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Research Article

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Biogas Production by Co-digestion: A Possible Alternative to the Threat of Climate Change

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Abstract It is widely share that climate change could have consequences on agricultural production systems worldwide. Sustained predictions are made on the rise in ppm of greenhouse gases and on the rise in temperature. Among these gases, CO_2 is particularly studied because it is on the essential elements involved in the photosynthesis of plants. A change in ppm could have consequences for food crops in general.

The same is true for an increase in temperature with a possible impact on the ability of crops to adapt to this increase. Alternatives are being tested to reduce the use of fossil fuels, the main causes of climate change, by introducing anaerobic digestion (biogas) of organic compound. On the other hand, agriculture practiced in the northern countries is timing more and more towards energy crops such as corn to serve as a substrate for anaerobic digestion. Previous studies have shown a negative effect of climate change on the future of biogas produced from energy crops.

To limit the damage caused by climate change, codigestion can be a reliable solution. In our study for about 52 g of organic matter (OM) and replacing water by sludge in each digester, we were able to obtain 37,8 NL of biogas with 62% of methane with corn silage and cow dung.

Keywords codigestion, biogas, energy crops, climate change

Introduction

The study of mixed 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 unsteady-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 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 humidification's 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 mixed 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 an inlet hot air with a temperature of Tc on the left then crossing through the tube whose walls are brought to a constant temperature Tp.

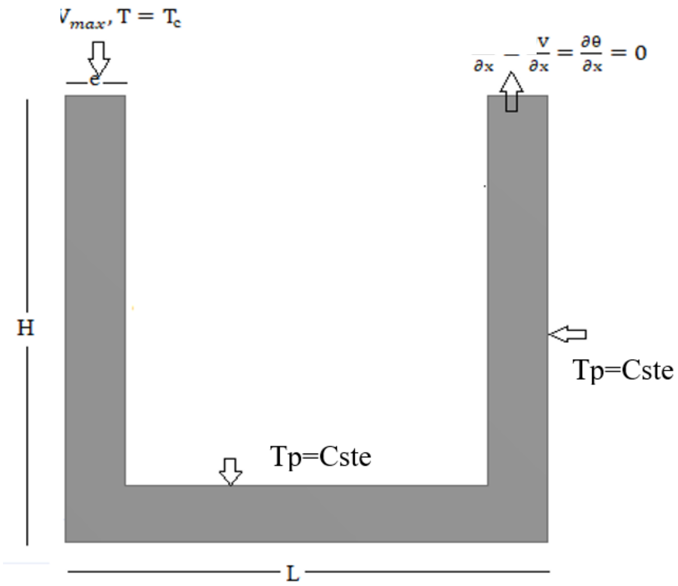

Figure 1: Geometry of the problem

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

- Fluid flow and heat transfer are incompressible, two-dimensional and the laminar regime.
- The thermophysical properties of the fluid (μ, Cp, and k) are constant.
- Viscous dissipation is negligible. There is no heat source.
- 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 mixed 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

$U = V = \theta = 0$

Boundary conditions

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

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

On the walls: $U = 0;\ V = 0\ and\ \theta = cste$

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 spatial discretization is done by a finite volume method with the SIMPLE algorithm while a purely implicit scheme is adopted for the temporal discretization [7]. The mesh adopted in this study is a uniform mesh according to the two vertical and horizontal directions. Practically, 40851 nodes are localized and 40000 elements are found at the level of the enclosure. The varied time step being 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]: σk = 1.0; σω,1 = 2.0; σω,2 = 1.17; γ2 = 0,44; β2 = 0,083; β∗ = 0,09.

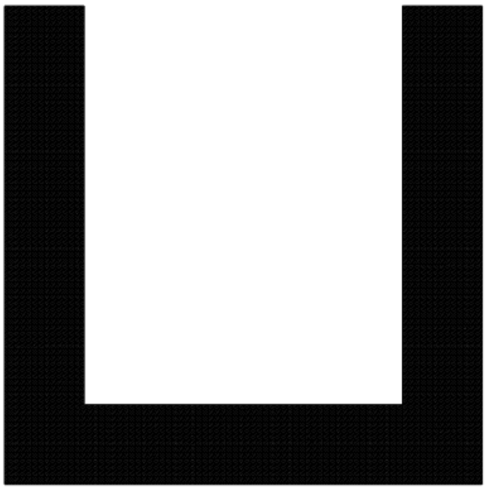

Figure 2: Mesh

Results and Discussion

We present the results and discuss the dynamic and thermal behavior of mixed convection in a U-shaped cavity. The simulation was performed for Prandtl (Pr=0.72) at the occurrence of air and different Rayleigh numbers (Ra= 10^3, 10^4, 10^6, 10^7). For t=5, the streamlines are almost parallel to each other and to the walls of the channel and from the vertical column of the channel towards the outlet with a small variation of the velocity intensity. The isotherms represented at the entrance show that the heat exchange between the walls and the ventilation air is limited in the left vertical column (at the entrance). We notice that the isotherms are much more concentrated at the entrance of the exchanger. This means that the strong temperature variations take place at this level. Indeed, the very low ventilation velocity at the inlet promotes rapid heat transfer in the vertical column. For the low Rayleigh number, we find a small variation of the velocity through the cavity.

