IJESRT: 11(10), October, 2022 ISSN: 2277-9655

International Journal of Engineering Sciences & Research Technology

(A Peer Reviewed Online Journal)

Impact Factor: 5.164

IJESRT

Website: www.ijesrt.com **Mail:** editor@ijesrt.com

Correspondence Address: 116, Sukhdev Nagar Ext-1, Airport Road, Indore – 452005, Madhya Pradesh, INDIA

IJESRT
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

NUMERICAL STUDY OF THE DYNAMIC AND THERMAL BEHAVIOR OF TWO FLOWS OF A FLUID IN A HEAT EXCHANGER

Omar Ngor THIAM [*1], Oumar DRAME [1], Samba DIA, Mamadou Lamine SOW and Cheikh MBOW

[*] Fluid Mechanics and Transfer Laboratory, Physics Department, Sciences and Technologies Faculty, Cheikh Anta DIOP University, Dakar-Fann, Senegal

Email: omarngorthiam@gmail.com /omarngor.thiam@ucad.edu.sn

DOI: 10.5281/zenodo.7238662

ABSTRACT

In this work, it is a numerical study of the dynamic and thermal behavior of two counter-current flows of a fluid in a heat exchanger. The exchanger has the shape of two rectangular channels of height H separated by an exchange surface through which the exchanges between the two fluids take place by conduction and of the same length L. We have written the transfer equations which govern our problem by closing it with the boundary conditions. To solve it, we used the finite volume method and wrote a calculation code under Ansys-Fluent. We have determined the variation of isotherms and isocurrents in the exchanger and the variation of the temperature on the exchange wall.

KEYWORDS: heat exchanger, finite volume, Ansys Fluent, discretization

1. INTRODUCTION

Forced convection is a heat transfer phenomenon linked to fluids, its presence simultaneously influences the thermal and dynamic fields of the flow, the problem thus coupled, finds its importance in many industrial applications and more particularly in heat exchangers, solar collectors, the reversal of turbomachines, nuclear reactors and electronic components.

In the literature, there are analytical, numerical and experimental studies on the dynamic and thermal behavior of fluid flow in complex geometries and which are related to our study. [1]

Starting with K.M.Kelkar and S.V.Patankar [2], these authors performed a numerical study for a fluid in laminar forced convection between two parallel planar walls with baffles. Z.X. Yuna et al [3] experimentally studied a case of a duct with periodic rectangular fins along the main flow direction. Gustao Urquiza Beltran [4] made a study of flow in rotating cylindrical cavities. Bazdizi Tehrani and M.Naderi Abadi [5] presented a numerical analysis of the dynamic and thermal behavior of a fluid flowing in a pipe fitted with rows of baffles. Tsay et al [6] have numerically studied the improvement of the heat transfer of a flow in a channel equipped with a vertical baffle. R. SAIM et al [7] performed a numerical simulation of the dynamic behavior of a turbulent flow crossing a pipe fitted with flat and or trapezoidal transverse baffles. Nabila GUENDOUZ and R. SAIM [8] carried out a numerical study of a two-dimensional incompressible turbulent flow of air crossing a rectangular pipe equipped with fins in the form of transverse diamonds. TAOURIT FARIDA [9] studied the dynamic and thermal behavior of two fluid flows in a heat exchanger without and with fins. R. SAIM et al [10] presented a numerical study of the dynamic and thermal behavior of a turbulent air flow in a horizontal channel of rectangular section provided with transverse baffles. Bénamar BOUHACINA [11] studied the dynamic and thermal behavior of a bi-tubular heat exchanger equipped with fins intended for the storage of solar energy. A.Youcef and R. Saim [12] studied the dynamic and thermal behavior of two flows in a heat exchanger in the simple case and with fins in the fluid flow stream. Marni Sandid Abdelfatah and Ismaili Abdelkader [13] studied the dynamic and thermal behavior of two flows in a heat exchanger in the simple case and with fins in the fluid flow path. Bernardo et al [14] studied a compact heat exchanger with a row of aligned tubes embedded in an aluminum foam numerically to find the dimensions of the foam region which represents an optimization between the improvement of the transfer rate of

