Numerical Investigation on the Fluid Flow and Heat Transfer in the Entrance Region of Wavy Channel

Nabou Mohamed\textsuperscript{a,*}, Biara Ratiba Wided\textsuperscript{b}, El Mir Mohamed\textsuperscript{a}
Missoum Abd el karim\textsuperscript{a} Bouanini Mohamed\textsuperscript{a}.

\textsuperscript{a}Laboratory ENERGARID, University of Béchar B.P. 417, Algeria
\textsuperscript{b}Lecturer, University of Béchar B.P. 417, Algeria

Abstract

In this study, the fluid flow and heat transfer in the entrance region in a converging-diverging channel with sinusoidal wall corrugations are investigated. Numerical solutions are obtained using the control-volume finite-difference method. Development of the hydrodynamic, thermal fields, Nusselt number and viscous constraint are presented for different flow rates, wall corrugations severity and Prandtl number values. In the channel entrance zone, the viscous constraint tangential as well as the local Nusselt number are characterized by a very fast decrease and their amplitudes increase with increasing the wall corrugations and the Reynolds number. The periodicity character as well as the maximal velocity are influenced by variations of the rate corrugation and the Reynolds number.

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Key words: Forced convection, Laminar flow, Entrance region, Wavy channel.

1. Introduction

The exchanger plates have been originally studied to meet the needs of milk industry, and then used in different branches especially in the fields of medical and biomedical engineering, nuclear, naval and chemistry. The channel with a periodically convergent- divergent section is one of means used to increase efficiently the heat transfer. Many experimental researches have been done to establish methods of study

* Corresponding author. Tel.: +0-000-000-0000 ; fax: +0-000-000-0000 .
E-mail address: mnabou10@yahoo.fr.
and conception of this type of exchangers, mainly for the sake of improving the thermic transfer by maintaining drip pressures at a very low level.

| Nomenclature       | Greek symbols                     |
|--------------------|----------------------------------|
| A                  | amplitude of wavy surface        |
| C                  | average separation distance between wavy walls |
| Cp                 | specific heat of fluid at constant pressure |
| L                  | wavelength of the wavy wall |
| Nuₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜₜ verteaux et al. [1] have investigated experimentally the behavior flow and local heat transfer for laminar and transitional flows in sinusoidal wavy passages. The heat transfer experiments confirm that instabilities observed in the low visualisation experiments cause a heat transfer enhancement in the wavy channel. Asako et al. [2, 4, 5] have determined numerically heat transfer and pressure drop characteristics in a corrugated channel with the rounded angles. The duct boundaries were approximated by a cosine function.

Wang and Vanka [6] have studied the rates of heat transfer for flow through a periodic array of wavy passages. They found that the average Nusselt numbers for the wavy wall channel were only slightly larger than those for a smooth channel. Bourouis and Prevost [7] studied the influence of the wavy wall geometry on the internal hydrodynamics and the low arrangement for the case of fully developed flow in a pipe under laminar conditions. Wang and Chen [8] analyzed rates of heat Transfer for flow through a sinusoidally curved converging-diverging channel. Results show that the amplitudes of the Nusselt number and the skin-friction coefficient increase with an increase in the Reynolds number and the amplitude-wavelength ratio. Fontes et al. [12] have investigated experimentally the developing laminar flow non-isothermal and non-Newtonian in the entrance region of short tubes with heating and cooling conditions. Rahmena et al. [13] studied numerically the bidimensional laminar flow and heat transfer through an array of parallel lat plates with finite thickness. Results indicate than for all cases, as the low develops, the friction coefficient approaches a constant value, which resembles the fully developed low condition in the duct. For plates with a uniform and constant heat lux, convection heat transfer along the entrance region varies with Reynolds number, blockage ratio and Prandtl number. Magno et al. [14] employed the generalized integral transform technique in non-Newtonian fluids within a parallel plates.
channel. It was verified that for values of power-law indices greater than unity, lower values for the Nusselt numbers in the entrance region are obtained. Consequently, in most investigations either numerical or experimental, the low is always supposed to take place in the part of the channel, far from the entrance and the exit, so that it can be considered established and periodic. However, in real situations the low field in the channel includes not only the fully developed region but also the entrance region.

In this paper a numerical analysis is performed to study the laminar forced convection in the entrance region of a wavy-channel which fourteen undulations. Transfers equations are solved using the ADI method and Thomas algorithm. The linkage between the velocity and pressure variables is handled by the PISO algorithm. The effects of Prandtl number, Reynolds number and the amplitude of the sinusoidal profile of the wall on the transfers in the entrance region of this channel are studied. Besides, we follow the flow of the entrance where the velocity profile is initially rectangular or uniform (flow in block) and to see from what section of the entrance the flow becomes periodic.