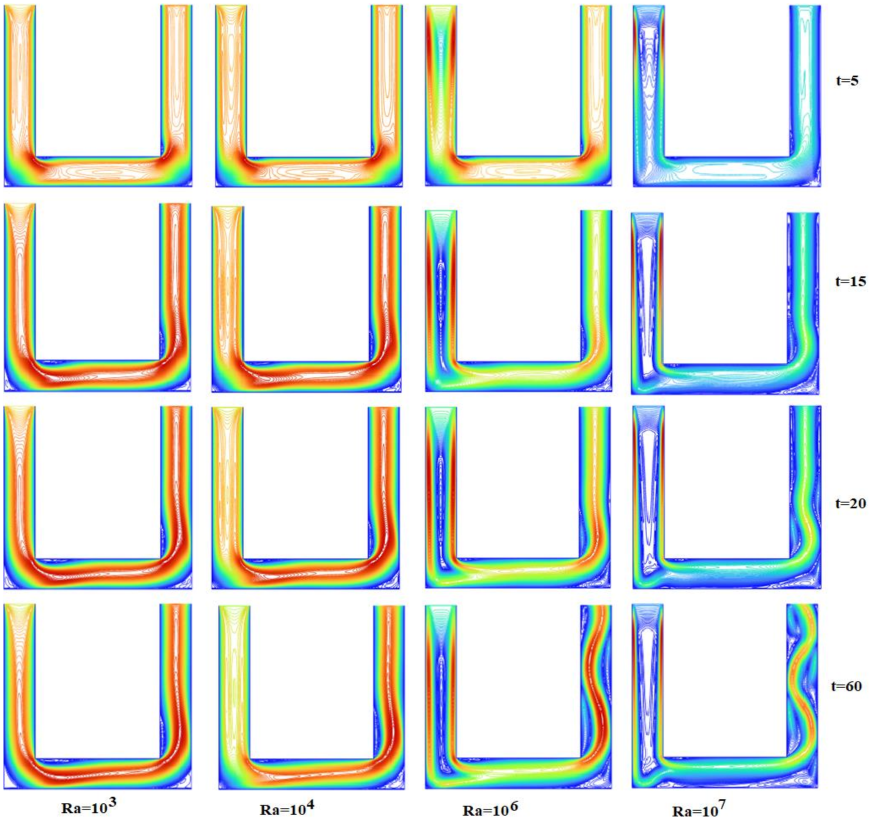

Figure 3: Variation of streamlines as a function of time with different Rayleigh numbers

As the time increases (t>20), the streamlines start with a very slight deformation at the exit to the u-shape. By increasing Ra, we see at the corners an appearance of the contraction zone of the flow but that in the right corner is stronger than that of 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). It should be remembered that it is at the level of the elbow that the temperature gradient is very high. The velocity is strong on the horizontal column where there is no difference between the pressure effect and the gravitation effect. Note also that the two convection modes operate in the opposite direction, right column and in the same direction, left side. At the corners of the tubes, the zones of narrowing are accentuated as Ra increases until the appearance of the cells of training at the level of the two corners, the structure of the flow becomes relatively complex. This new cell is more and more intense compared to the one on the right given the change in flux as the Rayleigh number is increased. The temperature gradients are remarkable only at the inlet of the exchanger. The maximum values are noted at the level of the elbow. This is more noticeable when we zoom in on the surface. 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 air particles. It is also an area of turbulence and pressure drop.

