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RESEARCH ARTICLE

NUMERICAL STUDY OF FORCED CONVECTION OF AIR IN A U-SHAPED CAVITY

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Abstract

In this work, we numerically study laminar flows in forced convection in a "U" -shaped channel. Our study was based on a finite volume method by the fluent code carried out for a Reynolds number varying in the range 100<Re<700, and the Rayleigh number $Ra=10^5$ and Prandtl number Pr=0.72. As a function of the Reynolds number, the results obtained are presented in the form of the thermal and dynamic field, the variation of the Nusselt number and the variation of the speed and average temperature at the outlet of the channel as a function of Re also. We see a significant improvement in heat exchange along all the walls of the channel as the number increases Re.

Introduction:-

The study of forced convection flows in a U-shaped channel is of practical interest in various fields, such as the pre-conditioning of air for its cooling, the so-called Canadian well (or air-ground heat exchanger). The use of geothermal cooling techniques is an alternative and makes it possible to afford ecological air conditioning.

This bibliographic research study presents a synthesized review of theoretical, analytical, numerical and experimental studies on air-ground heat exchangers.

BARTOLOMEU [1] is devoted to this work on the performance of an air-ground type heat exchanger. This pipe-type heat exchanger buried in the ground was designed at the ITP experimental station in Romillé. Its principle is based on temperature exchanges between the ground and the air circulating in the network of buried tubes. Proper sizing of this system is necessary to optimize its performance, which is analyzed throughout the year, distinguishing between winter and summer seasons.

Fudholi et al [2] studied the effect of mass flow rate, number and height of fins on efficiency by involving steady-state energy balance equations on the longitudinal fin absorber of solar collectors. The procedure for the theoretical resolution of the energy equations using a matrix inversion method and making some algebraic rearrangements. They saw that the efficiency of the collector increases leaking as the number and height of the fins increases.

Mebarki et al [3] made a study of the performance of an air-ground exchanger which was undertaken by way of analytical modelling. They had first validated their model of the ground temperature and the air temperature in the

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exchanger, then they analyzed the influence of a few parameters, namely: the depth, the diameter and the length of the tube on the temperature. inside the exchanger.

NEBBAR et al [4] assessed the potential of using so-called surface geothermal energy and the appropriate technology for its exploitation by determining ground temperature variations at different depths as well as determining ground temperature variations. air at the outlet of the exchanger, considering in this study the permanent flow of a Newtonian and incompressible fluid in a tube of circular section assuming that the dynamic regime is established. They showed the variation in air temperature at the outlet of the exchanger as a function of the geometric, thermal and site parameters of the exchanger, the characteristics of the medium, the passage geometry and the inlet parameters. and exit from the exchanger.

The work of KABORE [5] concerns the study of an air-ground heat exchanger intended for the cooling of a habitat in the Sahelian zone. It led to an analytical study by Fourier transform in order to understand the notions of damping and phase shift. He realized an experimental device in Ouagadougou. He also modeled on the COMSOL software the heat exchanges that take place in an air-ground heat exchanger and then the nodal method and a discretization of the equations by an implicit finite difference method for a numerical study of the cooling of a habitat using an air-ground heat exchanger.

Cuny [6] carried out a numerical study based on a 2D finite element modeling of an air-ground exchanger by evaluating the energy performance according to the different soil humidifications and the different rain scenarios. The results showed the interest of using a very humid coating soil to significantly increase the energy performance of the air-soil exchanger.

Mathematical Approach

This work is based on a numerical study of the dynamic and thermal behavior of forced convection in a U-shaped cavity (figure 01). It is a U-shaped tube filled with a Newtonian fluid at the occurrence of air. The tube is ventilated by a hot air inlet with a temperature of Tc on the left then crossing the tube and the horizontal walls are adiabatic (q=0).

