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Research Paper

Study of mixed convection in a horizontal cylindrical pipe partially filled with a porous medium saturated with a single fluid

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A B S T R A C T

The three-dimensional numerical simulation of laminar mixed convection heat transfer with and without porous medium in a cylindrical duct was carried out in this study. The cylinder is divided into three parts. The first and the third parts are adiabatic, while the central part is subjected to a constant heat flux. The effects of Darcy number, porosity, permeability and Nusselt number on heat transfer were considered. This problem is modeled by the Navier-stokes equations in the fluid region and by the Darcy-Brinkman-Forchheimer equation in the porous region, and for the thermal field the energy equation is used. The resolution of these equations was performed by the computational fluid dynamics code Ansys15.2 software using finite volume approach. The obtained results showed an accumulation of the hot fluid with stratification in the upper part of the pipe while the lower region of the pipe remained at practically constant temperature. The effect of the porous matrix is very noticeable on the velocity profile. The insertion of a porous high permeability medium significantly improves heat transfer.

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1. Introduction

The mixed convection heat transfer in a porous medium includes several important physical effects for example, the effect of a porous medium due to inertial forces at the pore scale on the momentum, on the energy and on the mass transport. This phenomenon has been studied in depth for several geometric configurations and various boundary conditions. The Brinkman-Forchheimer couple model (generalized model) is the most used by researchers in this type of problem.
The matrix of the porous medium is not so simple to model and requires careful attention. Indeed, there are effects which have a significant interest on heat transfer in porous medium. By way of example, mention may be made of the variation in porosity, thermal and mass dispersion, variation in the physical properties of a fluid such as viscosity and thermal conductivity, etc.
The objective of our work, which falls within the same context, consists in the simulation of heat transfer by mixed convection in a porous medium, in a permanent laminar regime. The model studied is a horizontal cylinder containing a porous layer and placed under simplified conditions.
For several decades, transfer phenomena in porous medium have attracted the attention of researchers and have been the subject of numerous studies. A review of the literature shows that there is a large number of numerical and theoretical works devoted to the study of mixed convection in a porous medium. We can cite the following works:
S.V. Patankar et al (1978) studied fully developed mixed convection in a horizontal pipe with non-uniform heat flux at the solid – fluid interface. Uniform heating is imposed on the upper half of the pipe while the other half is adiabatic. Bottom heating provides a more intense secondary flow which increases the Nusselt number over that of forced convection. On the other hand, in the case of top-applied heating, the superposition in the temperature profiles reduces secondary flow and heat transfer [15].
C. Abid et al [1994] studied an experimental, analytical and numerical approach of mixed convection in a horizontal cylindrical pipe traversed by a fluid in laminar regime and subjected to a continuous and uniform heat source. The results obtained show the existence of two zones of regimes: the hydrodynamic establishment of a secondary flow characterized by the presence of two counter-rotating transverse rollers, and the thermal establishment characterized by a rise in the average temperature of the wall while maintaining a non-uniform temperature distribution in a cross section. [16].

K. M. Khanafer et al (1999) have studied numerically the mixed convection inside a cavity containing a porous medium in the presence of an internal heat source and whose upper wall is movable. They note a suppression of the current lines by the presence of a porous medium (decrease in the Darcy number). For low Richardson numbers, the effect of the internal Rayleigh number becomes insignificant on the flow structure as the isotherms are strongly affected [17].

The double diffusive mixed convection in a porous rectangular cavity saturated by a fluid was studied numerically by Khanafer and Vafai (2002). The two horizontal walls are fixed and maintained at different temperatures and at different concentrations. On the other hand, the two vertical walls are adiabatic and move at a constant velocity. The finite volume method is used to discretize the transport equations. The results obtained indicate that the Darcy number, Lewis number and Richardson number have a great effect on heat and mass transfer. [18]

V. Elaprolu et al (2008) carried out a numerical study of mixed convection in a porous cavity where the vertical walls are kept respectively at constant cold and hot temperatures and move upwards by the same speed. As for the horizontal walls are fixed and adiabatic. The authors used the Brinkmane-Wooding model. The main results show that the decrease in the Darcy number causes a suppression of convection cells near the vertical walls. In addition, the mean Nusselt number decreases as the Richardson number increases [19].

N. Guerroudj et al (2010) studied numerically the laminar mixed convection in a two-dimensional horizontal channel equipped with porous blocks of various shapes. The results obtained show that the shape of the porous blocks can significantly change the flow and heat transfer characteristics [20].

M. Benmerkhi (2011) carried out a numerical study of heat transfer in a channel partially filled with porous material. The channel is divided into three equal parts. The first and the third parts are adiabatic and the second is maintained at a constant temperature. The presence of porous medium considerably increases heat transfer [21].

P. C. Huang et al (2012) carried out a numerical study on the laminar mixed convection of an upward flow in a vertical duct provided by several heat sources enveloped by a porous material. The study is based on the effect of Darcy number, Reynolds number, Grachof number and conductivity ratio on the flow structure and the heat transfer. They found that the insertion of the porous shell cools the heat sources well. They also found the existence of a critical value of the Darcy number at which the heat transfer is maximal [22].

A. Bourouis et al (2014) numerically analyzed the mixed conjugated convection in a cavity partially occupied by a vertical or horizontal porous layer. The Brinkman-Wooding model is used to describe the flow in the porous layer. The effect of the Richardson number and the thermal conductivity ratio of the porous-fluid medium is examined. They concluded that the rate of heat transfer along the hot wall is a decreasing function of the thermal conductivity ratio unlike that along the fluid-porous layer interface which is an increasing function with the ratio of the thermal conductivity [23].

F. Guellai (2015) have numerically analyzed the forced convection in a porous horizontal cylinder subjected to a constant partial heat flux. the study is based on the effect of the inlet flow velocity, porosity, thermal conductivity of the porous material and the heat flux imposed on the wall [24].

K. Maarka et al (2019) carried out a numerical study of the mixed convection of a water flow in laminar regime, developed in a heated cylinder by a constant and uniform heat flux. The obtained results confirmed that the forces of gravity generate two thermo-convective rollers along the pipe. Thus, the increase in the Grashof number indicates an increase in dynamic and thermal fields. On the other hand, an increase in the Reynolds number induces a decrease in temperature [25].

2. Model description

2.1 Geometric configuration and computational domain

Three dimensional simulations are carried out on a horizontal cylindrical pipe. Fig.1 1illustrates the physical model and the relevant parameters of the designed pipe. The length of the pipe (L) and the diameter (D) equals $2\ m$ and $0.01\ m$, respectively, with a thickness (e) of $0.2\ mm$. This pipe is made up of three parts. The central part with a length of $1m$ is heated by a constant flux and the two other parts which delimit the central one are adiabatic.

