Simulation of unsteady heat and mass transport with heatline and massline in a partially heated open cavity

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1. Introduction

Air-conditioning systems are very important on thermal comfort and life quality. Energy consumption is very high in these kinds of systems. Thus, the efficiency of the designed systems is a valuable subject for their sustainability and cost. In those systems, heat and mass transfer occurs in general simultaneously. Besides, the analysis of the flow distribution associated with the heat and mass transfer is very important for the design and efficiency of such AC systems.

The flow, heat and mass transfer in cavities with inlet and outlet ports are analyzed for these kinds of problems in various previous works. Liu et al. [1] studied the simultaneous transport of heat and moisture in a partially open enclosure with a thick wall. They used heatlines and masslines visualization techniques to simulate heat and moisture transport. They observed that the heat transfer potential, mass transfer potential, and volume flow rate can be promoted or inhibited. The effective parameters are wall materials and size as well as thermal and moisture Rayleigh number.
Transient laminar forced convection heat transfer leading to periodic state within a square cavity with inlet and outlet ports due to an oscillating velocity at the inlet port is presented by Saeidi and Khodadadi [2]. They indicated that the mean Nusselt numbers on the four walls clearly exhibit large amplitudes of oscillation and periodicity for St = 0.1 and increasing of St number, the amplitudes of oscillation on various walls are degraded. In another work, they presented forced convection results by investigating location of inlet and outlet ports [3]. Rahman et al. [4] studied the effects of heat generation and Reynolds and Prandtl numbers are studied for the same geometry [5]. Liu et al. [6] modeled numerically the indoor air quality with a new window-type air conditioner. They observed that the reduction of indoor pollutant levels can be accomplished either by increasing the fresh air ratio, or by decreasing the strength of indoor heating source. Oztop [7] worked on a mixed convection heat transfer in an enclosure with inlet and outlet ports and observed in particular that the location of the outlet port affects significantly the heat transfer and fluid flow. Besides, the inclination effects is an important parameter for natural

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
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<tbody>
<tr>
<td>Br</td>
<td>buoyancy ratio</td>
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<tr>
<td>c</td>
<td>dimensional concentration of species</td>
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<td>C</td>
<td>dimensionless species concentration</td>
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<tr>
<td>D</td>
<td>species diffusivity</td>
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<tr>
<td>g</td>
<td>gravitational acceleration</td>
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<td>Gr</td>
<td>Grashof numbers</td>
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<tr>
<td>H</td>
<td>height of the cavity</td>
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<tr>
<td>h</td>
<td>sizes of inlet and outlet</td>
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<tr>
<td>L</td>
<td>length of the cavity</td>
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<tr>
<td>Le</td>
<td>Lewis number</td>
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<tr>
<td>Ls</td>
<td>length of the heat and mass sources</td>
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<tr>
<td>Nu</td>
<td>Nusselt number</td>
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<tr>
<td>n</td>
<td>unit normal to the surface</td>
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<tr>
<td>p</td>
<td>dimensional pressure</td>
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<tr>
<td>P</td>
<td>dimensionless pressure</td>
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<tr>
<td>Pr</td>
<td>Prandtl number</td>
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<td>Re</td>
<td>Reynolds number</td>
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<tr>
<td>Sh</td>
<td>Sherwood number</td>
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<tr>
<td>T</td>
<td>dimensional temperature</td>
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<tr>
<td>t</td>
<td>time</td>
</tr>
<tr>
<td>u, v</td>
<td>dimensional velocity components</td>
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<tr>
<td>V</td>
<td>((U, V)) dimensionless velocity components</td>
</tr>
<tr>
<td>x, y</td>
<td>dimensional Cartesian coordinates</td>
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<tr>
<td>X, Y</td>
<td>dimensionless Cartesian coordinates</td>
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Greek symbols

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<tr>
<td>(\alpha)</td>
<td>thermal diffusivity</td>
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<tr>
<td>(\tau)</td>
<td>dimensionless time</td>
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<tr>
<td>(\beta_T)</td>
<td>thermal expansion coefficient</td>
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<tr>
<td>(\beta_c)</td>
<td>compositional expansion coefficient</td>
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<tr>
<td>(\nu)</td>
<td>kinematic viscosity</td>
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<tr>
<td>(\theta)</td>
<td>dimensionless temperature</td>
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<tr>
<td>(\rho)</td>
<td>mixture density</td>
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<tr>
<td>(\psi)</td>
<td>streamfunction</td>
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<tr>
<td>(\Gamma)</td>
<td>general dependent variable</td>
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<tr>
<td>(\nabla^2)</td>
<td>Laplacian operator</td>
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<tr>
<td>(\zeta)</td>
<td>heatfunction</td>
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<tr>
<td>(\xi)</td>
<td>massfunction</td>
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Subscripts