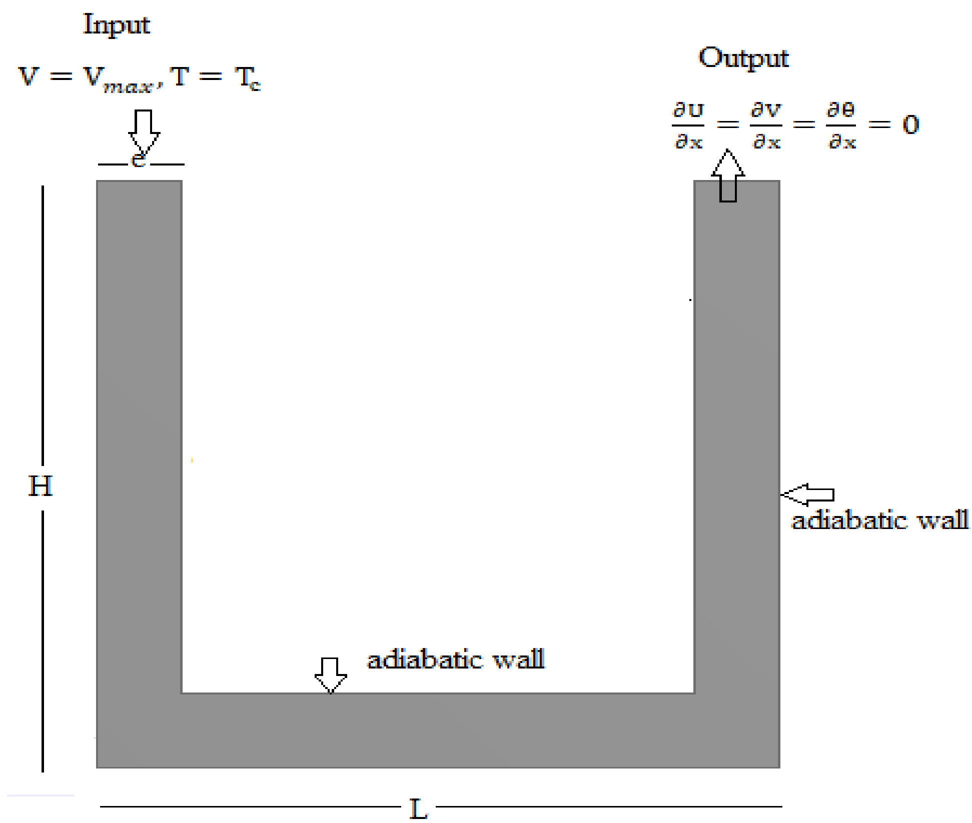

Figure 01:- Geometry of the problem.

Modeling of the studied system is based on the following simplifying assumptions:

1. Fluid flow and heat transfer are incompressible, two-dimensional and the laminar regime.
2. The thermophysical properties of the fluid (μ, Cp, and k) are constant.
3. Viscous dissipation is negligible. There is no heat source.
4. Boussinesq approximation is valid, it consists in considering that the density variations are negligible at the level of all the terms of the momentum equations $(\rho = \rho_0)$, except at the level of the gravity term. The variation of the density ρ as a function of temperature is given by: $\rho - \rho_0 = \rho_0 \beta (T - T_0)$

   ρ_0: the density of the fluid at the inlet temperature T_0

   β: the volume expansion coefficient of the fluid

By introducing the simplifying assumptions, the system of equations which govern the flow of the forced convection in Cartesian coordinates are written in the dimensionless form as follows:

Continuity equation

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0$$

Momentum equations

$$\frac{\partial U}{\partial t} + U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y} = -\frac{\partial P}{\partial x} + \frac{1}{Re}\left(\frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2}\right)$$

$$\frac{\partial V}{\partial t} + U\frac{\partial V}{\partial x} + V\frac{\partial V}{\partial y} = -\frac{\partial P}{\partial y} + \frac{1}{Re}\left(\frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2}\right) + \frac{Gr}{Re^2}\theta$$

Energy equation

$$\frac{\partial \theta}{\partial t} + U\frac{\partial \theta}{\partial x} + V\frac{\partial \theta}{\partial y} = \frac{1}{Re.\,Pr}\left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2}\right)$$

Initial conditions

$$U = V = \theta = 0$$

Conditions to the limits

At the input: $U = 0;\ V = \theta = 1$

To the output : $\frac{\partial U}{\partial x} = \frac{\partial V}{\partial x} = \frac{\partial \theta}{\partial x} = 0$

On the walls: $U = 0;\ V = \frac{\partial \theta}{\partial x} = 0$

The heat transfer calculations under the open cavity are measured in terms of the average Nusselt number at the vertical and horizontal walls as follows:

$$Nu = \frac{1}{\Delta T}\frac{\partial \theta}{\partial x}$$

The average Nusselt number is obtained by integrating the local Nusselt numbers over the entire length of the heated wall:

$$\overline{Nu} = \frac{1}{S}\int_S Nu\,dS$$

Numerical Method

The transfer equations described are nonlinear and coupled partial differential equations. Due to their complexity, these equations are solved by numerical techniques. The diagram used is simple for a mesh of 80*160. The time step being varied from 10^{-3} to 10^{-4}.

