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# NUMERICAL INVESTIGATION OF HEAT AND MASS TRANSFER OF HUMID-AIR INSIDE AN OPEN CAVITY: PARAMETRIC STUDY

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## ABSTRACT

Numerical results of turbulent natural convection and mass transfer in an open enclosure for different aspect ratios (AR = 0.5, 1, and 2) with a humid-air are carried out. Mass fraction and local Nusselt number were proposed to investigate the heat and mass transfer. A heat flux boundary conditions were subjected to the lateral walls and the bottom one make as an adiabatic wall, while the top area was proposed as a free surface. Effect of Rayleigh numbers ($10^6 \leq Ra \leq 10^{13}$) on natural convection and mass flow behavior are analyzed. The governing equations are solved using CFD Fluent code based on the SIMPLE algorithm. The results revealed a significant enhancement in the mass and heat transfer rates near the vertical walls. The results showed that the cavity with an aspect ratio of AR = 2 has a significant enhancement to raise the rates of both heat and mass transfer. When the Rayleigh number increases, maximum heat transfer rates were observed due to the fluid flow becomes more vigorous. However, mass transfer improves as the Rayleigh number decreases.

**Keywords**: *Open enclosure, Natural convection, CFD, Mass fraction, Turbulent flow.*

## 1. INTRODUCTION

Enhancement of thermal turbulent flow occurs in multiple applications of different mechanisms and systems such as (solar energy collectors, evaporation, drying, distillation, cooling processes of electronic compounds (Keshtkar *et al.* (2020); Alvarado-Juárez *et al.* (2015) and Saleem *et al.* (2020)).

Mass and heat transfer induced by temperature and concentration gradients are generally applied in physical processes that involve convective and diffusive transport of chemical species within physical technology. Thereby, these heat and concentration gradients are considered to be assisting or opposing to enhancing the performance of energy system, depending on the boundary conditions of the problem. To understand the behaviour of the heat transfer flow, many researchers have tried to analyze solutions within closed or open cavities extensively in the past. They used heat flux or temperature gradient field to the walls of the cavity as boundary conditions. They have done several experiments and numericals works (Ampofo and Karayiannis (2003); Altaç and Ugurlubilek (2016) and Olanrewaju *et al.* (2021)) for more structure configurations as rectangular and square cavities, and non-structure enclosures as: sinusoidal protuberances walls and inclined.

Scientist's aims to achieve the cooling and/or the heat transfer performances depending on turbulent. Many of them relied on explaining the results by methods of flow visualization inside the cavity by using CFD software to explain more the flow pattern. They have studied the effect of Rayleigh number (Vasiliev *et al.* (2016); Choudhary *et al*. (2019) and Zainab *et al.* (2022)) and Reynolds number on the behaviour of free, forced and mixed convection. Further conditions for mass transfer and heat fluid motion, different studies have been conducted for square cavities and rectangular shapes in a closed system. Béghein *et al*. (1992) have studied the heat and mass transfer induced by natural convection in a square cavity in a steady state. The authors have proposed a two-dimensional cavity with different heated walls. They have investigated the effect of the solute momentum on the rates of heat and mass transfer, as well as, the effect of the Lewis number on fluid motion. They have suggested correlations between the properties of dimensional quantities and rates of thermal and mass transfer.

To analyze the influence of horizontal walls on heat and mass transfer, Costa (1997) has investigated the natural double diffusion convection in a square cavity with horizontal walls. He characterized the mass transfer parameters through the local and global Nusselt and Sherwood numbers over all the walls of the geometry. Sezai and Mohamad (2000) have simulated a three-dimensional flow in a cubic cavity filled with a binary fluid subjected to opposing concentration and temperature gradient imposed along the vertical walls, they are considered the walls as adiabatic and impermeable. They have compared the 3D to 2D and they found a satisfactory agreement for the low values of Rayleigh number.

Many researchers chose the rectangular configuration area (Lee and Hyun (1990), (1991)) to study the heat and mass transfer in natural convection. Jang *et al.* (2003) have examined a numerical study of the effects of irregular vertical surfaces on flow characteristics of natural convection and of mass transfer for ondulatory surfaces. Their results involving the analysis of velocity, concentration, temperature and Nusselt number along the wavy shape. Laguerre *et al.* (2009) have studied numerically and experimentally on heat moisture transfer and air flow in a rectangular cavity. They found that significant influence of radiation on both cold and hot walls and the air velocity increases with contribution moisture at the bottom of cavity. Sun *et al.* (2011) have discussed the effect of humidity on the natural heat flow and their characteristics between vertical walls in a differentially heated cavity.

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For air-$CO_2$ mixture, Nikbakhti *et al.* (2012) numerically study have conducted the mass transfer for the working fluid contained in a rectangular enclosure with heated active sides. Serrano-Arellano *et al.* (2013) have presented a CFD study of heat and mass transfer by natural convection in a 2D area with an air-$CO_2$ mixture. To analyze the heat performances, the authors investigated the effect of the buoyancy ratio on mass and heat transfer for different regimes. The thermal air quality behaviour of a higher concentration of $CO_2$ in the vertical walls was analyzed under the assisted and the opposite flow conditions. For different values of the Rayleigh number were considered. They observed that the temperature gradient inside the cavity for turbulent flow was more homogeneous for all cases of flow. Xamán *et al.* (2011) have analyzed the heat and mass transfer inside a ventilated cavity in laminar flow. To study the thermal behaviour and the air quality inside a cavity, a comparative work was analyzed concerning the location of the exit space of the fluid mixture, considering three different values for the source of $CO_2$ contaminant. The air inlet space was located below the right vertical side of the cavity. The location of the mixture outlet space was considered in four different positions.

The air-humid transfer for natural convection in a 2D rectangular surface with a solid section is a study investigated by Hu *et al.* (2017). They have dealt with a set of different variables: Rayleigh thermal number, thickness of the partition, inclination, and buoyancy ratio.

A numerical study for the impact of a heated plate on a double-diffusive free convection in a square closed cavity was carried out by Muthtamilselvan *et al.* (2017). The vertical walls are maintained at low temperatures while the horizontal walls are adiabatic. They found that the vortex viscosity values reduced both heat and mass transfer rates. A new vision of a mass transfer behaviour of the moisture under the effect of buoyancy-driven for low turbulent flow was carried out by Iyi and Hasan (2019). They have used different values of mass fraction and temperature gradient varied from 20°C to 60°C. The authors concluded that the vapor content in the humid air has an important impact on the heat and current fields inside the enclosure and the variation of the vertical wall temperature gradients have also shown the effect of the moisture concentration inside the cavity. The percentage of the difference in the heat transfer changed significantly depending on the mass fraction of air moisture and the heat gradient between the vertical sides. Jiang *et al.* (2020) have examined the air-steam mixture flow of natural convection inside a square cavity with vertical wall temperature and concentration gradients. They proposed a Prandtl number equal to 1.32 and Rayleigh number $4.05\times10^5$, the hot wall temperature TH and cold wall temperature TC. The influence of species spread was investigated with Soret and Dufour mixture effect and it was found that for this binary mixture these two effects have a little contribution.

