Heat and mass transfer by natural convection around a hot body in a rectangular cavity

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KEYWORDS
Heat and mass transfer; Natural convection; Cavity.

Abstract. The simultaneous heat and mass transfer by natural convection around a hot body in a rectangular cavity is investigated numerically. The cavity is filled with air and the ratio of body’s length to enclosure’s length is assumed to be constant at 1/3. The differential equations for continuity, momentum, energy and mass transfer are solved using the Patankar technique. The results are displayed in the form of isotherms, isoconcentrations and streamlines, and the effects of Rayleigh number, Lewis number and buoyancy ratio on average Sherwood and Nusselt numbers are investigated. The study covers a wide range of Rayleigh numbers, Lewis numbers, and buoyancy ratios. It is observed that by increasing Lewis number, the average Sherwood number increases and the average Nusselt number decreases. Additionally, by increasing the absolute value of buoyancy ratio, the average Sherwood and Nusselt numbers enhance. This work presents a novel approach in this field in the light of geometry and the range of dimensionless numbers.

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1. Introduction

Natural convection has become the subject of many experimental and numerical studies in recent years due to its widespread applications in engineering and industry, such as cooling of electronic systems, float glass manufacturing, solar ponds and metal solidification processes. Davis [1] investigated a benchmark for numerical solution on natural convection of air in a square enclosure. Ostrach [2] traced out the rich diversity of natural convection problems in science and technology. The numerous studies have focused extensively on convective flows driven by the density inversion effect [3-10].

The steady-state flow structure, temperature and heat transfer in a square enclosure, heated and cooled on opposite vertical walls and containing cold water, are numerically investigated by Lin and Nansteel [3]. Nansteel et al. [4] studied the natural convection of water in the vicinity of its maximum density in a rectangular enclosure in the limit of small Rayleigh number. They observed that the strength of the counter rotating flow decreases with decreasing aspect ratio. Hossain and Rees [8] studied the natural convection in an enclosure with heat generation. In this analysis, when the heat generation parameter is sufficiently strong, the circulation of the flow is reversed.

Lee and Ha [11] numerically investigated natural convection in a horizontal enclosure with a conducting body. They compared the results of the case of conducting body with those of adiabatic and neutral isothermal bodies. They showed that when the dimensionless thermal conductivity is 0.1, a pattern of fluid
flow and isotherms and the corresponding surface- and 
time-averaged Nusselt numbers are almost the same as 
the case of an adiabatic body.

Das and Reddy [12] studied natural convection 
flow in a square enclosure with a centered internal 
conducting square block both having an inclination 
angle using SIMPLE algorithm. They considered an 
angle of inclination in the range of 15-90 degrees and 
ratio of solid to fluid thermal conductivities of 0.2 
and 5. Sheikhzadeh et al. [13,14] studied the steady 
mageto-convection around an adiabatic body inside a 
square enclosure.

The simultaneous heat and mass transfer occurs 
in many processes in industry and engineering equip-
ment and environmental applications, involving the 
transport of water vapor and other chemical contami-
ants across enclosed spaces. Comparison of the scales 
of the two buoyancy terms in the momentum equation 
suggests three classes of flows:

(i) Heat transfer driven flows when the buoyancy 
effect, due to heating from the sides, is dominant;
(ii) Combined heat and mass transfer driven flows 
when both buoyancy terms are important;
(iii) Mass transfer driven flows, when the buoyancy, 
due to heating from the sides, is negligible.

The distinction between these flows can be made using 
the buoyancy ratio ($N$): $|N|=O(1)$ for class (i), $|N|\sim 
O(1)$ for class, (ii), and $|N|=O(1)$ for class (iii).

Gebhart and Pera [15], using the similarity 
method, investigated the laminar flows which arise in 
fluids due to the interaction of the force of gravity 
and density differences caused by the simultaneous 
diffusion of thermal energy and chemical species. Bejan 
[16] presented a fundamental study of laminar natural 
convection in a rectangular enclosure with heat and 
mass transfer from the side. He used scale analysis 
to determine the scales of the flow, temperature and 
concentration fields in boundary layer flow for all values 
of Prandtl and Lewis numbers. He investigated the case 
of $N = 0$ to study the heat-transfer-driven flows. Wee 
et al. [17] investigated numerically and experimentally 
the same problem for both horizontal and vertical 
cavity. The experimental technique employed two 
porous plastic plates as two cavity walls allowing the 
imposition of simultaneous moisture and temperature 
gradients.