ISSN: 2277-9655
Impact Factor: 5.164
CODEN: IJESS7

heat and increased pumping power. M. Chen et al [15] studied the dynamic behavior of a high temperature printed circuit heat exchanger in steady state and transient. S. Ali and M. Baccar [16] have numerically studied the hydrodynamic and thermal behaviors in an innovative SSHE (scraped surface heat exchanger) with helical ribbon for Bingham fluids, and developed a heat transfer correlation. Z.X. YUAN et al [17] have numerically investigated laminar flow and heat transfer in parallel plate channels, with adiabatic perturbations of periodic bars using BFC technique. Bekkouche Mohamed Ismail and Trari Mansour [1] carried out a comparative numerical study between two symmetrical thermal conditions (flow, temperature) applied in heat exchangers. Benouadfel Billal et al [18] studied the contribution to the thermal and dynamic study of a plate heat exchanger. Lopez et al [19] numerically studied the hydraulic and thermal effects as a function of the location of normal baffles inside a 3D channel. Acharya et al [20] experimentally studied the dynamic and thermal behavior of turbulent and transient flows in pipes in the presence of obstacles and ribs. Yong-Gang Lei et al [21] experimentally and numerically studied the hydrodynamic characteristics and the heat transfer in a heat exchanger with a single helical baffle.

2. MATHEMATICAL FORMULATION

Problem geometry

The exchanger has the shape of two rectangular channels of height H, separated by an exchange surface through which the exchanges between the hot fluid and the cold fluid take place by conduction and of the same length L. The lower and upper walls of this exchanger are assumed to be adiabatic. The hot fluid (which is hot air) with temperature T_{ce} and speed U_{ce} enters the lower channel on the left and exits at temperature T_{cs} on the right. The cold fluid (which is cold air) enters the upper channel on the right with a temperature T_{fe} at a speed U_{fe} and exits at temperature T_{fs} on the left. The flow channels are traversed by the same fluid (air) but of different temperature in parallel currents and reverse to each other, so it is indeed a counter-current heat exchanger

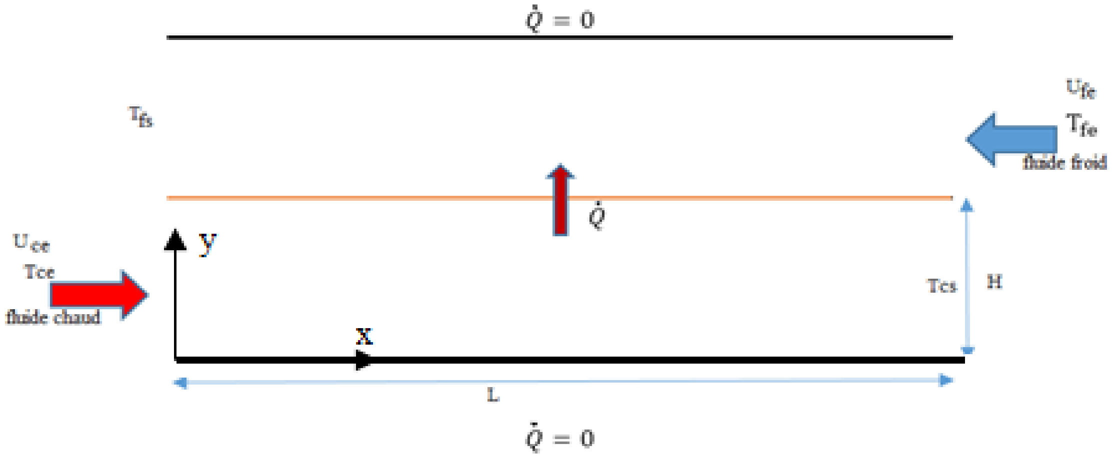

Figure 01: Geometry of the problem

Simplifying assumptions

In order to simplify the general transfer equations above, we make the following simplifying assumptions:

- The flow is stationary, incompressible, two-dimensional and laminar
- The physical properties of the fluid (Cp, μ, K) which are respectively: the specific heat, the dynamic viscosity and the thermal conductivity are considered constant.
- The variations of the density with temperature are negligible in all the terms of the conservation equations except in the term of the bulk forces, which here represents the Boussinesq approximation. Density varies linearly with temperature:
  $\rho = \rho_0[1 - \beta(T - T_0)]$
  ρ_0: Density of the fluid at the reference temperature T_0
  β: The thermal expansion coefficient of the fluid

