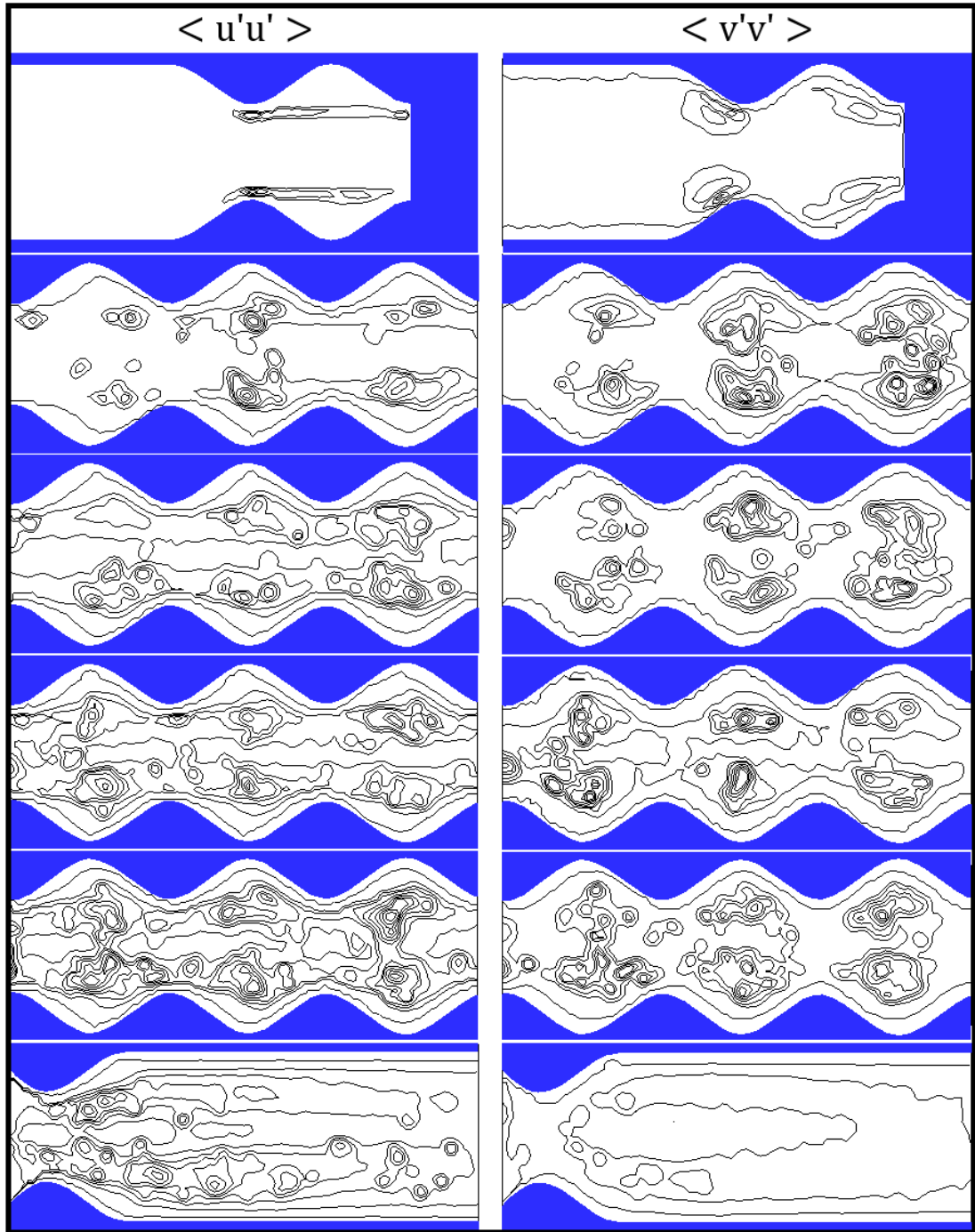


Figure 4.25. Patterns of streamwise Reynolds normal stress $|\langle (u'u')_{\min} \rangle| = 0.01$, $\langle \Delta(u'u') \rangle = 0.04$, transverse Reynolds normal stress $|\langle (v'v')_{\min} \rangle| = 0.01$, $\langle \Delta(v'v') \rangle = 0.04$ at $Re=3750$, for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 180 degree

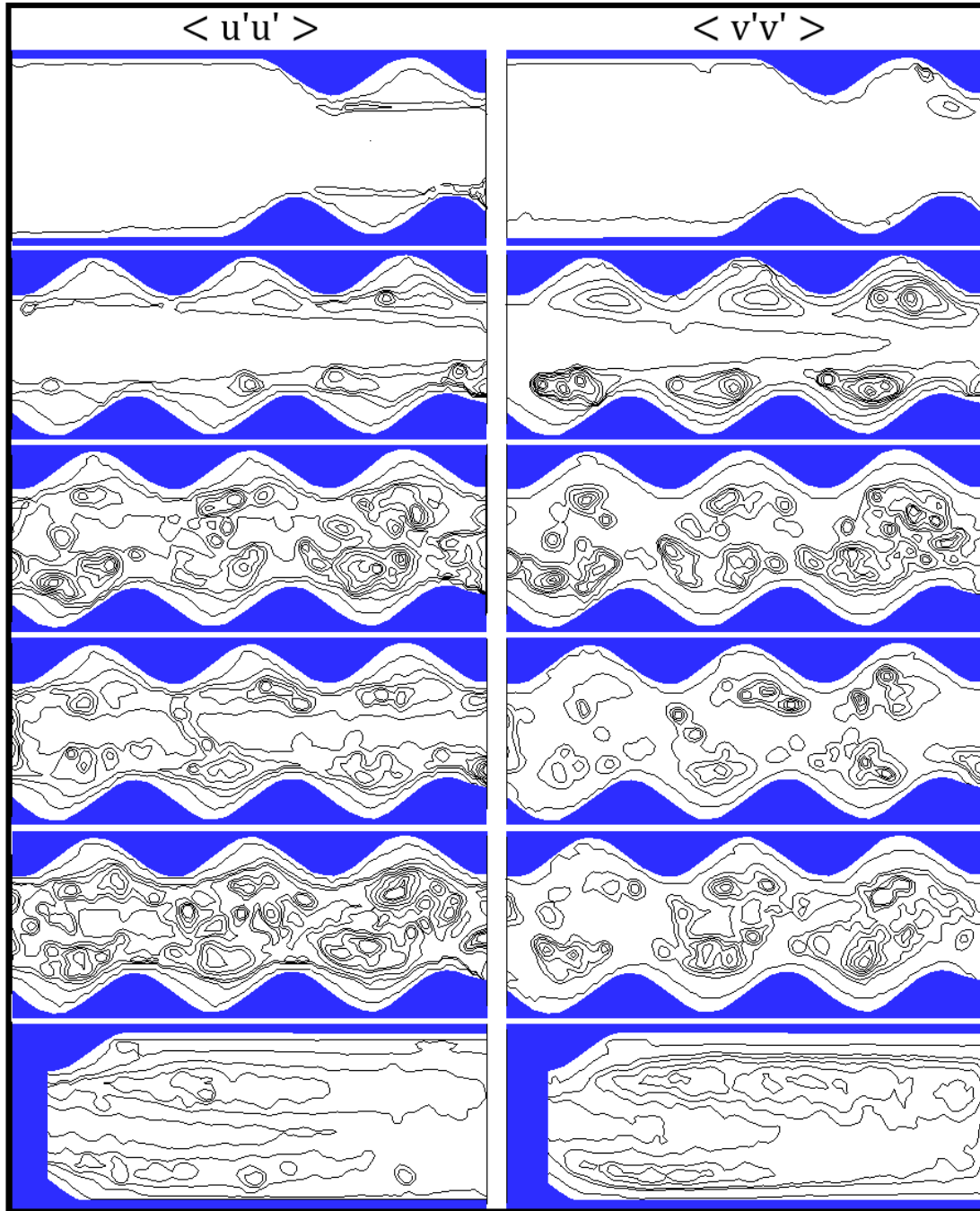


Figure 4.26. Patterns of streamwise Reynolds normal stress $|\langle u'u' \rangle_{\min}| = 0.01$, $\langle \Delta(u'u') \rangle = 0.04$, transverse Reynolds normal stress $|\langle v'v' \rangle_{\min}| = 0.01$, $\langle \Delta(v'v') \rangle = 0.04$ at $Re=3750$, for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 90 degree

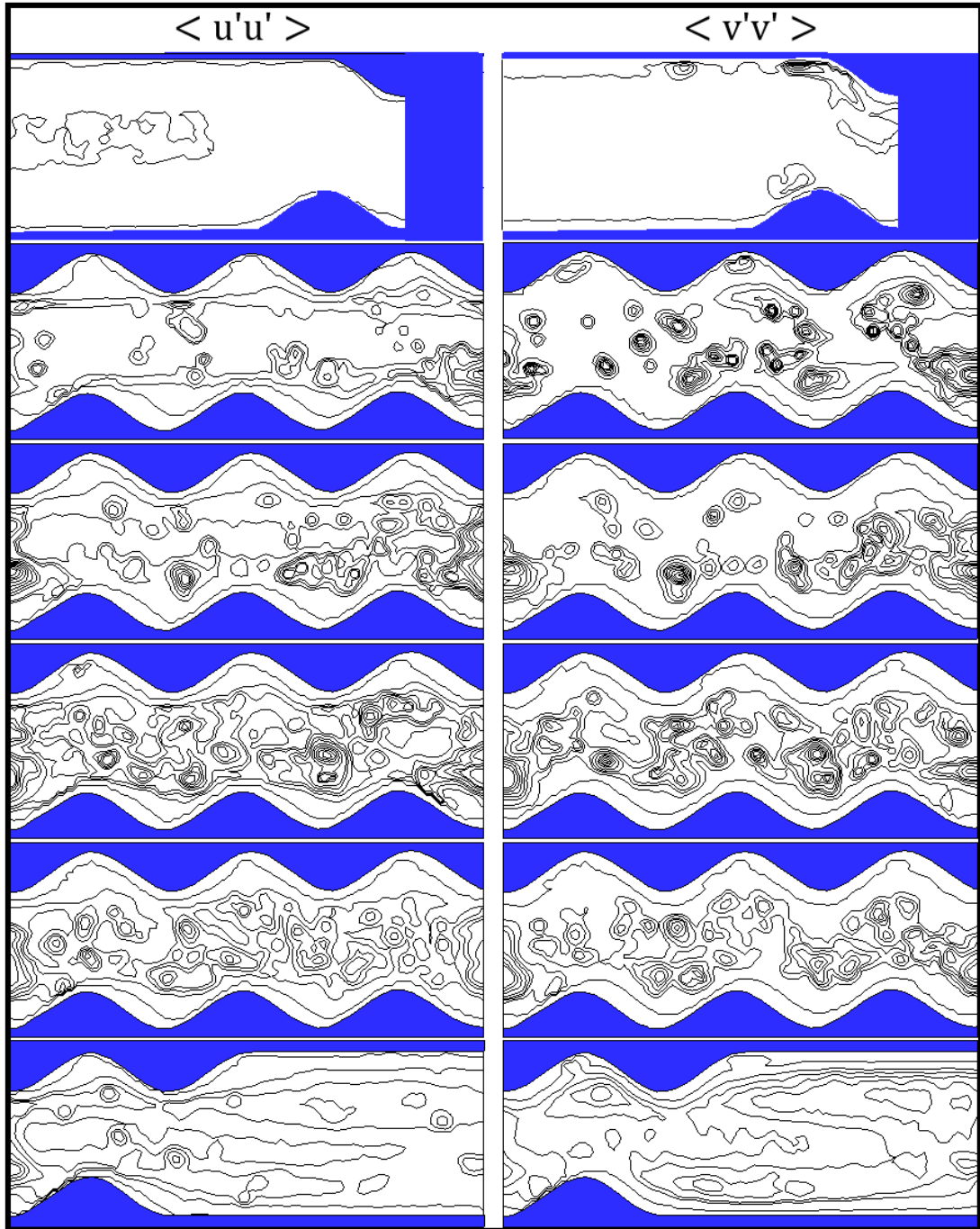


Figure 4.27. Patterns of streamwise Reynolds normal stress $|\langle u'u' \rangle_{\min}| = 0.01$, $\langle \Delta(u'u') \rangle = 0.04$, transverse Reynolds normal stress $|\langle v'v' \rangle_{\min}| = 0.01$, $\langle \Delta(v'v') \rangle = 0.04$ at $Re=3750$, for channel with $H_{\min}/H_{\max}=0.54$ and phase angle, ϕ of 0 degree

Turbulent kinetic energy, TKE contours for Reynolds number 3750 through channels with $H_{\min}/H_{\max} = 0.54$ and the phase angles, ϕ of 180° , 90° , and 0° are presented in figures 4.28, 4.29, and 4.30, respectively. On the other hand, figure 4.31 shows the turbulent kinetic energy, TKE contours at the Reynolds number 3750 for channel with $H_{\min}/H_{\max} = 0.72$ and the phase angles, ϕ of 180° . Same as previous sections, six field of views representing the turbulence kinetic energy, TKE contours of cavities are presented consecutively starting from the first cavity of the channel until the last cavity of the channel.

In fluid dynamics, turbulent kinetic energy, TKE is defined as the mean kinetic energy per unit mass associated with eddies in turbulent flow. Turbulent kinetic energy, TKE is the important parameter for the heat transfer applications since it is a measure of turbulence intensity which is directly related to the heat transfer in boundary layers. In turbulent flow, eddies exhibit additional transverse motion which enhancing the rate of energy and momentum exchange between them. Therefore, heat transfer and the friction coefficient increase with turbulent flow. The flow in the boundary layer becomes more turbulent when the turbulent kinetic energy, TKE increases with time.

Turbulent kinetic energy, TKE and the momentum transfer rate increases as a function of Reynolds number, Re . Reynolds number is defined as the ratio of the inertial forces to viscous forces. Hence, more fluid fluctuations are observed at a higher Reynolds number due to the weaker viscous forces. In addition, Nusselt number, Nu which is defined as the ratio of convective to conductive heat transfer across the boundary increases with the Reynolds number, Re due to the more active convection. On the other hand, additional pressure drop occurs with the Reynolds number, Re . Therefore, thermal performance factor, TPF should be taken into account in order to define optimum operating conditions.

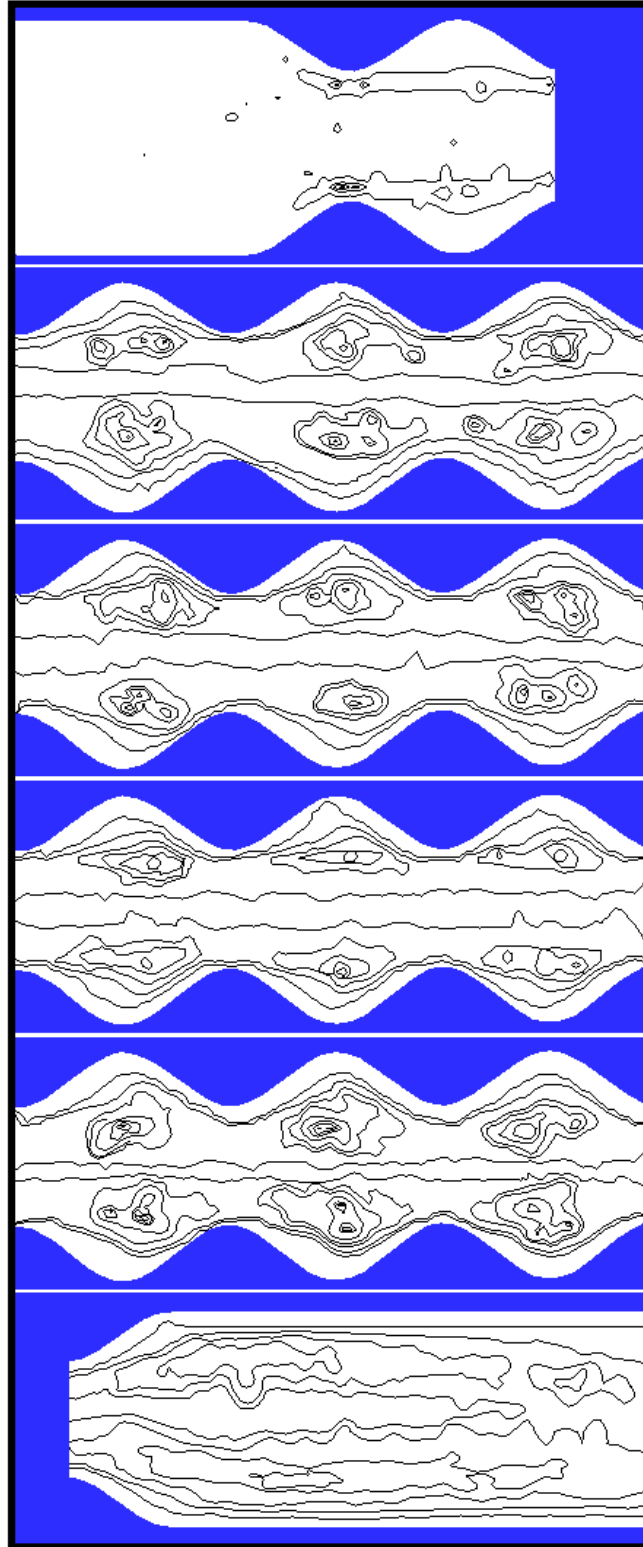


Figure 4.28. Patterns of Turbulent kinetic energy $|\langle \text{TKE} \rangle_{\min}| = 0.01$, $\langle \Delta(\text{TKE}) \rangle = 0.04$ at $\text{Re} = 3750$, for channel with $H_{\min}/H_{\max} = 0.54$ and phase angle, ϕ of 180 degree

In this study turbulent kinetic energy, TKE is calculated as follow:

$$TKE = \frac{3}{4} (\langle u'u' \rangle + \langle v'v' \rangle)$$

Where $\langle u'u' \rangle$ and $\langle v'v' \rangle$ are streamwise Reynolds normal stress and transverse Reynolds normal stress, respectively.