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<tr>
<td>(\overline{a})</td>
<td>average</td>
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<tr>
<td>c</td>
<td>referring concentration</td>
</tr>
<tr>
<td>h</td>
<td>higher value</td>
</tr>
<tr>
<td>l</td>
<td>lower value</td>
</tr>
<tr>
<td>p</td>
<td>referring pressure</td>
</tr>
<tr>
<td>T</td>
<td>referring temperature</td>
</tr>
<tr>
<td>i</td>
<td>inlet</td>
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Transmit transient laminar forced convection heat transfer leading to periodic state within a square cavity with inlet and outlet ports due to an oscillating velocity at the inlet port is presented by Saeidi and Khodadadi [2]. They indicated that the mean Nusselt numbers on the four walls clearly exhibit large amplitudes of oscillation and periodicity for St = 0.1 and increasing of St number, the amplitudes of oscillation on various walls are degraded. In another work, they presented forced convection results by investigating location of inlet and outlet ports [3]. Rahman et al. [4] studied the effects of heat generation and Reynolds and Prandtl numbers are studied for the same geometry [5]. Liu et al. [6] modeled numerically the indoor air quality with a new window-type air conditioner. They observed that the reduction of indoor pollutant levels can be accomplished either by increasing the fresh air ratio, or by decreasing the strength of indoor heating source. Oztop [7] worked on a mixed convection heat transfer in an enclosure with inlet and outlet ports and observed in particular that the location of the outlet port affects significantly the heat transfer and fluid flow. Besides, the inclination effects is an important parameter for natural
convection heat transfer in partially or fully open inclined cavities as indicated by Bilgen and Oztop [8]. Sourtiji et al. [9] observed that the oscillations of incoming flow in a cavity with inlet and outlet ports may represent a good way to improve the heat transfer.

Heatlines and masslines visualization techniques have been used in several studies to analyze the heat and mass transport characteristics. In this context, the study of Kimura and Bejan [10] may be the pioneer of this subject. Mobedi and Oztop [11] visualized the heat transport in a conjugate natural convection problem. They observed that using of heatline technique is the appropriate way to understand the heat transfer behavior in both solid and fluid. Heatline based heat flow visualization is analyzed by Basak et al. [12] in a square cavity. Zhao et al. [13] presented the heatlines, streamlines and masslines visualization in a square object inserted cavity. They indicated that the

Fig. 1. Physical configuration for the problem with boundary conditions (a) case 1, (b) case 2, (c) case 3.
streamlines, heatlines and masslines provide a more practical way to visualize the results than the customary ener-
genetical systems.

Numerical visualization of heat and mass transport for convective heat transfer by streamlines and heatlines are studied by Deng and Tang [14]. Then, they extended this work to mixed convection heat transfer [15]. They conducted an optimization analysis on the ventilation system for different outlet location to simulate an air-conditioning system. Celik and Mobedi [16] performed visualization of heat flow in a vertical channel with fully developed mixed convection.

The finite element method is a powerful technique that has been used in many studies to solve and visualize flow with heat and mass transfer for regular and curvilinear geometries as [17–20]. It was used also to solve double-diffusive natural convection problems [21–23].

Stavrakakis et al. [26] developed a new model to optimize window design for thermal comfort in naturally ventilated buildings. They made a single-room, rural-type prototype to make tests. They used both CFD and Artificial Neural Networks (ANN). Finally, they obtained optimal window designs, which correspond to the best objective variables for both single and several activity levels. In their similar work [27] they made an optimization study for window-openings design. Other studies with ventilated buildings can be found in [28–30].

The main objective of the present work is to examine transient behavior of the heat and mass transfer due to mixed convection in a ventilated cavity with partially heated and humidified. Three different cases are tested according to location of outlet port while inlet port is fixed. Results will be presented for different Grashof numbers and dimensionless time. Also, as a novelty result time dependent heat and mass transport patterns will be discussed using heatline and massline techniques.