The flow is described in a Cartesian coordinate system in which x, y and z are span wise, normal and stream-wise coordinates, respectively. The fluid used in our simulations is the water. The fluid enters the pipe with axial velocity and uniform temperature.

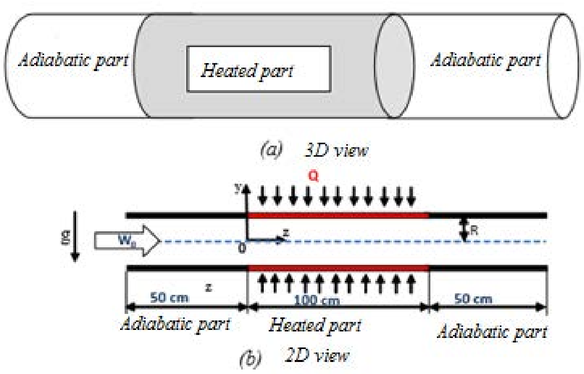


Fig 1 Physical model and relevant parameters of designed pipe [1]

2.2 Mathematical models, governing equations and boundary conditions

The resolution of this problem consists in determining the velocity and temperature fields at each point of the domain occupied by the fluid in the pipe. This is why we are going to establish the basic equations governing mixed convection. The present model is derived based on continuum flow approach and steady-state conservation equations of mass, momentum and energy. The fluid flow is assumed incompressible, laminar, permanent and three-dimensional. Moreover, radiation and gravity effects are ignored in this study and the solid porous matrix is permeable, isotropic and completely saturated with the fluid. The resulting governing equations can be written as follow:

Continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

- *x-Momentum equation :*

$$\rho_f\left(\frac{1}{\zeta}\frac{\partial u}{\partial t}+\frac{1}{\zeta^2}\left(\vec{V}.\vec{\nabla}\right)u\right)=-\frac{\partial p}{\partial x}+\mu_{eff}\nabla^2 v-\frac{\mu}{k}u-\frac{\rho_f}{\sqrt{k}}F\left|\vec{V}\right|u \tag{2}$$

- *y-Momentum equation:*

$$\rho_f\left(\frac{1}{\zeta}\frac{\partial v}{\partial t}+\frac{1}{\zeta^2}\left(\vec{V}.\vec{\nabla}\right)v\right)=-\frac{\partial p}{\partial y}+\mu_{eff}\nabla^2 v-\frac{\mu}{k}v-\frac{\rho_f}{\sqrt{k}}F\left|\vec{V}\right|v \tag{3}$$

- *z-Momentum equation*:

$$\rho_f\left(\frac{1}{\zeta}\frac{\partial w}{\partial t}+\frac{1}{\zeta^2}\left(\vec{V}.\vec{\nabla}\right)w\right)=-\frac{\partial p}{\partial z}+\mu_{eff}\nabla^2 w-\frac{\mu}{k}w-\frac{\rho_f}{\sqrt{k}}F\left|\vec{V}\right|w+\rho_f\vec{g} \tag{4}$$

- *Energy equation:*

$$(\rho c)_m\frac{\partial T}{\partial t}+(\rho c)_f\left(\vec{V}.\vec{\nabla}\right)T=\vec{\nabla}.\left(K_{eff}.\vec{\nabla}T\right) \tag{5}$$

All the equations that govern our problem can be written in the following conservative form:

$$\underbrace{A_\varphi\frac{\partial\varphi}{\partial t}}_{1}+\underbrace{\left(\vec{V}\cdot\vec{\nabla}\right)\varphi}_{2}=\underbrace{\vec{\nabla}\left(\Gamma_\varphi\vec{\nabla}\varphi\right)}_{3}+\underbrace{S_\varphi}_{4} \tag{6}$$

With (1): Time term; (2): Convective term; (3): Diffusion term; (4): Source term. Or : Γ: is the diffusion coefficient and φ: is corresponding to the dependent parameters u, v, w and T. In the following table, we give the definitions of φ as well as all the coefficients Γ, A_φ and the source terms that govern our problem.

Table 1. List of governing equations.

Equations	φ	A_φ	Γ_φ
Continuity	1	0	0
x-momentum	u	ε	$\varepsilon R_V P_r$
y-momentum	v	ε	$\varepsilon R_V P_r$
z-momentum	w	ε	$\varepsilon R_V P_r$
Energy	T	σ	R_K

2.3 Initial and boundary conditions

The required initial and boundary conditions at the inlet, outlet, and at the solid-liquid interface, according to Fig. 1, are introduced as follow:

For : $t = o$; $u = 0$; $v = 0$; $w = w_0,\ T = T_i$

For : $t > 0$

Inlet boundary : $For\ x = 0\ and\ 0 < y < R$:

$u = 0$; $v = 0$; $w = 0.072 m/s$; $T = 293K$

Outled boundary : (Pressure out let) $\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial w}{\partial z} = 0,\ T = T_s, \frac{\partial T}{\partial z} = 0$

Heated wall : $For\ y = R\ \ and\ \ 0.5m < x < 1.5m$

$q = 3082\ W/m^2\ and\ \ u = v = w = 0$

2.4 Numerical procedures and parameter definitions

In the present study, the computational fluid dynamics code Ansys – Fluent is used. As the quality of the mesh plays a very important role in the precision and the stability of any numerical study. Several tests are carried out in order to validate a chosen numerical method. Structured non-uniform grids are utilized to discretize the computational domain in which a three-dimensional hexahedral mesh is used (Fig. 2).

Cells are appropriately fine in order to improve accuracy. The conservation equations are discretized using finite volume method based on SIMPLEC algorithm. The governing equations are solved iteratively until the convergence criterion satisfied which is defined when the scaled residuals of the momentum, continuity and energy equations attain a value less than 1.0×10^{-8}, 1.0×10^{-5} and. 10×10^{-7} respectively. To present the results of numerical simulations the subsequent parameters are defined. The Reynolds number (Re) based on the characteristic length (L) of pipe is defined as follow.

$$Re = \rho VL / \mu \tag{7}$$

Where ρ is the fluid density, V is fluid velocity, L is the characteristic length and μ is the fluid dynamic viscosity.