The convective flux coefficients are derived by a second-order upwind scheme while the diffusive flux coefficients are obtained by central differentiation. The turbulent impulse and compensated heat fluxes in this problem were modeled by a K-w SST confined flow viscosity model which is a combination of the K-w model near walls and K-Ɛ at the core of the flow. The velocity and pressure fields are coupled using the SIMPLE algorithm [7]. For better convergence, all numerical tests are performed with convergence threshold residuals for momentum, continuity and energy equations equal to 10^{-6}. In addition to the above equations, we have also solved the equation for the turbulent kinetic energy and its dissipation ω. The model contains six empirical constants whose following values are assigned according to some suitable studies in the literature [8, 9]:

$\sigma_k = 1.0; \quad \sigma_{\omega,1} = 2.0; \quad \sigma_{\omega,2} = 1.17; \quad \gamma_2 = 0{,}44; \quad \beta_2 = 0{,}083; \quad \beta^* = 0{,}09.$

Results And Discusion:-

We present the results and discuss the dynamic and thermal behavior of forced convection in a U-shaped cavity. We will present the isocurrents and isotherms for the different parameters such as the Rayleigh number Ra= 10^5, the Prandtl number (Pr=0.72) at the occurrence of air and different Reynolds numbers (Re = 100, 300, 490 and 700).

For Re=100 (figure 03), the isocurrents are almost parallel to each other and to the walls of the channel and from the vertical column of the channel towards the exit they start at a very slight deformation in the shape of a snake with a very high velocity at the exit. The boundary lines laminate the walls except in the lower corners of the channel, where there is flow separation. The isotherms represented on the left show that the heat exchange between the walls and the ventilation air is limited in the vertical column on the left (at the input). Indeed, the very low ventilation speed (therefore low flow) promotes rapid heat transfer in the vertical column. We observe that the fluid is practically isothermal at the same temperature as the walls in the rest of the channel.

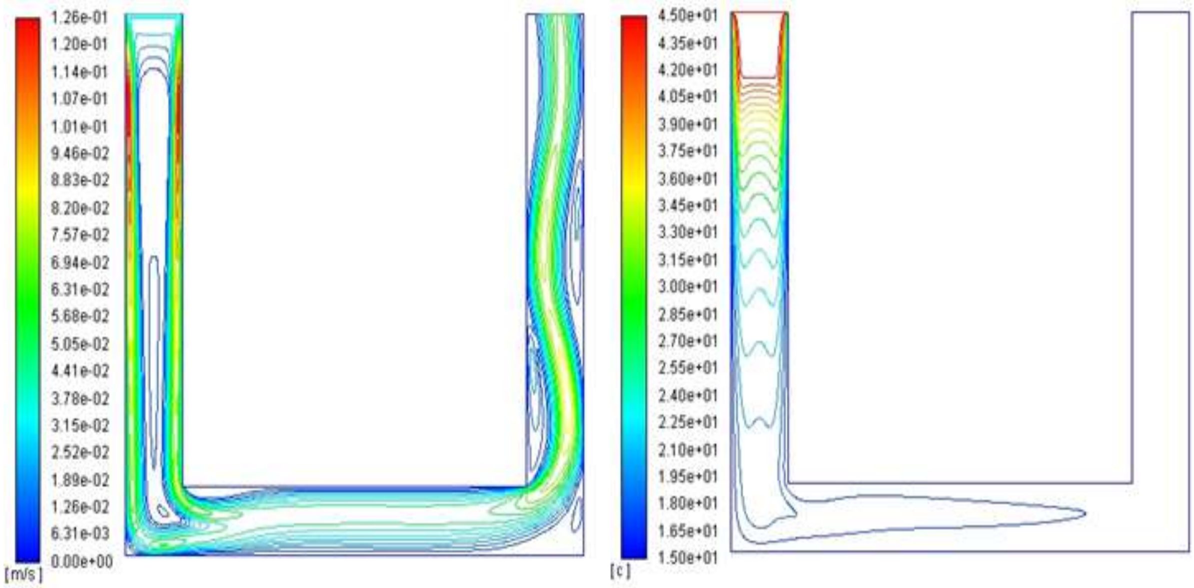

Figure 03:- Streamlines (left) and Isothermal lines (right) for Ra=10^5, Re=100.