Parametric studies of the differentially heated enclosure to treat the natural convection flows that contained a mixture of air and carbon dioxide was done by Koufi *et al.* (2019), Their work conducted for a laminar system to analyze the effect of change in buoyancy rate on the mass transfer. The governing equations are solved by the Finite Volume Method based on the SIMPLE algorithm. They investigated the effect of buoyancy ratio on hydrodynamic and thermal transfer. Keshtkar *et al.* (2020) have proposed a new procedure for numerical work of solar system. They have considered a latent heat source of evaporation/ condensation in the energy equation. Their results proved that the properties are uniformly distributed. The relative humidity is almost 100% in the whole enclosure which implies that the air is saturated everywhere. However, the sudden change in the temperature or mass fraction is near the evaporating surface. Kushawaha *et al.* (2020) have investigated the free convection of thermo-solute with mass lines and thermal flow in a rectangular area. The vertical sides are isotherms while the others are adiabatic. The study was conducted for the following variables: Prandtl number ($Pr$) Rayleigh number ($Ra$), buoyancy ratio (N) and Lewis number (Le). They found that these parameters significantly affect the overall rates of heat and solute transfer. Saleem *et al.* (2020) have exposed a new solar trapezoidal cavity to enhance the distillation of an air-vapor mixture under the effect of double-diffusive natural convection filled. The bottom wall is fixed at hot temperature Th and high concentration Ch, the temperature of inclined wall that kept at Th and low concentration Cc. The physical properties of the mixture are obtained in Poling *et al.* (2001). They calculated the effect of governing parameters as Reynolds number, Buoyancy ratio (N) and Rayleigh number ($Ra$). The governing equations were solved by adopting the finite volume method. They obtained that the total mass and heat transfer rates enhance with expanding each of Reynolds and Rayleigh number. When Rayleigh number increases the boundary layer thickness of heat increases too, because the buoyancy forces are dominated. Despite the importance of the mass transfer flow mentioned above, there are only a few research efforts on the natural convection of open cavities. Terrell and Newell (2007) have an experimental study conducted to do a parametric combined thermal flow caused by mass transport and buoyancy in open enclosure for various aspect ratios. The results of retained condensate on mass movement areas due to water thermal stability were also examined with no remarkable impact.

For open square enclosures, Arbin and Hashim (2014) have investigated numerically in double- diffusion natural convection with constant concentrations and temperatures were imposed along the right and the left walls. They found that the top thermal flow is the most effective for heat transfer while the middle heating is most stable for mass transfer. Wang *et al*. (2020) have simulated the effect of buoyancy ratio, Rayleigh number, the Souret, the Dufour coefficients and Lewis number, on heat and mass transfer in an open cavity. The horizontal walls of the square cavity are adiabatic and the left wall is kept high temperature Th and high concentration Ch. The right side is free surface. They have concluded that changing Lewis number have a little influence on Sherwood and Nusselt number. But when Rayleigh number increases Lewis number increases gradually.

In this research paper, we investigate the natural convection inside an open cavity through humid-air passes for turbulent flow in order to characterize the impact of Rayleigh number and aspect ratios on the heat and mass transfer at heat flux subjected to the vertical walls.

## 2. MATHEMATICAL MODELING

### 2.1 Geometrical Description

The physical problem considered is shown schematically in Fig. 1. A humid-air occupied in two-dimensional rectangular enclosure with aspect ratios. The vertical parallel walls are subjected to uniform flux (q) and the bottom side is considered impermeable and adiabatic. The top domain defined as free surface. The dimension of open fluid at bottom left wall is equal to 20% from the height of the enclosure. Different cases of aspect ratios (AR=H/L=0.5, 1 and 2) were proposed.

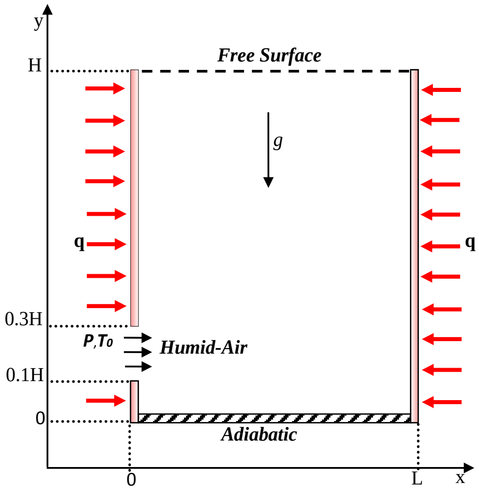


**Fig. 1** A schematic representation of an open cavity.

## 2.2 Governing Equations

The governing equations for heat and mass transfer inside this open cavity are the continuity, momentum, energy and concentration of humid-air liquid equation for natural convection (Sévéléder and Petit (1989) :

- Continuity equation :

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

- Momentum equation :

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial x}+\upsilon\left(\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial x^2}\right) \tag{2}$$

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial y}+\upsilon\left(\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial x^2}\right)+g\beta\left(T-T_0\right) \tag{3}$$

- Energy equation :

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

- Concentration equation :

$$\frac{\partial C}{\partial t}+u\frac{\partial C}{\partial x}+v\frac{\partial C}{\partial y}=D\left(\frac{\partial^2 C}{\partial x^2}+\frac{\partial^2 C}{\partial y^2}\right) \tag{5}$$

Where $\upsilon$ is the kinematic viscosity, $\beta$ is the thermal expansion coefficient, $P$ is the fluid pressure, $\alpha$ and $D$ are the thermal and mass diffusivity, respectively. All governing equations of heat and mass transfer in the proposed open cavity were solved in turbulent regime using a CFD code. A second-order upwind scheme was selected to solve the concentration and momentum equations. The computations were simulated to be converged at $10^{-7}$ of root mean square residual values.

## 2.3 Turbulence Model

Launder and Spalding (1972) proposed a standard simple model of turbulence called $\boldsymbol{k-\varepsilon}$, which based on the turbulent kinetic energy equations $\boldsymbol{k}$, and the rate of dissipation of turbulent kinetic energy $\boldsymbol{\varepsilon}$. It's used to analyze the flow and heat transfer calculations due to its robustness, and consistent precision.