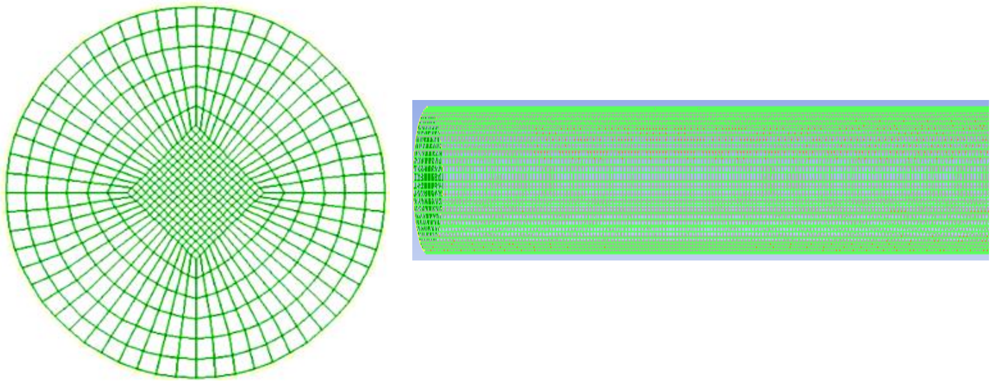


Fig. 2 Pipe Mesh

The dimensionless formatting of the conservation equations reveals dimensionless numbers characterizing the studied problem. These parameters are the Prandtl number (Pr), the Grashof number (Gr), the Rayleigh number (Ra), the Darcy number (Da) and the dimensionless thickness of the porous layer ($\eta = e / H$). Additionally, we express the heat transfers on active surfaces by the dimensionless Nusselt number (Nu) defined by:

$$Nu = hL/\lambda \tag{8}$$

The Nusselt number, which is a dimensionless quantity, can be interpreted physically as being the ratio of the temperature gradient in the fluid in immediate contact with the surface to the reference temperature gradient. In practice, the Nusselt number is a convenient measure of the convection heat exchange coefficient because, once its value is known, the convection heat exchange coefficient can be calculated.

3. Grid independency and model verification

Three grids with different sizes are employed for the grid independence test on the cylinder configuration. The grid used have sizes of 116 866 (coarse), 1 352 676 (intermediate), and 5262 444 (extremely fine). The results of the average velocity for different grid sizes is presented in table 2. It is noticed that there is no difference in the computed results between the intermediate and the extreme fine grids. Hence, by a compromise between required accuracy, time usage and computation costs, the intermediate is selected for the simulations.

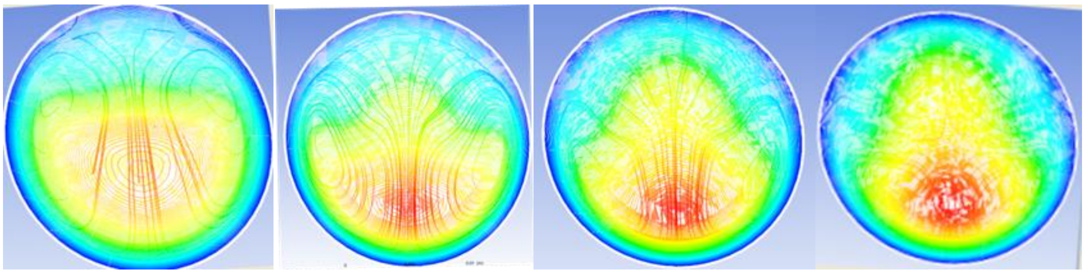
Table 2. Mesh effect on the average velocity.

Nodes number	116866	1352676	5262444
Average velocity	0.089456	0.097461	0.097255
Error (%)	4.29701909	0.032107438	0.0312777281

4. Results and discussions

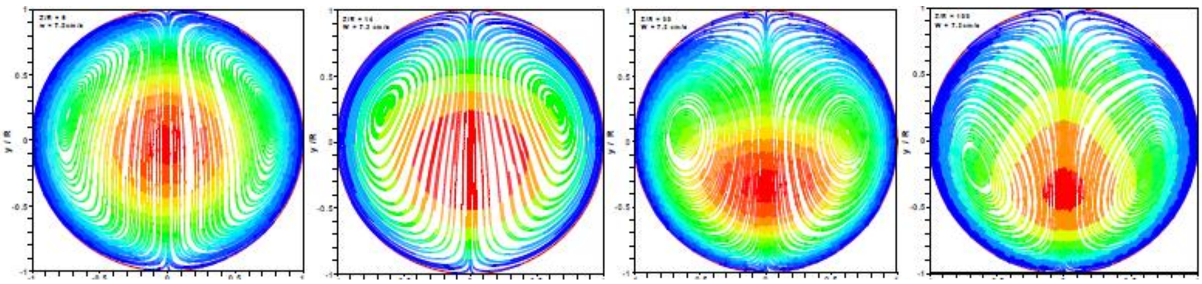
4.1 Mixed convection in a cylinder without porous medium

Concerning the dynamic field, the structure of the mean flow is illustrated by the streamlines of the cross sections of the cylinder, which are inferred from the normal components of the velocity (u and v). Calculations confirmed the presence of two convective rollers along the heated cylinder (Fig. 3) which are produced by the effect of Archimedean thrust forces. This result is in good agreement with those of C. Abid et al [16] and those of M. Benmerkhi [21].



Z/R=50 Z/R=100 Z/R=140 Z/R=160

Fig. 3 Streamlines for different cross sections.



Z/R=6 Z/R=14 Z/R=30 Z/R=100

Fig. 4 Streamlines for different cross sections, C. Abid et al. [16]

Since our goal is to increase heat transfer, the variation of the temperature in the radial direction is given by the isotherm profiles represented in figure 5. In the vicinity of the wall, the fluid heats up and rises towards the upper part of the pipe and that of the central region moves downwards, this is the effect of natural convection. Consequently, the fluid in the upper part of the pipe becomes hotter than that in the lower part and the minimum temperature takes its position at the bottom of the pipe.