The contours of dimensionless turbulent kinetic energy, $\langle TKE \rangle$ are presented in figure 4.28 for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angles, ϕ of 180° at the Reynolds number 3750. Minimum and incremental values are taken as 0.01 and 0.04, respectively. Dimensionless turbulent kinetic energy, $\langle TKE \rangle$ contours are almost symmetrical with respect to the central axis since the upper and lower walls are symmetric to each other. Turbulent kinetic energy, $\langle TKE \rangle$ increases as the flow goes from the first cavity to last cavity due to the sinusoidal channel wall shape. The maximum turbulent kinetic energy, $\langle TKE \rangle$ values occur along the shear layers at the channel throats since the flow velocity increase due to the converging parts. On the other hand, the minimum turbulent kinetic energy, $\langle TKE \rangle$ values are observed in the wake regions. As can be seen in figure 4.28, the size of the core region where the lower values of TKE take place decreases until the 6th cavity. The flow after the 6th cavity has moderate TKE values in the core region for all the waves. Higher TKE contours are seen in the last cavities of the channel and they expands after corrugations. These results show that turbulent kinetic energy, $\langle TKE \rangle$ in the corrugated channel increases in the flow direction.

It can be seen from the TKE equation given above, TKE is dependent on both streamwise Reynolds normal stress, $\langle u'u' \rangle$ and transverse Reynolds normal stress, $\langle v'v' \rangle$. On the other hand, figures 4.25, 4.26, and 4.27 show that TKE contours are strongly dependent on streamwise Reynolds normal stress, $\langle u'u' \rangle$. This results indicate that the side walls did not affect the core flow regions especially in streamwise direction at the center of the channel since the channel manufactured wide enough and the laser sheet passes through the midplane of the channel.

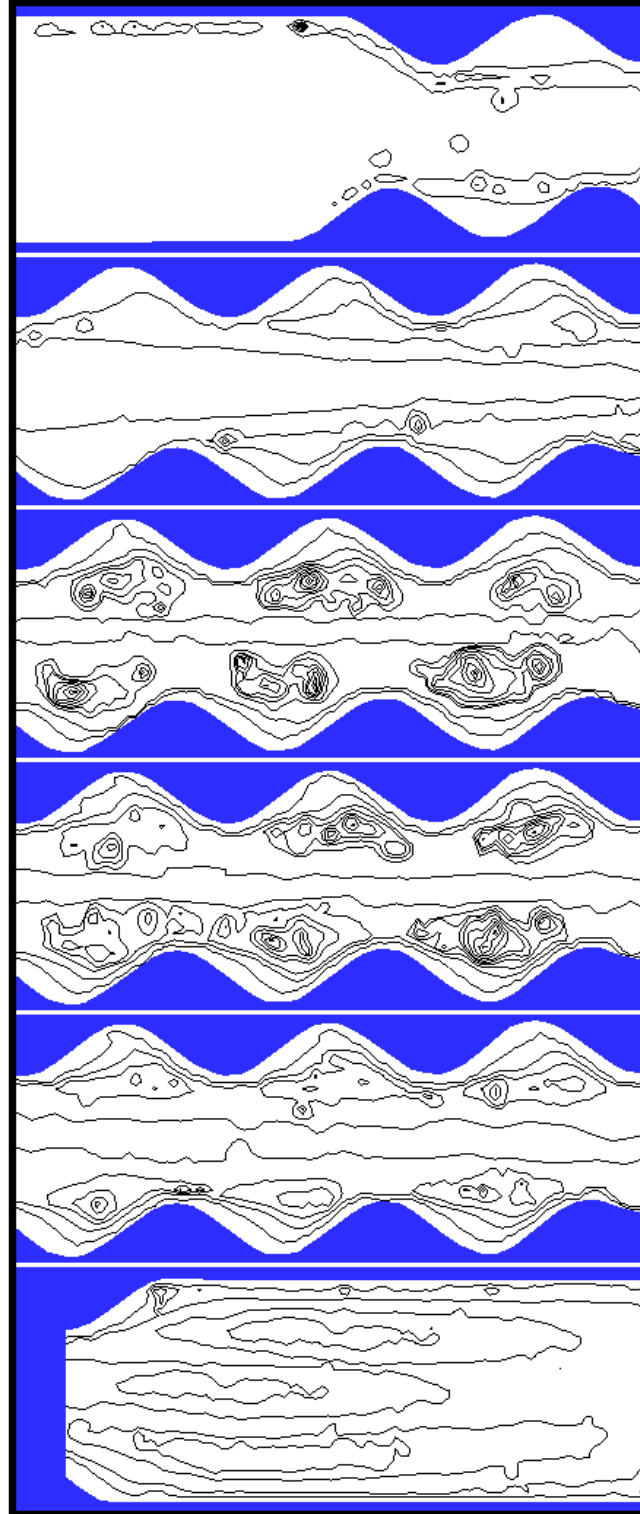


Figure 4.29. Patterns of Turbulent kinetic energy $|\langle (TKE)_{min} \rangle| = 0.01$, $\langle \Delta(TKE) \rangle = 0.04$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 90 degree

The contours of dimensionless turbulent kinetic energy, $\langle \text{TKE} \rangle$ are presented in figure 4.29 for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angles, ϕ of 90° at Reynolds number 3750. Minimum and incremental values are taken as 0.01 and 0.04, respectively. Turbulent kinetic energy, $\langle \text{TKE} \rangle$ values are lower at the inlet of the channel, and increases in the flow direction due to the effects of corrugations. TKE values at the center of the core region has smaller than TKE values at the shear layers that occur between core and wake flow regions. The size of the lower values core region decreases from inlet until 6th cavity. Highest TKE contours are clearly seen in the 5th FOV, and their sizes expand in the 6th FOV. The magnitude of TKE in the channel with 90° phase shift, ϕ is higher than that in the channel with 180° phase shift.

Figure 4.30 shows the turbulent kinetic energy, $\langle \text{TKE} \rangle$ contours for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angles, ϕ of 0° at the Reynolds number 3750. Minimum and incremental values are taken as 0.01 and 0.04, respectively. As previously stated that turbulent kinetic energy, $\langle \text{TKE} \rangle$ increases as the flow goes from the first cavity to last cavity of the sinusoidal corrugated channels. Very low magnitude of TKEs are observed in the core region along the center of the first five cavities. These regions become narrower in the flow direction, and higher value contours take place at the center after the 5th wave. Higher value of TKEs are observed at the inlet and outlet of the FOV 3, FOV 4, FOV 5, and FOV 6.

The contours of dimensionless turbulent kinetic energy, $\langle \text{TKE} \rangle$ are presented in figure 4.31 for the channel with $H_{\min}/H_{\max} = 0.72$ and the phase angles, ϕ of 180° at the Reynolds number, $Re=3750$. Minimum and incremental values are taken as 0.01 and 0.04, respectively. Since the magnitude of velocity is lower and the distance between the upper and lower walls is greater, the magnitude of TKE is much lower than those of the previous cases as can be easily seen in figure 4.31. Higher value TKE contours occur along the shear layers in between core and wake flow regions and expand towards the center of the channel as the flow goes from inlet to outlet.

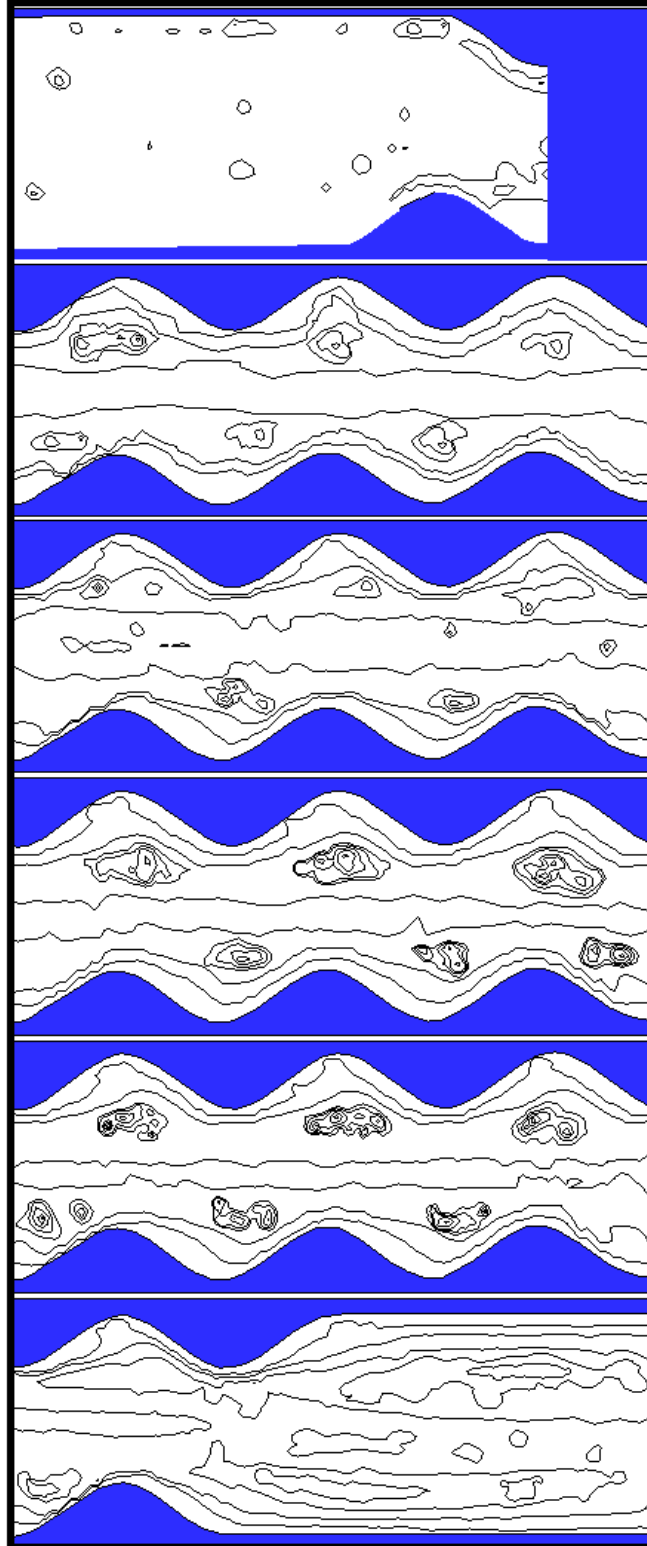


Figure 4.30. Patterns of Turbulent kinetic energy $|\langle (TKE)_{min} \rangle| = 0.01$, $\langle \Delta(TKE) \rangle = 0.04$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.54$ and phase angle, ϕ of 0 degree

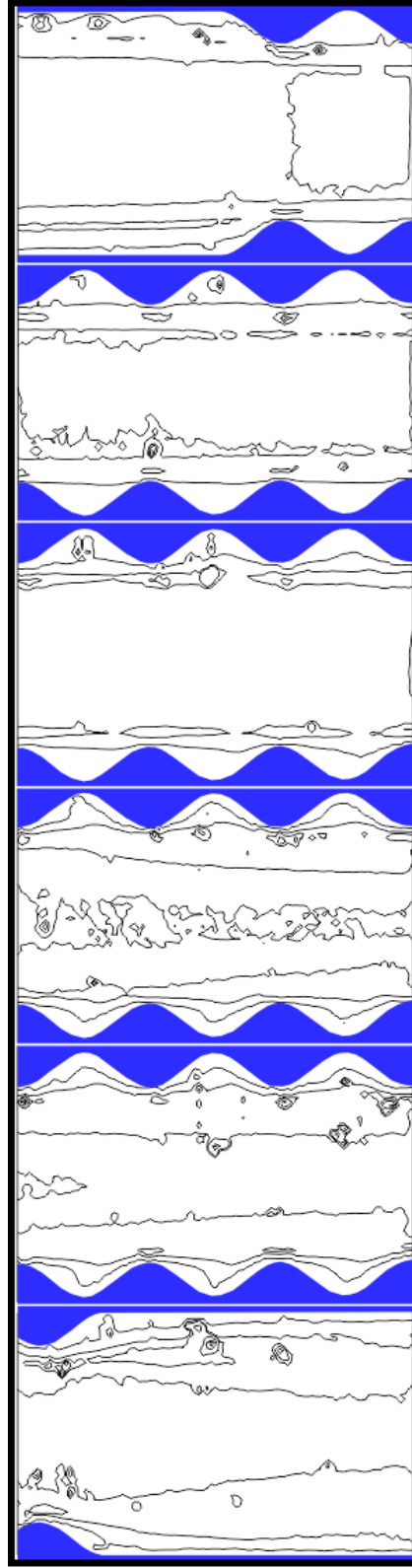


Figure 4.31. Patterns of Turbulent kinetic energy $|\langle (TKE)_{min} \rangle| = 0.01$, $\langle \Delta(TKE) \rangle = 0.04$ at $Re=3750$, for channel with $H_{min}/H_{max}=0.72$ and phase angle, ϕ of 180 degree

4.1.5. Reynolds Shear Stress Contours of Corrugated Channels

In this section, Reynolds shear stress, $\langle u'v'/U^2 \rangle$ correlations obtained from the results of the Particle Image Velocimetry (PIV) measurements are discussed for sinusoidal corrugated channels for different channel heights, H and phase angles, ϕ .

In fluid dynamics, in order to account turbulent fluctuations in fluid momentum, the component of the total stress tensor which is called Reynolds stress is obtained from the averaging operation over the Navier-Stokes equations. The velocity fluctuation can be described as a function of time. Turbulence will give rise to velocity fluctuations in both the horizontal (velocity u) and vertical (velocity v) directions characterized by u' and v' . The associated momentum fluctuation is called the turbulent stress or Reynolds stress. Similar to the viscous case, the Reynolds stress can be expressed as being the product of eddy viscosity.

The root-mean-square of streamwise, $\langle u_{rms} \rangle$ and transverse, $\langle v_{rms} \rangle$ components of velocity and Reynolds stress correlation were calculated as follows:

$$u_{rms} \equiv \langle u \rangle_{rms} \equiv \left[\frac{1}{N} \sum_{n=1}^N [u_n(x, y) - \langle u(x, y) \rangle]^2 \right]^{1/2}$$

$$v_{rms} \equiv \langle v \rangle_{rms} \equiv \left[\frac{1}{N} \sum_{n=1}^N [v_n(x, y) - \langle v(x, y) \rangle]^2 \right]^{1/2}$$

$$\langle u'v' \rangle = \frac{1}{N} \sum_{n=1}^N [u_n(x, y) - \langle u(x, y) \rangle] [v_n(x, y) - \langle v(x, y) \rangle]$$

Where N (1000) is the number of the instantaneous velocity field images taken during the PIV experiments for one set.

Time-averaged velocity vector fields, $\langle v \rangle$ Reynolds stress, $\langle u'v'/U^2 \rangle$ correlations, and other turbulent statistics were calculated by taking an average of 1000 detailed instantaneous velocity images.

. In turbulence, pressure and flow velocity vary rapidly in space and time. Fluctuations provide additional mechanism for momentum and energy transfer since

they improve the interaction of the fluid particles which have different momentum. In addition, momentum transfer mostly occurs by convection instead of diffusion due to the swirling eddies. Thus, heat transfer and the friction coefficient increase in turbulent flow due to the additional transverse motion enhancing the rate of energy and momentum exchange.