Nishimura et al. [18] studied numerically the 
oscillatory double-diffusive convection in a rectangular 
enclosure with combined horizontal temperature and 
concentration gradients. Natural convection flow, re-
sulting from the combined buoyancy effects of thermal 
and mass diffusion in a cavity with differentially heated 
sidewalls, was numerically studied by Snoussi et al. [19]. 
They considered a wide range of Rayleigh numbers 
and used finite-element method. They showed that 
mass and temperature fields are strongly dependent 
on thermal Rayleigh number and the aspect ratio of 
the cavity. Chouikhi et al. [20] studied numerically the 
natural convection flow resulting from the combined 
buoyancy effects of thermal and mass diffusion in 
an inclined glazing cavity with differentially heated 
side walls. The double-diffusive natural convection of 
water in a partially heated enclosure with Soret and 
Dufour effects is numerically studied by Nithyadevi 
and Yang [21]. The effect of various parameters such 
as thermal Rayleigh number, buoyancy ratio number, 
Schmidt number and Soret and Dufour coefficients 
on the flow pattern, heat and mass transfer was 
depicted. Nikbalhti and Rahimi [22] studied double-
diffusive natural convection in a rectangular cavity with 
partially thermal active side walls. They found that in 
aiding flow, heat transfer increases by increasing the 
buoyancy ratio. Their results also showed that in 
opposing flow, with increasing buoyancy ratio until 
unity, heat transfer decreases.

Al-Amini et al. [23] and Teamah and Magh-
lany [24] studied the heat and mass transfer in a lid-
driven cavity. More recently, Mahapatra et al. [25] an-
alyzed the effects of uniform and non-uniform heating 
of wall(s) on double-diffusive natural convection in a 
lid-driven square enclosure.

In the present study, natural convection around 
a hot body in a square cavity filled with air is 
studied numerically, using the finite volume method. 
Simultaneous heat and mass transfer and the effects of 
pertinent parameters such as buoyancy ratio, Rayleigh 
and Lewis numbers on the flow are studied. Despite 
numerous works in this field, to the best of authors' 
knowledge, no similar study has been carried out to 
simulate simultaneous heat and mass transfer in the 
current geometry with the range of buoyancy ratio, 
Rayleigh and Lewis numbers used in this study.

2. Mathematical model

A schematic diagram of the enclosure with coordinate 
system and boundary conditions is shown in Figure 1. 
Both body and enclosure walls are held at constant 
but different temperatures and concentrations: high 
temperature and concentration ($T_b$, $C_b$) for the 
body and low temperature and concentration ($T_c$, $C_c$) for 
the cavity are considered. In this study, the ratio of $W/L$ 
is maintained constant at 1/3, and the body is located 
in the center of the cavity.

The Boussinesq approximation holds, meaning that 
density is linearly proportional to both temperature 
and concentration,

$$\rho = \rho_0 (1 - \beta \Delta T - \beta M (c - c_e)).$$  \hspace{1cm} (1)

With these assumptions, two dimensional laminar
governing equations including continuity, momentum, concentration and energy in steady state conditions can be written as:

Continuity equation:

\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0. \]  

(2)

x-momentum equation:

\[ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + v \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right). \]  

(3)

y-momentum equation:

\[ u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + v \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + g \beta T (T - T_c) + \beta_M (c - c_c). \]  

(4)

Energy equation:

\[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right). \]  

(5)

Concentration equation:

\[ u \frac{\partial c}{\partial x} + v \frac{\partial c}{\partial y} = D \left( \frac{\partial^2 c}{\partial x^2} + \frac{\partial^2 c}{\partial y^2} \right). \]  

(6)

Using the dimensionless variables:

\[ X = \frac{x}{L}, \quad Y = \frac{y}{L}, \quad U = \frac{uL}{\alpha}, \quad V = \frac{vL}{\alpha}, \quad P = \frac{pL^2}{\rho \alpha^2}, \quad \theta = \frac{T - T_c}{T_h - T_c}. \]  

(7)

the dimensionless form of the equations become:

\[ \frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0. \]  

(8)

\[ U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \text{Pr} \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right), \]  

(9)

\[ U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \text{Pr} \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + \text{Ray} \text{Pr}(\theta + N), \]  

(10)

\[ U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}. \]  

(11)

\[ U \frac{\partial C}{\partial X} + V \frac{\partial C}{\partial Y} = \frac{1}{\text{Le}} \left( \frac{\partial^2 C}{\partial X^2} + \frac{\partial^2 C}{\partial Y^2} \right). \]  

