Cooperating Thermosolutal Convection in Fluid Annular Cylindrical Cavities with Low and Moderate Radii Ratio

Brahim El moustaine*, Abdelkhaled Cheddadi*
* « Thermal Systems and Real Flows » ERSTER-EMI
Mohammadia School of Engineers, Mohammed V University,
P.O.Box 765, Agdal, Rabat, Morocco

Abstract— We study the problem of thermosolutal convection in an annular cavity bounded by two very long cylinders, coaxial and isothermal, filled with a binary fluid considered incompressible and viscous. The partial differential equations modeling the problem are discretized using the finite difference method with ADI scheme. We focus our numerical simulation on the study of effects the control parameters (thermal Rayleigh number $10^3 \leq Ra_T \leq 10^4$, Lewis number $0.1 \leq Le \leq 20$ and buoyancy ratio $0 \leq N \leq 30$) on the flow structure, heat and mass transfer within the annular cavity.

It is found that the increase of $Ra_T$ number enhances the flow intensity, and improves the heat and mass transfer. On the other hand, in the ranges of variation of the control parameters, the flow structure remain unicellular. The Lewis number $Le$ affects strongly the variation of heat and mass transfer. Also, there is a remarkable effect on the same quantities when we vary the aspect ratio $R$ in the interval $1.2 \leq R \leq 1.6$.

Index Term— Thermosolutal convection, Heat transfer, Mass transfer, Annular medium, Finite differences,

I. INTRODUCTION

The study of thermosolutal natural convection has generated the interest of many scientists and industrialists. The research in this area, spread over a little more than a century. A considerable amount of work has been undertaken, following the discovery of the phenomenon of natural convection in the Bénard experiments [3] and the theoretical analysis of Rayleigh [13] in the early twentieth century to the present.

On the one hand, much research has been done on modeling the phenomenon of pure thermal convection for various cavities. For example, Shi et al. [15] applied finite difference-based lattice BGK model to simulate numerically natural convection and heat transfer in a horizontal concentric annulus bounded by two stationary cylinders with different temperatures. Cheddadi et al. [4] described, numerically and experimentally the free convection flows in a horizontal annulus. They pointed out the existence of a critical Rayleigh number related to the bifurcation phenomenon, with two types of solution: unicellular and bicellular flows. Shahraei [14] studied the fluid dynamic and thermal fields for numerical simulations of buoyancy-driven flows in a vertical eccentric annulus using the penalty finite element method.

On the other hand, many scientists undertook the modeling of the thermosolutal convection in confined fluid mediums. We can cite: Béghein et al. [1] who studied the double diffusion convection and the influence of the buoyancy ratio on the rate of heat and mass transfer, in a square cavity filled with air. Mamou et al. [9] presented an analytical modeling and numerical study of double diffusion natural convection in a fluid contained in a rectangular cavity. Beji et al. [2] used the finite volume method to study double diffusion natural convection in a vertical annulus. Papanicolaou and Belessiotis [12] studied the natural convection, heat and mass transfer in a trapezoidal enclosure to highlight the effect of control parameters on the flow, and on the rate of heat and mass transfer. Chen et al. [7] used Boltzmann model to study the double diffusion natural convection in a vertical eccentric space, different modes obtained in the cases of co-operating or opposed thermal and solutal forces for a single radii ratio $R = 2$. Nakamura et al. [11] found the approximate solution (theoretically) for the vertical ice melting with the transient double effects of temperature and concentration. Cheddadi et al. [5] described the heat and mass transfer by thermosolutal convection in a horizontal annular space, different modes obtained in the cases of co-operating or opposed thermal and solutal forces for a single radii ratio $R = 2$. Nakamura et al. [11] found the approximate solution (theoretically) for the vertical ice melting with the transient double effects of temperature and concentration, and proposed a simple formula to evaluate the melting mass. Masuda et al. [10] studied analytically the double-diffusive convection in a porous medium due to the opposing heat and mass fluxes on the vertical walls. Recently, Cheddadi et al. [6] presented the influence of the mesh on the numerical results and the influence of thermal and solutal buoyancies of double diffusion in an annular cylindrical space with an aspect ratio $R = 2$. 
II. PHYSICAL MODEL AND MATHEMATICAL FORMULATION