The shear stress of the laminar flow is defined as $\tau = -\mu \frac{d\bar{u}}{dy}$ where $\bar{u}(y)$ is the velocity profile of the turbulent flow. However, it is shown from the experiments that the shear stress is much higher due to the turbulent fluctuations between core and wake flow regions. So the turbulent shear stress should be considered as two parts. The laminar component is considered for the friction between layers, and the turbulent component is considered due to the friction between the fluctuating fluid particles. Therefore, the total shear stress in turbulent flow can be defines as follow:

$$\tau_{\text{total}} = \tau_{\text{laminar}} + \tau_{\text{turbulent}}$$

The root mean square of streamwise, $\langle u_{\text{rms}} \rangle / U_{\infty}$ and transverse velocity components, $\langle v_{\text{rms}} \rangle / U_{\infty}$ normalized by free-stream velocity, U_{∞} in all geometries at the Reynolds number of 3750 are presented in figures 4.32-4.34. The first and second columns of figures show the normalized streamwise velocity components, $\langle u_{\text{rms}} \rangle / U_{\infty}$ and the normalized transverse velocity components, $\langle v_{\text{rms}} \rangle / U_{\infty}$, respectively. The minimum and incremental values of both velocity components were taken as 0.001 and 0.04, respectively. As can be seen from the figure 4.32, the contours of both the normalized streamwise velocity components, $\langle u_{\text{rms}} \rangle / U_{\infty}$ and the normalized transverse velocity components, $\langle v_{\text{rms}} \rangle / U_{\infty}$ are symmetrical with respect to the channel axis since the upper and lower walls are symmetric along the centerline for the 180° phase angle, ϕ .

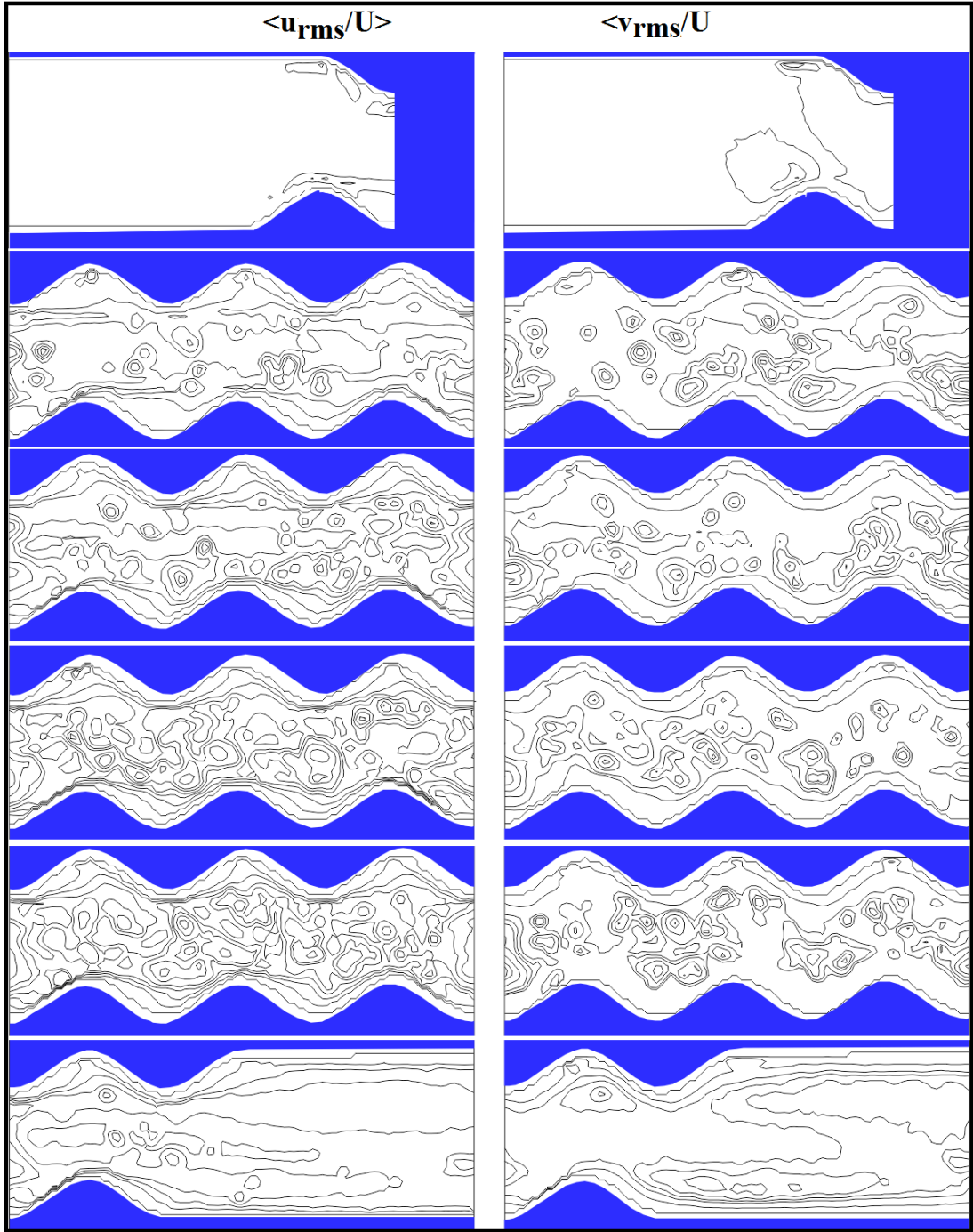


Figure 4.32. The root mean square of streamwise $|(\langle u_{rms} \rangle / U_{\infty})_{min}| = 0.001$, $\Delta(\langle u_{rms} \rangle / U_{\infty}) = 0.04$ and transverse velocity components, $|(\langle v_{rms} \rangle / U_{\infty})_{min}| = 0.001$, $\Delta(\langle v_{rms} \rangle / U_{\infty}) = 0.04$ for the channel with $H_{min}/H_{max} = 0.54$ and the phase angle, ϕ of 180° at $Re = 3750$

The contours of the root mean square of streamwise, $\langle u_{rms} \rangle / U_\infty$ and transverse velocity components, $\langle v_{rms} \rangle / U_\infty$ for the channel with the phase angles, ϕ of 90° and 0° are presented in figures 4.33 and 4.34, respectively. The root mean square of streamwise and transverse velocity components are higher at the center of the core region than at the near wall of the core region. The size of the lower value contours in the core region decreases in the flow direction.

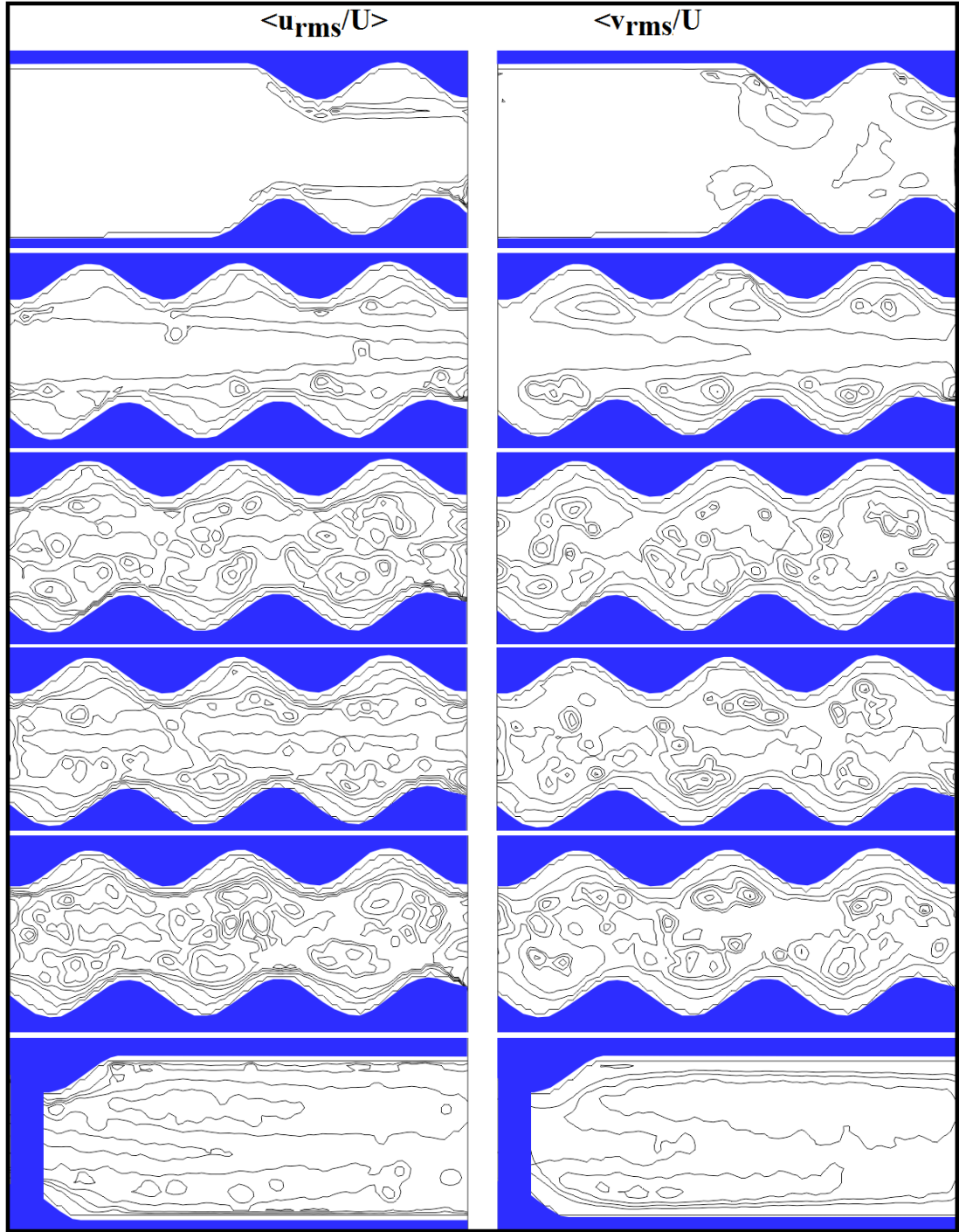


Figure 4.33. The root mean square of streamwise $|(\langle u_{rms} \rangle / U_\infty)_{min}| = 0.001$, $\Delta(\langle u_{rms} \rangle / U_\infty) = 0.04$ and transverse velocity components, $|(\langle v_{rms} \rangle / U_\infty)_{min}| = 0.001$, $\Delta(\langle v_{rms} \rangle / U_\infty) = 0.04$ for the channel with $H_{min}/H_{max} = 0.54$ and the phase angle, ϕ of 90° at $Re = 3750$

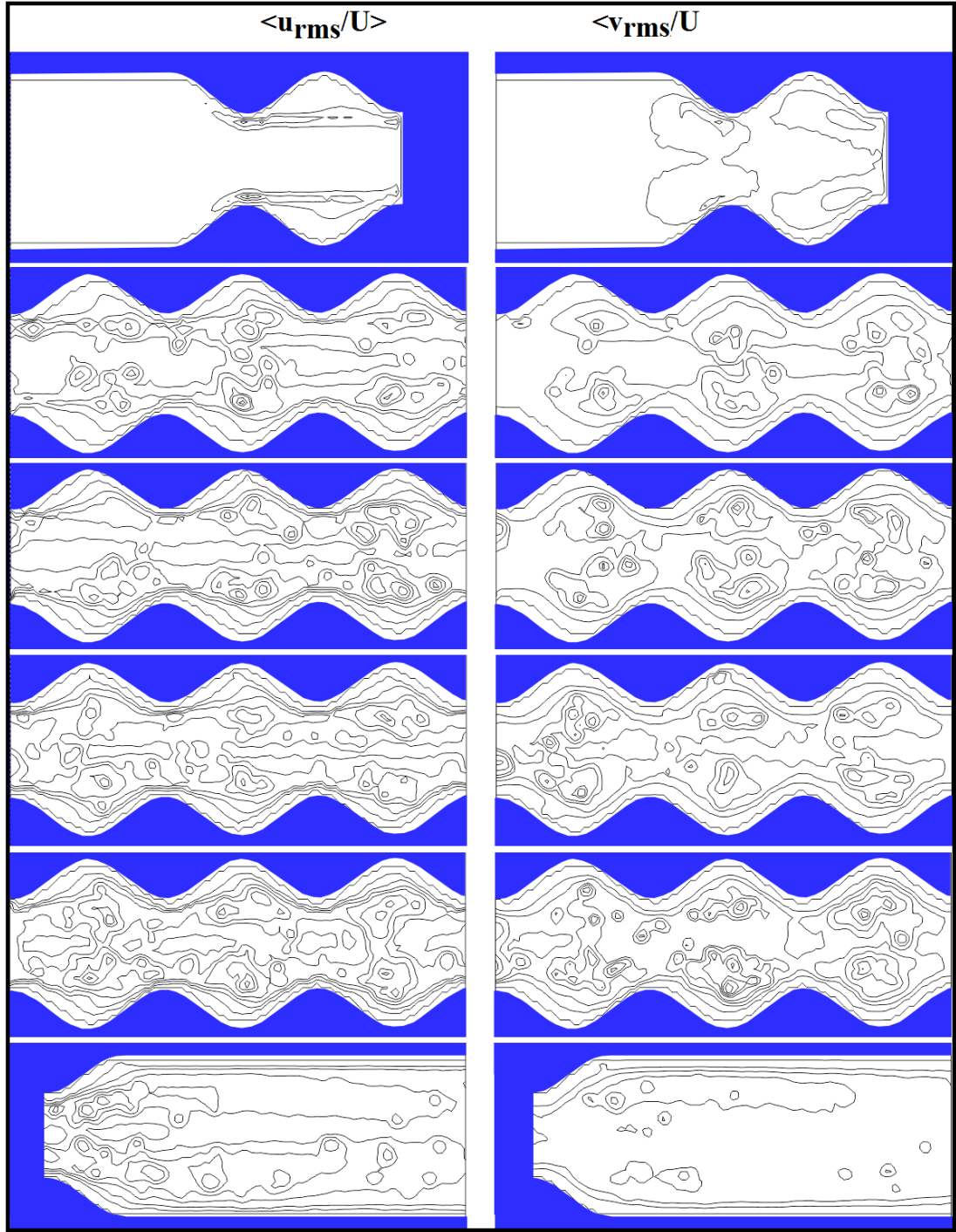


Figure 4.34. The root mean square of streamwise $|(\langle u_{rms}/U_{\infty} \rangle)_{min}|=0.001$, $\Delta(\langle u_{rms}/U_{\infty} \rangle)=0.04$ and transverse velocity components, $|(\langle v_{rms}/U_{\infty} \rangle)_{min}|=0.001$, $\Delta(\langle v_{rms}/U_{\infty} \rangle)=0.04$ for the channel with $H_{min}/H_{max} = 0.54$ and the phase angle, ϕ of 0° at $Re=3750$

The contours of the Reynolds shear stress, $\langle u'v'/U^2 \rangle$ at the Reynolds number, $Re=3750$ for channels with $H_{min}/H_{max}=0.54$ and the phase angles, ϕ of 180° , 90° , and 0° are presented in figures 4.35, 4.36, and 4.37, respectively. On the other hand, figure 4.38 shows the Reynolds shear stress, $\langle u'v'/U^2 \rangle$ contours at the Reynolds number, Re of 3750 for channel with $H_{min}/H_{max}=0.72$ and the phase angles, ϕ of 180° . The minimum and incremental values of dimensionless Reynolds shear stress, $\langle u'v'/U^2 \rangle$ contours were taken as ± 0.05 and 0.005, respectively. The solid black lines show the positive Reynolds shear stress, $\langle u'v'/U^2 \rangle$ contours, whereas red dashed lines represent negative contours. As previously mentioned, the contours of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ are combination of root mean square of normalized streamwise velocity components, $\langle u_{rms} \rangle/U_\infty$ and root mean square of transverse velocity components, $\langle v_{rms} \rangle/U_\infty$.

The contours of the Reynolds shear stress, $\langle u'v'/U^2 \rangle$ for the channel with the phase angles, ϕ of 180° are presented in figure 4.35. Negative and positive contours, which take place in the lower and upper parts of the channel, respectively are symmetrical with respect to the channel axis due to the geometrical symmetry of corrugated channel. Reynolds shear stress, $\langle u'v'/U^2 \rangle$ increases gradually as the fluid moves through the channel due to the sinusoidal corrugation of channel. The maximum Reynolds shear stresses occur along the shear layers at the channel throats since the flow velocity increase due to convergence of the channel. On the other hand, the minimum Reynolds shear stresses, $\langle u'v'/U^2 \rangle$ are observed in the wake regions. As can be seen in the figure 4.35, the size of the core region where the lower values of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ take place decreases until the 6th cavity. The flow after the 6th cavity has moderate Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values in the core region for all the cavities. The size of the contours remain same after the 6th cavity.