By increasing Re, we see at the corners an appearance of flow contraction zones (figures 04, 05 and 06) but that in the right corner is stronger than that in the left corner. It is caused by the opposition of flow to the gravity of the fluid (the reversal of the direction of the flow). Bends at the elbow are caused by the change in direction of the air, which leads to a loss of pressure. It should be remembered that it is at the level of the elbow that the temperature gradient is very high. Note also that the two convection modes operate in the opposite direction, right column and in the same direction, left side. The areas of narrowing are accentuated as Re increases until the appearance of the entrainment cells at the level of the two corners, the structure of the flow becomes relatively complex.

The thermal field relative to these numbers of Re, shows that the cooling of hot air is done along the walls of the channel more and more as Re increases. A boundary layer type flow is noticeable in the right column when Re> 490.

There is a significant decrease in air temperature as it passes through the elbow. The temperature gradients are remarkable only at the inlet of the exchanger. The maximum values are noted at the level of the elbow. This strong variation of the gradient at the level of the elbow, can be explained by the change of direction of the air (from the vertical to the horizontal). This change of direction creates a shock at the level of theair particles. It is also an area of turbulence and pressure drop. We notice that whatever the depth of the ground, the air temperature decreases from the inlet to the outlet of the tube (depending on the H/L ratio).

The outlet air temperature decreases by (outlet temperature value), then increases almost linearly for the rest of the time until it reaches (average outlet temperature value for the different Reynolds numbers). This means that over time the temperature of the air at the outlet of the exchanger is lower than that of the air at the inlet (298 K). In other words, the air cools over time. This thermal behavior is explained by the fact that the air-tube system gives up its heat along the tube and over time.

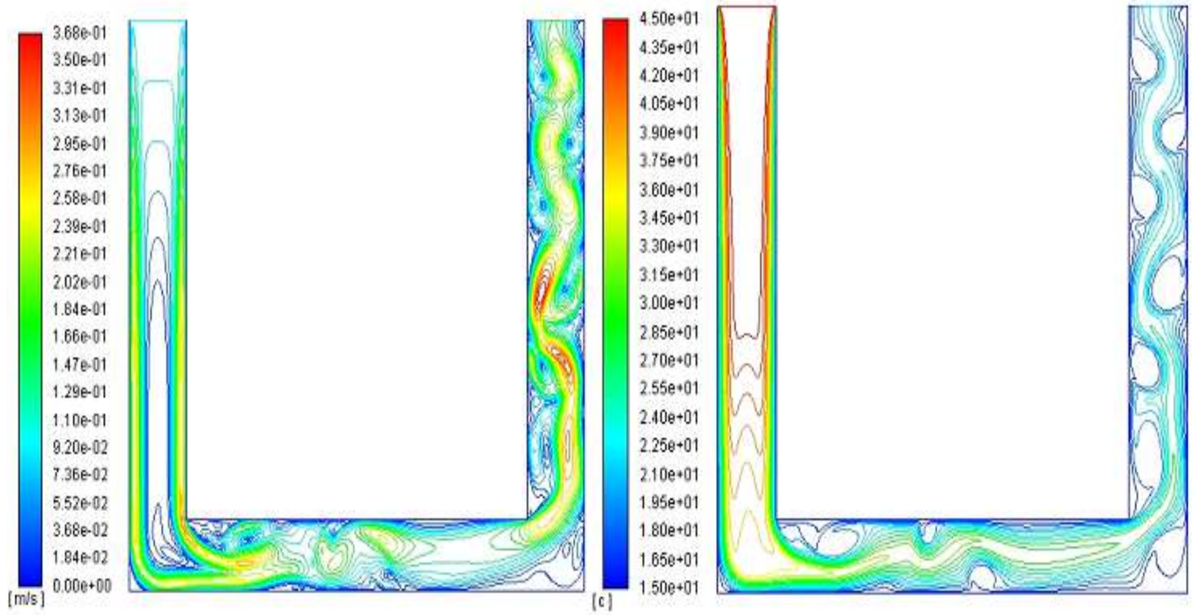

Figure 04:- Stream lines (left) and isothermal lines (right) for Ra=10^5, Re=300.