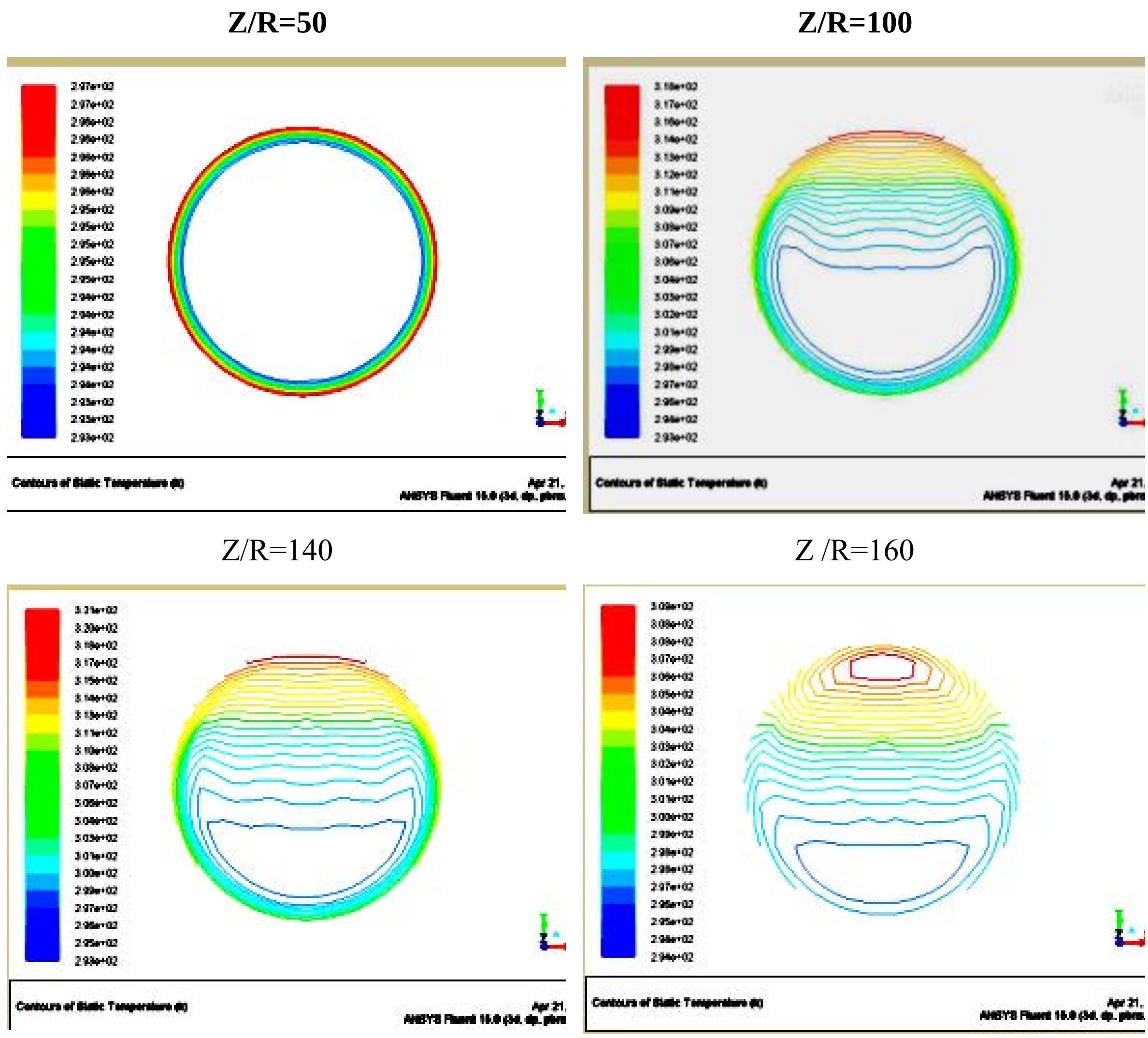


Fig. 5 Isothermal profiles for different axial positions.

Fig. 6 shows the variation of the local Nusselt number along the axial direction in the heated part of the pipe. The Nusselt number takes its maximum value at the duct entrance for $Z = 0.6\ m$, which characterizes an improvement in the wall-fluid heat transfer, also we notice that this number undergoes a sudden drop due to the decrease in the temperature gradient, then it stabilizes towards the channel outlet. This is the fully developed thermal region (steady state) where the Nusselt number no longer depends on Z.

4.2 Mixed convection in a cylinder containing a porous medium

This study was carried out for two different porous medium (sand and slate) with different porosities ε, different thermal conductivities λ and different heat capacities Cp. The imposed heat flux on the wall is assumed to be always constant.

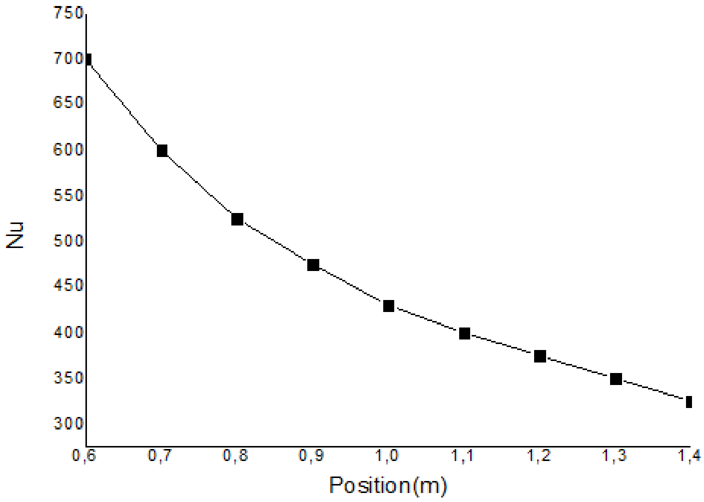


Fig. 6 Evolution of the Nusselt number along the heated part.

4.2.1 Case of sand

In Fig. 7, we have plotted the vertical profile of the horizontal velocity at some arbitrarily chosen axial positions. The effect of the porous matrix is very noticeable on the velocity profile. In fact, it changes from a parabolic profile in the fluid medium to a flat profile in the porous substrate. The Reynolds number also tends to flatten the velocity profiles and increase the settling length of the dynamic regime as it increases. This result is in good agreement with that of Guellai F [24].

In this case of sand, it is observed that for the low values of the Forchheimer's coefficient the variation of the horizontal velocity near the walls is due to the Brinkman term (Poiseuille profile), but when moving away from the walls the velocity profiles become flat under the effect of the Forchheimer term, i.e. for strong values of the Forchheimer coefficients. The effect of the decrease in the Darcy number on the flow structure appeared on the velocity profile which becomes increasingly flat [32].