The contours of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ for the channel with $H_{min}/H_{max}=0.54$ and the phase angles, ϕ of 90° and 0° at the Reynolds number, Re of 3750 are presented in figure 4.36 and 4.37, respectively. Minimum and incremental values are taken as ± 0.05 and 0.005, respectively. Reynolds shear stress,

$\langle u'v'/U^2 \rangle$ values are lower at the inlet of the channel, but it increases while fluid moves further downstream in the flow direction due to effect of corrugations. Reynolds shear stress, $\langle u'v'/U^2 \rangle$ at the center of the core region are smaller than Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values close to the near wall of region. The size of the lower values core region decreases from inlet until 6th cavity. In addition, ± 0.05 contours in the cavities widen until 5th cavity. Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values are higher in the channel with 90° phase shift, ϕ comparing to the 180° case.

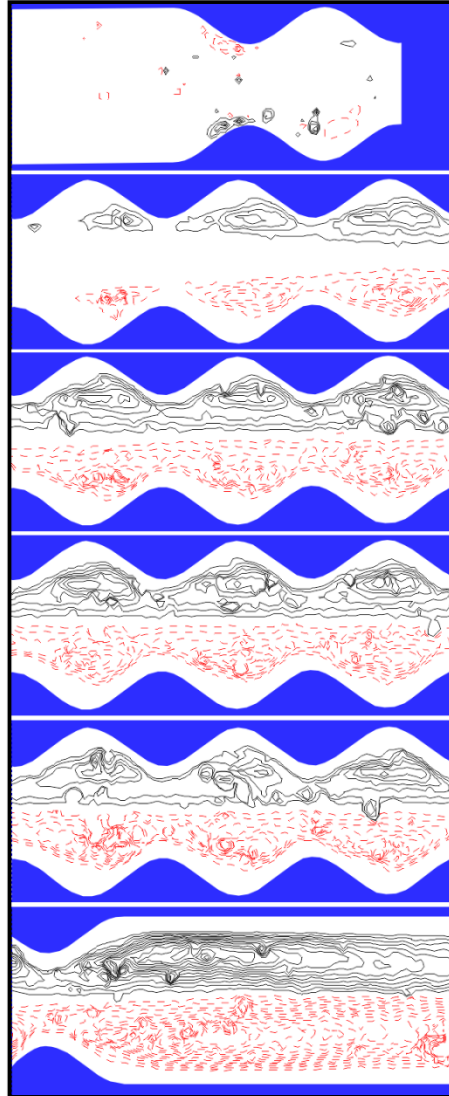


Figure 4.35. Reynolds shear stress, $|\langle (u'v'/U^2)_{\min} \rangle| = 0.001$, $\langle \Delta (u'v'/U^2) \rangle = 0.001$ for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angle, ϕ of 180° at $Re=3750$

Figure 4.37 shows Reynolds shear stress, $\langle u'v'/U^2 \rangle$ contours for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angle, ϕ of 0° at the Reynolds number, Re of 3750. As previously reported that Reynolds shear stress, $\langle u'v'/U^2 \rangle$ increases as fluid moves from the first cavity to the last cavity of sinusoidal corrugated channels. Very low magnitudes of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ are observed in the main flow region for example, along the central axis of the first five cavities. These regions become narrower in the flow direction, and higher magnitude of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ take places at the center after the 5th cavity. These magnitudes of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ do not change downstream of the 5th cavity. Higher values of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ are observed at inlet and outlet of the FOV 3, FOV 4, FOV 5, and FOV 6.

In order to investigate the effects of channel height, Reynolds shear stress, $\langle u'v'/U^2 \rangle$ contours for the channel with $H_{\min}/H_{\max} = 0.72$ are obtained. Figure 4.38 shows the Reynolds shear stress, $\langle u'v'/U^2 \rangle$ distributions for the phase angle, ϕ of 180° at the Reynolds number, Re of 3750. Although similar patterns are observed, Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values are lower due to the lower velocity values and the greater distance between upper and lower walls. Very low Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values are seen in the first three FOV, and domain of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ become in the region downstream of FOV 3 until FOV 6. The main flow regions, where the Reynolds shear stress, $\langle u'v'/U^2 \rangle$ values are very low, is larger comparing to the $H_{\min}/H_{\max} = 0.54$ case since the effects of wavy walls become weaker. In addition, the size of the very low values core region decreases in the flow direction. Moreover, the contours of Reynolds shear stress, $\langle u'v'/U^2 \rangle$ in figure 4.38 show that the contours cannot become stabile in 13 wave long for this channel height and Reynolds number, Re .

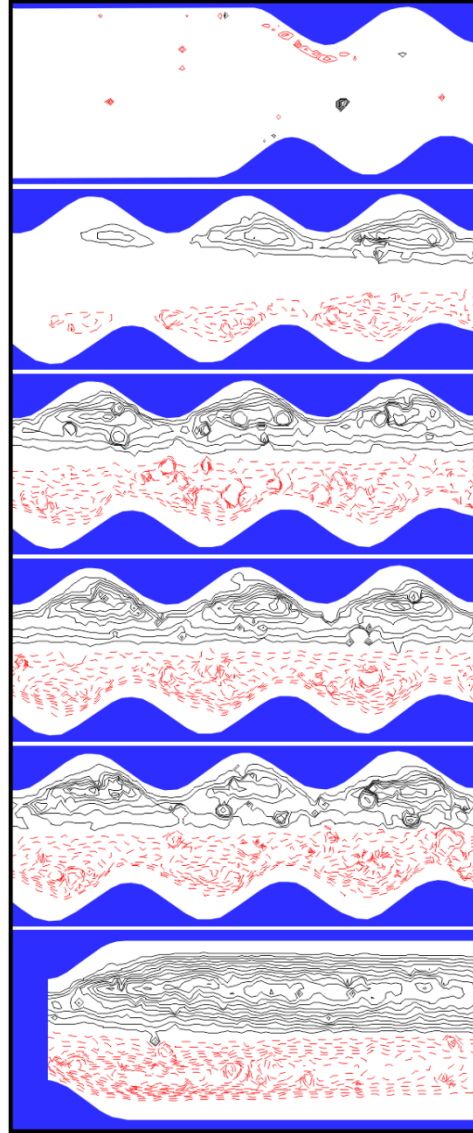


Figure 4.36. Reynolds shear stress, $|\langle (u'v'/U^2)_{\min} \rangle| = 0.001$, $\langle \Delta(u'v'/U^2) \rangle = 0.001$ for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angle, ϕ of 90° at $Re=3750$

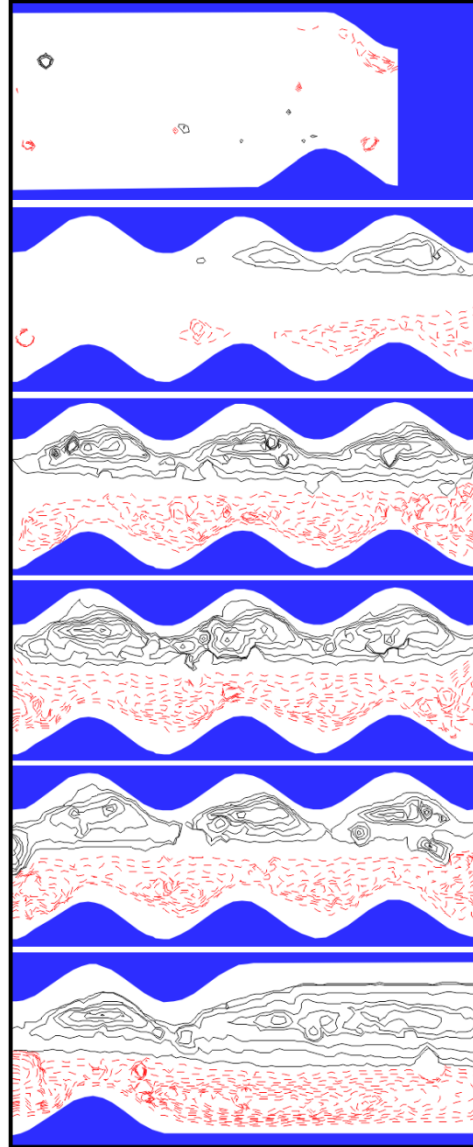


Figure 4.37. Reynolds shear stress, $|\langle (u'v'/U^2)_{\min} \rangle| = 0.001$, $\langle \Delta(u'v'/U^2) \rangle = 0.001$ for the channel with $H_{\min}/H_{\max} = 0.54$ and the phase angle, ϕ of 0° at $Re=3750$.

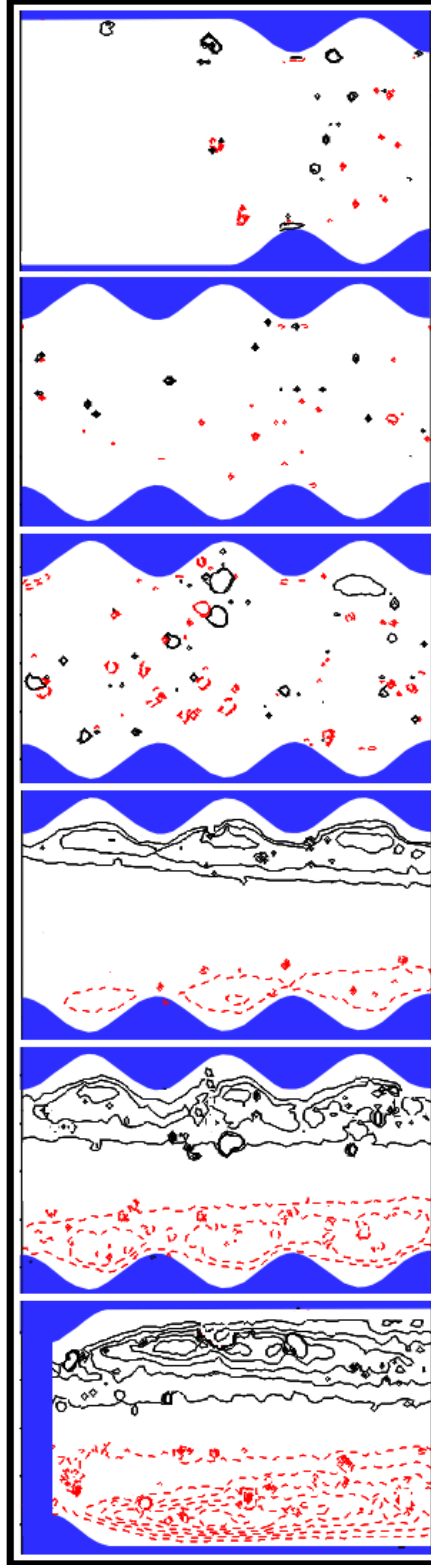


Figure 4.38. Reynolds shear stress, $|\langle (u'v'/U^2)_{\min} \rangle| = 0.001$, $\langle \Delta(u'v'/U^2) \rangle = 0.001$ for the channel with $H_{\min}/H_{\max} = 0.72$ and the phase angle, ϕ of 90° at $Re=3750$.

4.1.6. Validation of the Data Sets

The experiments of this study were performed for investigation of flow field in corrugated channel using the Particle Image Velocimetry (PIV) technique. Time-averaged flow data are determined by taking an average of 1000 PIV images. Two sets of 1000 images were taken for different time periods in order to determine the discrepancy of the data sets.

The experimental time-averaged streamwise normalized velocity profiles at the Reynolds number, Re of 3750 for channel with $H_{min}/H_{max}=0.54$ and the phase angle, ϕ of 180 degree are presented in figure 4.39 for two data sets. The velocity profiles are obtained from the widest cross-section of the 3rd cavity in the xy -plane. The velocity profile is almost symmetrical with respect to the centerline of the channel. The maximum velocity occurs at the centerline, and negative velocities at the upper and lower regions of the cavities are observed due to the secondary flow structures. The velocity profile in the boundary represents reverse flow which shows separated flow. Figure 4.39 shows that velocity profiles obtained from different time periods agree completely with each other. It is also concluded that 1000 images are sufficient to get time-averaged velocity data.

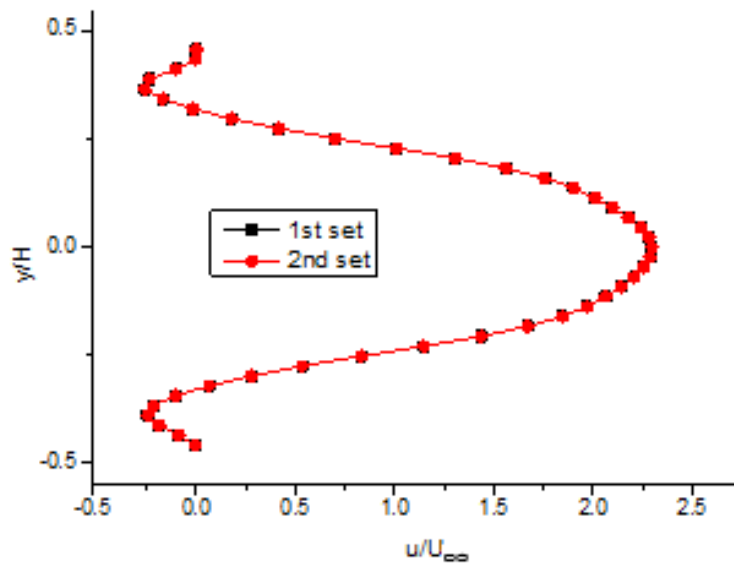


Figure 4.39. Dimensionless streamwise velocity component, u/U_{∞}

Figure 4.40 shows the dimensionless Turbulent Kinetic energy, $\langle \text{TKE} \rangle$ obtained from the widest cross-section of the 3rd wave of the sinusoidal corrugated channel with $H_{\min}/H_{\max}=0.54$ and the phase angle, ϕ of 180 degree at the Reynolds number, Re of 3750 for two data sets. As can be seen in figure 4.40, there is a small difference in Turbulent Kinetic Energy, $\langle \text{TKE} \rangle$ values of 1st and 2nd sets which is reasonable.

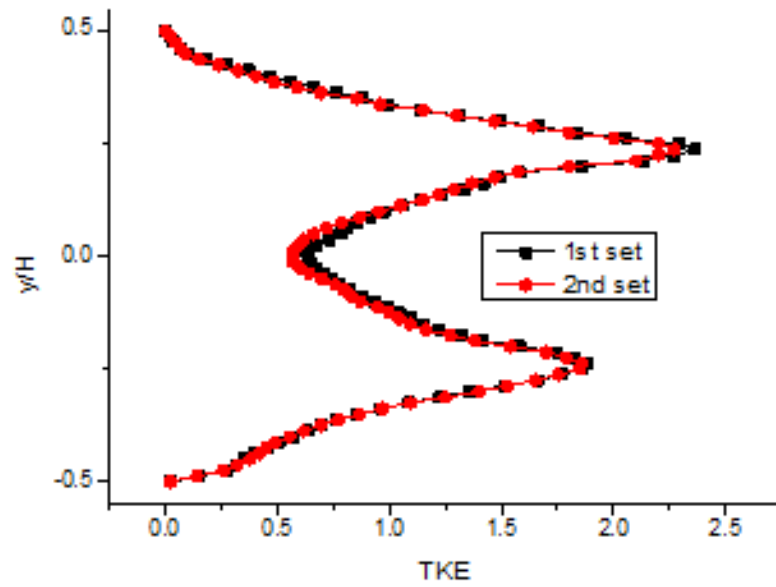


Figure 4.40. Dimensionless Turbulent Kinetic energy, $\langle \text{TKE} \rangle$

4.2. Numerical Results

In this section, effects of the channel height, H and phase shift, ϕ on heat transfer enhancement of the sinusoidal corrugated channel are discussed for three Reynolds number, Re . The optimum geometrical configuration and operating condition of corrugated channel are determined by considering the thermal performance factor (TPF).

4.2.1. Effects of the Channel Height on Heat Transfer Enhancement of Corrugated Channel

In this subsection, effects of the channel height on the flow characteristics and heat transfer enhancement in the sinusoidal corrugated wavy channels are investigated. Hydrodynamic of water flow in corrugated channels with three channel heights H_1 ($H_{\min}/H_{\max}=0.36$), H_2 ($H_{\min}/H_{\max}=0.54$), and H_3 ($H_{\min}/H_{\max}=0.72$) are studied and the obtained results are compared with each other and that of the parallel plate channels in order to demonstrate the channel height effects on the thermal-hydraulic characteristics. The numerical simulations has been done for Reynolds numbers, Re of 2500, 5000, and 7500.