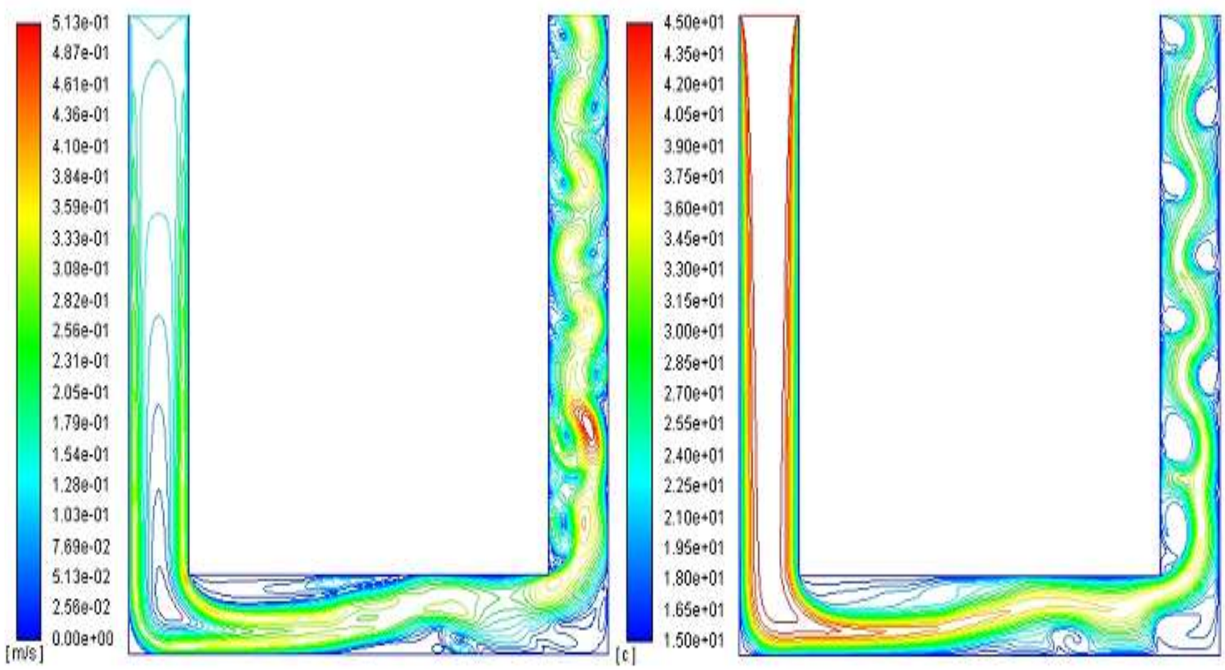

Figure 05:- Stream lines (left) and isothermal lines (right) for Ra=10^5, Re=490.