Dimensionalization

By introducing the simplifying assumptions, the transfer equations which govern our problem are written under dimensionless with the reference quantities:

U_0: average characteristic speed along the x axis
V_0: average characteristic speed along the y axis

L: The characteristic size of the domain (m)
H: the height of the flow channels in (m)

$$\tilde{x} = \frac{x}{L},\ \tilde{y} = \frac{y}{H}\ \tilde{u} = \frac{u}{U_0},\ \tilde{v} = \frac{v}{V_0} = \frac{LV}{HU_0}, \tilde{P} = \frac{P}{\rho U_0^2}, \tilde{T} = \frac{T - T_0}{T_c - T_f}$$

$B = \frac{H}{L}$: which is the form factor.

- Conservation of mass equation:

$$\frac{\partial \tilde{u}}{\partial \tilde{x}} + \frac{\partial \tilde{v}}{\partial \tilde{y}} = 0 \quad \text{(i)}$$

- Momentum equation:

$$\tilde{u}\frac{\partial \tilde{u}}{\partial \tilde{x}} + \tilde{v}\frac{\partial \tilde{u}}{\partial \tilde{y}} = -\frac{\partial \tilde{P}}{\partial \tilde{x}} + \frac{1}{Re}\left(\frac{\partial^2 \tilde{u}}{\partial \tilde{x}^2} + \frac{1}{B^2}\frac{\partial^2 \tilde{u}}{\partial \tilde{y}^2}\right) \quad \text{(ii)}$$

$$\tilde{u}\frac{\partial \tilde{v}}{\partial \tilde{x}} + \tilde{v}\frac{\partial \tilde{v}}{\partial \tilde{y}} = -\frac{1}{B^2}\frac{\partial \tilde{P}}{\partial \tilde{y}} + \frac{1}{R_e}\left(\frac{\partial^2 \tilde{v}}{\partial \tilde{x}^2} + \frac{1}{B^2}\frac{\partial^2 \tilde{v}}{\partial \tilde{y}^2}\right) + Ri\tilde{T} \quad \text{(iii)}$$

- Heat equation

$$\tilde{u}\frac{\partial \tilde{T}}{\partial \tilde{x}} + \tilde{v}\frac{\partial \tilde{T}}{\partial \tilde{y}} = \frac{1}{Pe}\left(\frac{\partial^2 \tilde{T}}{\partial \tilde{x}^2} + \frac{1}{B^2}\frac{\partial^2 \tilde{T}}{\partial \tilde{y}^2}\right) \quad \text{(iv)}$$

- Dimensionless boundary conditions

Solving the system of dimensionless equations obtained previously requires the incorporation of dimensionless initial and boundary conditions.

- Inputs: the input temperatures are imposed

$x = 0$ and $0 < y < H$ $\qquad$ $x = L$ and $H < y < 2H$

$$\tilde{U} = \tilde{T} = 1, \tilde{V} = 0 \qquad \tilde{U} = 1, \tilde{T} = \tilde{V} = 0 \quad \text{(v)}$$

- Outputs: the regime is established

$x = 0$ and $H < y < 2H$ $\qquad$ $x = L$ and $0 < y < H$

$$\frac{\partial \tilde{U}}{\partial \tilde{x}} = \frac{\partial \tilde{V}}{\partial \tilde{x}} = \frac{\partial \tilde{T}}{\partial \tilde{x}} = 0 \qquad \frac{\partial \tilde{U}}{\partial \tilde{x}} = \frac{\partial \tilde{V}}{\partial \tilde{x}} = \frac{\partial \tilde{T}}{\partial \tilde{x}} = 0 \quad \text{(vi)}$$

- Walls: $y = 0$ and $0 < x < L$; $y = 2H$ and $0 < x < L$; $\qquad$ $y = H$ and $0 < x < L$

$$\frac{\partial \tilde{T}}{\partial \tilde{y}} = 0 \qquad \frac{\partial \tilde{T}}{\partial \tilde{y}} = -\frac{hH}{K} = Nu \quad \text{(vii)}$$