A. System description

We study the problem of thermosolutal convection in an annular space bounded by two very long cylinders, coaxial, horizontal and isotherm, filled with a binary fluid considered incompressible and viscous. It is assumed that both gradients of temperature and concentration are negligible along the axis direction of the annular medium, that is to say the problem is considered to be bidimensional. The surface of the inner cylinder is maintained at constant uniform temperature and solute concentration $T_i$, $C_i$, respectively. The surface of the outside cylinder is maintained at $T_o$, $C_o$ where $T_i > T_o$ and $C_i > C_o$ (Fig. 1).

Nomenclature

$R$ : Aspect ratio of the cavity, $r_o/r_i$
$r$ : Polar radial coordinate
$\theta$ : Polar angle
$U$ : Dimensionless radial velocity
$V$ : Dimensionless tangential velocity
$C$ : Dimensionless solute concentration
$T$ : Dimensionless temperature
$D$ : Mass diffusivity, $m^2.s^{-1}$
$N$ : Ratio of buoyancy forces
$Nu$ : Nusselt number
$Pr$ : Prandtl number
$Ra_T$ : Thermal Rayleigh number based on the inner radius
$Ra_C$ : Thermal Rayleigh number based on the gap width
$Sh$ : Sherwood number
$Le$ : Lewis number
$g$ : Acceleration of gravity, $m.s^{-2}$
$\rho$ : Density, $Kg.m^{-3}$
$\alpha$ : Thermal diffusivity, $m^2.s^{-1}$
$\nu$ : Kinematic viscosity, $m^2.s^{-1}$
$\beta_T$ : Coefficient of thermal expansion, $K^{-1}$
$\beta_C$ : Coefficient of solutal expansion, $K^{-1}$
$\psi$ : Dimensionless stream function
$\omega$ : Dimensionless vorticity

Subscripts

$i$ : Inner
$o$ : Outer
$T$ : Thermal
$C$ : Solutal

B. Simplifying hypotheses

- The fluid is Newtonian and incompressible and satisfying the Boussinesq hypothesis.
- With $\rho = \rho^0 (1 - \beta_T (T - T^0) - \beta_C (C - C^0))$ (1) $T^0$, $C^0$ and $\rho^0$ are the values of temperature, concentration and density taken as reference.
- The fluid flow within the cavity is laminar.
- The thermophysical properties of the fluid are constant over the temperature and concentration range considered. And work induced by viscous forces and pressure is negligible.
- The energy transfer by radiation and the Soret and Dufour effects are neglected.

C. Dimensionless Equations

The non-dimensional equations governing the two-dimensional problem are written in polar coordinates $(r, \theta)$ using the stream function-vorticity formulation $(\psi, \omega)$. The problem is governed by the following equations written in dimensionless form:

- **Momentum conservation:**

  \[
  \frac{\partial \omega}{\partial t} + U \frac{\partial \omega}{\partial r} + V \frac{\partial \omega}{\partial \theta} - Pr \left( \frac{\partial^2 \omega}{\partial r^2} + \frac{1}{r} \frac{\partial \omega}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \omega}{\partial \theta^2} \right) = Ra_T Pr \left[ \left( \frac{1}{r^2} \frac{\partial T}{\partial \theta} \cos \theta + \frac{\partial \psi}{\partial r} \sin \theta \right) + N \left( \frac{1}{r^2} \frac{\partial C}{\partial \theta} \cos \theta + \frac{\partial \psi}{\partial r} \sin \theta \right) \right] \quad (2)
  \]

- **Energy conservation:**

  \[
  \frac{\partial T}{\partial t} + U \frac{\partial T}{\partial r} + V \frac{\partial T}{\partial \theta} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} \quad (3)
  \]

- **Solute conservation:**

  \[
  \frac{\partial C}{\partial t} + U \frac{\partial C}{\partial r} + V \frac{\partial C}{\partial \theta} = \frac{1}{Le} \left( \frac{\partial^2 C}{\partial r^2} + \frac{1}{r} \frac{\partial C}{\partial r} + \frac{1}{r^2} \frac{\partial^2 C}{\partial \theta^2} \right) \quad (4)
  \]