The generic form of this model is :

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x}(\rho k u)=\frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]+G_k+G_b-\rho\varepsilon-Y_M+S_k \tag{6}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x}(\rho\varepsilon u)=\frac{\partial}{\partial y}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial y}\right]+C_{\varepsilon 1}.\frac{\varepsilon}{k}\left(G_k+C_{\varepsilon 3}G_b\right)-C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k}+S_\varepsilon \tag{7}$$

Where; $\mu_t$ is eddy viscosity, $\sigma_k$ and $\sigma_\varepsilon$ are the turbulent Prandtl numbers for $k$ and $\varepsilon$, respectively. $S_k$ and $S_\varepsilon$ are user-defined source terms. $G_k$ represents the kinetic energy production of turbulence due to velocity gradients, $G_b$ is the kinetic energy production of turbulence due to buoyancy, $Y_M$ represents the contribution of the expansion of fluctuations, $C_\mu$=0.09, $C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$=1.92, $\sigma_k$=1, $\sigma_\varepsilon$=1.3.

## 2.4 Boundary Conditions

The boundary conditions associated with the problem are:

- $x=0;\ 0\le y\le 0.1H$
  $$u=v=0, C=0, q=5w \tag{8}$$
- $x=0;\ 0.1H\le y\le 0.3H$
  $$u=v=0, C=0, q=0, T=T_0, P=P_0 \tag{9}$$
- $x=0;\ 0.3H\le y\le H$
  $$u=v=0, C=0, q=5w \tag{10}$$
- $x=L;\ 0\le y\le H$
  $$u=v=0, C=0, q=5w \tag{11}$$
- $y=0;\ 0\le x\le L$
  $$u=v=0, \frac{\partial T}{\partial y}=\frac{\partial C}{\partial y}=0 \tag{12}$$
- $y=H;\ 0\le x\le L$
  $$u=v=0, C=0, T=T_0, \Delta P=\rho gH \tag{13}$$

The following variables are given dimensionless:

$$T^*=(T-T_C)/(T_H-T_C),\quad X^*=x/L,\ Y^*=y/L \tag{14}$$

The Probability Density Function PDF (%) is a function that expresses the density of a variable parameter, giving its potential value at a given point out of the sum of the probabilities in a given medium or cavity.

## 2.5 Meshing Sensitivity Analysis

The interest region was covered with meshing grid that a further refinement in vertical sides. The different grid were studies executed; 140×140, 160×160, 180×180 and 200×200 for the cavity with aspect ratio is AR= 1. From Table 1, it's clear that the 180×180 elements produce satisfactory results, for a maximum error of 0.02% compared to other element number, for maximum values of velocity components u, v and their locations.

**Table 1** Mesh sensitivity test for maximum of x and y-velocities, and maximum temperature.

| | *140×140* | *160×160* | *180×180* | *200×200* |
|---|---|---|---|---|
| $u_{max}$ *(m/s)* | 0.100652<br>(0.03%) | 0.100668<br>(0.01%) | 0.100673<br>(**0.00%**) | 0.100678 |
| $v_{max}$ *(m/s)* | 0.087004<br>(0.05%) | 0.086937<br>(0.03%) | 0.0869814<br>(**0.02%**) | 0.086962 |
| $T_{max}$ *(K)* | 304.914<br>(0.005%) | 304.92<br>(0.003%) | 304.9246<br>(**0.001%**) | 304.929 |

## 3. RESULTS AND DISCUSSIONS

We investigate parametrically the effect of aspect ratios (AR=0.5, 1 and 2) and Rayleigh number ($10^6 \leq Ra \leq 10^{13}$) on the heat and mass transfer of humid-air flow for turbulent inside an open enclosure. The thermo-physical properties of the working fluid are proposed by Poling *et al.* (2001), see Table 2.

### 3.1 Mass Transfer Validation

In order to ensure the CFD efficiency, the analysis are compared for the case of heat and mass transfer in turbulent (Ra = $10^7$ with different buoyancy ratio) of closed square cavity, where the computational results, regarding a flow visualization for isotherms, streamlines, and iso-concentrations, are compared with those obtained by Koufi *et al.* (2019), as shown in Fig. 2.

**Table 2** Thermo-physical properties of the Humid-air, Poling *et al*. (2001).

| $\rho_0$ $[Kg.m^{-3}]$ | $Cp_0$ $[J.Kg^{-1}.K^{-1}]$ | $\lambda_0$ $[W.m^{-1}.s^{-1}]$ | $\mu_0$ $[Kg.m^{-1}.s^{-1}]$ | $D$ $[m^2.s^{-1}]$ |
|---|---|---|---|---|
| 1,15 | 1005 | $2{,}65.10^{-2}$ | $1{,}85.10^{-5}$ | $2{,}59.10^{-5}$ |

A second validation was investigated with the results existing in the literature Béghein *et al* (1992), Sezai and Mohamad (2000), and Xamán *et al*. (2011), concerning the Nusselt number as function of buoyancy ratio (N). The validations are satisfactory and exhibited good agreements as can be seen in Table 3