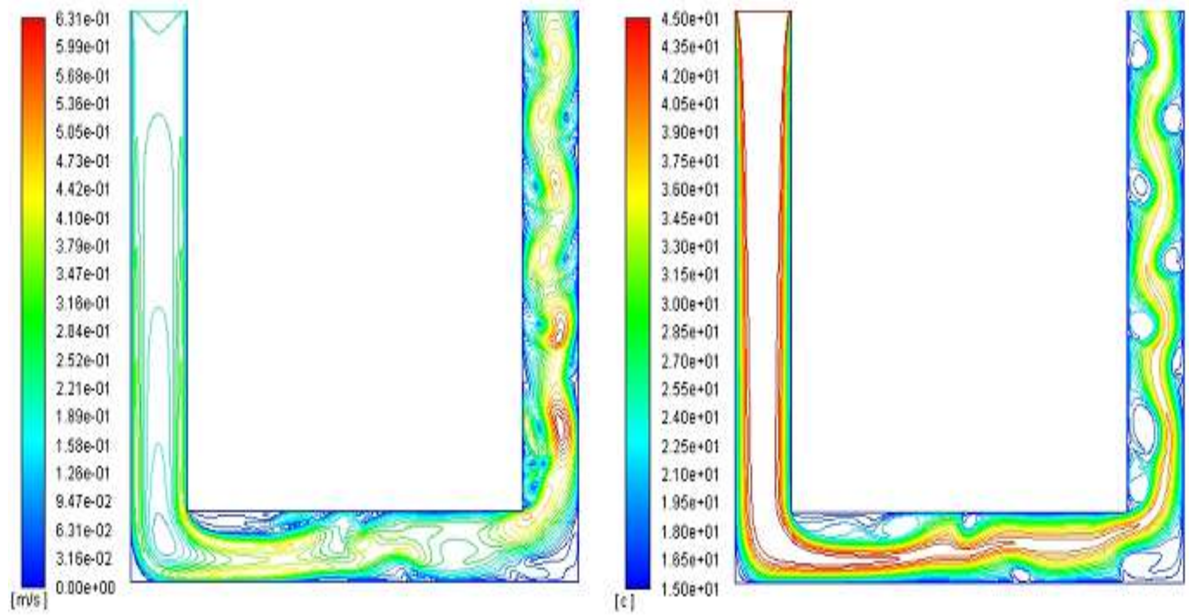

Figure 06:- Stream lines (left) and isothermal lines (right) for Ra=10^5, Re=700.

The Nusselt number characterizes the convective exchange between the air and the wall of the cavity tube of the U shape. Thus, in order to better understand the heat exchange between the external wall of the tube, we have shown the Nusselt number for the left, right and bottom outer wall of the tube. We note that from the entrance of the well (x= 0, 3) there is a thermal shock between the warmer entering area and the cooler wall of the well. This temperature difference between the surface and the wall gives a large increase in the Nusselt number at this level. As one enters the well, the area gradually loses its temperature approaching the temperature of the well, which justifies this decrease in the Nusselt number

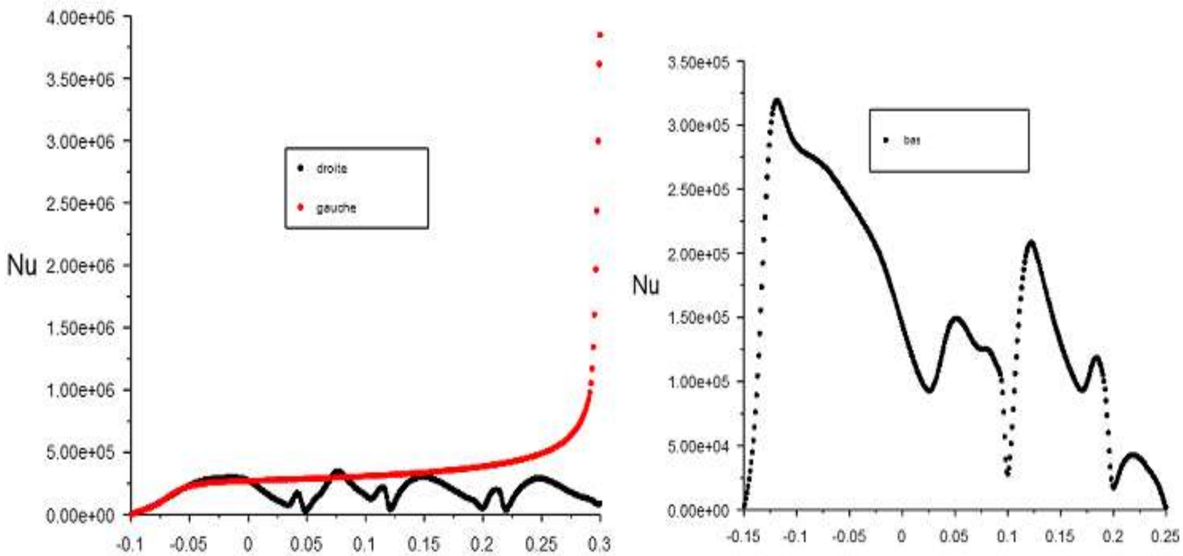

Figure 07:- Variation of Nusselt number for Ra=10^5, Re=700.

Conclusion:-

We have numerically studied the forced convection flows in a U-shaped channel representing a Canadian well for air cooling. The studied configuration is numerically simulated by the fluent code with direct resolution of the Navier–Stockes equations. The results obtained for different Reynolds numbers are very significant and give much information on the dynamic and thermal behavior of the exchanger.

At low Re, the heat exchange is limited to the right vertical column of the channel. Most of the channel is almost isothermal.
- When Re increases, there is a significant improvement in the heat exchange along all the walls of the channel.
- If one continues to increase Re, entrainment cells appear and cause a bottleneck at the corners of the channel.