2. Problem formulation and numerical solution

We consider in our study a channel, schematized on Fig.1, periodically convergent-divergent, with sinusoidal undulation parallel plates and the temperature of the lower plate is double of that upper plate. The smooth parts to the exit of the channel are adiabatic. The fluid flow in the channel will be supposed bi-dimensional; hence the height of plates is supposed larger in relation to their spacing. In addition, we consider the forced and stationary flow of a fluid (water or air) Newtonian, laminar, incompressible and constant physical properties.

The flow is governed by equations the Navier-Stokes. In addition the retained simplifying hypotheses, we disregarded the action of the weight in the movement equation and effects of the viscous dissipation and the thermal expansion in the heat equation. The system can be written in cartesian coordinates in the following dimensionless form:

\[
\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0
\]

\[
U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re} \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right)
\]

\[
U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re} \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right)
\]
\[
U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{1}{Re \cdot Pr} \left( \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right)
\] (3)

Where dimensionless variables are defined by:

\[
\begin{align*}
X &= \frac{x}{L}, \\
Y &= \frac{y}{L}, \\
A &= \frac{a}{L}, \\
\theta &= \frac{T - T_0}{T_w - T_0}, \\
U &= \frac{u}{U_e}, \\
V &= \frac{v}{U_e}, \\
P &= \frac{P}{\rho U_e^2}, \\
Re &= \frac{U_e L}{\nu}, \\
Pr &= \frac{\mu C_p}{\lambda}
\end{align*}
\] (4a-i)

As it is shown in Fig.1 the wall profile in the region \(0 < X < L_0\) is given by the following equation:

\[
y = \frac{c}{2} + a \sin \left( \frac{\pi x}{L} \right)
\] (5)

Thus, the conditions to the corresponding limits are the following:

- For lower wall:
  \[
  U = V = 0
  \] (6a-b)

  \[
  \theta = 2 \quad \text{with} \quad X \leq L_0
  \] (7)

- For upper wall:
  \[
  U = V = 0
  \] (8a-b)

  \[
  \theta = 1. \quad \text{with} \quad X \leq L_0
  \] (9)

- At the channel entrance:
  \[
  U = 1; V = 0; \theta = 0
  \] (10a-c)

At the exit of the smooth part of the channel all gradients are null; although this boundary condition is strictly valid only when the flow is fully developed, its use in other low conditions is also admissible for computational convenience, provided that the outlet boundary is located in a region where the low is in the downstream direction and sufficiently far from the region of interest [13].

Once the thermal and dynamic fields are determined, several important quantities can be deduced. The local Nusselt number relative to the corrugated surface is defined by the relation:

\[
Nu_x = \frac{\left( \frac{\partial T}{\partial n} \right)_{w}}{T_w - T_0}
\] (11)
With \( n \) the normal to the wall for the sinusoidal part.

The average Nusselt number is defined by the following relation:

\[
\overline{N_{u_m}} = \frac{1}{L_S} \int_0^{L_X} \left( \frac{N_u X}{1 + Y^2} \right)^{\frac{1}{2}} dX \quad \text{where} \quad L_S = \int_0^{L_X} \left( 1 + Y^2 \right)^{\frac{1}{2}} dX
\]  

(12)

Integrals of equations (17) and (18a) are calculated by the Simpson method [17]. The viscous constraint tangential to the wall can be calculated by the following expressions:

\[
\tau_w = \mu \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)_w
\]  

(13)

To determine the flow dynamic field, a simulation by a fluid dynamic calculation (C. F. D.) has been used. The general equation is discretized according to the Patankar procedure [19]. The discretization in finite volume employed uses an integral form of the general equation; the calculation domain is subdivided in a set of small elementary control volumes (or cells) on which the integration is applied. Finally, this discretization produces an algebraic equation system. The method numeric ADI (Alternating Direction Implicit) [20] has been used for the resolution of the equation system. The PISO algorithm (Pressure- Implicit with Splitting of Operators) [18] has been applied to solve the coupled governing equation system. The convergence criterion for all calculations, represented by the relative magnitude of the error \( \varepsilon \), was grouped, respectively by velocity fields, pressure and temperature by the following relation:

\[
\varepsilon = \left| \frac{\Phi^{n+1} - \Phi^n}{\Phi^{n+1}} \right| < 10^{-5}
\]  

(14)

Where \( \Phi \) represents the components axial and transversal of the velocity, pressure and temperature.