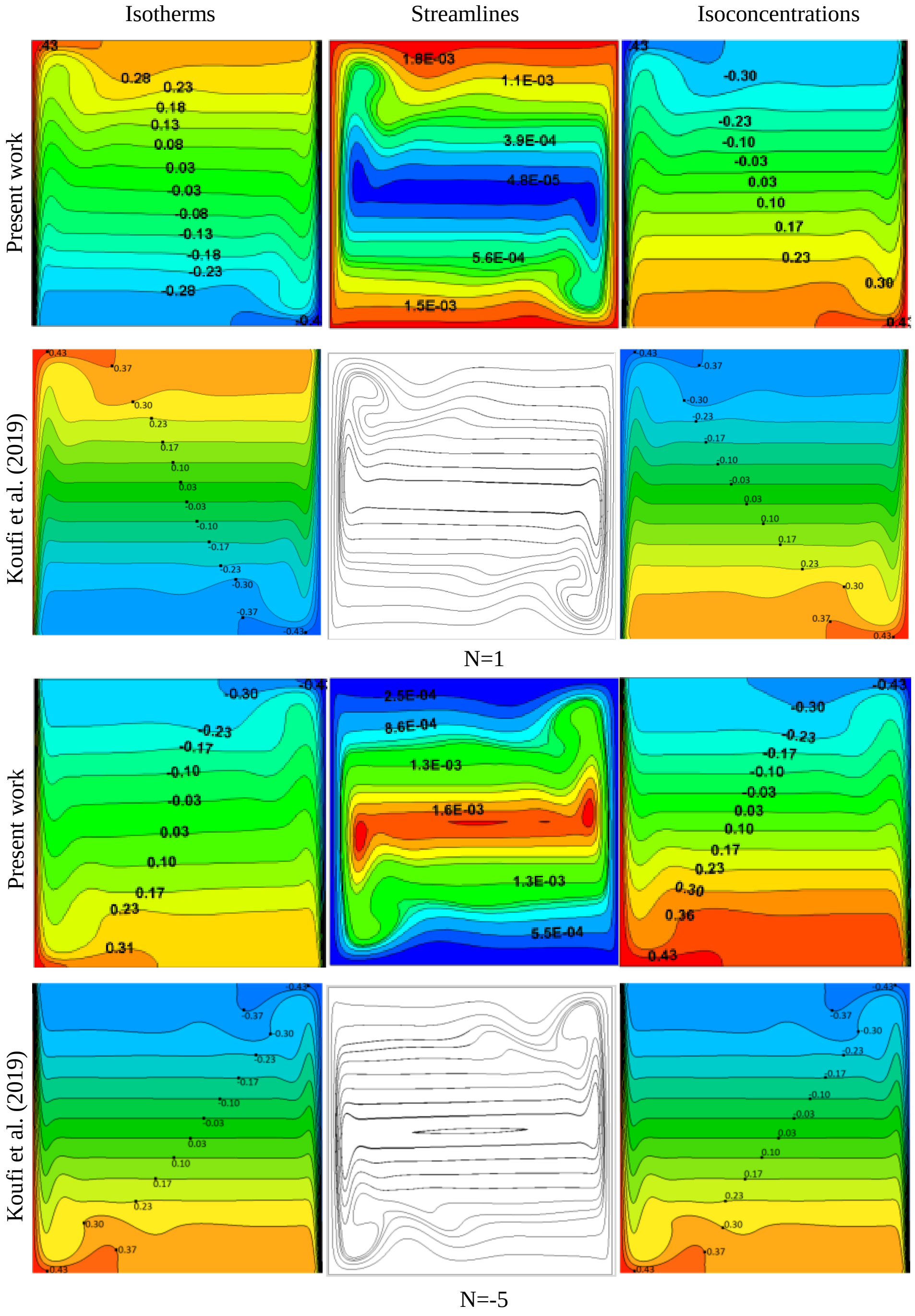


**Fig. 2** Qualitative validation; Comparaison of isotherm, streamline and iso-concentration contours for Ra = $10^7$ and N = 1 and -5.

**Table 3** Quantitative validation; Nusselt number as function of bouancy ratio (N).

| *Nu* | | | |
|---|---|---|---|
| *N* | -0.5 | -0.8 | -5.0 |
| Present work | 13.6 | 10.3 | 23.2 |
| Xamán *et al*. (2011) | 13.6 | 10.6 | 23.7 |
| Sezai and Mohamad (2000) | 13.5 | 10.5 | 23.5 |
| Béghein *et al.* (1992) | 13.6 | 10.6 | 23.7 |
| Err % For Xamán *et al.* (2011) | 0.7% | 1.9% | 1.2% |

## 3.2 Discussion

To analyze the hydrodynamic behavior and the heat transfer characteristics, an isotherms and streamlines visualisation illustrates here for the cases of AR = 0.5, 1, and 2 were illustrated in Figs. 3-5. The main flow tends to cross the cavity by a semi vertical direction, that make a strong vortex at the bottom right wall of each configuration case. For high Rayleigh numbers, a small change in the thermal field, wherever the thermal stratification in the cavities are absent and the intensity of the vortex becomes vigorous. This means that the mass transfer is accelerated. For AR = 0.5, the small cell found at the middle free surface is lost by the effect of enhanced buoyancy forces.

Another clockwise small rotating vortex was created at the left bottom wall, where its size becomes increasingly powerful, while the high temperatures are localized in close spaces near the humid left wall, which corresponds to the concentration of the thermal boundary layer.

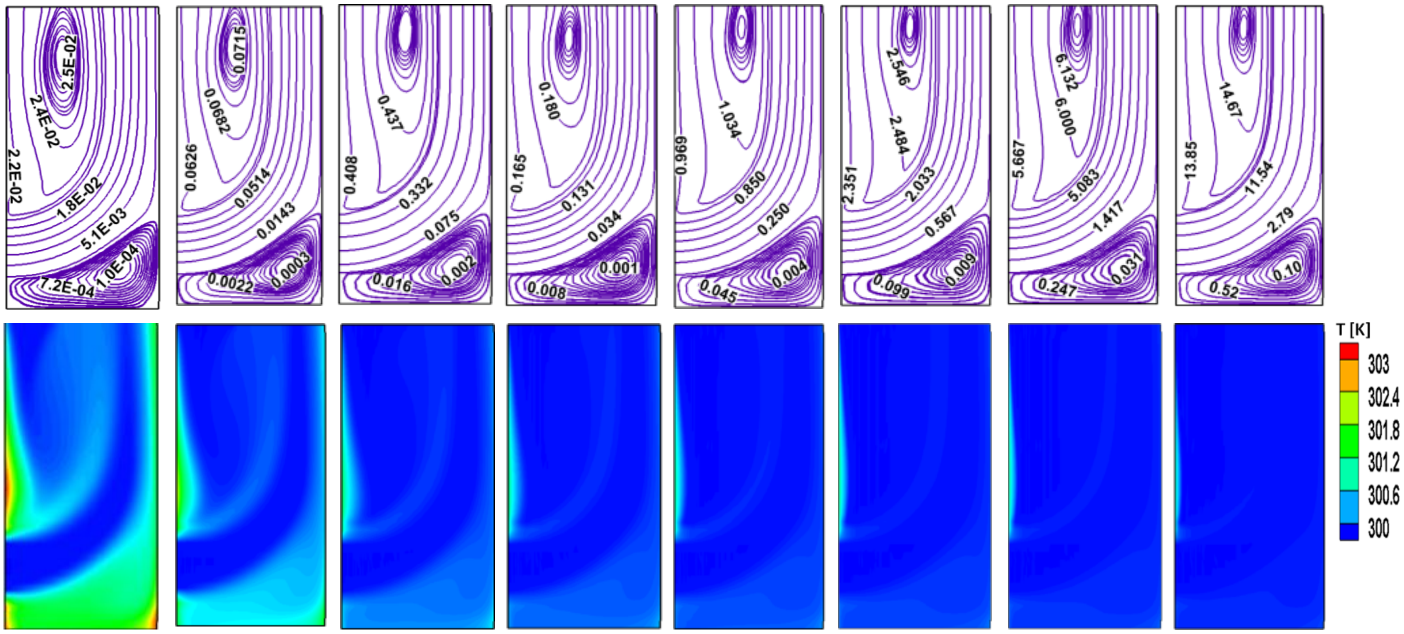
For square enclosure, three cells started to develop compared to AR = 2 geometries. Fig. 6 displays the profiles of u- and v-velocities at Y* = 0.5 and X* = 0.5 for different Rayleigh number, respectively. We got that, for all Rayleigh number, the maximum of u- and v-velocity components drive closer to the right wall, and the u- and v-velocity decrease gradually with increasing in Rayleigh number. Which leads to the secondary vortex development as can be seen in the contours of Fig. 6. At AR = 1, there is no remarkable change in velocities intensity position.