Nomenclature

Latin letter	
C_p:specific heat capacity of fluid[J/kg.K]	e: thickness of the tube[m]
g : acceleration due to gravity$[m.s^{-2}]$	Gr : Grashof number$Gr = \frac{\alpha g L^3 \Delta T}{\nu^2}$
H: tube height[m]	h: coefficient of transfer of heat by convection $h = \frac{q}{\Delta T} \left[Wm^{-2}K^{-1}\right]$
$k$fluid thermal conductivity [W/m K]	L: length of the tube[m]
Nu: Nusselt number $Nu = \frac{hL}{\lambda}$	$\overline{Nu}$: Average Nusselt number
P : Pressure$[Pa]$	Pr : Prandtl number $Pr = \frac{\nu}{\alpha}$
q: heat flux density$\left[Wm^{-2}\right]$	Ra: Rayleigh number $Ra = \frac{\alpha g L^3 \Delta T}{\lambda \nu} = Gr.Pr$
Re : Reynolds Number $Re = \frac{\rho U e}{\mu} = \frac{Ue}{\nu}$	t : dimensionless time[s]
T: Tempertaure [K]	T_0 : temperature at the initial moment$\left[K\right]$
U,V: dimensionless velocity components in the transformed plane	x, y, z : cartesian coordinates$\left[m\right]$
Greek symbols	
α: thermal diffusivity$\left[m^2 s^{-1}\right]$$\alpha = \frac{k}{\rho C_p}$	β: thermal expansion coefficient$\left[K^{-1}\right]$
μ : dynamicviscosity$[kg.m^{-1}s^{-1}]$	ν :kinematic viscosity$[m^2.s^{-1}]$
θ: dimensionless temperature$\left[K\right]$	Δt : time step$\left[s\right]$
λ: thermal conductivity$\left[Wm^{-1}K^{-1}\right]$	ρ:density of fluid$[kg.m^{-3}]$

References:-

1. D. BARTOLOMEU, Performance of an air-ground type heat exchanger, University J. Fourier, Grenoble, Vol. 28, No3, 4p, 2005.
2. Ahmad Fudholi, KamaruzzamanSopian, MohdHafidz Ruslan, Mohd. Yusof Othman and Muhammad Yahya, Thermal Efficiency of Double Pass Solar Collector with Longitudinal Fins Absorbers, American Journal of Applied Sciences 8 (3): 254-260, 2011
3. B. Mebarki, B. Draoui, S. Abdessemed A. Keboucha, S. Drici et A. Sahli, Etude d’un système de climatisationintégrant un puitscanadien dans les zones arides, cas de Béchar, Revue des Energies Renouvelables Vol. 15 N°3 (2012) 465 – 478
4. H. NEBBAR, O. HAMDI, N. MOUMMI, A. BRIMA, Etude de comportementthermique d’un échangeurenterré air/sol. Expérimentation de Biskra (Algérie), Le 3ème SéminaireInternational sur les Energies Nouvelles et Renouvelables, Unité de Recherche Appliquéeen Energies Renouvelables, Ghardaïa – Algérie 13 et 14 Octobre 2014
5. Boureima KABORE, Sié KAM, Amadou KONFE et Dieudonné Joseph BATHIEBO; Étudeanalytique du fonctionnementannuel d'un puitsprovençal pour la détermination des différences de température, des flux thermiques et des périodes de refroidissement dans une habitation à Ouagadougou; AfriqueSCIENCE 13(1) (2017) 118 - 128
6. Mathias Cuny, Study of the impact of geological and climatic conditions on the energy efficiency of surface geothermal systems, Unique Doctoral Thesis, University of Strasbourg, September 2017

7. E. F. NOGOTOV, Application of Numerical Heat Transfer, Mc Graw Hill Book Company, New York, 1978
8. Menter F. R.: Zonal Two Equations k-w Turbulence Models for Aerodynamic Flows, AIAA paper 93-2906, 24th Fluid Dynamics Conference, July 6-9, 1993, Orlando Florida.
9. Ntinas, G.K., Shen, X., Wang, Y. and Zhang, G. (2018) Evaluation of CFD Turbulence Models for Simulating External Airflow around Varied Building Roof with Wind Tunnel Experiment. Building Simulation, 11, 115-123. https://doi.org/10.1007/s12273-017-0369-9.