2. Considered problem

The study consists of the analysis of the two dimensional transient behavior of laminar mixed convective flow with heat and mass transfer in open cavities. Three different configurations are studied (Fig. 1). For all configurations, constant temperature and constant concentration are partially applied on the bottom left of the vertical wall of the cavity. All other boundaries are assumed adiabatic and impermeable (zero concentration gradient). The inlet port is located on the right bottom wall. In case 1, outlet hole is located on left wall near the ceiling, in case 2, outlet port is located on the right side of the ceiling. The outlet port is located on the middle of the ceiling in case 3.

3. Governing equations

The solution domain of the present study is given in Fig. 1(a)–(c). The governing equations are those expressing the conservation of mass, momentum, energy and concentration transports in the enclosure. The fluid and transported pollutant are assumed to be perfectly mixed inside the cavity. To simplify analysis, the following assumptions are made for this mixture

![Fig. 2. Grid distribution of the considered geometry.](image-url)
inside the cavity: (i) the double-diffusive mixed convection is two-dimensional and unsteady; (ii) the mixture is Newton–Fourier fluid, which flows in laminar regime and does not experience any phase change; (iii) the mixture is incompressible but expands or contracts under temperature and/or concentration changes. Also, all thermo physical properties of the mixture are taken to be uniform overall the cavity, except for the density variation in the buoyancy term in Boussinesq approximation. The mixture density in the buoyancy term can thus be obtained \[ q = q_L \left( \beta_T \frac{T - T_L}{T_L} + \beta_c \frac{c - c_L}{c_L} \right) \]

where \( q_L, T_L, c_L \) are the reference density, temperature and concentration. According to Thermodynamics \( \beta_T = -\frac{1}{\rho_L \left( \frac{\partial \rho}{\partial T} \right)_{P, c}} \) and \( \beta_c = -\frac{1}{\rho_L \left( \frac{\partial \rho}{\partial c} \right)_{P, T}} \) are the volumetric thermal and concentration expansion coefficients respectively. In energy conservation

Fig. 3. Comparison of the (a) present model with (b) the results of Deng et al. [26].
analysis, it is assumed that thermal levels are small and similar enough so that the thermal radiation heat transfer between
the heat source and the incoming mixture is negligible. The energy term due to viscous dissipation and change of temper-
atture due to work of pressure forces are not considered. Besides, transfer of energy by inter-diffusion of species as well as
Soret and Duffour effects are not considered here [25]. Taking into account the above mentioned assumptions the
non-dimensional forms of the governing conservation equations are as follows:

\[ \nabla V = 0 \]  
\[ \frac{\partial V}{\partial \tau} + (V \cdot \nabla)V = -\nabla P + \frac{1}{Re} \nabla^2 V + \frac{GrT}{Re^2} \left( \frac{\partial \theta}{\partial X} + Br \frac{\partial C}{\partial X} \right) \]  
\[ \frac{\partial \theta}{\partial \tau} + V \cdot \nabla \theta = \frac{1}{Re Pr} \nabla^2 \theta \]

Fig. 4. Effect of dimensionless time \( \tau \) on (a) streamlines, (b) isotherms, (c) iso-concentration, (d) heatline and (e) massline for the case 1 at \( Gr = 10^5 \).
\[ \frac{\partial C}{\partial \tau} + \nabla \cdot \mathbf{V} = \frac{1}{\text{LeRePr}} \nabla^2 C \]  

(4)

where \( \mathbf{V} = (U, V) \) is the dimensionless velocity vector, \( P \) the dimensionless acting pressure, \( \text{Re} = \frac{u_l}{v} \) is the Reynolds number, \( \text{Br} = \frac{\text{Gr}^k}{\text{Gr}^l} \) is the buoyancy ratio, \( \text{Le} = \frac{a}{D} \) is the Lewis number, and \( \text{Pr} = \frac{\alpha}{\nu} \) is the Prandtl number.

Eqs. (1)–(4) were cast in dimensionless form by using the following:

\[
\tau = \frac{\tau}{L^2}, \quad (X, Y) = \left( \frac{x}{L}, \frac{y}{L} \right), \quad \mathbf{V} = (U, V) = \left( \frac{u}{u_l}, \frac{v}{u_l} \right), \quad P = \frac{(p + \rho g y) L^2}{\rho v^2}, \quad \theta = \frac{T - T_L}{T_h - T_L} \quad \text{and} \quad C = \frac{c - c_L}{c_h - c_L} \]  

(5)

The corresponding initial and boundary conditions for the above problem are given by:

\begin{align*}
\tau &= 0.01 \\
\tau &= 0.1 \\
\tau &= 1
\end{align*}

Fig. 5. Effect of dimensionless time \( \tau \) on (a) streamlines, (b) isotherms, (c) iso-concentration, (d) heatline and (e) massline for the case 1 at \( \text{Gr} = 10^6 \).
For $\tau = 0$