To understand more the behavior of secondary flow inside the enclosures, the effect of the aspect ratio on the hydrodynamics of fluid flow for different Rayleigh number are presented in Fig. 7 and Fig. 8. It can be observed that the intensity of the velocities is strongly dominant when Rayleigh number increases for all cases of aspect ratio. Because the configuration of AR=2 have the tallest height compared to the other geometries, a strong fully flow was created from the opening to the top free surface with vigorous vortex created at the right bottom side.

Figure 9 presents the dimensionless temperature profiles at the midlle line for a fixed aspect ratios (AR = 0.5, 1 and 2) with various Rayleigh number. For low Rayleigh number, the heat transfer is considerably enhanced for the case of AR = 2, where the temperature profile concentrated to the vertical walls. The heat flow is almost negligible in the middle of the enclosure, as depicted by the dimensionless temperature is zero. The distribution of temperature follows a perturbed profile and enhancing the convection process with the increment in Rayleigh number.

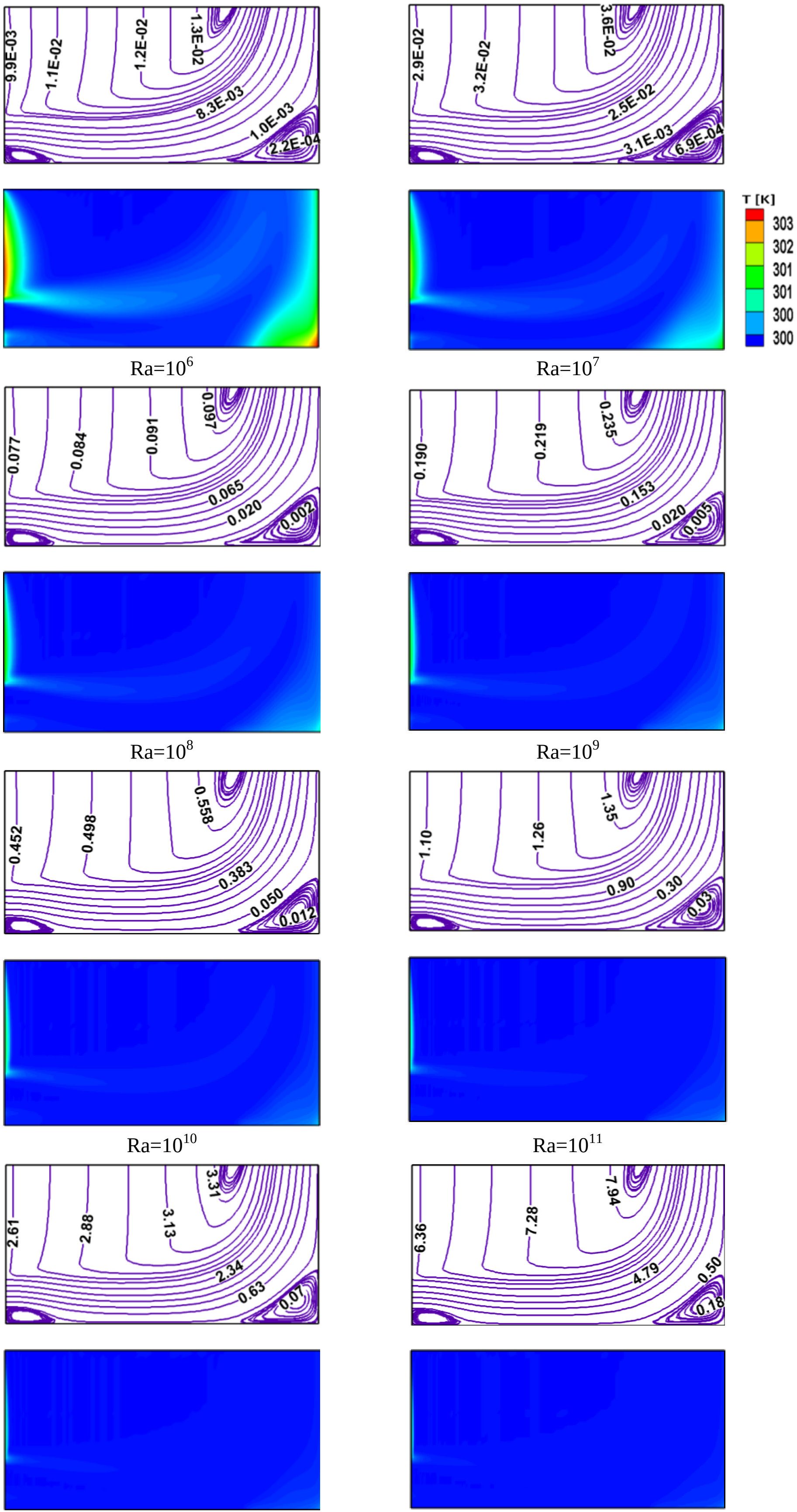
Effect of Rayleigh number with fixed wall heat flux on the enhancement of thermal flow and mass transfer are presented in Fig. 10 and Fig. 11, respectively. The local Nusselt number and mass fraction indicate the efficiency of heat and mass transfer process. Because the height of the cavity has a significate role, the heavier heat and mass transfer were found at the aspect ratio equal 2.

For low values of Rayleigh number, the mass fraction of the humid-air enhance while the Nusselt number decreases due to the slowly moving fluid. The heat transfer increases with Rayleigh number augments, shows in optimum thermal performance due to the vigorous turbulent flow.



Ra=$10^6$ Ra=$10^7$ Ra=$10^8$ Ra=$10^9$ Ra=$10^{10}$ Ra=$10^{11}$ Ra=$10^{12}$ Ra=$10^{13}$

**Fig. 3** Contours of streamlines (top) and isotherms (bottom) at AR = 2 for different Rayleigh number ($10^6 \leq Ra \leq 10^{13}$).



Ra=$10^6$ Ra=$10^7$

Ra=$10^8$ Ra=$10^9$

Ra=$10^{10}$ Ra=$10^{11}$

Ra=$10^{12}$ Ra=$10^{13}$

**Fig. 4** Contours of streamlines (top) and isotherms (bottom) at AR = 0.5 for different Rayleigh number ($10^6 \leq Ra \leq 10^{13}$).