The Nusselt number (Nu) is a dimensionless number used to characterize the type of heat transfer between a fluid and a wall. We define an average Nusselt number from the global heat exchange coefficient (h) associated with the global heat flux over the entire exchange wall. [22]

$$Nu = \frac{1}{\Delta T}\frac{\partial T}{\partial \tilde{y}} \qquad \overline{Nu} = \frac{1}{S}\int Nu\, ds \quad \text{(viii)}$$

3. NUMERICAL APPROACH:

The heat transfer equations described are nonlinear and coupled partial differential equations. Due to their complexity, these equations are solved by numerical techniques. The spatial discretization is done by a finite volume method while a purely implicit scheme is adopted for the temporal discretization.
To our knowledge, there is no general analytical solution for our problem,
or experimental work. In order to verify the accuracy of the numerical results obtained in this work, a validation of the numerical code was made by taking into account certain numerical studies. The "Fluent" code was used to simulate the transport of the flow and the evolution of the temperature. Our results will therefore be compared with those of Taourit [9]. In figure 02 and 03, we can see the shape of the velocity field distribution with the following data for the two hot and cold fluids:

- The hot fluid flow 0.003 kg/s at the temperature T_{fe}=355K.
- Cold fluid flow 0.008 kg/s at temperature T_{fe}=300K

ISSN: 2277-9655
Impact Factor: 5.164
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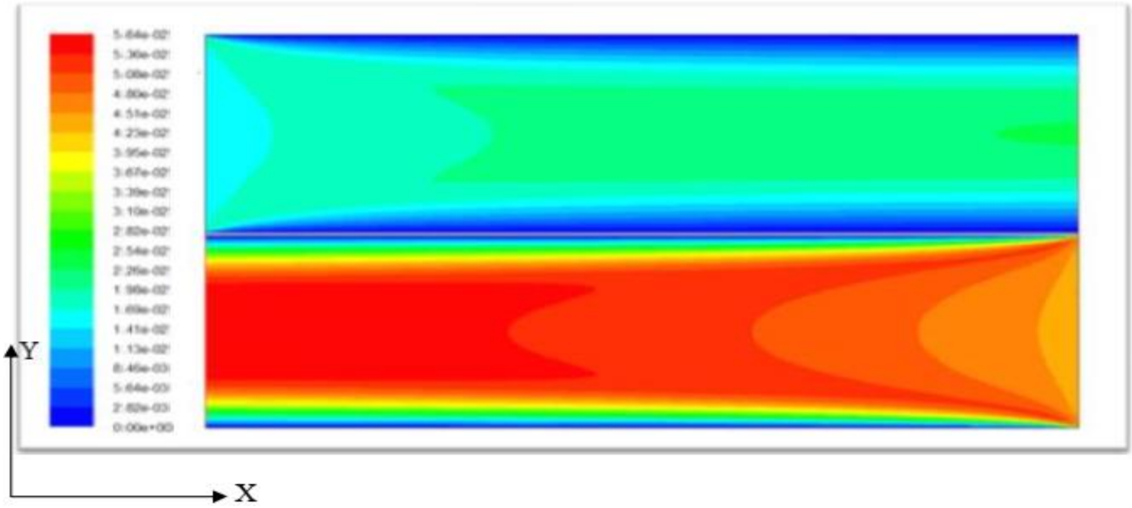

Figure 02: Axial Velocity Field Distribution in Taourit Channel Length [9]