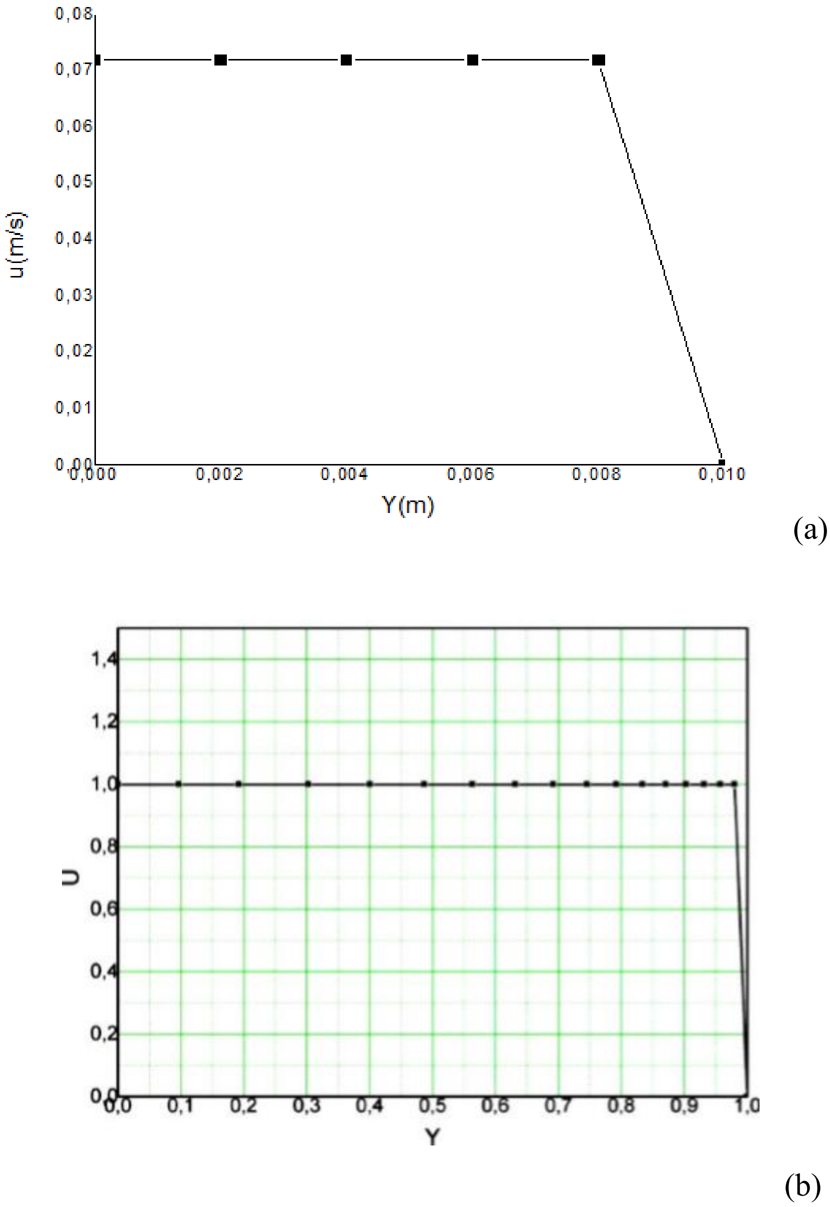


Fig. 7 Vertical profile of the horizontal velocity.

(a) The present study, (b) study of Guellai F [24]

Fig. 8 shows the fluid temperature variation at the wall. The temperature of the fluid gradually increases from the inlet of the heated part (297K) to the outlet of this part (318K) to give a temperature gain ΔT of 21° along the flow direction. This result shows a clear improvement in heat transfer due to the increased convection near the walls. This result is in good agreement with that of Guellai F. [24].

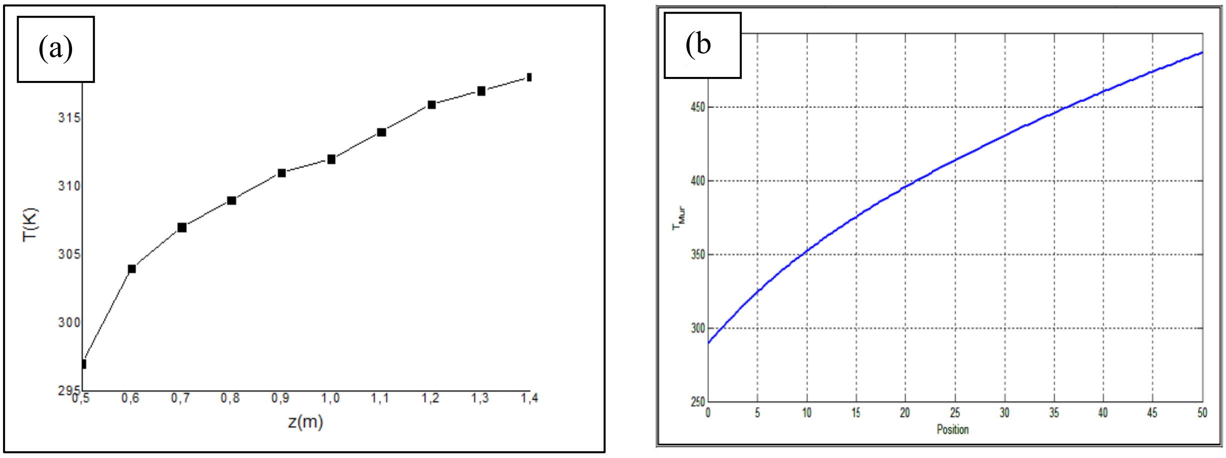


Fig. 8 Fluid temperature variation at the all

(a)Present steady ($\varepsilon = 0.4, q_w = 7712.7$ W/m^2, $\lambda = 0.3347\ W/m.K$)

(b)Guellai F. [24] ($\varepsilon = 0.4, q_w = 500$ W/m^2, $\lambda = 0.3347\ W/m.K$)

We present in Fig. 9 the evolution of the local Nusselt number. As, the boundary conditions of the cylinder remain the same for all the studied cases, therefore the local Nusselt number of the first and the third part is always zero, since the walls are always adiabatic. In the same figure, in the transfer zone, there is a rapid axial drop in the local Nusselt number from 800 to an asymptotic value of 400 which remains higher than that of the fluid in the case of the channel without porous medium (see Fig. 4), this change is a consequence of the effect of convection (decrease in the temperature gradient).

The comparison of the local Nusselt number for a cylinder with and without porous medium is presented in Fig. 10. Since the conductivity of water is greater than the conductivity of sand, the Nusselt number ($Nu = hD/\lambda$) is important in heat transfer with porous medium (conduction effect).