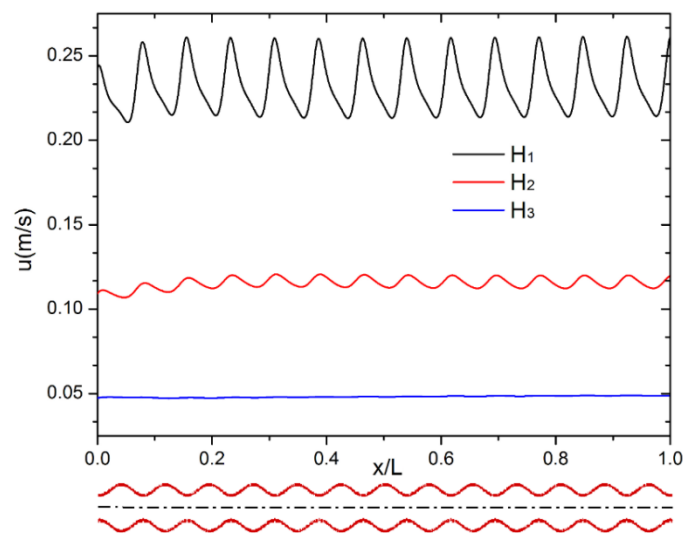


Figure 4.41. Distributions of streamwise velocities, u along the channel axis with 180° phase angle, ϕ for $Re=2500$

Figure 4.41 presents streamwise velocity, u distributions along the axis in the corrugated channels with 180° phase angle for H_1 , H_2 and H_3 . It is seen in this figure that the maximum velocity is observed for H_1 since the distance between the upper and lower walls are shortest. Distributions of streamwise velocity, u component illustrates that streamwise velocity, u profile along the axis of the channels are periodic. H_1 has the highest amplitude of the periodic velocity, u distributions among all cases. However, streamwise velocity, u for H_3 seems almost straight line along the channel axis since the amplitude of sinusoidal wavy walls are very small with respect to channel height. It can also be said that periodicity of streamwise velocities, u eventually become stabile in the region downstream of second cavities. The streamwise velocity, u decreases in diverging sections and increases in converging sections of the wavy channel.

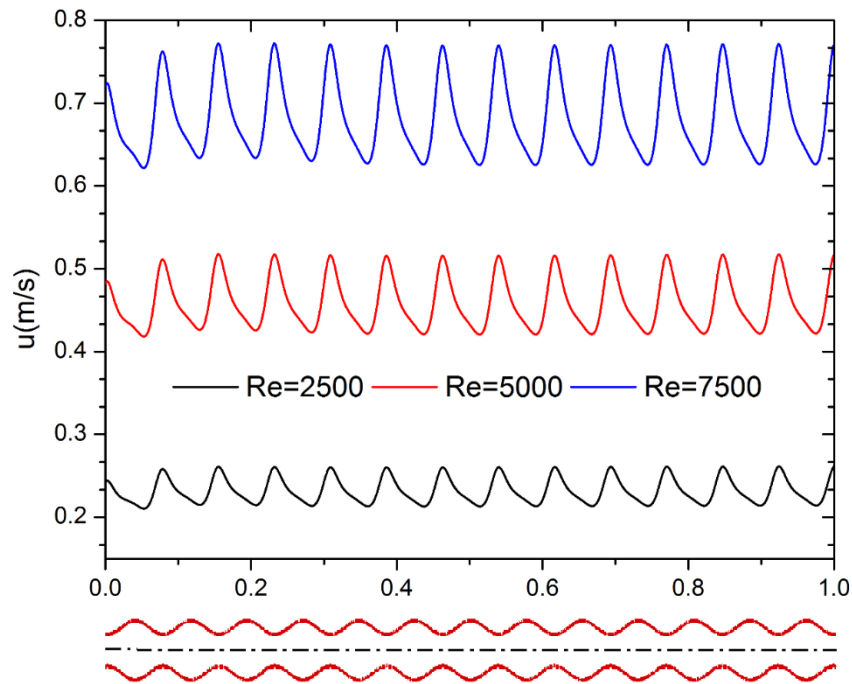


Figure 4.42. Distributions of streamwise velocities, u along the axis of channel H_1 with 180° phase angle, ϕ

The amplitude of streamwise velocity, u variation on the axis also depends on the Reynolds number, Re . The amplitude of the streamwise velocity, u on the axis is

small for low Reynolds number such as $Re=2500$ and become bigger as the Reynolds number, Re increases as can be seen from figure 4.42.

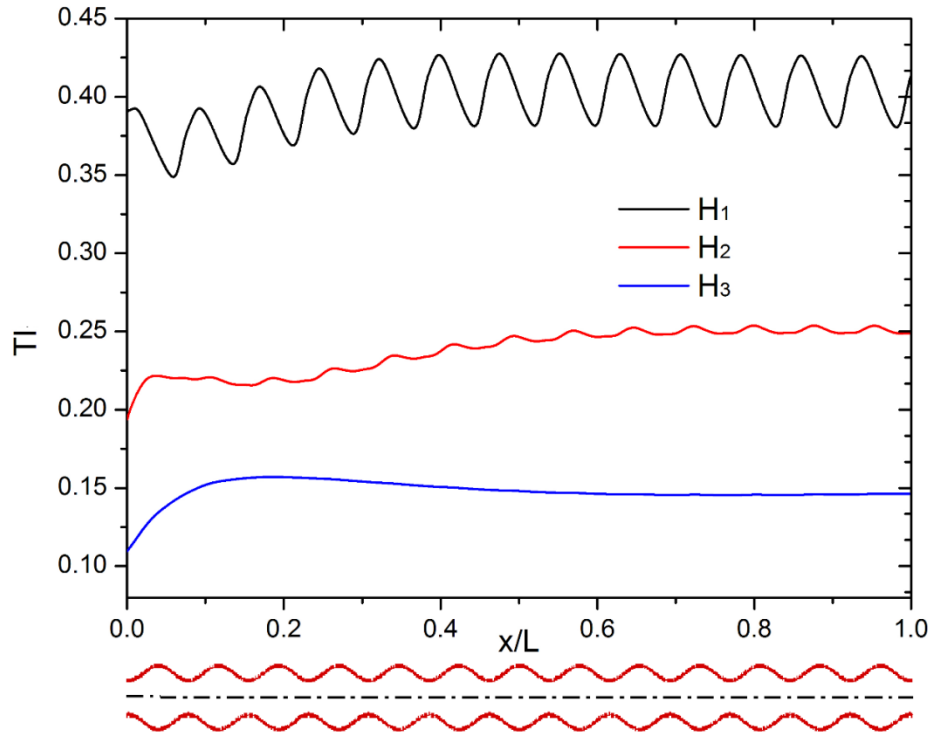


Figure 4.43. Distributions of turbulence intensities, TI along the channel axis with 180° phase angle, ϕ for $Re=2500$

Using sinusoidal corrugation shaped walls instead of parallel plates lead to magnify turbulence intensity through the corrugated channel. The rate of turbulence intensity varies as a function of the channel height. As can be seen from figure 4.43, by decreasing the distance between upper and lower walls increases the rate of turbulence intensity through the corrugated channel. Increasing the flow mixing and turbulence intensity improve the thermal transport through the corrugated channel with pressure drop penalty. Hence, additional considerations are required to find optimum design parameters.

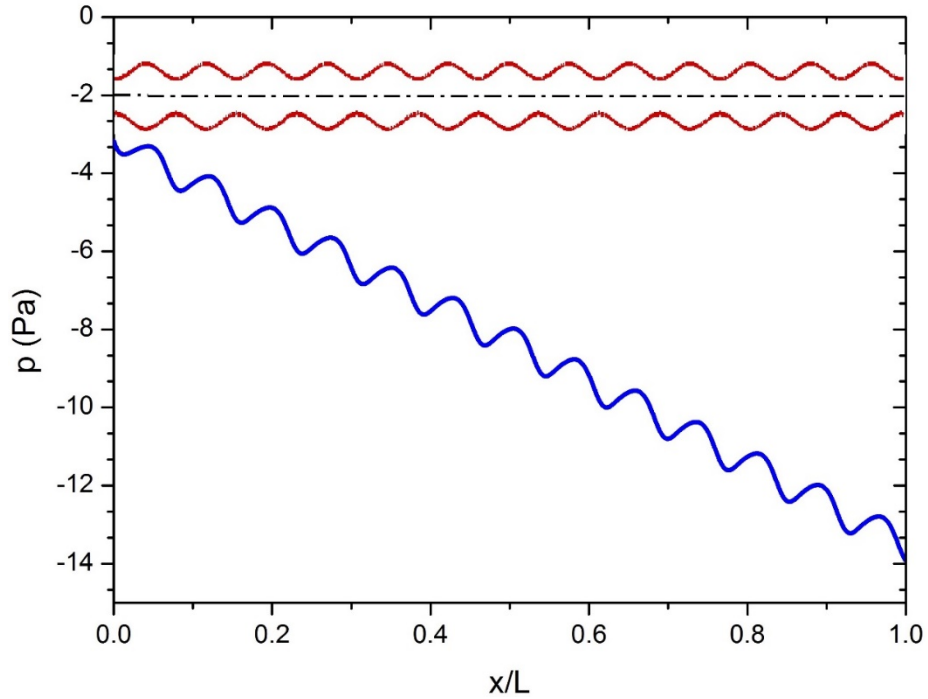


Figure 4.44. Axial static pressure, p distribution along the channel axis with 180° phase angle, ϕ for $Re=2500$

Figure 4.44 illustrates the pressure distribution along the axis in the wavy part of the sinusoidal corrugated channel with a configuration of H_2 ($H_{\min}/H_{\max}=0.54$) and phase shift, $\phi = 180^\circ$ at the Reynolds number, Re of 2500. As the flow moves towards the converging section of the channel, a rapid pressure drop occurs in the flow direction due to the contraction of core flow region. On the other hand, since the core flow expands after the narrowest cross section, axial static pressure increases rapidly as the fluid moves from the converging section to the diverging section. The periodic variations in axial static pressure occur throughout the sinusoidal corrugated channel since the channel geometry changes periodically too. Therefore, axial static pressure distribution along the wave repeats identical variation through the other waves. Moreover, the rate of pressure variations depends on the wave parameters (amplitude and wavelength), the channel height, H and the Reynolds number, Re .

Corrugated channels are normally used to enhance heat transfer. The distance, H between the upper and lower walls of such channels are important parameter that influence heat transfer performance. The effects of the variations of channel height of

the tested corrugated channels on the average Nusselt number, $\overline{Nu}$ are discussed in this subsection.

The influences of the flow structure in corrugated channels with different channel heights on the averaged Nusselt number, $\overline{Nu}$ for 0° phase shift is illustrated in figure 4.45. For certain phase shift angle, ϕ , averaged Nusselt number, $\overline{Nu}$ is seen to increase significantly with increasing the Reynolds number, Re . It is also shown that the averaged Nusselt number, $\overline{Nu}$ obtained in the case of higher channel height, H is lower than that from lower ones for a given phase shift angle, ϕ due to the reduction in the velocities and hence the wall velocity gradients cause an increase in the thermal boundary layer thickness. The maximum heat transfer occurs for the channel H_1 ($H_{\min}/H_{\max}=0.36$) at the Reynolds number of 7500. As can be seen in figure 4.45, H_1 has the maximum increment rate of averaged Nusselt number, $\overline{Nu}$ with the Reynolds number for 0° phase shift angle, ϕ . As the fluid flows through the corrugated surface, the fluid re-circulates or/and the swirl flows are generated in the corrugation troughs. The onset and growth of recirculation zones promote the mixing of fluid in the boundary layer enhancing the heat transfer.

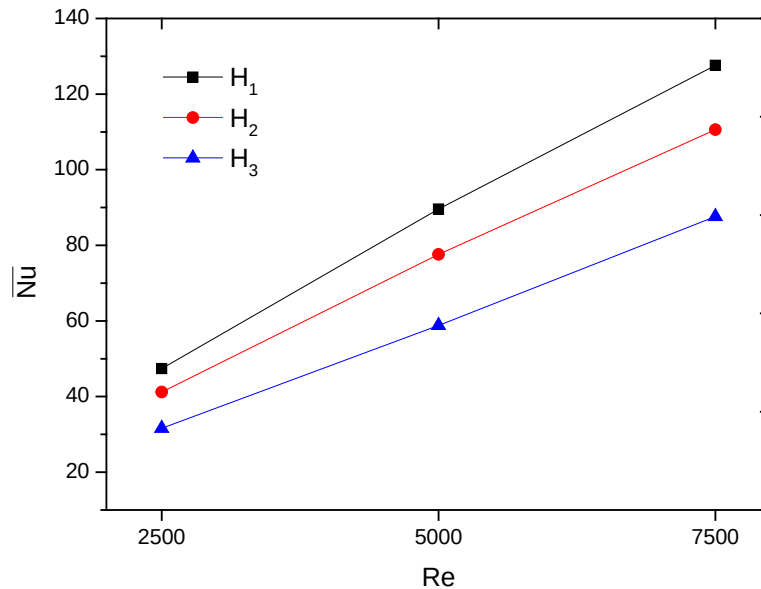


Figure 4.45. Variations of the averaged Nusselt numbers, $\overline{Nu}$ with the Reynolds number, Re for the corrugated channels with phase angle, $\phi=0^\circ$

The friction factor, f is a main hydraulic characteristics of the flow and is influenced by development of turbulence flow in corrugated channels. The friction factor, f of corrugated channels mainly affected by the Reynolds number, Re and channel configurations with respect to channel height, H and phase shift angle, ϕ . Friction factor, f as a function of the Reynolds number, Re for flow in 0° phase shift angle, ϕ corrugated channel at different channel height $H_{\min}/H_{\max}=0.36, 0.54$, and 0.72 are plotted in 4.46. This figure shows that friction factor, f gradually decreases with increasing Reynolds numbers, Re for any value of channel height, H . In addition, the friction factor, f gets higher values as the channel becomes narrower. The friction factors, f for all corrugated channel configurations have much higher than that for the straight channel since corrugations promote unstable formed vortices in the core of corrugated channel's flow, causing a significant increase of pressure drops. It can be seen that the channel height, $H_{\min}/H_{\max}=0.36$ has the highest friction factor, f . On the other hand, the lowest value of friction factors, f are observed at $H_{\min}/H_{\max}=0.72$ for all Reynolds numbers considered.

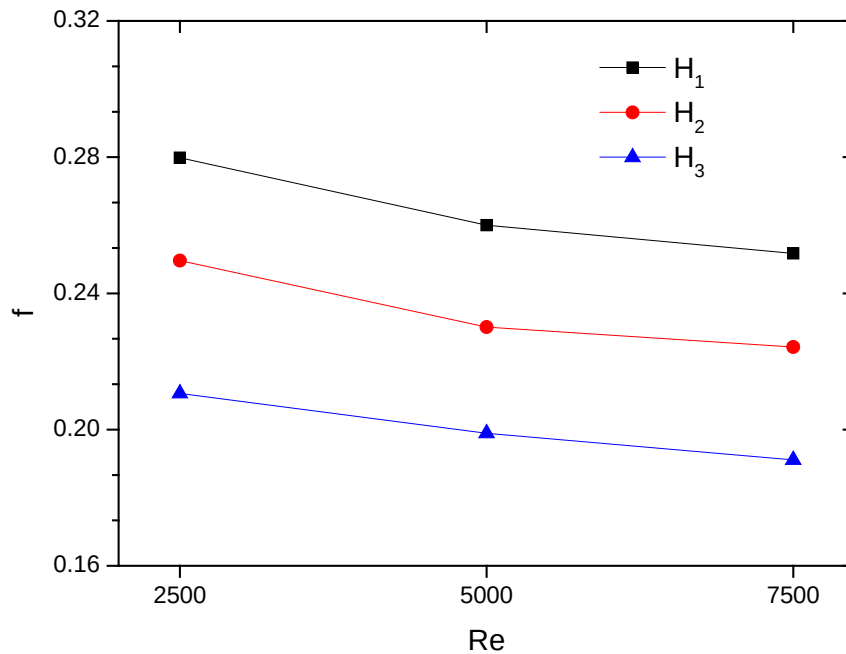


Figure 4.46. Variations of the friction factors, f with the Reynolds number, Re for the corrugated channels with phase angle, $\phi=0^\circ$

4.2.2. Effects of the Phase Shifts, ϕ on the Heat Transfer Enhancement in Corrugated Channels

In this subsection, effects of the phase shift angles, ϕ on the flow characteristics and heat transfer enhancement in corrugated sinusoidal wavy channels are studied. The water flow in corrugated channels with three phase shift angles, ϕ of 0° , 90° , and 180° are simulated and the obtained results are compared with each other and that of the parallel plate channels in order to get the phase shift effects on the thermal-hydraulic characteristics. The numerical simulations has been done for Reynolds numbers 2500, 5000, and 7500. The streamwise velocity, u distributions and the turbulence intensity, TI along the channel axis as a function of Reynolds number for different phase shift angles, ϕ are presented in figure 4.47. All the results are obtained from the wavy part of the channel H_1 .

As shown in this figure, using wavy plates develops harmonic velocity distributions on the axis of the channel. For all the phase shift angles, ϕ the wavelengths and amplitudes of the velocity become stabile after certain distances. The distance obtained for the same wavelength and amplitude is shortest for 180° phase shift angle, ϕ since the upper and lower walls are symmetrical from the beginning of the channel. It is also noted that the amplitude of the velocity distribution depends on the phase shift, ϕ and the Reynolds number, Re . The amplitude of the velocity distribution increases with phase shift angle, ϕ and the Reynolds number, Re . Highest amplitude is observed for the channel with 180° phase shift at $Re=7500$. Corrugated plates lead turbulence along the axis of the channel. In figure 4.47, strong dependence of turbulence intensity on the phase shift angle, ϕ and the Reynolds number, Re is also noticed. For all the geometric configurations tested, turbulence intensity rate increases with the Reynolds number.