The grid sharpness precision test has been made to establish an independent numerical solution grid. In Fig. 2 we show a test of precision, the latter is made for grids 355x16, 431x21 and 565x31 to \( \text{Re} = 500, A = 0.3 \) and \( \text{Pr} = 6.69 \) for the channel lower wall. While comparing grids 565x31 and 431x21, we notice that the viscous constraint tangential shapes of both grids are practically confounded. There are some gaps in Nusselt numbers in the groove of every channel undulation; it shows that more grids are required to solve the temperature field to get a better result for the Nusselt number. Therefore, as shown on Fig. 2, results of the grid 431x21 of our study with a refined grid especially very close to walls and those of the grid.
3. Results and Discussions

Calculations have been done in such a way to make the dimensionless parameters vary alternately, one by one, which are the profile wall amplitude $A$, the Reynolds number $Re$ and the Prandtl number $Pr$. The aim is to see the influence of each of these parameters on the one hand on the flow establishment, on the thermal transfer and on the viscous constraint tangential.

4. Periodicity and Establishment of the Hydrodynamic Regime

At the channel entrance the flow is in block, that means that the axial velocity is uniform and the transverse component is null. We notice, while following the fluid flow along the channel that its profile alters. This modification is mainly due to the action of the viscosity strengths. As the fluid velocity decreases close to the walls neighborhood, because of the viscous friction, it increases to the neighborhood of the symmetry axis, since the layers that are in contact with the walls are stopped. In the vicinity zone of the axis the viscous effects are weak and the velocity profile keeps a maximal value of the flow; this maximal value increases with $X$ when we move away from the entrance; the uniform velocity distribution in the central part accelerates so that the mass flow rate preserves itself. The thickness of the stopped fluid layers increases more and more until it reaches the channel axis; the hydrodynamic boundary layers join themselves, whence the parabolic profile settles and becomes stable at the end of a length $L_m^e$ called entrance length. The mechanical entrance length for the smooth channel is very close to the entrance (see Fig.3.). Far from the entrance, once the regime becomes established, the maximal velocity in the case of the smooth channel and for all the range of the Reynolds number is $U_{max} = 1.5$, this last is compatible with $U_{max} = 1.5U_m$ data by [21] and constitute a validation of our work (to the entrance of our channel $U_m = 1$). From this distance, there will be any no more variation of the axial component according to the $X$ axis, as for the component transverse $V$, it is null, therefore the condition of establishment is verified: $\partial U / \partial X; V = 0$

With the increase of the wall amplitude, the convergent section of the channel decreases indeed. It provokes a reduction of the section of passage of the fluid more and more. Besides, following the action of the viscosity strengths layers in contact with walls are braked, whereas in the central part, they accelerate so that the debit preserves itself.

Fig. 3. Axial velocity distribution to the four sections of the second and third undulation for $A = 0.4$, $Pr = 6.69$ and $Re = 500$. 

565x31 are compatible.
5. Flow and strengths of shearing

Variations of amplitudes of the wavy channel and the Reynolds number have an important effect on the dynamic growth of the flow field. The intensity of swirls grows with the increase of amplitude, which generates a good contribution to the transfer. The increase of the Reynolds number produces a similar effect with a channel to fixed corrugation. We notice that the cell's swirl occurs and increases its size with the increase of the Reynolds number. Concerning the weak rates of corrugation the Reynolds number doesn't have an influence on the flow, and the fluid circulates along the channel without recirculation while merely adopting the wavy shape of passages.

The evolution of the viscous constraint tangential with the rate of corrugation and the Reynolds number is represented in Fig. 4. In the region of entrance of the channel, the viscous constraint tangential is characterized by a very fast decrease. In the first period of the channel ($0 \leq X \leq 0.02$), we notice that the decrease becomes very small in the divergent section with the reduction of the Reynolds number. But from the second period and far from the exit of the channel, the contrary effect occurs. Far from the entrance and the exit of the channel, where the regime is established, the viscous constraint tangential increases in the convergent sections of the channel with the increase of the rate of corrugation and the Reynolds number, which is in conformity with numerical results of [8].

The evolution of the viscous constraint tangential is represented by a harmonic curve, this last has the same frequency that of the wavy surface. Concerning a smooth parallel channel and far from the entrance and the exit, the viscous constraint tangential is independent of the Reynolds number and stayed constant along the channel; but it undergoes the same phenomenon that of a waved channel in the entrance region. For different rates of corrugation and Reynolds numbers, maximal and minimal values of the viscous constraint tangential are located respectively to positions of minimal and maximal sections of the wavy channel. Besides, this maximum increases with the increase of the Reynolds number and the rate of corrugation becomes sharper. But the minimum decreases with the increase of these two parameters and the curve becomes flat. As it is shown in streamlines curves, where there is formation of the swirls and from the flow reversing, the viscous constraint tangential is negative in grooves of the wavy channel.