Entire domain: $U = V = 0, \ 0 = C = 0$

For $\tau > 0$

At inlet: $U = -1, \ V = 0, \ 0 = 0, \ C = 0$

At the outlet: $0 = 0, V = 0, \ 0 = 0, \ 0 = 0$

At walls: $U = V = 0, \ 0 = 0 = C = 1, \ heat and contaminant sources$

$U = V = 0, \ 0 = 0 = C = 0, \ elsewhere$

The heat and mass transfer calculations within the enclosure are measured in terms of the average Nusselt and Sherwood numbers at the heat and contaminant sources. The latters are defined as follows:

$$\text{Nu}_{av} = -\frac{1}{L} \int_0^{L/L} \frac{\partial \theta}{\partial X} \, dY \quad \text{and} \quad \text{Sh}_{av} = -\frac{1}{L} \int_0^{L/L} \frac{\partial C}{\partial X} \, dY$$

(6)
To illustrate the influence of the governing dimensionless parameters on (i) the flow structure in the domain, the fluid motion is displayed using the dimensionless streamlines as a streamfunction ($\psi$) obtained from velocity components $U$ and $V$. (ii) heat and contaminant transport characteristic in the domain through the dimensionless heat-line and massline as a heatfunction ($\eta$) and massfunction ($\zeta$) respectively. The relation between streamfunction ($\psi$) and velocity components, heatfunction ($\eta$) and temperature, massfunction ($\zeta$) and concentration are respectively defined [26] as:

$$U = \frac{\partial \psi}{\partial Y}, \quad V = -\frac{\partial \psi}{\partial X}$$

$$-\frac{\partial \zeta}{\partial X} = V \theta = \frac{1}{RePr} \frac{\partial \theta}{\partial Y}, \quad \frac{\partial \zeta}{\partial Y} = U \theta = \frac{1}{RePr} \frac{\partial \theta}{\partial X}$$

Fig. 7. Effect of dimensionless time $\tau$ on (a) streamlines, (b) isotherms, (c) iso-concentration, (d) heatline and (e) massline for the case 2 at $Gr = 10^5$. 

To illustrate the influence of the governing dimensionless parameters on (i) the flow structure in the domain, the fluid motion is displayed using the dimensionless streamlines as a streamfunction ($\psi$) obtained from velocity components $U$ and $V$. (ii) heat and contaminant transport characteristic in the domain through the dimensionless heat-line and massline as a heatfunction ($\eta$) and massfunction ($\zeta$) respectively. The relation between streamfunction ($\psi$) and velocity components, heatfunction ($\eta$) and temperature, massfunction ($\zeta$) and concentration are respectively defined [26] as:

$$U = \frac{\partial \psi}{\partial Y}, \quad V = -\frac{\partial \psi}{\partial X}$$

$$-\frac{\partial \zeta}{\partial X} = V \theta = \frac{1}{RePr} \frac{\partial \theta}{\partial Y}, \quad \frac{\partial \zeta}{\partial Y} = U \theta = \frac{1}{RePr} \frac{\partial \theta}{\partial X}$$
3.1. Grid distribution and validation

The solution of the above set of governing equations with the appropriate initial and boundary conditions had been obtained using Galerkin weighted residual method of finite element formulation. The description of the numerical model can be found in our previous work [20–22]. In this method, the solution domain is discretized into non-uniform triangular elements. Then the non linear governing equations are transferred into a system of integral equations. Gauss quadrature method carries out the integration involved in each term of these equations. The nonlinear algebraic equations thus obtained

\[
- \frac{\partial \zeta}{\partial X} = VC - \frac{1}{\text{LeRePr}} \frac{\partial C}{\partial Y}, \quad \frac{\partial \zeta}{\partial Y} = UC - \frac{1}{\text{LeRePr}} \frac{\partial C}{\partial X}
\] (9)
are modified by imposition of initial and boundary conditions. These modified nonlinear equations are transferred into linear algebraic equations by using Newton’s method. Finally, these linear equations are solved by using triangular factorization method.

Grid distribution and validation of the code are given in Figs. 2 and 3, respectively. Triangular cells are used to obtain grid distribution. Totally, 3872 nodes and 2025 elements grid dimension is used. Obtained results are compared with the results from the literature of Deng et al. [26]. It is found that results of present code shows good agreement with the literature as shown in Fig. 3.