The vorticity $\omega$ is related to the stream function $\psi$ by the relation:

\[
\omega = - \frac{1}{r^2} \frac{\partial^2 \psi}{\partial \theta^2} - \frac{1}{r} \frac{\partial \psi}{\partial r} - \frac{\partial^2 \psi}{\partial r^2} \quad (5)
\]

In these equations, four characteristic dimensionless numbers appear:

- $Ra_T = \frac{\beta_T \Delta T r_i^3}{\nu}$ the thermal Rayleigh number,
- $N = \frac{Ra_C}{Ra_T}$ the ratio of buoyancy forces,
- $Pr = \frac{\nu}{\alpha}$ the Prandtl number,
- $Le = \frac{\alpha}{\beta_C}$ the Lewis number.
Boundary conditions

The above equations are subject to the following conditions:

\[
\begin{align*}
\text{For } r = 1, & \quad \psi = 0, T = C = 1 \quad \text{and } U = V = 0, \\
\text{for } r = R & \quad \psi = 0, T = C = 0 \quad \text{and } U = V = 0
\end{align*}
\]

(6)

On the vertical plane containing the axis of the cylinders:

\[
\frac{\partial \psi}{\partial r} = 0 \quad \text{and} \quad \psi = 0
\]

(7)

For the vorticity \( \omega \) we consider the following boundary conditions:

\[
\begin{align*}
\text{for } r = 1 \quad & \quad \text{and } 0 \leq \theta \leq \pi : \quad \frac{\partial^2 \psi}{\partial r^2} + \omega = 0; \\
\text{or } \theta = \pi \quad & \quad \text{and } \quad r_1 \leq r \leq r_o : \quad \omega = 0
\end{align*}
\]

(8)

D. Numerical method and numerical code validation

The mathematical equations modeling the thermosolutal convection in annular medium are discretized by the centered finite difference method, with the Alternating Direction Implicit (ADI) scheme. The results will be presented using a thermal Rayleigh number based on the gap width \( Ra_T^f \), defined by:

\[
Ra_T^f = (R - 1)^3 Ra_T.
\]

(9)

The heat and mass transfer rates are evaluated by an arithmetic average of local Nusselt number and local Sherwood number on hot and cold walls, defined as follows:

\[
\begin{align*}
Nu & = \frac{-\log(R)}{2\pi} \left( \frac{\pi}{0} \frac{\partial T}{\partial r} \bigg|_{r=1} d\theta + R \int_0^\pi \frac{\partial T}{\partial r} \bigg|_{r=R} d\theta \right) \\
Sh & = \frac{-\log(R)}{2\pi} \left( \frac{\pi}{0} \frac{\partial C}{\partial r} \bigg|_{r=1} d\theta + R \int_0^\pi \frac{\partial C}{\partial r} \bigg|_{r=R} d\theta \right)
\end{align*}
\]

(10)

To validate the numerical code developed, we simulated the problem of thermosolutal convection where ratio of buoyancy forces is null \( N = 0 \), \( Le = 1 \) that is in the case of pure thermal convection. In table 1, the code has been validated by comparison with numerical and experimental results.

Then in the case \( R = 2.0 \) , \( Pr = 0.7 \) and for different value of thermal Rayleigh number appearing in Tab. 2, we compared the thermal transfer and maximum value of the stream function evolutions with numerical study based on the classic formulation pressure-velocity fields.

The results obtained by the present model agree quantitatively well with the data available in the literature. The annular medium is discretized with a uniform mesh on tangential and radial directions. In our study, the numerical results are obtained for a mesh (60x60). We chose this mesh which gives accurate results with optimal calculation time. The Prandtl number value is fixed, \( Pr = 0.7 \).

III. RESULTS AND DISCUSSION

A. Effects of the aspect ratio \( R \) and thermal Rayleigh number \( Ra_T^f \)

To investigate the effect of aspect ratio \( R \) and the thermal Rayleigh number \( 10^3 \leq Ra_T^f \leq 10^4 \) on thermosolutal convection (flow intensity, heat and mass transfer), we simulated our problem for the following three aspect ratios of the cavity \( R = 1.6 \), \( R = 1.4 \) and \( R = 1.2 \), in the case where buoyancy forces cooperate \( (N = 2) \). The results are compared with the reference case \( N = 0 \) (pure thermal convection). The Lewis number has been taken \( Le = 1 \) as to guarantee equality between the thermal diffusivity and mass diffusivity.