6. Distribution of the Temperature Field and Nusselt Numbers

For the elaborate work, we signal the existence of two important parameters that affects the transfer of heat in channels of wavy walls: the conventional development of the flow and effects of undulation amplitudes.

Distributions of temperatures reflect the growth of the heat transfer by convection due to corrugations of walls and swirls' lows. With the development of the recirculation in the hollow regions, there is a considerable effect of the layer thermal limits and the field of flow shows regions of thermal mixture. The temperature is governed by modes of heat transfers by convection, and conduction for the weak Reynolds
numbers. The growth of the Reynolds number that accompanies an intensification of heat transfer by
convection provokes a reduction of temperature. This reduction is more important when Reynolds number
is raised on one hand and the distance to the entrance is big on the other hand. For different Reynolds
numbers and different rates of corrugation, the evolution of the local Nusselt number according to the
abscissa is characterized then by a very fast decrease to the channel entrance. This evolution tends toward
a constant value which increases with the growth of the Reynolds number. The predominance of
convective transfers is accompanied by a shape of the local Nusselt number that has the tendency to
describe that of the wall. Since the convergent section is characterized by a gradient of velocity that
increases the rate of heat transfer; we notice that the local Nusselt number takes a maximal value in the
convergent section of every undulation than that in the divergent section. This maximal value increases
with the increase of the Reynolds number and decrease while moving away from the channel entrance. On
the other hand the number of minimal local Nusselt is located upstream inside the maximal section of
every undulation of the channel. Variations of the average Nusselt number for corrugations’ different rates
according to the Reynolds number are represented in Figs.5, 6, 7 and 8.

![Fig. 5. Distribution of the viscous constraint tangential for Pr = 6.69.](image)

According to the drawn curves, we distinctly notice the increase of the heat transfer with the increase
of the Reynolds number for the two numbers of Prandtl. For water, the heat transfer increases with the
reduction of the corrugation rate. The upper wall of corrugated channel is characterized by a good transfer
compared to the smooth channel for a Reynolds number lower to 1200. But the lower wall looses this
characteristic for the big rates of corrugation (A >0.3). For air and a Reynolds number lower to 1300, the
heat transfer increases with the increase of the corrugation rate for the upper wall of the channel.

In addition, the smooth wall is characterized by a good heat transfer compared to big rate of
corrugation (A >0.2). We can also note that transfers by diffusion are more important in air than in water.
The increase of the Prandtl number associate with a growth of heat transfer.
7. Conclusion

The forced convection of the bidimensional and stationary low, in the entrance region, of a fluid Newtonian through a sinusoidally curved converging-diverging channel was numerically investigated to isotherm walls of which the temperature of the lower wall is the double of that of the upper wall. The development of the low has been followed by the channel entrance (flow in block) until its establishment. In addition, the thermal conditions are imposed since the side of entrance; therefore hydrodynamic and thermal establishments are interdependent. Fields of velocity and of temperature, Nusselt number and the viscous constraint tangential are presented for the different rates of low ($100 \leq \text{Re} \leq 1500$) and different geometries ($0 \leq A \leq 0.5$) for the case of water and air. This study permitted us to verify that when the low becomes established its velocity is 1.5 times that of the entrance and stay constant for a smooth channel and that the length of entrance increases with the increase of the Reynolds number. The thermal establishment took place far from the entrance contrary to the hydrodynamic establishment. For a wavy channel, the characters of periodicity and the maximal velocity are influenced by rates of corrugation and flow. The variations of the amplitude of channel plates and the Reynolds number have an important effect on the dynamic growth of the low field. The intensity of the swirl grows with the rate of corrugation and/or of the Reynolds number, what generates a good contribution to the transfer of heat. To the weak rates of corrugation and far from the entrance region, the fluid circulates along the channel without recirculation while merely adopting the undulatory shape of passages. In the entrance region, the viscous constraint tangential and the Nusselt number are characterized by a very fast decrease. But far from this zone where the low is established, their harmonic curves have the same frequency that that of the wavy surface of the channel. Besides, their maximal and minimal values are respectively located to positions of minimal and maximal sections of the channel. These values increase with the increase of the amplitude and/or of the Reynolds number. The growth of the Reynolds number that is accompanied by an intensification of heat transfer by convection provokes a reduction of temperature. This reduction is important when the Reynolds number is raised and the distance of entrance is big. Transfers by diffusion are more important in air than in water. So, the increase of the Prandtl number is accompanied by a growth of heat transfer.

Fig. 7. Distribution of the viscous constraint tangential for $Pr = 6.69$.

Fig. 8. Distribution of the average Nusselt number relative to the upper wall for $Pr = 0.708$. 
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