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# COMBINED HEAT AND MOISTURE CONVECTIVE TRANSPORT IN A PARTIAL ENCLOSURE WITH MULTIPLE FREE PORTS

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#### Abstract and keywords

#### Abstract

Combined heat and moisture transport in an enclosure with free ports has been investigated numerically. Enclosed moist air interacts with the surrounding air through these free-vented ports. The governing conservation equations were solved numerically using a control volume-based finite difference technique. Appropriate velocity boundary conditions at each ports are imposed to achieve overall mass conservation across this system. Air, heat and moisture transport structures are visualized respectively by streamlines, heatlines and masslines. Effects of buoyancy ratio, thermal Rayleigh number on convective heat/moisture transfer rate and flow rate across each free-vented port are discussed. Particularly, Numerical results demonstrate that the convective heat and moisture transport patterns and transport rates on horizontal ports greatly depend on properties of porous medium, while the air exchange rate on vertical port is almost unaffected by the buoyancy ratios for most situations.

#### Keywords

Enclosure flow, Natural convection, Heatlines and masslines

#### Nomenclature

NOMENCLATURE									
AR	aspect ratio (W/H)	Greek Symbols							
d	size of port	α	thermal diffusivity						
D	mass diffusivity (dimensionless port size)	$\beta_{\rm t}$	thermal expansion coefficient						
Da	Darcy number $(K/H^2)$	$\beta_{\rm s}$	expansion coefficient with mass fraction						
8	gravitational acceleration	v	kinematic viscosity						
Н	height of the enclosure	ρ	fluid density						
k	thermal conductivity of the porous medium	τ	dimensionless time						
Κ	permeability of the porous medium	Ψ	dimensionless streamfunction						
Le	Lewis number $(\alpha/D)$	ξ	dimensionless heatfunction						
М	dimensionless volumetric flow rate	η	dimensionless massfunction						
Ν	buoyancy ratio								
Nu	overall Nusselt number	Subscript	Subscripts						
р	fluid pressure	L	left side						
Pr	Prandtl number $(\nu/\alpha)$	max, min	maximum, minimum						
Ra	thermal Rayleigh number	0	reference						
S	dimensional mass fraction	S	solutal						
Sh	overall Sherwood number	t	Thermal						
Т	temperature	T, R, B	top, right and bottom vents						
<i>u</i> , <i>v</i>	velocity components								
W	width of the enclosure	Superscripts							
<i>x</i> , <i>y</i>	Cartesian coordinates	*	dimensional variable						

Main body

#### **1. Introduction**

Natural convection in porous medium has been studied extensively in past decades, due to it is frequently encountered in chemical transport in packed-bed reactors, melting and solidification of binary alloys, grain storage, food processing and storage, contaminant transport in ground water, crystal growth, the migration of moisture through air contained in fibrous insulation, and etc [1].

In past studies, single component natural convection (thermal convection) in porous enclosures has been paid broadly attentions [1-4]; whereas, multiple component natural convection, particularly the combined heat and moisture transfer in porous enclosures, has received relatively few attentions. Typically, Chamkha and his coauthors numerically investigated the double diffusive buoyancy driven convection in an enclosure filling with porous medium. The effects of cavity inclination, internal heat generation or absorption, thermal and solute buoyancy forces, and Darcy number on the flow field and heat transfer potential were studied in details, which models the heat and moisture convection in a grain storage [5, 6]. Kumar and his coauthors performed numerical study of coupled heat and mass transfer in porous enclosures with wavy geometries and thermally stratified boundary conditions, effects of stratification and wavy function on aiding and opposing flows were particularly analyzed [7, 8]. Costa numerically investigated double diffusive natural convection in a parallelogrammic porous enclosure filled with moist air, the heat and mass transfer characteristics of parallelogrammic porous enclosure are analyzed on the thermal Rayleigh number, aspect ratio, and inclination angle, both for the situations of combined and opposite heat and moisture flows through the enclosure [9]. Zhao et al. numerically investigated the double diffusive natural convection in a porous enclosure with the simultaneous presence of discrete heat and moisture sources, effects of permeability of porous medium, strip pitch, thermal and solutal Rayleigh numbers on the combined heat and mass transfer, particularly on the multiple steady motions, have been explored [10].

Observing from aforementioned studies, only the enclosed domain filling with porous medium is considered. Thinking essentially of actual engineering applications, such as food and grain storage, building construction elements, internal fluid motion could interact with ambient environment through openings or infiltrations, i.e., partial enclosures. In past decades, single component natural convection in partial enclosures has been investigated broadly, including inclination, port size, heating boundary conditions, fluid properties, coupling with radiation, etc [11-26].

In the present work, double diffusive natural convection in a cavity with multiple free openings, modeling the food and grain storages under natural environment, will be investigated numerically. To the authors' knowledge, this problem has never been studied. The porous medium considered here is modeled according to the Darcy-Brinkman formulation, which could account for the inertia effects and the no-slip boundary conditions on rigid boundaries [5, 6, 10, 27]. Additionally, visualization of the heat and solute transports, using streamlines, heatlines and masslines [9, 10, 27, 28, 29, 30], would be conducted in the present work.

In following sections, the physical model and mathematical formulation for the problem is first given. Subsequently, a numerical simulation of the full governing equations is carried out to study the transport structures and heat/mass transfer rates. Finally, the results from the numerical computations are discussed in details.