The heat and mass buoyancy forces cooperate. The numerical data displayed in Fig. 2 show that the increase in thermal Rayleigh number \( Ra_T^f \) always promotes fluid flow as the flow intensity represented by the maximum value of the stream function \( \psi_{\text{max}} \) increases.

On the other hand, it should be noted that \( \psi_{\text{max}} \) values decrease with increasing aspect ratio \( R \). For high thermal Rayleigh number \( Ra_T^f \) values, the aspect ratio effect on the fluid flow becomes remarkable. In addition, it is clear that increased buoyancy forces ratio improves fluid flow compared to the pure thermal convection \( N = 0 \).

From Fig. 3, it follows that there is an improvement of heat (and mass, since \( Le = 1 \)) transfer with the increase of thermal Rayleigh number. This increase is due to the increased importance of thermal and solutal forces. On the other hand, the decrease in the radii ratio \( R \), and hence in the thickness of the annular medium \( (R - 1) \), contributes to the weakening of heat (mass) transfer in both cases of cooperative buoyancy forces \( N = 2 \) and pure thermal convection \( N = 0 \).

Finally it is found in this case that the heat transfer evolution becomes independent of the stream function variation. It follows then that the thickness of the annular layer has more influence on the heat transfer improvement than has the flow intensity. Concerning the evolutions of heat transfer \( Nu \) and the maximum value of the stream function \( \psi_{\text{max}} \), we register with aspect ratio variation an effect opposite to that obtained with other control parameters \( (Ra_T, Le, N) \), as can be seen hereafter.

B. Effects of the Lewis number \( Le \) and the buoyancy ratio \( N \)

To investigate the effects of the Lewis number \( Le \) and the buoyancy ratio \( N \) on the flow intensity \( \psi_{\text{max}} \), and the evolutions of heat \( (Nu) \) and mass \( (Sh) \) transfer rates; we performed a series of numerical simulations for the aspect ratio of cavity \( R = 1.6 \), keeping in this section a fixed thermal Rayleigh number. The value of thermal Rayleigh number was set at \( Ra_T^f = 7 \times 10^3 \) that guarantee the development of both
heat and mass convection. The cooperating case of buoyancy forces is considered: \( N \geq 0 \).

- **Effect the Lewis number: \( 0.01 \leq Le \leq 20 \)**

  In this section we chose a buoyancy ratio \( N = 1 \). The numerical simulation shows in Fig. 4 that the maximum value of the stream function \( \psi_{\text{max}} \) rapidly decreases when increasing the Lewis number \( (Le \leq 5) \), and beyond this value \( \psi_{\text{max}} \) becomes independent of increased Lewis number \( Le \). It is observed in Fig. 5 that the flow consists of a single convective cell having a crescent shape, occupying a half-annular space and rotating anti-clockwise. The fluid rises against the hot inner cylinder. At the top, it becomes cooler and dives against the cold cylinder.

  This weakening of the flow intensity with increasing Lewis number is generated by the importance of the kinematic viscosity acting as a uniform resistance to the fluid flow into the cavity. This justification is approved by dimensionless analysis of the controls parameters, since increasing Lewis number could be obtained by enhancing the thermal diffusivity \( \alpha \). As the Prandtl number is kept fixed, this implies an increase in the viscosity \( \nu \). When \( Le \) takes large values, the flow intensity tends towards an asymptotic value \( \psi_{\text{max}} = 13 \) obtained in the flow of pure natural convection-type, see Fig. 2. With regard to the flow intensity, the case of thermosolutal convection \( Le \gg 1 \) and \( N = 1 \) is similar to the case of pure thermal convection \( N = 0 \).

  Figure 6 shows that the Nusselt number \( Nu \) decreases and the Sherwood number \( Sh \) increases as the Lewis number undergoes growth:

  - \( 0.01 \leq Le < 1 \), the Nusselt number is higher than Sherwood number, so the thermal convection is more important than the mass convection, this is also shown in Fig. 5.a where a distortion of isothermal lines and a stratification of isoconcentration lines can be noticed. The heat transfer rate decreases because the flow intensity becomes weaker, but increasing Lewis number improves mass transfer and promotes isomass lines distortion, indicating starting of the mass convection.