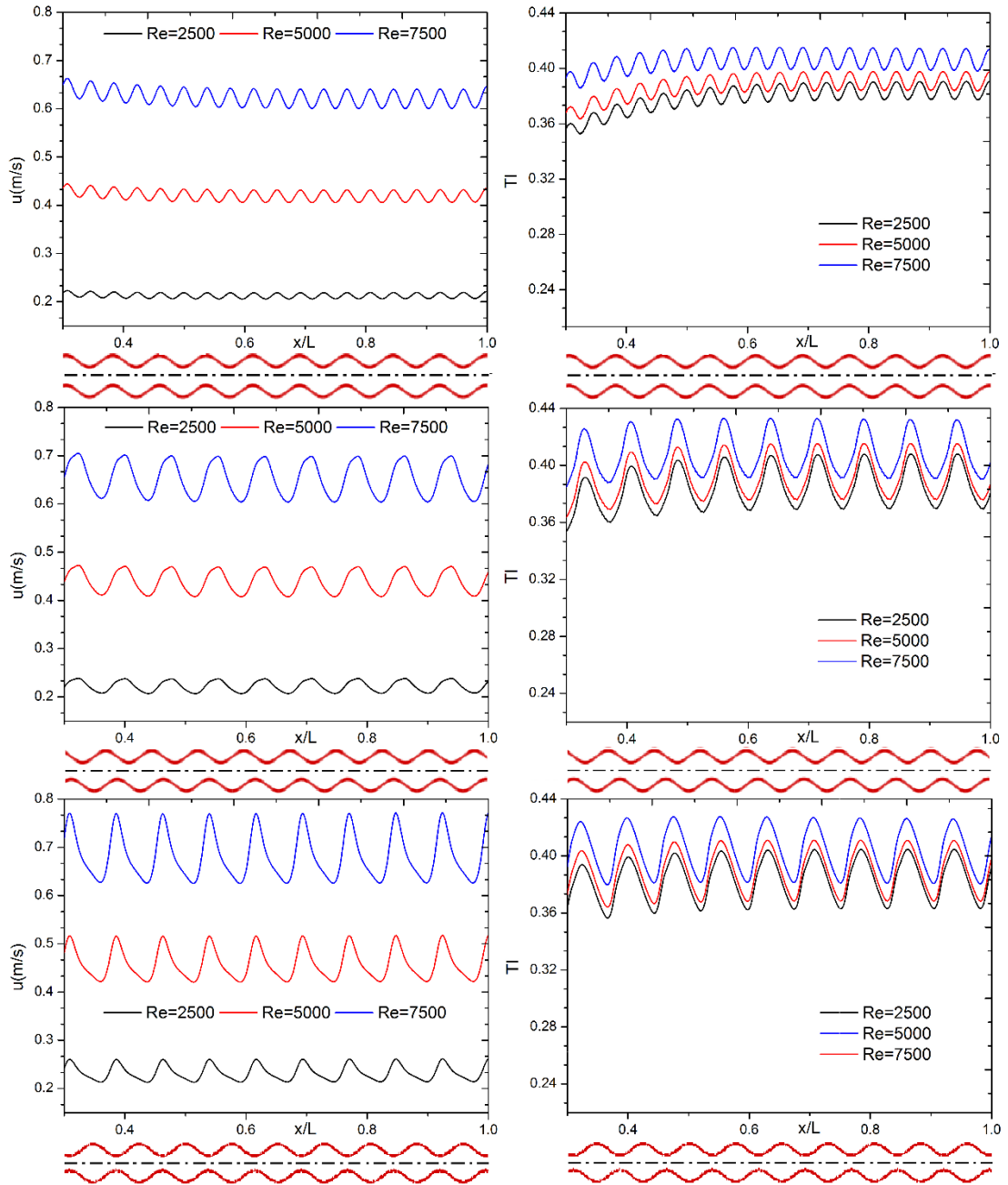


Figure 4.47. Distributions of streamwise velocities, u (left) and turbulence intensities, TI (right) along the channel axis

Figure 4.48 shows that the value of Nusselt number, Nu for straight channel is maximum at the inlet of channel and decrease slightly with axial distance along the channel wall. This is due to the fact that the boundary layer is thinner at the inlet of channel than at any distance along the channel.

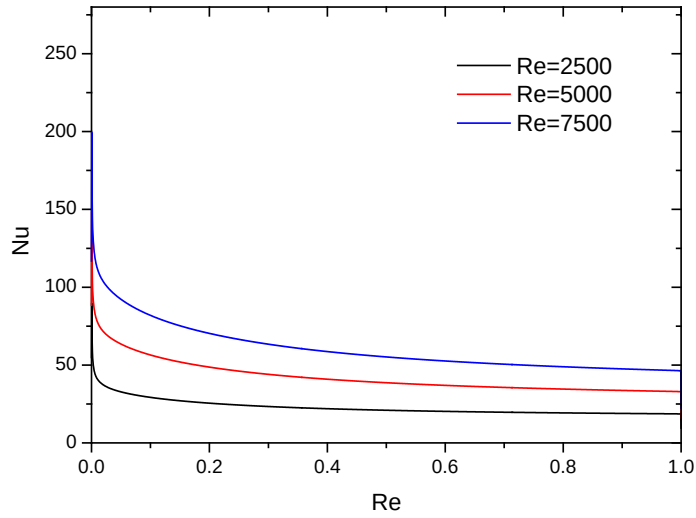


Figure 4.48. Variations of the local Nusselt numbers, Nu along the lower wall of the straight channel

The variations of the local Nusselt number, Nu with the Reynolds number, Re along the lower wavy walls of the corrugated channels with phase shift angles, ϕ of 0° , 90° , and 180° are shown in figure 4.49. The local Nusselt number, Nu is higher in the converging section of each wave than in the diverging section since as the fluid moves downstream, when the channel cross sectional areas over the throats decreases so this section has higher averaged velocity and velocity gradient, which increases the heat transfer ratio. The peaks of the local Nusselt number, Nu are observed at the channel throats due to the sudden increase in velocity gradient. In addition, local Nusselt number, Nu increases with the Reynolds number, Re for all phase shifts, ϕ . Although the local Nusselt number, Nu values change with the Reynolds number, Re similar trends are observed for certain phase shifts, ϕ . Moreover, the Nusselt number, Nu of corrugated channels for all phase shifts, ϕ are higher than that of the straight channel at all Reynolds numbers, Re .

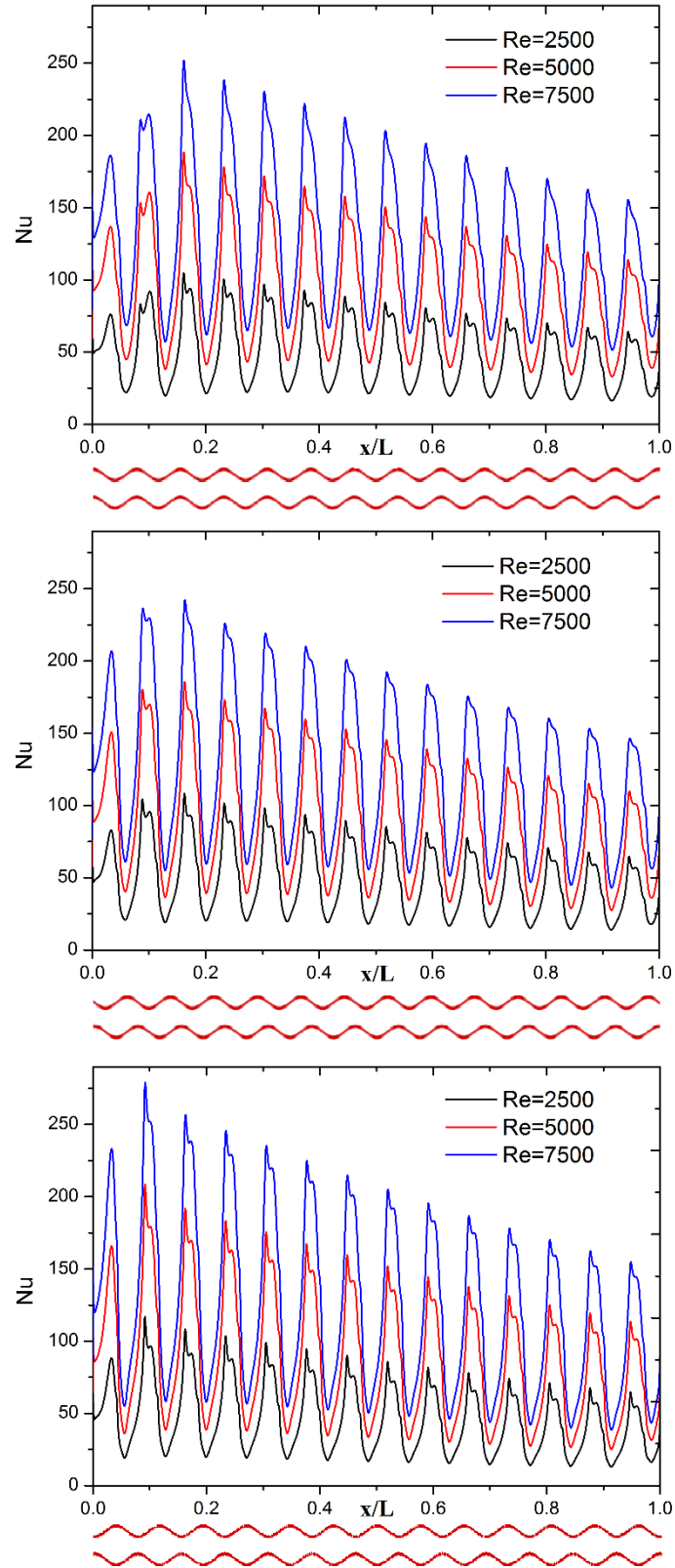


Figure 4.49. Variations of the local Nusselt numbers, Nu along the lower wavy walls for different phase shifts, ϕ of the channel H_1

The averaged Nusselt numbers, Nu as a function of Reynolds number, Re for both corrugated channels ($\phi=0, 90, 180$) and straight channel are presented in figure 4.50. It is shown in this figure that the average Nusselt number, $\overline{Nu}$ and the heat transfer from the wall enhance with an increase of the Reynolds number, Re since the temperature gradients in the boundary layer region near walls increase too. In addition, heat transfer enhancement due to using corrugated plates at a high Reynolds number, Re is higher comparing to the case of a low Reynolds number, Re since the mixing of fluid in corrugated channel improves with increasing Reynolds number. It is also observed that the averaged Nusselt number, $\overline{Nu}$ decreases as the phase shift angle, ϕ increases because the decrease in phase shift angle, ϕ causes larger region causing re-circulation or/and higher swirl flows intensity in the corrugation trough. Moreover, averaged Nusselt number, $\overline{Nu}$ values of corrugated channels for all phase shift angle, ϕ configurations have higher value than that of straight channel case. It can be clearly seen in figure 4.50, the channel with 0° phase shift angle, ϕ has highest averaged Nusselt number, $\overline{Nu}$ at all Reynolds number considered in this study.

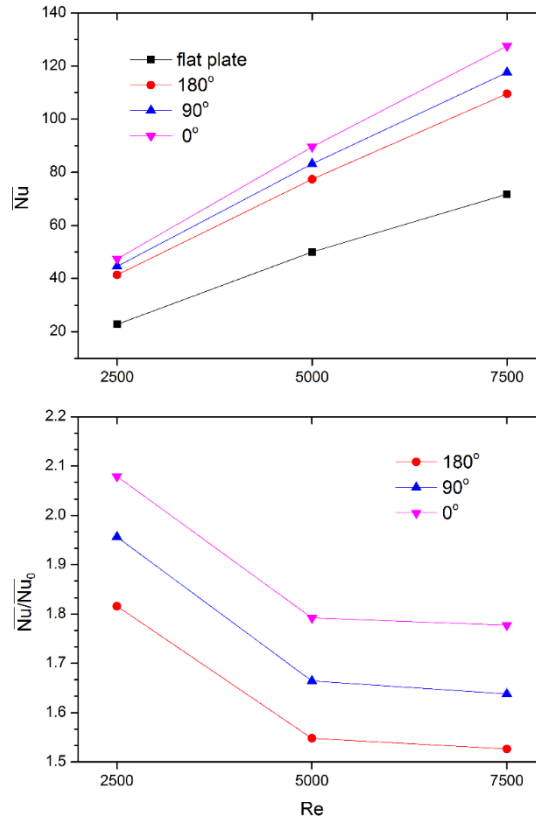


Figure 4.50. Variations of averaged Nusselt numbers, $\overline{Nu}$ with the Reynolds number, Re for corrugated channel H_1 with different phase shifts, ϕ and comparison with straight channel

Averaged Nusselt number, $\overline{Nu}$ in the corrugated channels enhances several folds, depending upon the values of Reynolds number, Re and phase shift angle, ϕ for given channel height. Heat transfer rates in corrugated channels of 0° , 90° and 180° phase shifts are increased by a factor of 1.78, 1.64, and 1.53 comparing to those of straight channel, respectively at the Reynolds number, Re of 7500. On the other hand, heat transfer enhancement rates increase up to 2.08, 1.96, and 1.82 for 0° , 90° and 180° phase shift angles, ϕ respectively at $Re=2500$.

Thermal transport is improved with increasing the rate of flow mixing which results in more pressure drop. The measured pressure drops among the simulated corrugated channels are used to predict the friction factors, f . The friction factor, f variations with Reynolds numbers, Re for straight channel and corrugated channels having $H_{min}/H_{max}=0.36$ at different phase shifts are shown in figure 4.51. The friction factor, f investigated through wavy channels are larger than that of straight channel

due to the suppression of viscous sub-layer. In addition, friction factor, f decreases with the increase of phase shift angle, ϕ .

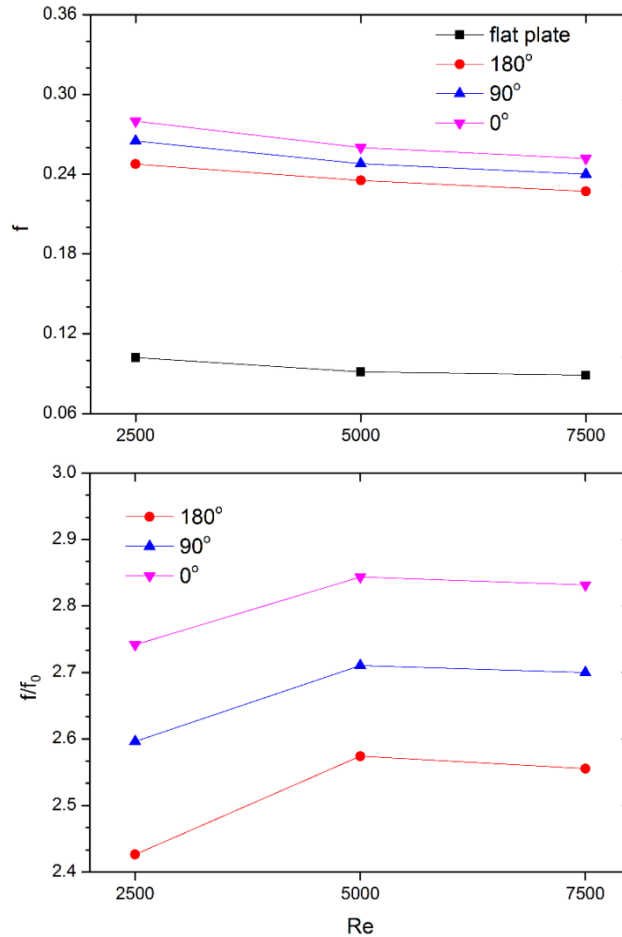


Figure 4.51. Variations of friction factors, f with the Reynolds number, Re for corrugated channel H_1 with different phase shifts, ϕ and comparison with straight channel

It can be observed from figure 4.51 that the friction factor, f decreases with increasing Reynolds numbers, Re for straight channel and all configurations of corrugated channels. For straight channel, an increase in the Reynolds number, Re results in decreasing the laminar sub layer thickness which reduces the friction factor, f . Using corrugations causes to block fluid flow and increase the pressure drop, Δp along the channel. Although the pressure drop, Δp increases with increasing Reynolds numbers, the friction factor, f decreases with an increase in Reynolds numbers since the friction factor, f depends on $\Delta p/\rho u^2$ and it is inversely

proportional with square of the velocity, u . The friction factor, f values for phase shift angle, $\phi = 0^\circ$ is the highest among all configurations for all considered Reynolds numbers, Re while the lowest values are observed for phase shift angle, $\phi = 180^\circ$. The lowest friction factor, f enhancement is observed at the Reynolds number, Re of 2500 for all phase shifts. The friction factor, f in corrugated channels of 0° , 90° and 180° phase shift angles, ϕ increase by a factor of 2.74, 2.59, and 2.43 compare to those of straight channel, respectively at the Reynolds number, Re of 2500.