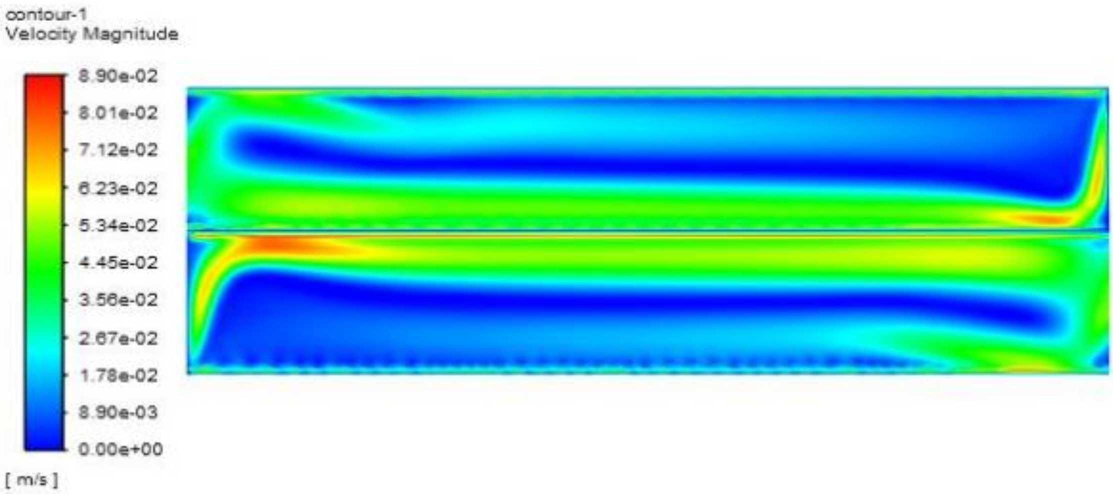

Figure 03: Velocity field distribution along the exchanger.

4. RESULTS AND DISCUSSION

Evolution of temperature and streamlines

Figures 09 and 10 show the evolution of isotherms and streamlines respectively in flow channels in laminar regime in countercurrent flow mode for respective Reynolds numbers (Re_c=1000 and Re_f=100) where the flow velocities are U_{ce}=0.146 m/s for the hot fluid and U_{fe}=0.0146 m/s for the cold fluid. It can be seen that there is a significant temperature difference between the wall in contact with the cold fluid and that with the hot fluid (ΔT=20 °K near). At the inlets of the exchanger, the temperature is not homogeneous in the rectangular space (hot fluid) and in the rectangular space (cold fluid) because of the temperature difference existing between the two fluids mainly due to the absence of the fluctuating speed which acts as a temperature homogenizing factor which allows perfect mixing with a single temperature. At the level of the temperature distribution, it shows two thin layers at the level of the two entrances to the channels, which begin to expand gradually until they disappear. This is due to the heat exchange between the two fluids through the exchange wall, the cold fluid has received the heat lost by the hot fluid. The particles of the cold fluid in the upper channel which receive heat become more and more mobile and the particles of the hot fluid in the lower channel which give up heat become less and less mobile. There is therefore interdependence between the speed and the temperature, the lower the speed, the more the hot fluid and the cold fluid have enough time to exchange heat.

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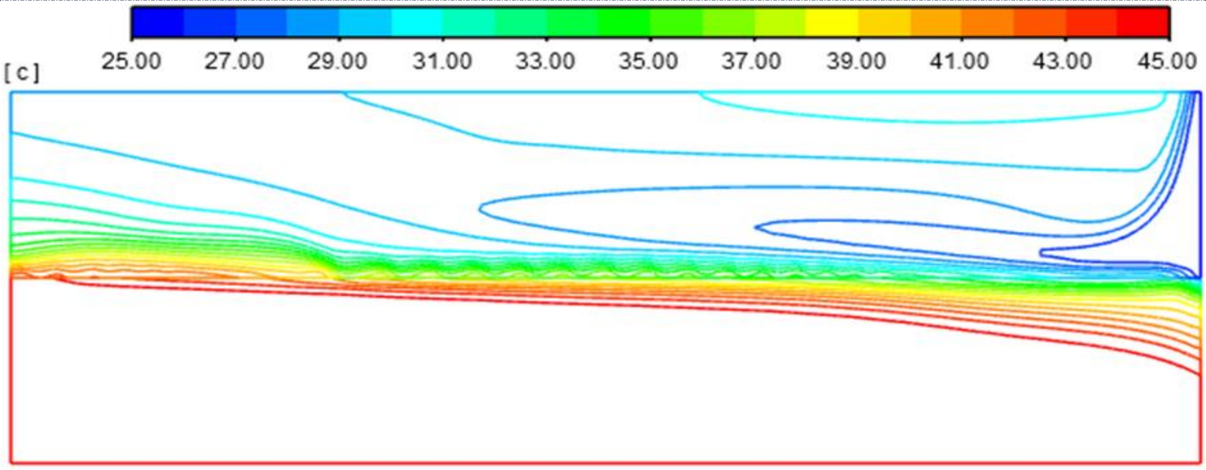