  - \( Le = 1 \), Nusselt and Sherwood numbers are equal \( Nu = Sh = 1.75 \) because the heat and mass diffusivities are similar. This equality between the heat and mass transfer rates is justified by the identical forms of isotherms and isoconcentration lines in the annular cavity observed in Fig. 5.b.

  - \( Le > 1 \), mass transfer becomes important and larger than heat transfer because a large deformation is observed in Fig. 5.c, 5.d in the isoconcentration lines, while there is a comparatively slight distortion of isothermal lines. Note that the Nusselt number is almost quasi-independent of the variation of Lewis number for \( Le > 10 \). The decrease in Nusselt number is due to the weakening of the flow; the \( \psi_{\text{max}} \) values decrease and approach those of pure thermal convection \( N = 0 \) as Lewis number becomes large.

So the effect of convection on the heat transfer rate becomes low. While the increase in Sherwood number can be explained by the intense amplification of mass Rayleigh number compared to thermal Rayleigh number with an amplification factor defined by \( N \times Le = Ra_c/Ra_T \).

- **Effect of the buoyancy forces \( N: \ 0 \leq N \leq 30 \)**

  In this part, we have examined the effect of varying the buoyancy ratio in the range \( 0 \leq N \leq 30 \), with \( Le = 1 \); this ensures equality between the heat and mass diffusivities, and thereafter eliminates the Lewis number effect. We keep the same values of other control parameters \( Ra_c = 7 \times 10^3 \) and \( R = 1.6 \).

  We note in Fig. 7 that increasing the ratio of buoyancy forces \( N \) promotes the flow intensity, because heat and mass forces cooperate and buoyancy effects are added. Secondly, during the increase of the buoyancy ratio \( N \), the convection cells conserve the rotation direction obtained in the case of the pure thermal convection. When the ratio \( N \) varies between 0 and 5, the values of \( \psi_{\text{max}} \) undergo high progression. And beyond the value \( N = 5 \), the evolution of \( \psi_{\text{max}} \) is globally linear with a smaller rate of variation. In Fig. 8, it is worth noting that in the case of high values of \( N \), a vortex appears inside and in the lower part of the convective cell.

  As the Lewis number was fixed to the value \( Le = 1 \), the heat and mass transfer evolve in a similar way. So, only the heat transfer rate is presented. It is observed in the Fig. 9 that the Nusselt (Sherwood) number increases with the variation of the buoyancy forces ratio \( N \), because heat and mass forces cooperate and buoyancy effects are added. As well there is a high growth in heat (mass) transfer in the area \( 0 \leq N \leq 5 \), and a relatively lower enhancement transfer in the area \( 5 < N < 30 \), in accordance with the trend depicted in the variation of the flow intensity.

  In Fig. 10, we plotted the local Nusselt number on both walls depending on the polar angle \( \theta \), for four values of the buoyancy forces ratio \( N \). Graph (a) and graph (b) illustrate respectively the local heat transfer on the hot wall \( Nu_1 = f(\theta, N) \) and on the cold wall \( Nu_o = f(\theta, N) \). The values of these rates give valuable information on the thermal layer thicknesses on the isothermal cylinders. Note that the rate \( Nu_1 \) reaches its maximum at \( \theta = 0 \) (thermal layers on the bottom part of the inner cylinder are very thin) and the rate \( Nu_o \) reaches its maximum at \( \theta = \pi \). Globally, it is shown that increasing buoyancy forces ratio \( N \) improves local heat transfer on both walls.

  For high values of \( N \), an extremum is observed in the inner transfer rate, near \( \theta = \pi/3 \). This is explained by the development in this angular region of a vortex within the great
convection cell, as noted above, and the resulting distortion in the isotherms as seen in Fig. 8(c-d).