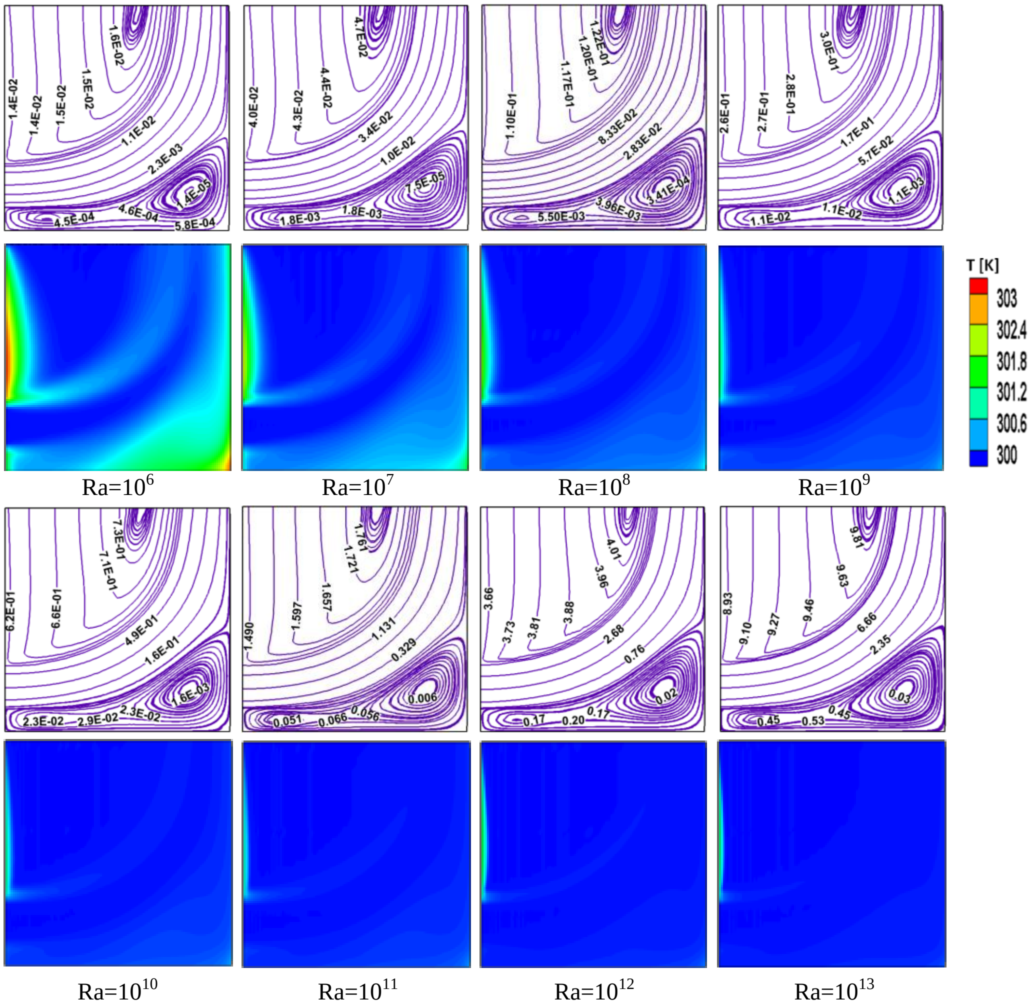


**Fig. 5** Contours of streamlines (top) and isotherms (bottom) at AR = 1 for different Rayleigh number ($10^6 \leq Ra \leq 10^{13}$).

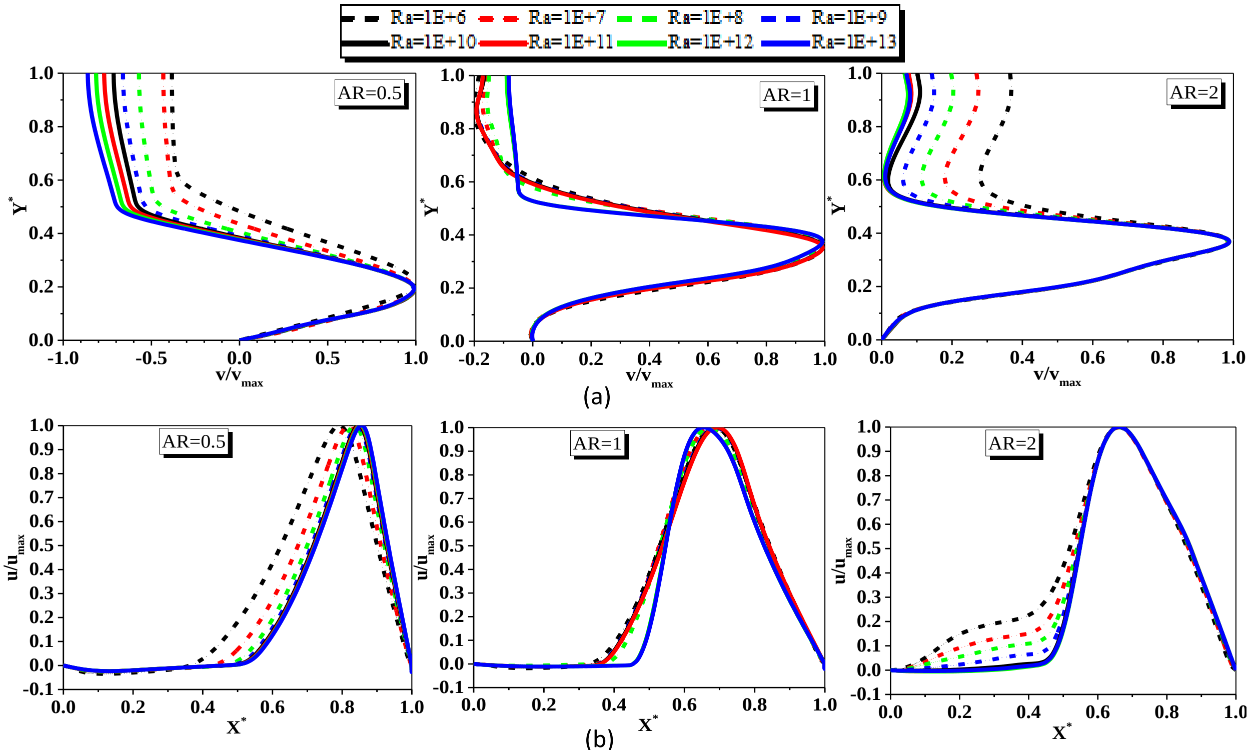


**Fig. 6** Effect of Rayleigh number on: (a) v-velocity at X* = 0.5 and (b) u-velocity at Y* = 0.5 for fixed aspect ratios (AR = 0.5, 1 and 2).