Figure 09: Evolution of isothermal lines

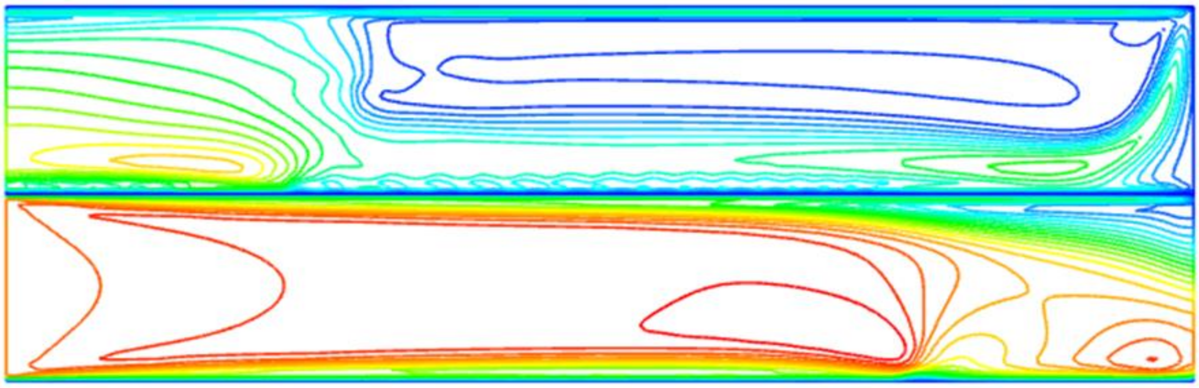

Figure 10: Evolution of streamline

Evolution of the temperature at the outlet of the channels

The variation of the outlet temperatures of the channel (upper) and the variation of the outlet temperatures of the channel (lower) as a function of the position are shown in figures (09 and 10). At the outlet of the channel containing the cold air (the upper channel) there is a sharp increase in the temperature of the cold air. We notice the opposite of this phenomenon at the exit of the channel containing the hot air (the lower channel) where we note a strong decrease in the temperature of the hot air. Therefore, the temperature exchanges between the two fluids are much greater at the channel outlets than inside the exchanger. There is a decrease in the air outlet temperature and the exchange surface of the heat exchanger during movement along the wall.

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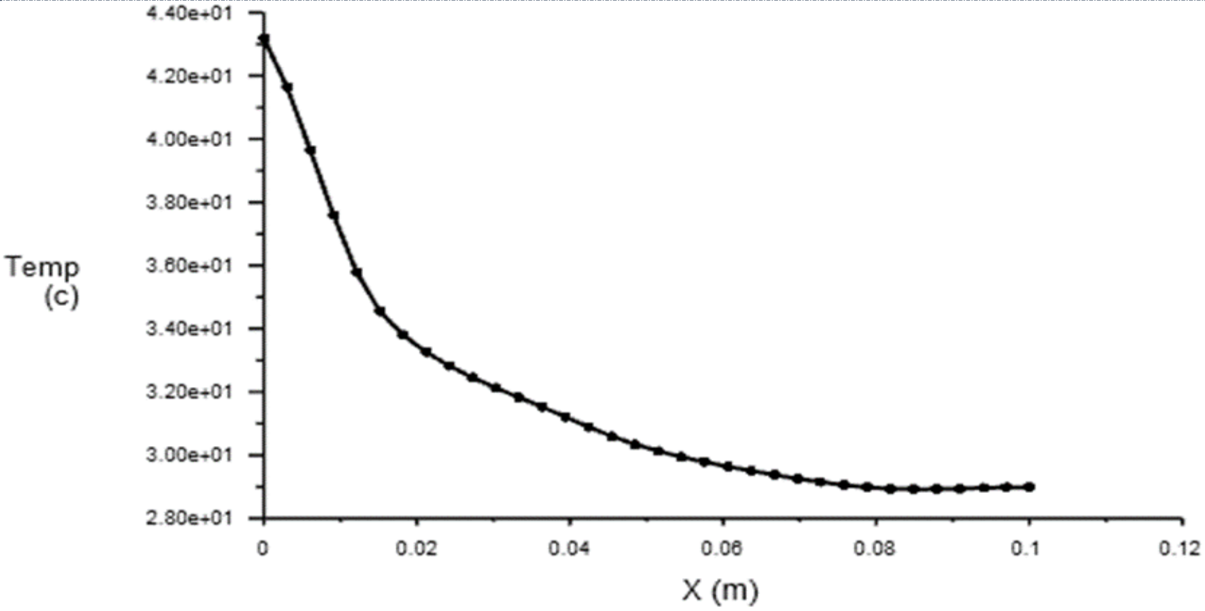