(12)

The thermal and mass Rayleigh numbers, Lewis number, Prandtl number, and buoyancy ratio in the above equations are defined as:

\[ \text{Ray} = \frac{g \beta_T L^3 \Delta T}{\alpha v}, \]  

\[ \text{Ra}_M = \frac{g \beta_M L^3 \Delta c}{D v}, \]  

\[ \text{Le} = \frac{\alpha}{D}, \]  

\[ \text{Pr} = \frac{v}{\alpha}, \]  

\[ N = \frac{\beta_M \Delta c}{\beta_T \Delta T}. \]  

(13)

where \( \beta_T \) and \( \beta_M \) are thermal and concentration expansion coefficients, \( \beta_T \) is positive but \( \beta_M \) can be negative or positive [16]:

\[ \beta_T = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial T} \right)_p. \]  

\[ \beta_M = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial c} \right)_p. \]  

(14)

The values of \( \beta_M \) for some species are presented by Gebhart and Pera [15].

Dimensionless boundary conditions are as follows:

\[ 0 \leq X \leq 1 \rightarrow \]

\[ U(X, 0) = U(X, 1) = V(X, 0) = V(X, 1) = 0, \]

\[ \theta(X, 0) = \theta(X, 1) = C(X, 0) = C(X, 1) = 0. \]  

\[ 0 \leq Y \leq 1 \rightarrow \]

\[ U(0, Y) = U(1, Y) = V(0, Y) = V(1, Y) = 0, \]

\[ \theta(0, Y) = \theta(1, Y) = C(0, Y) = C(1, Y) = 0. \]

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\[ \frac{L}{2} \leq \frac{W}{2} \leq \frac{L}{2} + \frac{W}{2} \rightarrow \]

\[ U \left( X, \frac{L}{2} - \frac{W}{2} \right) = U \left( X, \frac{L}{2} + \frac{W}{2} \right) \]

\[ = V \left( X, \frac{L}{2} - \frac{W}{2} \right) = V \left( X, \frac{L}{2} + \frac{W}{2} \right) = 0, \]

\[ \theta \left( \frac{L}{2} - \frac{W}{2} \right) = \theta \left( \frac{L}{2} + \frac{W}{2} \right) \]

\[ = C \left( \frac{L}{2} - \frac{W}{2} \right) = C \left( \frac{L}{2} + \frac{W}{2} \right) = 1. \]  

Local and average Nusselt and Sherwood numbers on the vertical left side hot wall are defined as follows:

\[ \text{Nu} = - \left( \frac{\partial \theta}{\partial X} \right) \bigg|_{X=\frac{L}{2}+\frac{W}{2}}, \]  

\[ \text{Nu}_a = - \frac{1}{A} \int_0^A \left( \frac{\partial \theta}{\partial X} \right) \bigg|_{X=\frac{L}{2}+\frac{W}{2}} dY, \]  

\[ \text{Sh} = - \left( \frac{\partial C}{\partial X} \right) \bigg|_{X=\frac{L}{2}+\frac{W}{2}}, \]  

\[ \text{Sh}_a = - \frac{1}{A} \int_0^A \left( \frac{\partial C}{\partial X} \right) \bigg|_{X=\frac{L}{2}+\frac{W}{2}} dY. \]

In general, Sherwood number represents the mass transfer strength like Nusselt number that symbolizes the heat transfer strength.

3. Numerical method

The governing non-linear equations with appropriate boundary conditions were solved by iterative numerical method, using finite volume technique. In order to couple the velocity field and pressure in momentum equations, the well-known SIMPLER (Semi-Implicit Method for Pressure-Linked Equations Revised) algorithm was adopted. Uniform grid is used for this problem and grid independency test was performed. The discretized equations were solved by the Gauss-Seidel method. The iteration method used in this program is a line-by-line procedure which is a combination of the direct method and the resulting Three Diagonal Matrix Algorithm (TDMA). The diffusion terms in the equations are discretized by a second order central difference scheme, while a hybrid scheme (a combination of the central difference scheme and the upwind scheme) is employed to approximate the convection terms. The under-relaxation factors for U-velocity, V-velocity, energy and concentration equations were 0.6, 0.6, 0.4 and 0.4, respectively. The solution is terminated until the following convergence criterion is met:

\[ \text{Error} = \frac{\sum_{j=1}^{m} \sum_{i=1}^{n} |\xi^{t+1} - \xi^{t}|}{\sum_{j=1}^{m} \sum_{i=1}^{n} |\xi^{t+1}|} \leq 10^{-10}, \]  

where \( m \) and \( n \) are the number of cells in \( X \) and \( Y \) directions, respectively, \( \xi \) is a transport quantity, and \( t \) is the number of iteration.