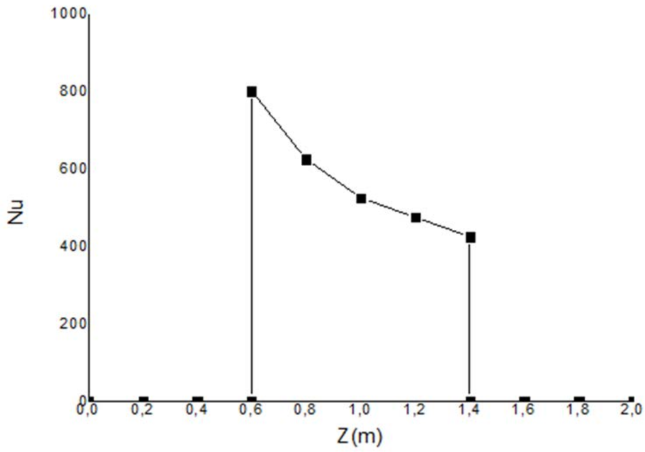


Fig. 9 Evolution of the local Nusselt number.

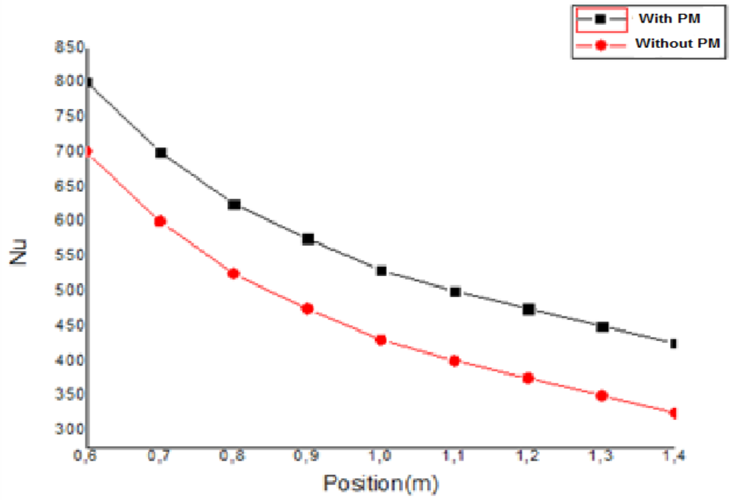


Fig. 10 Comparison of the local Nusselt number for the cylinder heat part.

4.2.2 Case of slate:

Fig. 11 shows the fluid temperature variation at the wall. We note that the temperature increases progressively from the inlet of the heated part ($295K$) to the outlet of this part ($308K$) to give a temperature gain ΔT of 13° along the flow direction while maintaining a constant heat flux of 7712.7W/m² on the central part of our cylinder ($z = 0.5 - 1.5\ m$). So there is a marked improvement in heat transfer due to the increased convection near the walls. Fig. 12 presents a comparison between the two cases of imposed flux. For the low flux the temperature at the inlet of the central part is $293K$ while the high flux gives an inlet temperature of $297K$. In addition to this temperature difference at the entrance of the heated part, the temperature gradient is not the same. For the high flux value ($7712.7W/m^2$) the temperature gradient is 21° and for the low flux ($3082W/m^2$) the temperature gradient is 15°.

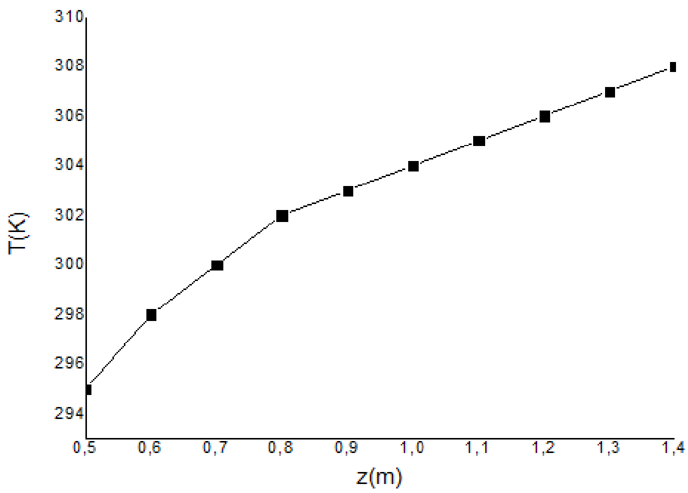


Fig. 11 Fluid temperature variation at the wall

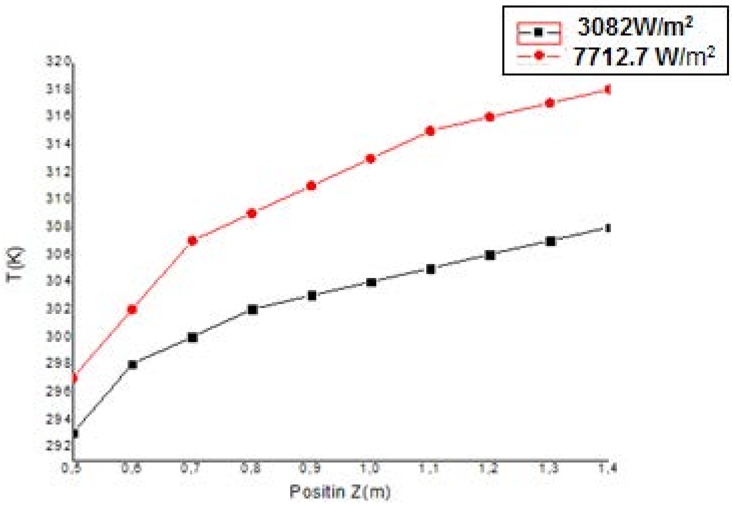


Fig. 12 Fluid temperature variation at the wall for the two imposed flux

Fig. 13 presents the vertical temperature profiles at the outlet of the heated part for a porous medium of sand and another of slate. The curves of outlet temperatures in this figure show that sand plays a very important role in increasing heat transfer unlike slate which is important in cooling. The decrease in the conductivity ratio (fluid-solid) leads to an increase in the global Nusselt number. For a small Darcy numbers and a very low permeability; the porosity and the conductivity will be low values and the porous medium then behaves like a solid (case of slate). While for a large Darcy numbers and a very high permeability; the porosity and the conductivity will be large values and the porous medium then behaves like a fluid (in the case of sand). The influence of permeability reflects the ease of movement of the fluid in the porous medium, that is, the fluid flows easily in this medium when the permeability is higher.