Figure 11: Evolution of the temperature at the outlet of the channel (upper) containing the cold air

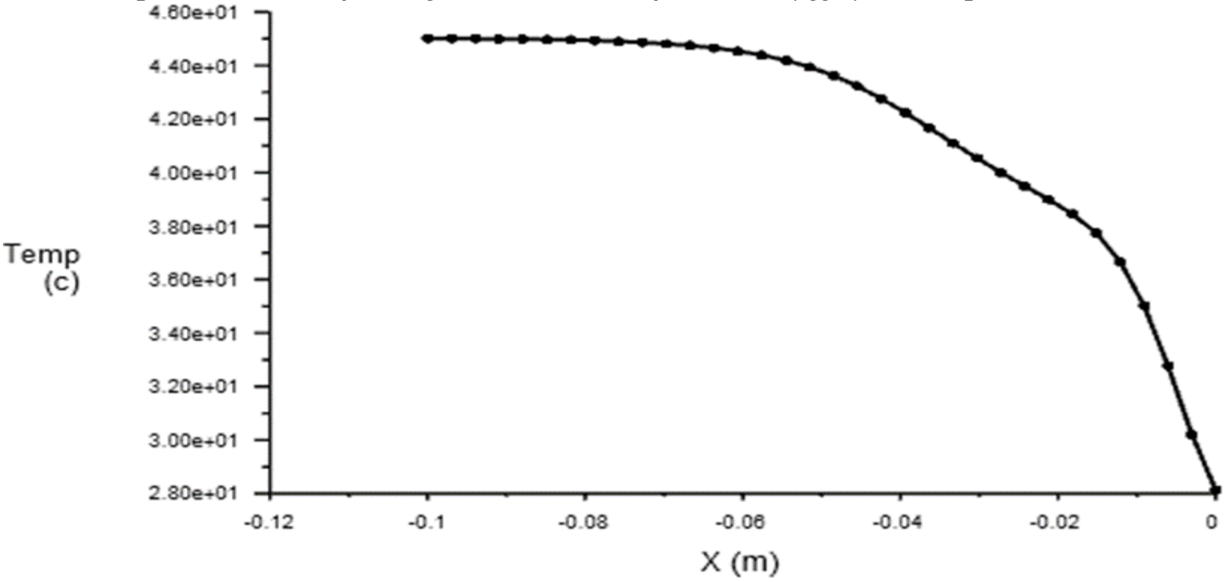

Figure 12: Evolution of the temperature at the outlet of the channel (lower) containing the hot air

Evolution of the temperature on the channel separation wall
For figure 11, it shows us the variation of the temperature on the wall separating the two channels. It is noted that an oscillating decrease in the temperature on the wall induces an increase in the exchange surface and a decrease in the external and global exchange coefficients of the exchanger. It confirms the exchanges of temperature and speed through the exchange wall.