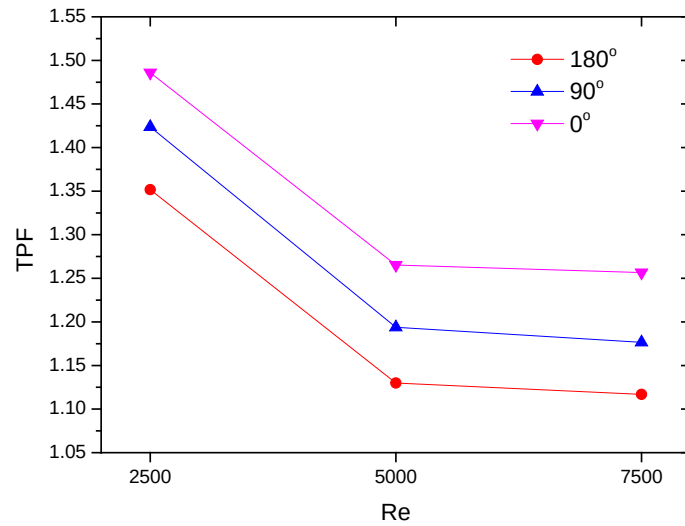


Figure 4.52. Variations of thermal performance factors (TPF) with the Reynolds number, Re for corrugated channel H_1 with phase shifts, ϕ of 0° , 90° and 180°

Improvements in heat transfer of flow in corrugated channels are also accompanied by increase in the frictional losses since corrugated channel disturbs the entire flow field. Therefore, it is necessary to evaluate the net enhancement obtained from using such channels. There are various methods according to the exchanger designs. In this study, the relative thermal-hydraulic performance enhancement is evaluated by thermal performance factor (TPF) expressed as:

$$TPF = \frac{Nu/Nu_0}{(f/f_0)^{1/3}}$$

The variations of the thermal performance factor (TPF) with phase shift, ϕ and the Reynolds number, Re are presented in figure 4.52. A high value of TPF indicates good thermal-hydraulic performance. This figure shows that thermal performance factor, TPF decreases with increasing the Reynolds number. For the tested three regimes of Reynolds numbers ($Re=2500$, 5000 and 7500), TPF decreases with increasing the phase shift angle, ϕ . The highest thermal performance factor, TPF is observed for flow in corrugated channel with phase shift angle, $\phi=0^\circ$. The maximum thermal performance factors, TPF at the Reynolds number, $Re=2500$ for the phase shift angles, $\phi=0^\circ$, 90° , and 180° are calculated as 1.48, 1.42, and 1.35, respectively. The channel of $\phi=0^\circ$ gives the best thermal performance factor, TPF at the low Reynolds number regime ($Re=2500$). Moreover, using corrugated channels at lower Reynolds numbers, Re seems to be more efficient than at higher Reynolds numbers, Re .

5. CONCLUSIONS

The main purpose of this study is to provide detailed information about flow characteristics and heat transfer enhancement of sinusoidal corrugated channels with different channel heights ($H_{\min}/H_{\max}=0.36, 0.54, 0.72$) and phase shift angles ($\phi=180^\circ, 90^\circ, 0^\circ$) experimentally and numerically. The experiments were performed for investigation of instantaneous and time-averaged structures of flow field in corrugated channel using the Particle Image Velocimetry (PIV) technique. Since it is difficult to take a detailed image of thirteen wave long channel in one view, the channel was divided into six field of views. On the other hand, thermal-hydraulic characteristics of water flow were investigated numerically for Reynolds numbers ranging from 2500 to 7500 using commercial software ANSYS FLUENT. The predicted hydraulic characteristics results of the numerical simulations were compared with the experimental results and found to be in good agreement.

Experimental time-averaged velocity contours were presented in order to compare the numerical results for straight channel and corrugated channels with phase angles of $180^\circ, 90^\circ$, and 0° , respectively. Since the core flow is forced to follow the waves of the surrounding geometry in the channel with phase angle, ϕ of 0° , a higher rate of interactions occur with recirculation. Hence $\phi=0^\circ$ is the case where significant mixing appear first among all shift angles, ϕ investigated in this study. In steady flow, the cool fluid passing straight through the center of the channel does not significantly mix with the warmer fluid near the walls of the channel. Since the warmer fluid becomes isolated in the large recirculating vortices filling the upper and lower furrows of each waves, the heat transfer does not enhance considerably in steady flow cases. As the Reynold number increases, the recirculation magnitude gets smaller and fresh fluid into the core region moves close to the wave furrows so that heat transfer enhances.

Using corrugated plates instead of flat plates are one of the most used passive techniques in compact heat exchanger designs. Lengthen of the flow path and presence of recirculation of flow in such channels improve heat transfer rate with more pressure drop compared to that of the parallel plate channel cases. In this

study, three parameters including channel phase shift angle, ϕ , channel height, H and Reynolds number, Re were systematically varied. The thermal performance factor (TPF) were used to obtain optimum operating conditions and design configurations of the sinusoidal corrugated channel. The following conclusions may be drawn:

- A significant heat transfer enhancement can be obtained with using sinusoidal corrugated channel as compared to the conventional parallel plate channel. The average Nusselt number, $\overline{Nu}$ of sinusoidal corrugated channel enhances up to 2.08 folds compared to the parallel plate channel depending on the Reynolds number, phase shift angle, ϕ , and height of the corrugated channel.
- Using corrugated channel enhances heat transfer with an increase in friction factor, f . The friction factor, f of sinusoidal corrugated channel increases up to 2.74 times compared to the parallel plate channel depending on Reynolds numbers, phase shift angles, ϕ , and height of the corrugated channel.
- Heat transfer of corrugated channel depends more on channel height, H than phase shift especially at high Reynolds number.
- The friction factor, f and Nusselt number, Nu decrease with the increase of the phase shift angles, ϕ for any channel heights considered.
- For the tested corrugation channels for all configurations, friction factor, f increases with increasing Reynolds numbers.
- For the tested corrugated channels with different channel heights ($H_{min}/H_{max}=0.36, 0.54, 0.72$) and phase shift angles ($\phi=180^\circ, 90^\circ, 0^\circ$) over three flow regions; $Re=2500$, $Re=5000$, and $Re=7500$, thermal performance factor (TPF) increased with decreasing the channel height and phase shift.
- Thermal performance factor (TPF) increases with decreasing Reynolds numbers.
- Among all configurations over the tested range of Reynolds numbers, the channel with $H_{min}/H_{max}=0.36$ and $\phi=0^\circ$ at the Reynolds number=2500 has highest thermal performance factor (TPF).

The numerical results needed to be validated with experimental results to draw conclusion related to heat transfer. Experimental and numerical results of the flow structures of flow in corrugated channel were compared and found to be in good agreement. Numerical simulations showed that using corrugations enhances heat transfer with pressure drop penalty.

REFERENCES

- ABED, A.M., SOPIAN, K., MOHAMMED, H.A., ALGHOUL, M.A., RUSLAN, M.H., MAT, S., AL-SHAMANI, A.N., 2015. Enhance heat transfer in the channel with V-shaped wavy lower plate using liquid nanofluids, *Case Studies in Thermal Engineering*, 5: 13-23.
- ABED, A.M., ALGHOUL, M.A., SOPIAN, K., MOHAMMED, H.A., MAJDI, H.S., AL-SHAMANI, A.N., 2015. Design characteristics of corrugated trapezoidal plate heat exchangers using nanofluids, *Chemical Engineering and Processing: Process Intensification*, 87: 88-103.
- ADRIAN, R.J., 1988. Review of particle image velocimetry research presented at the symposium on optical methods in flow and particle diagnostics, *Optics and Lasers in Engineering*, 9 (3-4): 317-319.
- ADRIAN, R.J., 1989. Particle-imaging techniques for experimental fluid mechanics, *Annual Review of Fluid Mechanics*, 23: 261-304.
- AHMED, M.A., SHUAIB, N.H., YUSOFF, M.Z., AL-FALAH, A.H., 2011. Numerical investigations of flow and heat transfer enhancement in a corrugated channel using nanofluid, *International Communications in Heat and Mass Transfer*, 38: 1368–1375.
- AHMED, M.A., SHUAIB, N.H., YUSOFF, M.Z., 2012. Numerical investigations on the heat transfer enhancement in a wavy channel using nanofluid, *International Journal of Heat and Mass Transfer*, 55: 5891-5898.
- AHMED, M.A., YUSOFF M.Z., SHUAIB, N.H., 2013. Effects of geometrical parameters on the flow and heat transfer characteristics in trapezoidal-corrugated channel using nanofluid, *International Communications in Heat and Mass Transfer*, 42: 69-74.
- AHMED, M.A., YUSOFF M.Z., NG, K.C., SHUAIB, N.H., 2014. Effects of corrugation profile on the thermal-hydraulic performance of corrugated channels using CuO-water nanofluid, *Case Studies in Thermal Engineering*, 4: 65-75.

- AKDAG, U., AKCAY, S., DEMIRAL, D., 2014. Sinusoidal wavy channels with Taylor-Goertler vortices, *International Communications in Heat and Mass Transfer*, 59: 17-23.
- ALI, H.H., HANAOKA, Y., 2002. Experimental study on laminar flow forced-convection in a channel with upper V-corrugated plate heated by radiation, *International Journal of Heat and Mass Transfer*, 45: 2107–2117.
- ALI, M., RAMADHYANI, S., 1992. Experiments on convective heat transfer in corrugated channels, *Experimental Heat Transfer*, 5: 175–93.
- ARSENYEVA, O., TOVAZHNYANSKY, L., KAPUSTENKO, O., KHAVIN, G., 2011. Investigation of the new corrugation pattern for low pressure plate condensers, *Applied Thermal Engineering*, 31: 2146–2152.
- ASAKO, Y., FAGHRI, M., 1987. Finite volume solutions for laminar flow and heat transfer in a corrugated duct, *Journal of Heat Transfer*, 109: 627–634.
- ASAKO, Y., NAKAMURA, H., FAGHRI, M., 1988. Heat transfer and pressure drop characteristics in a corrugated duct with rounded corners, *International Journal of Heat and Mass Transfer*, 31 (6): 1237-1245.
- BAHAIDARAH, H.M.S., 2009. A numerical study of fluid flow and heat transfer characteristics in channels with staggered wavy walls, *Numerical Heat Transfer, Part A: Applications: An International Journal of Computation and Methodology*, 51: 877-898.
- BAHAIDARAH, H.M.S., 2009. Fluid flow and heat transfer characteristics in sharp edge wavy channels with horizontal pitch, *Emirates Journal of Engineering Research*, 14 (1): 53-63.
- BAHAIDARAH, H.M.S., ANAND, N.K., CHEN, H.C., 2005. Numerical study of heat and momentum transfer in channels with wavy walls. *Numerical Heat Transfer, Part A*, 47: 417–439.
- BILEN, K., CETIN, M., GUL, H., BALTA, T., 2009. The investigation of groove geometry effect on heat transfer for internally grooved tubes. *Applied Thermal Engineering*, 29 (4): 753-761.

- CHANG, S.W., HUANG, B.J., 2012. Thermal performance of undulant narrow channel with skewed sinusoidal wall-waves and 45° ribs, *International Journal of Heat and Mass Transfer*, 55 (11-12): 3167-3177.
- CHOI, U.S., 1995. Enhancing thermal conductivity of fluids with nano-particles, *ASME Fluids Engineering Division*, 231: 99-103.
- CHOI, U.S., EASTMAN, J. A. November 12-17, 1995. Enhancing thermal conductivity of fluids with nanoparticles. *ASME International Mechanical Engineering Congress & Exposition*, San Francisco, CA.
- CIOFALO, M., STASIEK, J., COLLINS, M.W., 1996. Investigation of flow and heat transfer in corrugated passages—II. Numerical simulations, *International Journal of Heat and Mass Transfer*, 39, (1): 165-192.
- COMINI, G., NONINO, C., SAVINO, S., 2003. Effect of aspect ratio on convection enhancement in wavy channels, *Numerical Heat Transfer, Part A: Applications: An International Journal of Computation and Methodology*, 44 (1): 21-37.
- DESRUES, T., MARTY, P., FOURMIGUE, J.F., 2012. Numerical prediction of heat transfer and pressure drop in three-dimensional channels with alternated opposed ribs, *Applied Thermal Engineering*, 45–46: 52-63.
- DURMUS, A., BENLI, H., KURTBAS, I., GUL, H., 2009. Investigation of heat transfer and pressure drop in plate heat exchangers having different surface profiles, *International Journal of Heat and Mass Transfer*, 52: 1451–1457.
- EIMSA-ARD, S., PROMVONGE, P., 2008. Numerical study on heat transfer of turbulent channel flow over periodic grooves, *International Communication in Heat Mass Transfer*, 35: 844-852.
- ELSHAFEI, E.A.M., AWAD, M.M., EL-NEGIRY, E., ALI, A.G., 2010. Heat transfer and pressure drop in corrugated channels, *Energy*, 35 (1): 101-110.
- ERGIN, S., OTA, M., YAMAGUCHI, H., 2001. A numerical study of periodic turbulent flow through a corrugated duct, *Numerical Heat Transfer, Part A*, 40 (2): 139-156.

- ERRICO, O., STALIO, E., 2014. Direct numerical simulation of turbulent forced convection in a wavy channel at low and order one Prandtl number, *International Journal of Thermal Sciences*, 86: 374-386.
- FABBRI, G., 2000. Heat transfer optimization in corrugated wall channels. *International Journal of Heat and Mass Transfer*, 43: 4299-4310.
- FABBRI, G., ROSSI, R., 2005. Analysis of the heat transfer in the entrance region of optimized corrugated wall channel, *International Communications in Heat and Mass Transfer*, 32: 902-912.
- FAIZAL, M., AHMED, M.R., 2012. Experimental studies on a corrugated plate heat exchanger for small temperature difference applications, *Experimental Thermal and Fluid Science*, 36: 242-248.
- GARG, V.K., MAJI, P.K., 1988. Flow and heat transfer in a sinusoidally curved channel. *International Journal of Engineering Fluid Mechanics*, 1: 293-319.
- GOLDSTEIN, L., SPARROW, E.M., 1977. Heat/mass transfer characteristics for flow in a corrugated wall channel, *Transaction of the ASME, Journal of Heat Transfer*, 99:187-95.
- GRADECK, M., HOAREAU, B., LEBOUCHÉ, M., 2005. Local analysis of heat transfer inside corrugated channel, *International Journal of Heat and Mass Transfer*, 48: 1909-1915.
- GSCHWIND, P., REGELE, A., KOTTKE, V., 1995. Sinusoidal wavy channels with Taylor-Goertler vortices, *Experimental Thermal and Fluid Science*, 11 (3): 270-275.
- GURLEK, C., SAHIN, B., OZALP, C., AKILLI, H., 2008. Flow structures around a three-dimensional rectangular body with ground effect, *Wind and Structure*, 11 (5): 345-359.
- GURLEK, C., SAHIN, B., OZKAN G.M., 2012. PIV studies around a bus model, *Experimental Thermal and Fluid Science*, 38: 115-126.
- GUZMAN, A.M., CARNEDAS, M.J., URZUA, F.A., ARAYA, P.E., 2009. Heat transfer enhancement by flow bifurcations in asymmetric wavy wall channels, *International Journal of Heat and Mass Transfer*, 52: 3778-3789.

- HAN, X.H., CUI, L.Q., CHEN, S.J., CHEN, G.M., WANG, Q., 2010. A numerical and experimental study of chevron, corrugated-plate heat exchangers, *International Communications in Heat and Mass Transfer*, 37 (8): 1008-1014.
- HASANUZZAMAN, M., R. SAIDUR, ALI, M., MASJUKI, H.H., 2007. Effect of variables on natural convective heat transfer through V-corrugated vertical plates, *International Journal of Mechanical and Materials Engineering*, 2 (2): 109–117.
- HEIDARY, H., KERMANI, M.J., 2010. Effect of nano-particles on forced convection in sinusoidal-wall channel, *International Communications in Heat and Mass Transfer*, 37: 1520-1527.
- HERMAN, C., KANG, E., 2002. Heat transfer enhancement in a grooved channel with curved vanes, *International Journal of Heat Mass Transfer*, 45: 3741–3757.
- HEWITT, G.F., SHIRES, G.L., BOTT, T.R., 1994. *Process Heat Transfer*, CRC Press, Boca Raton, FL.
- HOSSAIN, M.Z., ISLAM, A.K.M.S., 2004. Fully developed flow structures and heat transfer in sine-shaped wavy channels, *International Communication Heat Mass Transfer*, 31 (6): 887-896.
- ISLAMOGLU, Y., PARMAKSIZOGLU, C., 2003. The effect of channel height on the enhanced heat transfer characteristics in a corrugated heat exchanger channel, *Applied Thermal Engineering*, 23:979–987.
- ISLAMOGLU, Y., PARMAKSIZOGLU, C., 2004. Numerical investigation of convective heat transfer and pressure drop in a corrugated heat exchanger channel, *Applied Thermal Engineering*, 24: 141-147.
- JAURKER, A.R., SAINI, J.S., GANDHI, B.K., 2006. Heat transfer and friction characteristics of rectangular solar air heater duct using rib–grooved artificial roughness, *Solar Energy*, 80: 895–907.
- JIN, D.X., LEE, Y.P., LEE, D.Y., 2007. Effects of the pulsating flow agitation on the heat transfer in a triangular grooved channel, *International Journal of Heat and Mass Transfer*, 50 (15–16): 3062–3071.