3.1. Code validation

To reach the grid independent solution, various grids \((21 \times 21, 61 \times 61, 81 \times 81 \text{ and } 101 \times 101)\) were tested for calculating the average Nusselt number on the hot walls (Table 1). The grid size of \( 81 \times 81 \) is adequately appropriate to ensure a grid-independent solution.

Figures 2 and 3 present a comparison between the present work and the results of Lee and Ha [11] and

<table>
<thead>
<tr>
<th>Grid</th>
<th>( 21 \times 21 )</th>
<th>( 61 \times 61 )</th>
<th>( 81 \times 81 )</th>
<th>( 101 \times 101 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nu_a</td>
<td>5.310</td>
<td>6.056</td>
<td>6.162</td>
<td>6.167</td>
</tr>
</tbody>
</table>

Figure 2. The comparison between isotherms for adiabatic body at \( \text{Ra} = 10^5 \) (left) and \( \text{Ra} = 10^6 \) (right); Lee and Ha [11] (solid lines), present work (dashed line).
the effect of buoyancy ratio, the thermal Rayleigh number is kept constant at $10^4$, and Lewis numbers are chosen 0.1, 1 and 10. At all Lewis numbers and buoyancy ratios, two primary vortices are formed at the right and left sides of the hot body, respectively. By increasing the absolute value of $N$ and $Le$ number, the maximum absolute values of stream function for these primary vortices increase and decrease, respectively. Discussion about the effect of $N$ on fluid flow, heat and mass transfer is done in three cases: $N > 0$, $N < 0$ and $N = 0$.

$N > 0$: At $Le=1$ and 10, by increasing the value of $N$ from 0 to 10, two secondary vortices are appeared at the top of the hot body; the strength of these secondary vortices are lower than that of the primary vortices. At $Le = 0.1$, the mass diffusivity is higher than the thermal diffusivity and for all $N$, isoconcentration lines are approximately parallel to the walls (Figure 4). As shown in Figure 4, by increasing the buoyancy ratio, isotherms become denser near the bottom wall of the hot body, and two weak isothermal plumes are appeared at the top corners of the body. According to Figure 7, it is observed that by increasing the buoyancy ratio, the average Nusselt number increases but the average Sherwood number is approximately constant. Figure 7 shows also that the values of average Nusselt number for $Le=0.1$ are higher than those of Sherwood number. At $Le=1$, plots of isoconcentration and isotherms are the same, and the average Nusselt number and Sherwood number are equal [11] and they increase with increasing $N$. According to Figure 5, two isothermal and isoconcentration plumes are appeared

4. Results and discussion

The present study considers the effects of buoyancy ratio ($-10 \leq N \leq 10$), Rayleigh number ($10^3 \leq Ra_T \leq 10^5$) and Lewis number ($0.1 \leq Le \leq 50$) on the isotherms, isoconcentrations, streamlines, average Nusselt number and average Sherwood number.

4.1. Effect of buoyancy ratio

Dependence of streamlines, isoconcentrations and isotherms on buoyancy ratio is shown in Figures 4, 5, and 6. Figure 7 shows the effect of $N$ and $Le$ number on average Sherwood and Nusselt numbers. To investigate
Figure 5. Isotherms, isoconcentrations and streamlines for $Ra = 10^4$ and $Le = 1$.

Figure 6. Isotherms, streamlines and isoconcentrations for $Ra = 10^4$ and $Le = 10$.

Figure 7. Effect of Lewis number and buoyancy ratio on average Sherwood and Nusselt numbers.
at the top corners of the body. At Le=10, the isoconcentration plumes are more obvious than the iso thermal plumes.

\( N < 0 \): At Le=1 and 10, by decreasing \( N \) from 0 to -10, two secondary vortices are appeared at the bottom of the hot body. At Le=10, the strength of the secondary vortices becomes more than that of the primary vortices. At Le=0.1, two weak isothermal plumes are appeared at the bottom corners of the hot body. It is also observed from Figure 7 that by decreasing \( N \) from -1 to -10, the average Nusselt number increases, but the average Sherwood number is approximately constant. At Le=1, two isothermal and isoconcentration plumes are appeared at the bottom corners of the hot body at \( N = -10 \) (Figure 5). By decreasing \( N \) from -1 to -10, the average Sherwood and Nusselt number increase (Figure 7). By decreasing \( N \), both heat and mass transfer are increased. At Le=10, the formed isoconcentration plumes, at the bottom corners of the hot body, are more obvious than the isothermal plumes. At \( N = -1 \) and Le=1 there is no convection flow, and therefore the average Nusselt number and Sherwood number have their minimum values at this condition. As Figure 7 shows, at low Le, \( \text{Sh}_n \) and \( \text{Nu}_n \) are not a function of \( N \).