IV. CONCLUSION

A numerical study has been conducted on the phenomenon of double diffusive natural convection occurring in a concentric horizontal annulus filled with a binary fluid with Prandtl number \( Pr = 0.7 \). The temperature and solute gradients that induce the fluid motion are imposed radially, and the buoyancy effects are cooperating. The flow is considered to be laminar and two-dimensional. The temperature and concentration distributions as well as the flow intensity vary depending on the controls parameters defining the characteristics of the problem (the thermal Rayleigh number \( Ra_T \), Lewis number \( Le \), ratio of buoyancy forces \( N \), and the geometrical aspect ratio \( R \) of the cavity).

The summary of the major results is as follows:

- The thermal Rayleigh number is the principal parameter generating the convective movement in the flow in the annular cavity. Increasing \( Ra_T \) improves the heat and mass transfer.
- As the ratio of buoyancy forces \( N \) is positive (cooperating forces), increasing \( N \) leads to an enhancement of the heat and mass transfer rates, respectively \( Nu \) and \( Sh \).
- Varying the Lewis number has opposite effects on thermal and solutal transfers. While \( Nu \) is decreasing gradually, \( Sh \) increases rapidly as \( Le \) is increased.
- The reduction in thickness of the annular space for \( Le = 1 \) contributes to the weakening of the heat (mass) transfer.

REFERENCES

Table I

<table>
<thead>
<tr>
<th>$Ra_T^f$</th>
<th>$Nu$</th>
<th>Experimental (Kuehn and Goldstein [8])</th>
<th>Numerical (Shi et al. [15])</th>
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<td>$2.38 \times 10^3, Pr = 0.716$</td>
<td>1.28</td>
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<td>$9.50 \times 10^3, Pr = 0.717$</td>
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<td>$3.20 \times 10^4, Pr = 0.717$</td>
<td>2.98</td>
<td>2.89</td>
<td>2.911</td>
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<tr>
<td>$6.19 \times 10^4, Pr = 0.718$</td>
<td>3.34</td>
<td>3.32</td>
<td>3.361</td>
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<tr>
<td>$1.02 \times 10^5, Pr = 0.718$</td>
<td>3.78</td>
<td>3.66</td>
<td>3.531</td>
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Table II

<table>
<thead>
<tr>
<th>$Ra_T^f$</th>
<th>Present work</th>
<th>Numerical (Cheddadi et al. [4])</th>
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<tr>
<td>$1.00 \times 10^3$</td>
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<td>$1.70 \times 10^3$</td>
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<tr>
<td>$1.00 \times 10^4$</td>
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Fig. 1. Problem scheme

Fig. 2. Variation of $\psi_{\text{max}} = f(Ra_T^f, R)$ for $Le = 1, N = \{0, 2\}$
Fig. 3. Variation of $N_u = f(Ra, R)$ for $Le = 1, N = \{0, 2\}$

Fig. 4. Effect of the Lewis number on $\psi_{\text{max}}$ for $R = 1.6, Ra^*_T = 7 \times 10^3$ and $N = 1$
Fig. 5. Streamlines (left), isotherms (middle) and isomass lines (right) for \( R = 1.6, Ra = 7 \times 10^3 \) and \( N = 1 \)

(a): \( Le = 0.01 \)  
(b): \( Le = 1 \)  
(c): \( Le = 10 \)  
(d): \( Le = 20 \)

Fig. 6. Effect the Lewis number on \( Nu, Sh \) for \( R = 1.6, Ra = 7 \times 10^3 \) and \( N = 1 \)
Fig. 7. Effect of the ratio $N$ on $\psi_{\text{max}}$ for $R = 1.6$, $Ra_T = 7 \times 10^3$ and $Le = 1$

(a): $N = 0$  
(b): $N = 1$  
(c): $N = 10$  
(d): $N = 30$

Fig. 8. Streamlines (left), isotherms / isomass lines (right) for $R = 1.6$, $Ra_T = 7 \times 10^3$ and $Le = 1$
Fig. 9. Effect of the ratio $N$ on $Nu$ for $R = 1.6$, $Ra_T = 7 \times 10^3$ and $Le = 1$

Fig. 10. Local heat transfer; (a) on the hot wall $Nu_i$, (b) on the cold wall $Nu_o$, for $R = 1.6$, $Ra_T = 7 \times 10^3$ and $Le = 1$