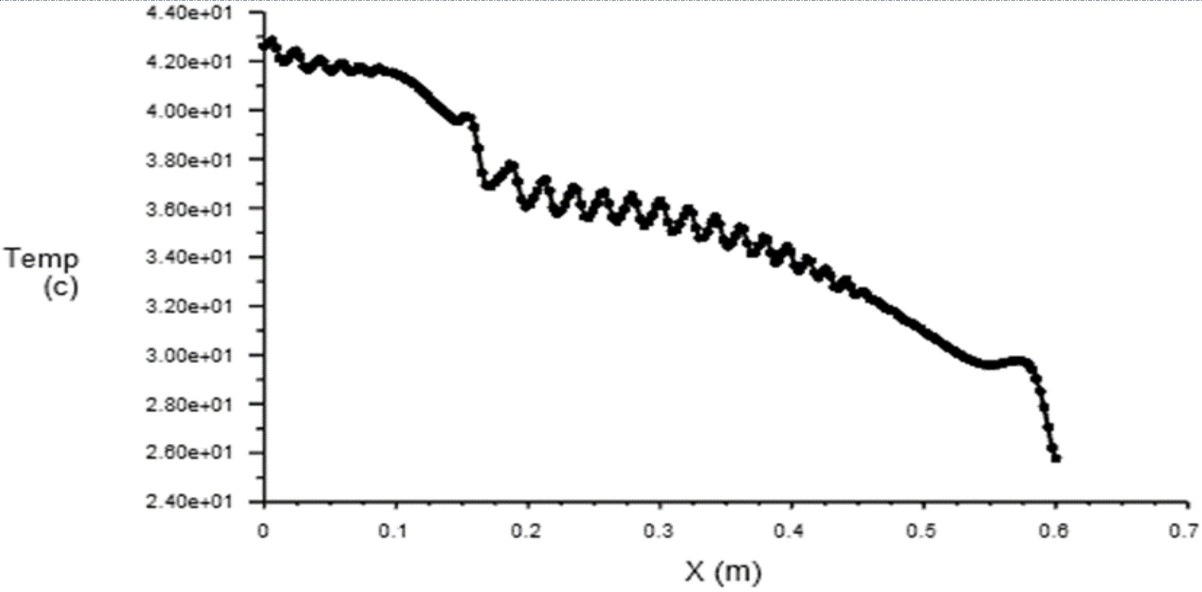

Figure 13: Evolution of the temperature on the exchange wall

5. CONCLUSION

The aim of the present study is to expand the knowledge base on the dynamic and thermal flow of a fluid in a heat exchanger. We have described the equations governing the phenomenon of forced convection in laminar and steady state in dimensional and dimensionless form and the boundary conditions of a countercurrent heat exchanger. These transfer equations which govern our problem are strongly coupled and nonlinear, their theoretical solutions are inaccessible, which is why we have resorted to a numerical resolution method which is the method of finite volumes. The choice of this method is justified by the fact that the numerical simulation software Ansys Fluent is based on this method. For the results obtained, we notice that the fluid velocity is an important factor for the heat exchange. Heat exchange is greater at low fluid flow velocities than at high velocities. The velocity profile depends reciprocally on the temperature.

6. ACKNOWLEDGEMENTS

his work is the brainchild of the Mechanics of Fluids and Transfers Laboratory, Physics Department, Sciences and Technologies Faculty, Cheikh Anta DIOP University, Dakar-Fann, Senegal.

Nomenclature	
Latin letter	
B: form factor	C_p : specific heat capacity of fluid J/kg. K
g : acceleration due to gravity	$\mathrm{Gr} = \frac{\beta g L^3 (T - T_0)}{\vartheta^2}$: Grashof number
h: coefficient of transfer of heat by convection	H : pipe height[m]
L : Tube length[m]	$\tilde{P}$: dimensionless pression
$P_r = \frac{\vartheta}{\alpha}$: Prandtl number	Nu: Nusselt number $Nu = \frac{hL}{\lambda}$
$\overline{Nu}$: Average Nusselt number	$Re = \frac{\rho U e}{\mu} = \frac{Ue}{\nu}$: Reynolds Number
$\mathrm{Ri} = \frac{\mathrm{Gr}}{\mathrm{Re}^2}$: Richardson's number	t : dimensionless time[s]
$\tilde{T}$: dimensionless Temperature	$\tilde{u}, \tilde{v}$: Dimensionless velocity component
$\tilde{x}, \tilde{y}$: Dimensionless coordinates	***Greek symbols***
α: thermal diffusivity [m^2s^{-1}]	β: thermal expansion coefficient [K^{-1}]
λ: thermal conductivity [$Wm^{-1} K^{-1}$]	ν : kinematic viscosity[m^2s^{-1}]
ρ: fluid density	Δt : time step [s]

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Impact Factor: 5.164
CODEN: IJESS7

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