\( N = 0 \): At \( N = 0 \), convection flow is formed only due to temperature gradient. One isoconcentration plume at the top of the hot body is appeared at Le=10.

Flow direction in the vortices for negative and positive values of \( N \) are shown in Figure 5; for instance at \( N = 10 \) (aiding effect of temperature and concentration gradients) and \( N = -10 \) (opposing effect of temperature and concentration gradients). It is observed that at \( N = -10 \), the primary vortex is CCW and the secondary vortex is CW at the right side of the hot body (in the direction caused by the predominant mass gradient), and at \( N = 10 \), the primary vortex is CW and the secondary vortex is CCW at the right side of the hot body.

4.2. Effect of Lewis number

The effect of Lewis number at Ra = 10^4 on plots of isotherms, isoconcentrations and streamlines for \( N = 1, \ 10 \) and \(-1\) are shown in Figures 8, 9 and 10, respectively. When Le=1, the isotherms and isoconcentrations are similar to each other due to the similarity between the energy and concentration equations. It is observed that by increasing Le, convection flow suppresses and so the maximum absolute value of stream function decreases. At \( N = 1 \) for Le > 10 there is an isoconcentration plume at the top of the hot body. At \( N = 10 \), two secondary vortices and two plumes are appeared at the top of the hot body at Le=1 and 10. The secondary vortices vanish at Le=30 and 50, and two plumes become one plume. It is observed from Figure 9 that isoconcentration plumes are more obvious than that of isothermal plume. It is found from Figure 7 that mass transfer increases when Le

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**Figure 8.** Isotherms, Isoconcentrations and streamlines for \( N = 1 \) and Ra = 10^4.
enhances as a result of densing solutal boundary layer (it is seen in Figure 8). On the other hand, $Nu_a$ reduces when $Le$ increases. As $Le$ increases, thermal diffusivity increases ($\alpha = k/\rho c_p$). It causes conduction heat transfer to be stronger at high $Le$ compared to the low $Le$ cases where convection is stronger and dominant. Because conduction has a minor effect on heat transfer in comparison with convection, $Nu$ decreases as $Le$ increases.

The effect of $Le$ on isotherms, isoconcentrations and streamlines at $N = -1$ is shown in Figure 10. The isotherms are approximately the same. It is observed that by increasing $Le$, the maximum absolute value of stream function increases.

The predicted values of $Sh_a$ and $Nu_a$ in the ranges investigated for $N$ and $Le$ and at $Ra = 10^6$ are correlated and presented in Table 2. The validity ranges for these correlations are $0.1 \leq Le \leq 50$ and $-10 \leq N \leq 10$.

### Table 2. Correlations of $Sh_a$ and $Nu_a$ in terms of $Le$ and $N$.  

<table>
<thead>
<tr>
<th>Range of $N$</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N &gt; 0$</td>
<td>$Sh_a = 7.3167Le^{0.237}N^{0.1103}$</td>
</tr>
<tr>
<td></td>
<td>$Nu_a = 7.0775Le^{-0.0764}N^{0.0818}$</td>
</tr>
<tr>
<td>$N \leq -1$</td>
<td>$Sh_a = 5.2186Le^{0.2370}N^{0.2418}$</td>
</tr>
<tr>
<td></td>
<td>$Nu_a = 6.4143Le^{-0.0527}N^{0.0937}$</td>
</tr>
</tbody>
</table>

4.3. Effect of Rayleigh number

Effect of Rayleigh number on plots of isotherms, isoconcentrations and streamlines, for instance at $N = 1$ and $Le=1$, are shown in Figure 11. It is observed, as expected, that convection flow becomes stronger by increasing $Ra$, and the secondary vortices are appeared at $Ra = 10^7$. Variations of $Sh_a$ and $Nu_a$ with respect to $N$ at various $Ra$ are shown in Figure 12. In this figure it is shown that $Nu$ and $Sh$ increase with Rayleigh number.