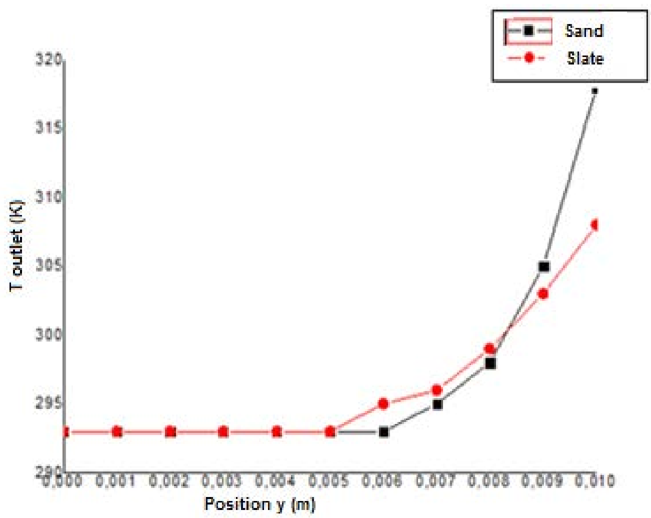


Fig. 13 Temperature profiles at the outlet of the heated part in the case of two porous mediums: sand and slate.

In order to study the effect of permeability on heat transfer we used five porous mediums: sand, slate, limestone, sandstone and clay. The simulation results show that the Nusselt number is inversely proportional to the increase in the permeability k (Fig. 14).

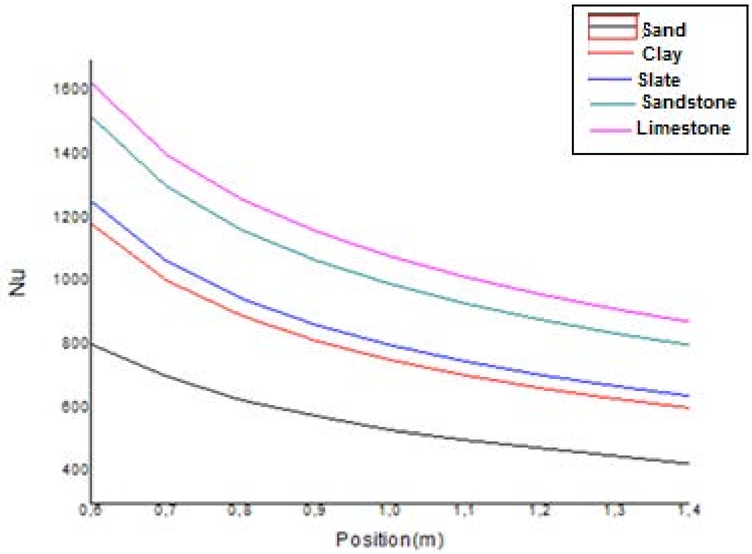


Fig. 14 Nusselt profile for different porous mediums.

The variation of the mean Nusselt number, as a function of the porous medium permeability is presented in Fig. 15. The results show that for low values of the permeability ($Da \leq 10^{-6}$) the Nusselt numbers remain practically constants and the porous medium in this range of Darcy number behaves like an impermeable area where the flow is almost negligible. A simple linear correlation of order 1 between the Nusselt number and the permeability is suggested: $\overline{Nu} = -3 \times 10^{13} K + 969.3434$ with an error deviation of 10%. The coefficient of this correlation is $R^2 \approx 0.7$ which explains a strong relationship between the mean Nusselt and the permeability of the porous medium.

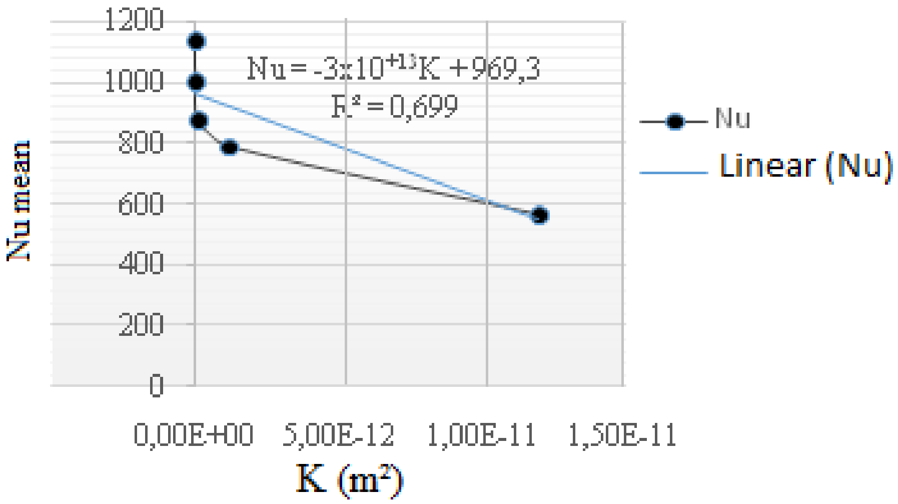


Fig. 15 Mean Nusselt profile as a function of permeability.

4.3 Mixed convection in the case of a nanofluid

In order to compare the heat transfer rate obtained in a partially heated cylinder in the presence of a porous medium, with the transfer rate when using $Al_2O_3 - water$ nanofluid , a numerical study of the mixed and laminar convection of a flow of $Al_2O_3 - water$ along a channel of circular section was realized. Numerical simulations were carried out for a single volume fraction of the nanoparticles ($\varphi = 4\%$), a Reynolds number of 790, a Richardson number of 0.5 and a Grachoft number of $3{,}12 \times 10^5$.

Fig. 16 represents the variation of the mean Nusselt number for the $Al_2O_3 - water$ nanofluid. The mean Nusselt number is found to grow almost linearly along the channel. The addition of nanoparticles improves the effective conductivity within the fluid and also improves heat transfer so the $Al_2O_3 - water$ nanofluid is a good conductor of heat.

The comparison of the results obtained in the case of $Al_2O_3 - water$ nanofluid and that of pure water with and without porous medium, shows that the mean Nusselt number of the nanofluid is greater than that of pure water even in presence of the porous medium. The addition of nanoparticle to the fluid leads to a higher heat transfer coefficient therefore a high mean Nusselt number.