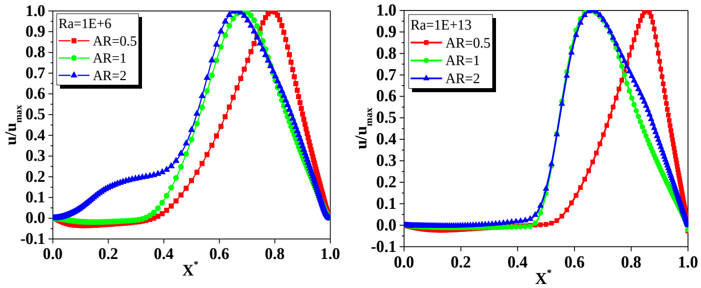


**Fig. 7** Effect of (AR) on u-velocity at Y* = 0.5 for Ra = $10^6$ and $10^{13}$.

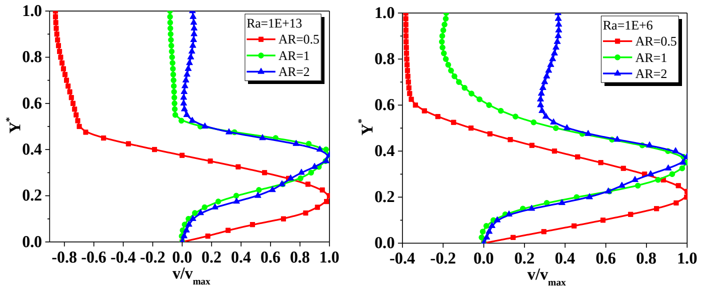
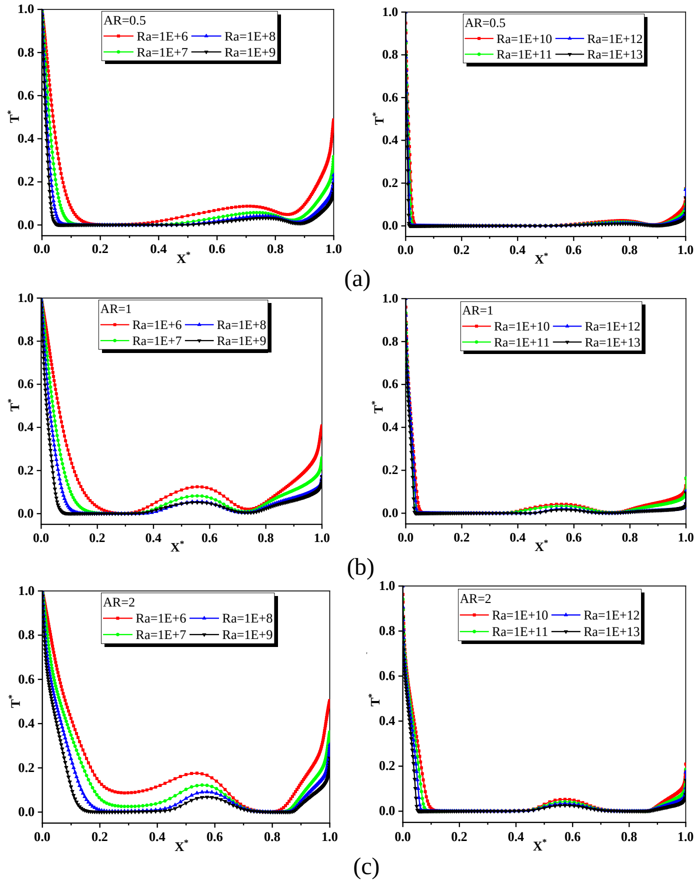


**Fig. 8** Effect of (AR) on v-velocity at X* = 0.5 for Ra = $10^6$ and $10^{13}$.



(a)

(b)

(c)

**Fig. 9** Static temperature profiles at Y* = 0.5 for (a) AR=0.5, (b) AR=1 and (c) AR=2 with various Rayleigh number ($10^6 \leq Ra \leq 10^{13}$).

Figure 12 depicts the probability density function (PDF) of mass fraction of humid-air with flow visualization contours for different configurations (AR = 0.5, 1 and 2) for Rayleigh numbers ($10^6$ to $10^{13}$). It can be observed that the configuration of AR = 0.5 has not achieved on the mass transfer process compared to the others form. This indicates that the use of this cavity in the mass transfer is not a satisfactory choice for turbulent. For AR = 2, the mass fraction is the best compared to AR = 1 and AR = 0.5. Thus, the PDF at the vertical walls (humid area) has the maximum values which are corresponding to the preferred mass transfer regarding to the low Rayleigh numbers.

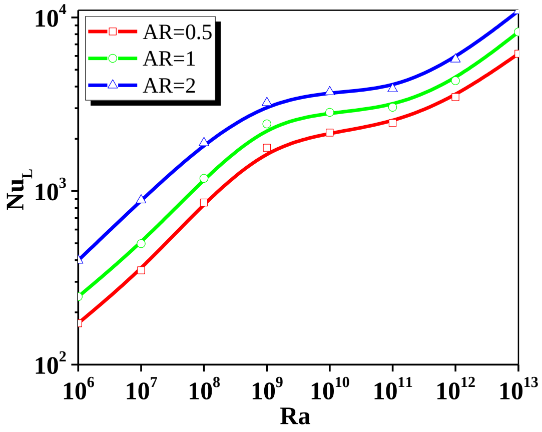


**Fig. 10** Evolutions of local Nusselt number as function of Rayleigh number with different aspect ratio (AR = 0.5, 1 and 2).

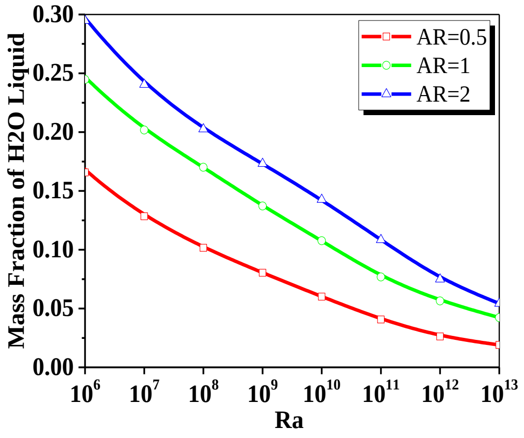


**Fig. 11** Mass fraction of $H_2O$ for various Rayleigh number with different aspect ratio (AR = 0.5, 1, and 2).

The contours of $H_2O$ mass fraction for all cases of aspect ratio shows that the mass transfer dominates more as Rayleigh number decreases. By looking at the isotherms and hydrodynamics contours, it was remarked that the mass and heat transfer process were less concentrated at the top free surface. The concentration of $H_2O$ shows almost similar patterns when Rayleigh number more than $10^{12}$ and increases significantly on the left wall of the enclosure, especially for the enclosure of AR =2.