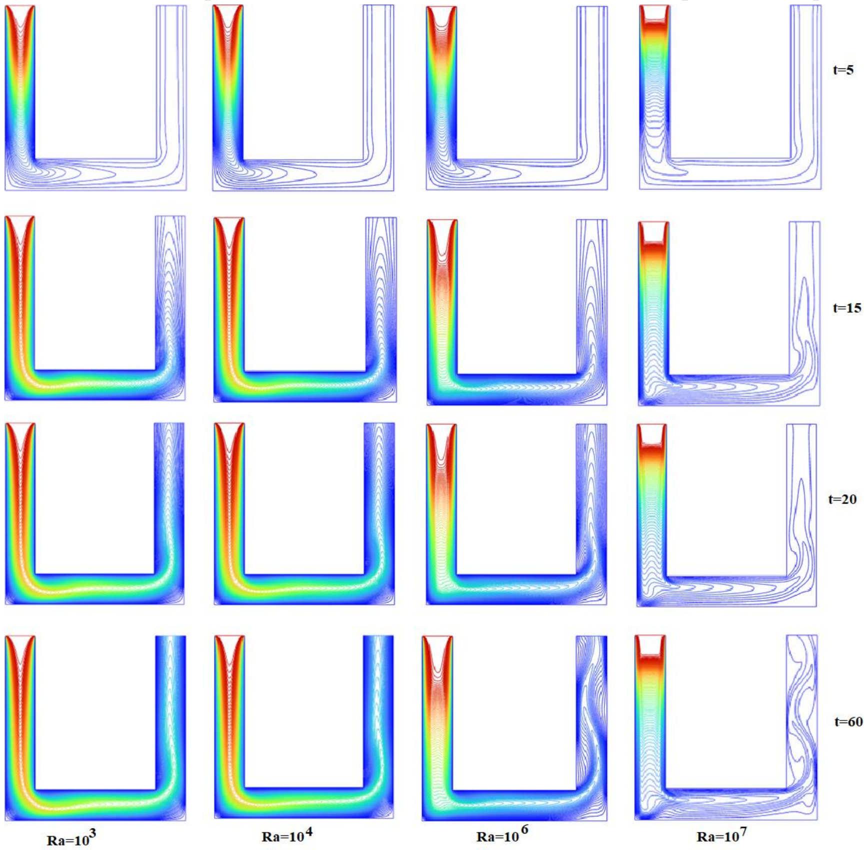

Figure 4: Variation of isotherms as a function of time with different Rayleigh numbers

The outlet air temperature is almost constant for the Rayleigh number Ra=10^5 (outlet temperature value) as there is not a large temperature variation in the column. By increasing the Rayleigh number, the temperature increases from the wall inside the tube and we see the formation of the thermal boundary layer. From the wall to the inside, we have a temperature gradient of 2°C. This means that over time the temperature of the air at the outlet of the exchanger is lower than that of the inlet air (298 K). This can be explained by the loss of thermal energy between the incoming (hot) air and the walls of the channel protected at a lower temperature than that of the incoming air. In other words, we are witnessing a cooling of the air at the exit of the channel. This thermal behavior is explained by the fact that the air-tube system gives up its heat along the tube and over time.

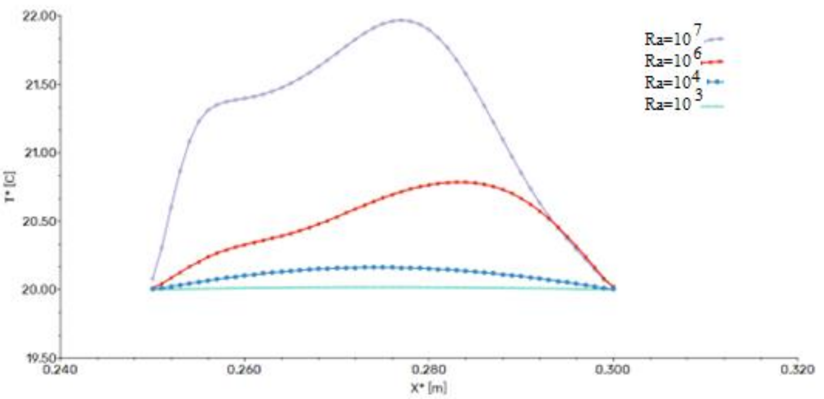

Figure 5: Variation of outlet temperature as a function of Ra

Conclusion

We have numerically studied mixed convection in a U-shaped channel for an unsteady flow representing a Canadian well for air cooling. After having introduced the simplifying hypotheses and then adimensionalized, the transfer equations are projected into the Cartesian coordinate systems with its boundary conditions. For the numerical resolution, the spatial discretization is done by a finite volume method with the SIMPLE algorithm while a purely implicit scheme is adopted for the temporal discretization. The results obtained for different Rayleigh numbers with a fixed Reynolds number are very significant for an unsteady flow and give a lot of information on the dynamic and thermal behavior of the exchanger such as:

- At low Ra, the heat exchange is limited to the left vertical column of the channel with a low velocity variation.
- When Ra increases, we see a significant improvement in thermal and dynamic exchange all the walls of the channel and the birth of the drive cells appear with a constriction 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 $Re=\frac{\rho Ue}{\mu}=\frac{Ue}{\nu}$	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$\left[m^2s^{-1}\right]$ $\alpha=\frac{k}{\rho\, C_p}$	β: thermal expansion coefficient$\left[K^{-1}\right]$
μ : dynamic viscosity$[\mathrm{kg.\,m^{-1}s^{-1}}]$	ν : kinematic viscosity$[\mathrm{m^2.\,s^{-1}}]$
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$[kg.m^{-3}]$

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