4. Results and discussion

A computational work is performed to investigate the unsteady heat and mass transfer in a cavity with different outlet ports. The cavity has a partial heater and humidifier in the left vertical wall. Heatline and massline are obtained to understand the direction of heat and mass transport inside the cavity. Three configurations are considered as explained in Fig 1. Thus, obtained results will be discussed under three subtitles according to studied cases.
4.1. Case I

For all cases, inlet position of the flow is the same and the flow inlets to the cavity from the bottom of the right vertical wall. In case I, the flow departs from the cavity, at the top region of the left vertical wall. Fig. 4 presents the effect of dimensionless time on streamlines, isotherms, iso-concentration, heatline and massline for the case 1 and Gr = 10^5. Streamline contours show that most of the flow moves diagonally from the inlet to outlet and remain flow goes from the right vertical wall to outlet by following the boundaries of the cavity at τ = 0.01. However, the flow distribution is affected by the left vertical source and a circulation near the heat and mass source starts. For τ = 0.1, this circulation cell is observed and it turns in clockwise direction near the top of partial source. Then, a larger circulation cell is formed at the middle of the cavity for τ = 1. In this case, entering fluid to the cavity follows the bottom and left vertical wall and goes to the outside from the outlet port. As given in boundary conditions that the temperature of heat source is higher than that of inlet fluid and remaining walls are adiabatic. Thus, temperature is distributed within the cavity starting from the source at τ = 0.01. Then, heated flow moves to the inside of the cavity

Fig. 10. Effect of dimensionless time τ on (a) streamlines, (b) isotherms, (c) iso-concentration, (d) heatline and (e) massline for the case 3 at Gr = 10^5.
with $\tau = 0.1$. For the steady regime, almost half of the fluid is heated inside the cavity and thermal boundary layer becomes larger. Heated fluid moves to the outlet port. As given in Fig. 4(c), iso-concentration exhibits almost similar variation with isotherms and plumelike distribution starting from left vertical wall to ceiling. Fig. 4(e) answers the question of how mass goes inside the cavity. Similarly, Fig. 4(d) shows the heat transport mechanism. Heatline distribution for $\tau = 0.1$ indicates that natural convection heat transfer becomes dominant. Streamlines and temperature and mass contours distributions with time are depicted in Fig. 5 for case 1 and for $Gr = 10^6$. At the beginning, $\tau = 0.01$, the fluid is confined near the heat and mass source. Then, a circulation induced by natural convection is formed near the source. It hits the flowing fluid to the half of the enclosure. Due to stronger effect of natural convection multiple circulation cells inside the fluid and flow strength becomes very low from inlet port. Temperature distribution develops starting from the source and temperature of the half bottom side of the cavity becomes same. On the contrary of isotherms, iso-concentration contours are cumulated near the left vertical wall as seen from Fig. 5(c). Heat and mass transport starts from the left bottom wall and they show plumelike distribution starting from left top corner inside to the cavity as given in Fig. 5(d) and (e).

![Fig. 11. Effect of dimensionless time $\tau$ on (a) streamlines, (b) isotherms, (c) iso-concentration, (d) heatline and (e) massline for the case 3 at $Gr = 10^6$.](image)
Fig. 6 shows the streamlines, isotherms, iso-concentration, heatlines and masslines for case 1 at Gr = 10^7. As seen from the streamlines, the flow goes almost diagonally from inlet to outlet except interruption (at $\tau = 0.01$) of flow near the partial heater due to occurrence of buoyant flow. This flow turns to a circulation near the partial heater for $\tau = 0.1$ and the main flow is pressed to right top corner. For $\tau = 1$, five circulation cells are obtained due to inlet and outlet ports. The fluid is captured flow inside the cavity. Although higher Grashof number, temperature distribution is stronger for $\tau = 1.0$. Bottom half of the cavity has constant temperature and thermal boundary layer becomes stronger around the heater and left wall (as given in Fig. 6(b)). Due to very high value of Grashof number heatline and massline contours show messy distribution starting from the beginning of time as given in Fig. 6(d) and (e).