## 4. CONCLUSIONS

We have examined the performance of the effective mass and heat transfer efficiency for natural convection of a humid-air inside open enclosures.

According to this work, the next conclusions can be given:

- Heat transfer efficiency for natural convection of a humid-air inside open enclosures is influenced by the Rayleigh number at the lateral walls for all cases of aspect ratio. While the mass transfer decreases, because of vaporization. Also, as the Rayleigh number increases, the flow visualization revealed that the vortex created in each enclosure has more vigorous intensity.

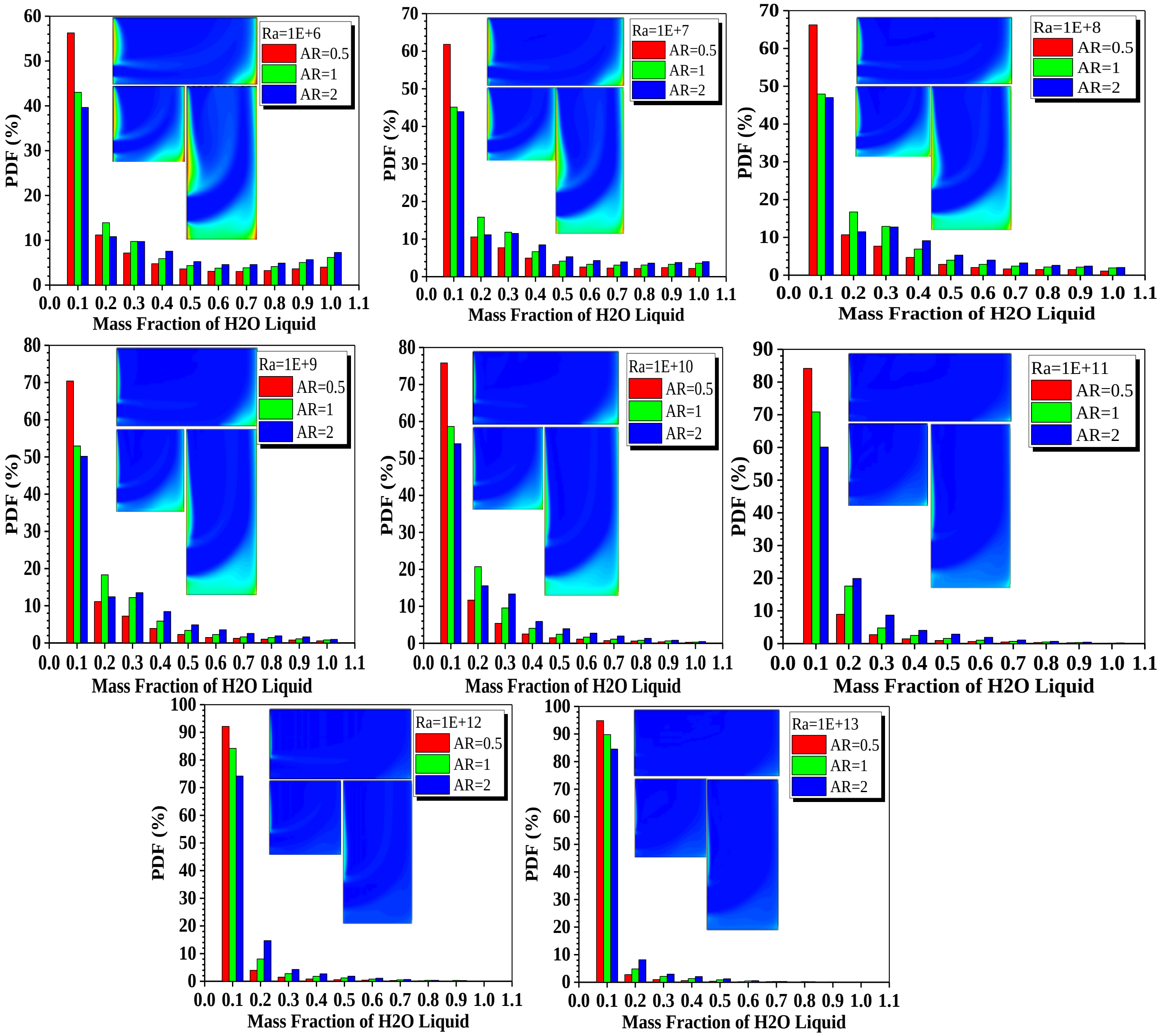


**Fig. 12** The probability density function (PDF) with mass fraction contours (Humid-air) as function of Rayleigh number ($10^6 \leq Ra \leq 10^{13}$) with different aspect ratio (AR = 0.5, 1 and 2).

- The opening surface has an enhancement advantage to the mass transfer, while the bottom wall was the most stable for a few mass transfer.
- It was found that at the higher value of aspect ratio, the mass transfer reduces for the square cavity compares to the preferable enclosure. Also, higher rates of Rayleigh number have more effects to increase the thermal enhancement rate and decrease the mass fraction of humid-air.
- As a consequence, the thermal and mass transfer performances in the cavity (AR=2) are more important compared to the other configurations for humid-air as working fluid.

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## NOMENCLATURE

| | |
|---|---|
| AR | enclosure aspect ratio |
| *Cp* | specific heat capacity, J. kg$^{-1}$. K$^{-1}$ |
| *g* | gravitational acceleration, m.s$^{-2}$ |
| *H* | enclosure height, m |
| *L* | enclosure width, m |
| $Nu_L$ | local Nusselt number |
| *P* | pressure, Pas |
| *Pr* | prandtl number |
| q | heat flux, w |
| Ra | Rayleigh number |
| *T* | temperature, K, C° |
| $T_{ref}$ | reference temperature, K, C° |

| | |
|---|---|
| T* | dimensionless temperature |
| u,v | velocity components, m.s$^{-1}$ |

*Greek Symbols*

| | |
|---|---|
| $\alpha$ | thermal diffusivity, m$^{2}$. s$^{-1}$ |
| $\beta$ | thermal expansion coefficient, K$^{-1}$ |
| $\varepsilon$ | turbulent dissipation rate |
| $\lambda$ | thermal conductivity, w.m$^{-1}$.K$^{-1}$ |
| $\mu$ | dynamic viscosity, kg. m$^{-1}$.s$^{-1}$ |
| $\upsilon$ | kinematic viscosity, m$^{2}$.s$^{-1}$ |
| $\rho$ | density of mixture, Kg.m$^{-3}$ |

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