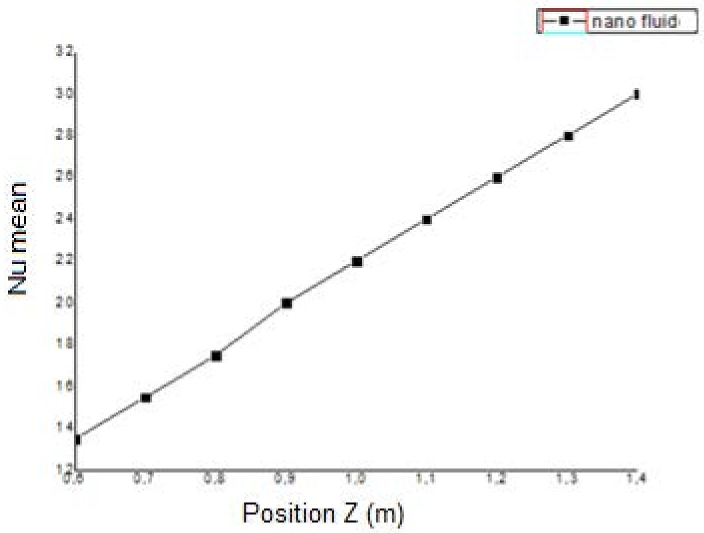


Fig. 16 Mean Nusselt number profile.

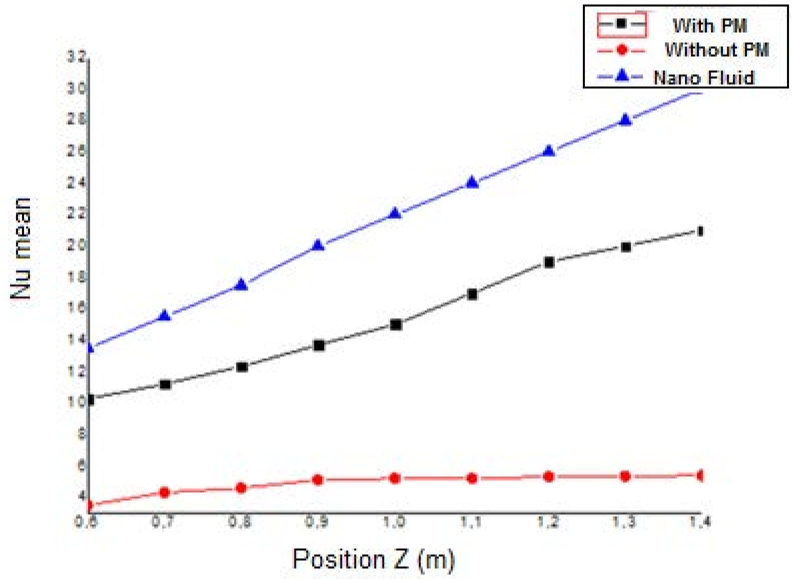


Fig. 17 Comparison of the mean Nusselt number for different cases

5. Conclusion

The importance of porous medium in industrial and technological processes is now well demonstrated and found various applications, such as cooling of electronic components, heat treatment of waste, design and manufacture of heat exchangers.

The numerical simulation of mixed convection heat transfer of a laminar flow, in a cylindrical duct with and without porous medium, was carried out in this study. The cylinder is divided into three parts. The first and the third parts are adiabatic, while the middle part is subjected to a constant heat flow. The effects of Darcy number, porosity, permeability and Nusselt number on heat transfer were considered.

The main objective of our study is the numerical simulation of mixed convection in a cylinder. To do this, we have divided our work into two parts, the first one concerning a fluid medium and the second part relating to a porous medium.

This three-dimensional problem is modeled by the Navier-stokes equations in the fluid region and by the Darcy-Brinkman-Forchheimer equation in the porous region, and for the thermal field we used the energy equation. The resolution of these equations was performed by the computational fluid dynamics code Ansys15.2 software using finite volume approach.

The results obtained showed the influence of several thermo physical parameters on the dynamic and thermal behavior of the flow in a partially heated duct with and without a porous medium. The results were presented in terms of velocity and temperature profiles as well as in terms of the Nusselt number. Our results show a good agreement with those of Benmerkhi M. [21], those of Abid et al [16] and those of Guellai F. [24].

The use and analysis of the various obtained results have led to a number of observations and findings which are:

- The presence of two convective rollers along the heated cylinder part which are produced by the effect of Archimedean thrust forces.
- An accumulation of hot fluid with stratification in the upper part of the pipe while the lower region of the pipe remains at almost constant temperature.
- The effect of the porous matrix is very noticeable on the velocity profile. Indeed, it goes from a parabolic profile in the fluid medium to a flat profile in the porous substrate.
- The decrease in the conductivity ratio (fluid - solid) leads to an increase in the Nusselt number.
- The insertion of a high permeability porous matrix improves heat transfer. The fluid flows easily in this medium when the permeability is higher.
- Increasing porosity decreases heat transfer in the porous medium.
- For a small number of Darcys and a very low permeability; the porosity and the conductivity will have low values and the porous medium then behaves like a solid (case of slate). However, For a large Darcy number and a very high permeability; the porosity and the conductivity will be large values and the porous medium then behaves like a fluid (in the case of sand).
- For low values of permeability, the Nusselt number remains almost constant and the porous medium in this Darcy number range behaves as an impermeable zone where the flow is almost negligible.
- The addition of nanoparticles improves the effective conductivity within the fluid and also improves heat transfer so the $Al_2O_3 - water$ nanofluid is a good conductor of heat.
- The comparison of the results obtained in the case of $Al_2O_3 - water$ nanofluid and that of pure water with and without porous medium, shows that the average Nusselt number of the nanofluid is greater than that of pure water itself in the presence of the porous medium

Nomenclature

F: Darcy permeability correction factor	α : Specific surface (m^2)
q: Heat flux (W/m^2)	η: Dimensionless thickness of the porous layer
h : Convection coefficient ($W.K^{-1}m^{-2}$)	λ : Thermal conductivity ($W.m^{-1}K^{-1}$)

S : Solid / fluid contact surface (m²)

T_p : Wall temperature (K)

T_f : Fluid temperature (K)

L: Characteristic length (m)

V: Fluid velocity ($m.s^{-1}$)

C_p : Specific heat capacity ($J.kg^{-1}K^{-1}$)

d: Pore diameter (m)

T_0 : Reference temperature (K)

K : Permeability (m^2)

$(u;v;w)$: Velocity component (m/s)

$(x;y;z)$: coordinate system (m)

ρ: Fluid density ($kg.m^{-3}$)

β: The fluid thermal volume expansion coefficient.

$\boldsymbol{\varepsilon}$: Porosity.

μ: dynamic viscosity of the fluid ($kg.m^{-1}.s^{-1}$)

$\boldsymbol{v}$: Kinematic fluid viscosity ($m^2.s^{-1}$)

Γ : Diffusion coefficient.

$\boldsymbol{\Gamma_\varphi}$: source term

τ : Tortuousness.

6. References

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