The predicted values of $Sh_a$ and $Nu_a$ over the ranges investigated for $N$ and $Ra$ at $Le=1$ are correlated and presented in Table 3. The validity ranges for these correlations are: $0.1 \leq Le \leq 50$ and $0.1 \leq Ra \leq 10^7$.
Figure 11. Effect of Rayleigh number on isotherms (and also isoconcentrations) and streamlines, for \( Le = 1 \) and \( N = 1 \).

Figure 12. Effect of Rayleigh number on average Nusselt and Sherwood numbers on hot wall at \( Le = 1 \) and various \( N \).

Table 3. Correlations of \( Sh \) and \( Nu \) in terms of \( Ra \) and \( N \).

<table>
<thead>
<tr>
<th>Range of ( N )</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( N &gt; 0 )</td>
<td>( Nu = Sh = 0.91526Ra^{0.2958} \lambda^{0.1557} )</td>
</tr>
<tr>
<td>( N \leq -1 )</td>
<td>( Nu = Sh = 0.66309Ra^{0.2002} \lambda^{0.4114} )</td>
</tr>
</tbody>
</table>

\(-10 \leq N \leq 10\). Figure 13 shows the comparison between numerical results and the correlations in Table 3.

5. Conclusion

In this article, simultaneous natural convection heat and mass transfer around a hot body in a rectangular air-filled cavity was studied numerically. According to the present numerical results, the following conclusions are drawn:

1. By increasing the absolute value of Bucancy ratio, the average Nusselt and Sherwood numbers increase.

2. By increasing Lewis number, the average Sherwood number increases but the average Nusselt number decreases.

3. By increasing Rayleigh number, the average Nusselt and Sherwood numbers increase at \( Le=1 \) and \( N = 1 \).

4. The predicted values of average Sherwood and Nusselt numbers in the range investigated for buoyancy ratio, Rayleigh and Lewis numbers are correlated.

Nomenclature

- \( C \) Concentration
- \( C \) Dimensionless concentration
CCW \hspace{1cm} \text{Counterclockwise} \\
CW \hspace{1cm} \text{Clockwise} \\
D \hspace{1cm} \text{Mass diffusivity} \\
H \hspace{1cm} \text{Enclosure height} \\
L \hspace{1cm} \text{Enclosure length} \\
Le \hspace{1cm} \text{Lewis number (Sc/Pr)} \\
N \hspace{1cm} \text{Buoyancy ratio} \\
Nu \hspace{1cm} \text{Nusselt number} \\
p \hspace{1cm} \text{Pressure} \\
P \hspace{1cm} \text{Dimensionless pressure} \\
Pr \hspace{1cm} \text{Prandtl number} \\
Ra_{M} \hspace{1cm} \text{Mass transfer Rayleigh number} \\
Ra_{T} \hspace{1cm} \text{Heat transfer Rayleigh number} \\
Sc \hspace{1cm} \text{Schmidt number} \\
Sh \hspace{1cm} \text{Sherwood number} \\
T \hspace{1cm} \text{Temperature} \\
u, v \hspace{1cm} \text{Components of velocity} \\
U, V \hspace{1cm} \text{Dimensionless velocity components} \\
W \hspace{1cm} \text{Body length} \\
x, y \hspace{1cm} \text{Cartesian coordinates} \\
X, Y \hspace{1cm} \text{Dimensionless Cartesian coordinates} \\

\text{Greek symbols} \\
\alpha \hspace{1cm} \text{Thermal diffusivity} \\
\Psi \hspace{1cm} \text{Streamfunction} \\
\theta \hspace{1cm} \text{Dimensionless temperature} \\
v \hspace{1cm} \text{Kinematic viscosity} \\
\beta_{T} \hspace{1cm} \text{Volumetric coefficient of thermal expansion} \\
\beta_{M} \hspace{1cm} \text{Volumetric coefficient of expansion with concentration} \\

\text{Subscript} \\
a \hspace{1cm} \text{Average} \\
c \hspace{1cm} \text{Cold wall} \\
h \hspace{1cm} \text{Hot wall} \\
T \hspace{1cm} \text{Thermal} \\
M \hspace{1cm} \text{Mass} \\

\text{References} \\


**Biographies**

**Ghanbar Ali Sheikhzadeh** is an Associate Professor of Mechanical Engineering Department at University of Kashan, Kashan, Iran. He received his PhD in Mechanical Engineering from Shahid Bahonar University of Kerman. His research works concern numerical analysis and application of heat transfer in nano-systems and other areas of thermal and fluid sciences. Dr. Sheikhzadeh has published many papers in journals and conferences in his research fields.

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