- KARAKUS, C., 2007. Investigation Of Tip Vortex Formation, Development And Merging Using Particle Image Velocimetry (PIV) Technique, Çukurova University Institute of Natural and Applied Sciences, ADANA, Ph.D. Thesis.
- KAKAÇ, S., LIU H., PRAMUANJAROENKIJ, A., 2002. Heat exchangers: selection, rating, and thermal design, Second Edition. CRC Press LLC.
- KEANE, R.D., ADRIAN, R.J., 1990. Optimization of Particle Image Velocimeters, Part I: Double Pulsed Systems, Measurement Science and Technology, 1: 1202-1215.
- KEANE, R.D., ADRIAN, R.J., 1991. Optimization of Particle Image Velocimetries, Part II: Multiple Pulsed System, Measurement Science and Technology, 2: 963-974.
- KEANE, R.D., ADRIAN, R.J., 1992. Theory of cross-correlation analysis of PIV images, Applied Scientific Research, 49: 191-215.
- KHOSHVAGHT-ALIABADI, M., 2014. Influence of different design parameters and Al_2O_3 -water nanofluid flow on heat transfer and flow characteristics of sinusoidal-corrugated channels, Energy Conversion and Management, 88: 96-105.
- KIM, M., BAIK, Y.J., PARK, S.R., SANG-RA, H., LIM, H. 2010., Experimental study on corrugated cross-flow air-cooled plate heat exchangers. Experimental Thermal and Fluid Science, 34: 1265–1272.
- KIM, S.K., 2001. An experimental study of developing and fully developed flows in a wavy channel by PIV. KSME International Journal 15: 1853–1859.
- KREITH, F., BOEHM, R.F, ET. AL., 1999. Heat and mass transfer, CRC Press LLC.
- KWON, H.G., HWANG, S.D., CHO, H.H., 2008. Flow and heat/mass transfer in a wavy duct with various corrugation angles in two dimensional flow regimes, Heat Mass Transfer, 45: 157–165.
- LANDRETH, C.C., ADRIAN, R.J., 1989. Measurement and refinement of velocity data using high image density analysis in Particle Image Velocimetry, Applications of Laser Anemometry to Fluid Mechanics, 484-497.
- LAOHALERTDECHA, S., WONGWISES, S., 2010. The effects of corrugation pitch on the condensation heat transfer coefficient and pressure drop of r-134a

- inside horizontal corrugated tube. *International Journal of Heat and Mass Transfer*, 53: 2924–2931.
- LAUNDER, B.E., SPALDING, D.B., 1974. The numerical computation of turbulent flows, *Computer Methods in Applied Mechanics and Engineering*, 3 (2): 269–289.
- LIN, J.H., HUANG, C.Y., SU, C.C., 2007. Dimensional analysis for the heat transfer characteristic in the corrugated channels of plate heat exchangers, *International Communications in Heat and Mass Transfer*, 34:304–312.
- LIU, S., SAKR, M., 2013. A comprehensive review on passive heat transfer enhancements in pipe exchangers, *Renewable and Sustainable Energy Reviews*, 19: 64–81.
- LORENZ, S., MUKOMILOV, D., LEINER, W., 1995. Distribution of the heat transfer coefficient in a channel with periodic transverse grooves, *Experimental Thermal and Fluid Science*, 11: 234–242.
- MAHMUD, S., ISLAM, A.K.M.S., FERROZ, C.M., 2003. Flow and heat transfer characteristics inside a wavy tube, *Heat and Mass Transfer*, 39 (5–6): 387–393.
- MAHMUD, S., ISLAM, A.K.M.S., MAMUN, M. A. H., 2002. Separation characteristics of fluid flow inside two parallel plates with wavy surface. *International Journal of Engineering Science*, 40: 1495–1509.
- MENDES, P., SPARROW, E. M., 1984. Periodically converging–diverging tubes and their turbulent heat transfer, pressure drop, fluid flow, and enhancement characteristics, *Journal of Heat Transfer*, 106: 55–63.
- METWALLY, H.M., MANGLIK, R.M., 2004. Enhanced heat transfer due to curvature-induced lateral vortices in laminar flows in sinusoidal corrugated-plate channels, *International Journal of Heat and Mass Transfer*, 47:2283–2292.
- MIRZAEI, M., SOHANKAR, A., DAVIDSON, L., INNINGS, F. 2014. 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: 432–446.

- MOHAMMED, H.A., ABED, A.M., WAHID, M.A., 2013. The effects of geometrical parameters of a corrugated channel with in out-of-phase arrangement, *International Communications in Heat and Mass Transfer*, 40: 47-57.
- NAPHON, P., 2007. Heat transfer characteristics and pressure drop in channel with v-corrugated upper and lower plates, *Energy Conversion and Management*, 48(5): 1516–24.
- NAPHON, P., 2007. Laminar convective heat transfer and pressure drop in the corrugated channels, *International Communications in Heat and Mass Transfer*, 34: 62–71.
- NAPHON, P., 2009. Effect of wavy plate geometry configurations on the temperature and flow distribution, *International Communication in Heat and Mass Transfer*, 36: 942–946.
- NICENO, B., NOBILE, E., 2001. Numerical analysis of fluid flow and heat transfer in periodic wavy channels, *International Journal of Heat and Fluid Flow*, 22:156– 167.
- NILPUENG, K., WONGWISES, S., 2006. Flow pattern and pressure drop of vertical upward gas–liquid flow in sinusoidal wavy channels, *Experimental Thermal and Fluid Science*, 30: 523–534.
- NISHIMURA, T., 1984. Flow characteristics of the polymeric system with high loading of inorganic Filler, *Kobunshi Ronbunshu*, 41 (7):413-420.
- NISHIMURA, T., BIAN, Y.N., MATSUMOTO, Y., KUNITSUGU, K., 2003. Fluid flow and mass transfer characteristics in a sinusoidal wavy-walled tube at moderate reynolds numbers for steady flow, *Heat and Mass Transfer*, 39: 239–248.
- NISHIMURA, T., KOJIMA, N., 1995. Mass transfer enhancement in a symmetric sinusoidal wavy-walled channel for pulsatile flow, *International Journal of Heat and Mass Transfer*, 38 (9): 1719-1731.
- NISHIMURA, T., MATSUNE, S., 1996. Mass transfer enhancement in a sinusoidal wavy channel for pulsatile flow, *Heat and Mass Transfer*, 32(1-2): 65-72.

- NISHIMURA, T., MATSUNE, S., 1998. Vortices and wall shear stresses in asymmetric and symmetric channels with sinusoidal wavy walls for pulsatile flow at low reynolds numbers, *International Journal of Heat and Fluid Flow*, 19: 583-593.
- NISHIMURA, T., MURAKAMI, S., ARAKAWA, S., KAWAMURA, Y., 1990. Flow observations and mass-transfer characteristics in symmetrical wavy-walled channels at moderate reynolds-numbers for steady flow, *International Journal of Heat and Mass Transfer*, 33: 835-845.
- O'BRIAN, J.E., SPARROW, E.M., 1982. Corrugated-duct heat transfer, pressure drop, and flow visualization, *Transaction of the ASME, Journal of Heat Transfer*, 104:410–416.
- OYAKAWA, K., SHINZATO, T., MABUCHI, I., 1989. The effects of the channel width on heat transfer augmentation in a sinusoidal wave channel, *JSME International Journal, Series II, Vol 32, 3*: 403-410.
- PEHLIVAN, H., TAYMAZ, I., ISLAMOGLU, Y., 2013. Experimental study of forced convective heat transfer in a different arranged corrugated channel, *International Communications in Heat and Mass Transfer*, 46: 106-111.
- PETHKOOL, S., EIAMSA-ARD, S., KWANKAOMENG, S., PROMVONGE, P., 2011. Turbulent heat transfer enhancement in a heat exchanger using helically corrugated tube, *International Communications in Heat and Mass Transfer*, 38 (3): 340-347.
- RAHIMI-ESBO, M., RANJBAR, A.A., RAMIAR, A., ARYA, A., RAHGOSHAY, M., 2012. Numerical study of turbulent forced convection jet flow in a converging sinusoidal channel, *International Journal of Thermal Sciences*, 59: 176-185.
- RAMGADIA, A.G., 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*, 55: 2494-2509.

- RAMGADIA, A.G., SAHA, A.K., 2013. Numerical study of fully developed flow and heat transfer in a wavy passage, *International Journal of Thermal Sciences*, 67: 152-166.
- ROSAGUTI, N.R., FLETCHER, D.F., HAYNES, B.S., 2007. Low-Reynolds number heat transfer enhancement in sinusoidal channels, *Chemical Engineering Science*, 62 (3): 694-702.
- ROZZI, S., MASSINI, R., PACIELLO, G., PAGLIARINI, G., RAINIERI, S., TRIFIRO, A., 2007. Heat treatment of fluid foods in a shell and tube heat exchanger: comparison between smooth and helically corrugated wall tubes, *Journal of Food Engineering*, 79: 249–254.
- RUSH, T.A., NEWELL, T.A., JACOBI, A.M., 1999. An experimental study of flow and heat transfer in sinusoidal wavy passages, *International Journal of Heat and Mass Transfer*, 42(9): 1541-1553.
- RUSS, G., BEER, H., 1997. Heat transfer and flow field in a pipe with sinusoidal wavy surface—I. Numerical investigation, *International Journal of Heat and Mass Transfer*, 40 (5): 1061–1070.
- RUSS, G., BEER H., 1997. Heat transfer and flow field in a pipe with sinusoidal wavy surface—II. Experimental investigation, *International Journal of Heat and Mass Transfer*, 40 (5): 1071-1081.
- SAHIN, B., OZTURK A.N., AKILLI, H., 2007. Horseshoe vortex system in the vicinity of the vertical cylinder mounted on a flat plate, *Flow Measurement and Instrumentation*, 18 (2): 57-68.
- SANIEI, N., DINI, S., 1993. Heat transfer characteristics in a wavy-walled channel, *Journal of Heat Transfer*, 6.115-788.
- SAVINO, S., COMINI, G., NONINO, C., 2004. Three-dimensional analysis of convection in two-dimensional wavy channels, *Numerical Heat Transfer Part A – Applications*, 46: 869–890.
- SAWYER, D., SEN, M., CHANG, H. C., 1998. Heat transfer enhancement in three dimensional corrugated channel flow, *International Journal of Heat and Mass Transfer*, 41: 3559–3573.

- SHENG, J., MENG, H., FOX, R.O., 2000. A large eddy PIV method for turbulence dissipation rate estimation, *Chemical Engineering Science*, 55: 4423-4434.
- SNYDER, B., LI, K.T., WIRTZ, R.A., 1993. Heat transfer enhancement in a serpentine channel, *International Journal of Heat and Mass Transfer*, 36 (12): 2965–2976.
- SPARROW, E.M., COMB, J.W., 1983. Effect of interval spacing and fluid flow inlet conditions on a corrugated wall heat exchanger, *International Journal of Heat and Mass Transfer*, 26 (7): 993–1005.
- STASIEK, J.A., COLLINS, M.W., CIOFALO, M., CHEW, P.E., 1996. Investigation of flow and heat transfer in corrugated passages-I. Experimental results, *International Journal of Heat and Mass Transfer*, 39 (1):149–164.
- STONE, K., VANKA, S. P., 1999. Numerical study of developing flow and heat transfer in a wavy passage, *Journal of Fluids Engineering*, 121: 713–719.
- SUI, Y., TEO, C.J., LEE, P.S., 2012. Direct numerical simulation of fluid flow and heat transfer in periodic wavy channels with rectangular cross-sections, *International Journal of Heat and Mass Transfer*, 55 (1–3): 73-88.
- SUI, Y., TEO, C. J., LEE, P.S., CHEW, Y.T., SHU, C., 2010. Fluid flow and heat transfer in wavy microchannels, *International Journal of Heat and Mass Transfer*, 53: 2760–2772.
- TANDA, G., VITTORI, G., 1996. Fluid flow and heat transfer in a two-dimensional wavy channel, *Heat and Mass Transfer*, 31 (6): 411-418.
- TOLENTINO, F.O., MENDEZ, R.R., GUERRERO, A.H., PALOMARES, B.G., 2008. Experimental study of fluid flow in the entrance of a sinusoidal channel, *International Journal of Heat and Fluid Flow*, 29: 1233-1239.
- TUNAY, T., SAHIN, B., OZBOLAT, V., 2014. Effects of rear slant angles on the flow characteristics of Ahmed bod, *Experimental Thermal and Fluid Science*, 57: 165-176.
- VICENTE, P.G., GARCIA, A., VIEDMA, A., 2004. Mixed convection heat transfer and isothermal pressure drop in corrugated tubes for laminar and transition flow, *International Communications in Heat and Mass Transfer*, 31: 651–662.

- WANG, C.C., CHEN, C.K., 2002. Forced convection in a wavy-wall channel, *International Journal of Heat and Mass Transfer*, 45: 2587–2595.
- WANG, C.C., FU, W.L., CHANG, C.T., 1997. Heat transfer and friction characteristics of typical wavy fin-and-tube heat exchangers, *Experimental Thermal and Fluid Science*, 14: 174-186.
- WANG, G., VANKA, S.P., 1995. Convective heat transfer in periodic wavy passages, *International Journal of Heat and Mass Transfer*, 38 (17): 3219-3230.
- WEBB, R.L. 1987. Enhancement of single-phase convective heat transfer, in *Handbook of single-Phase Convective Heat Transfer* (Eds. Kakaç, S., Shah, R.K., Aung, W.), John Wiley and Sons, New York.
- WEN, M.Y., HO, C.Y., 2009. Heat-transfer enhancement in fin-and-tube heat exchanger with improved fin design, *Applied Thermal Engineering*, 29: 1050–1057.
- WESTERWEEL, J., 1997. Fundamentals of digital particle image velocimetry, *Measurement Science and Technology*, 8: 1379-139.
- WILLERT, C., GHARIB, M., 1991. Digital particle image velocimetry, *Experiments in Fluids*, 10: 181-193.
- XIE, G.N., WANG, Q.W., ZENG, M., L.L.Q., 2007. Numerical investigation of heat transfer and fluid flow characteristics inside a wavy channel, *Heat and Mass Transfer*, 43 (7): 603-611.
- XUAN, Y. M., LI, Q., 2000. Heat transfer enhancement of nanofluids. *International Journal of Heat and Fluid Flow*, 21: 58-64.
- YANG, K.S., 2000. Large Eddy Simulation of turbulent flows in periodically grooved channel, *Journal of Wind Engineering and Industrial Aerodynamics*, 84: 47–64.
- YIN, J., YANG, G., LI, Y., 2012. The effects of wavy plate phase shift on flow and heat transfer characteristics in corrugated channel, *Energy Procedia*, 14: 1566-1573.
- ZHANG, J., KUNDU, J., MANGLIK, R.M., 2004. Effect of fin waviness and spacing on the lateral vortex structure and laminar heat transfer in wavy-

- plate-fin cores, *International Journal of Heat and Mass Transfer*, 47: 1719–1730.
- ZHANG, L., CHE, D., 2011. Influence of corrugation profile on the thermal hydraulic performance of cross-corrugated plates, *Numerical Heat Transfer, Part A*, 59: 267–296.
- ZIMMERER, C., GSCHWIND, P., GAISER, G., KOTTKE, V., 2002. Comparison of heat and mass transfer in different heat exchanger geometries with corrugated walls, *Experimental Thermal and Fluid Science*, 26 (2-4): 269-273.
- ZIMPAROV, V., 2001. Enhancement of heat transfer by a combination of threestart spirally corrugated tubes with a twisted tape, *International Journal of Heat and Mass Transfer*, 44: 551-574.
- ZIMPAROV, V., 2002. Enhancement of heat transfer by a combination of a single-start spirally corrugated tubes with a twisted tape, *Experimental Thermal and Fluid Science*, 25: 535-546.

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