### 4.2. Case 2

In this case, outlet port is located near the right of the ceiling. Results for this case and Gr = 10^5 are given in Fig. 7(a)–(e). The flow enters from the inlet hole at the bottom and it sweeps almost whole cavity at the beginning. For $\tau = 0.1$, a
small circulating cell is formed near heater and humidifier due to starting of natural convection and the main flow goes from bottom to top in a crescent shape. The flow behind the circulating cell is decreased the flow due to impinging phenomenon associated with the combined effect of natural convection and forced convection. The main cell moves to the middle of the cavity. Isotherms are distributed as parallel to the heated wall section and they show plumelike distribution above the bottom half. Iso-concentration contours show mushroom shape near the ceiling due to location of outlet port near the top corner. For lower Grashof numbers, thermal conduction and solutal diffusion becomes dominant. For \( \text{Gr} = 10^6 \), results are shown in Fig. 8(a)–(e). In this case a rain-drop shaped circulation cell is occurred at \( \tau = 0.01 \) and it starts to develop to the right top corner and some of the inlet flow goes from behind this circulation cell to outlet port. And, some of the flow goes directly to the outlet port. For \( \tau = 1 \), multiple cells are formed due to weaken of inlet flow. Isotherms are distributed from the left and it move to the inlet port with increasing non-dimensional time. Iso-concentration contour show similar distribution but it mostly sits near the top and outlet port. Heatlines and masslines in Fig. 8(d) and (e) indicate that heat and mass are distributed from the partial heater to ceiling and they spread to inside the cavity due to impinging flow and circulation inside to the cavity. Heatline results are supported by [26]. For the highest value of Grashof number, namely \( \text{Gr} = 10^7 \), all results are presented in Fig. 9(a)–(e). Flow strength becomes stronger with increasing of Grashof number and a flow occupies all volume and a circulating cell is observed near the outlet hole. Temperature of the fluid inside the cavity increases as seen from Fig. 9(b). Similar situation can be seen from
iso-concentration contours (Fig. 9(c)). The heatline and massline contours are cumulated above the half upper part of the enclosure near the outlet port. It shows again messy distribution due to strong heat and mass transport and high flow velocity as illustrated in Fig. 9(d) and (e).

4.3. Case 3

Outlet port is located onto the middle of the ceiling in this case. Fig. 10(a) illustrates the contours to simulate the thermal system. At the beginning of time, heat and mass transfer due to partial heater and humidifier is very weak and it becomes stronger with increasing time. If this case compares with case 1 and case 2 with same value of Grashof number, it is seen that the location of outlet port is effective on the shape of main circulation cell. The fluid with warmer temperature tries to move up but the flow goes from outlet port restricted on this flow movement. Due to this flow movement temperature distribution and concentration tend be same distribution and they move to the outlet port with flowing fluid. Both iso-concentration and massline show mushroom shaped distribution as seen from Fig. 10.
As well known that the heatline and massline, namely heatfunction and massfunction, both are composed of convection and diffusion terms (see Figs. 11 and 12).

4.4. General observations on heat and mass transfer

In this chapter, results for the validation of unsteady heat and mass transfer by presenting average Nusselt number and Sherwood number. Both heat transfer and mass transfer are affected from the location of the outlet port. In this context, Fig. 13 shows variation of average Nusselt and Sherwood number with time for different Grashof numbers. As seen from the figure, both of these values are decreased with time and they reach steady state depends on value of Grashof numbers. The flow, heat and mass transfer reach steady state after short time from the beginning for \( \text{Gr} = 10^7 \). The maximum average Nusselt and Sherwood number for case 1 are obtained around 11 and 35, respectively. Fig. 14 gives results for case 2. Trend of these values with time is almost same for three different cases. Case 3 presents the highest values from the heat and mass transfer point of view. Fig. 15 shows the results in similar way with Fig. 13 and 14 for case 3. Trend is almost same.

As a general comparisonal figure, Fig. 16 is presented to compare effects of outlet ports (cases) on heat and mass transfer at different dimensionless time and \( \text{Gr} = 10^7 \). These figures show that each case follow each other with increasing time and the highest value of heat and mass transfer is obtained for the case 1.
5. Conclusions

A numerical study has been performed by using finite element method to simulate the heat and mass transport in a square cavity with inlet and outlet ports and partial heater and humidifier. The important findings can be drawn from the results as

- Both heatline and massline are valuable techniques to analyze unsteady heat and mass transport problems and construct efficient design in thermal systems.
- Average Sherwood and Nusselt numbers are decreased with Grashof numbers and time for all considered cases. These non-dimensional numbers become almost constant with time for the highest value of Grashof number.
- When outlet port located near the right side of the ceiling, namely case 1, the highest heat and mass transfer are observed.
- Heat and mass transfers reach steady-state regime in short time with increasing Grashof number.

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