#### 2. Physical model and problem statements

The physical domain under investigation is a two-dimensional fluid-saturated Darcy-Brinkman porous enclosure (see Fig. 1). The rectangular enclosure is of width W and height H, and the Cartesian coordinates (x, y), with the corresponding velocity components (u, v), are indicated herein. It is assumed that the third dimension of the enclosure is large enough so that the fluid, heat and mass transports are two-dimensional. Three ports, respectively of size  $d_T$ ,  $d_R$ , and  $d_B$ , are respectively centrally-imposed on the top, bottom and right boundaries of the enclosure, where ambient fluid could enter the enclosure or enclosed fluid could effuse into the ambient. The left wall maintains higher and constant temperature and concentration,  $t_1$  and  $s_1$ , while the rest of cavity walls are insulated and impermeable. Ambient fluid reservoir is maintained lower temperature and concentration constants,  $t_0$  and  $s_0$ . Gravity acts in the negative y-direction.

The porous matrix is assumed to be uniform and in local thermal and compositional equilibrium with the saturating fluid. Thermophysical properties are supposed constant. The flow is assumed to be laminar and incompressible. Viscous dissipation and porous medium inertia are not considered, and the Soret and Dufour effects are neglected. Density of the saturated fluid mixture is assumed to be uniform over all the enclosure, exception made to the buoyancy term, in which it is taken as a function of both the temperature *t* and concentration *s* through the Boussinesq approximation [13],

$$\rho = \rho_0 [1 - \beta_t (t - t_0) - \beta_s (s - s_0)] \tag{1}$$

Where  $\rho_0$  is the fluid density at temperature  $t_0$  and concentration  $s_0$ , and  $\beta_t$  and  $\beta_s$  are the thermal and concentration expansion coefficients, respectively. Subscript 0 refers to the condition over the ambient fluid reservoir.

By employing the aforementioned assumptions into the macroscopic conservation equations of mass, momentum, energy and species, a set of dimensionless governing equations is obtained as,

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0$$
(2)

$$\frac{\partial U}{\partial \tau} + \frac{\partial UU}{\partial X} + \frac{\partial VU}{\partial Y} = -\frac{\partial P}{\partial X} + \sqrt{\frac{\Pr}{Ra}} \left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) - \sqrt{\frac{\Pr}{Ra}} \frac{U}{Da}$$
(3)

$$\frac{\partial V}{\partial \tau} + \frac{\partial UV}{\partial X} + \frac{\partial VV}{\partial Y} = -\frac{\partial P}{\partial Y} + \sqrt{\frac{\Pr}{Ra}} \left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) - \sqrt{\frac{\Pr}{Ra}} \frac{V}{Da} + (T + NS)$$
(4)

$$\frac{\partial T}{\partial \tau} + \frac{\partial UT}{\partial X} + \frac{\partial VT}{\partial Y} = \frac{1}{\sqrt{Ra \operatorname{Pr}}} \left( \frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} \right)$$
(5)

$$\frac{\partial S}{\partial \tau} + \frac{\partial US}{\partial X} + \frac{\partial VS}{\partial Y} = \frac{1}{Le\sqrt{Ra}\Pr} \left(\frac{\partial^2 S}{\partial X^2} + \frac{\partial^2 S}{\partial Y^2}\right)$$
(6)

The following dimensionless variables are used,

$$(X, Y) = (x, y)/H, (U, V) = (u, v)/V_{\rm r}, \tau = \tau^*/(H/V_{\rm r})$$
(7a)

$$P = (p + \rho_0 g y) / (\rho_0 V_r^2), T = (t - t_0) / \Delta t, S = (s - s_0) / \Delta s$$
(7b)

$$V_{\rm r} = (g\beta_{\rm t}\Delta tH)^{1/2}, \Delta t = t_1 - t_0, \Delta s = s_1 - s_0$$
(7c)

Where H,  $V_{\rm r}$ ,  $H/V_{\rm r}$ ,  $\Delta t$ , and  $\Delta s$  are the scales for length, velocity, time, temperature and concentration respectively. From the boundary and geometry of the enclosure, it appears that the present problem is simultaneously governed by additional dimensionless geometry parameters, cavity aspect ratio *AR*, and sizes of three openings  $D_{\rm T}$ ,  $D_{\rm R}$  and  $D_{\rm B}$ ,

which are defined respectively as,

$$AR = W/H, D_{\rm T} = d_{\rm T}/H, D_{\rm R} = d_{\rm R}/H, D_{\rm B} = d_{\rm B}/H$$
 (8)

Equations (2)-(6) indicate that the present problem is governed by the following dimensionless parameters, namely, Prandtl number Pr, Darcy number Da, the thermal Rayleigh number Ra, the solutal to thermal buoyancy ratio N and Lewis number Le defined as,

$$Pr = v/\alpha, Da = K/H^2, Ra = g\beta_t \Delta t H^3/v\alpha, N = \beta_s \Delta s/\beta_t \Delta t, Le = \alpha/D$$
(9)

Where *v* is the kinematics viscosity of the fluid,  $\alpha$  and *D* respectively are the thermal and molecular diffusivities of the combined fluid plus solid porous matrix medium, *K* is the permeability of the porous medium, and *g* is the acceleration due to gravity. Since the particle Reynolds number is considered being less than unity in this work, the Forchheimer inertia term has been dropped from the momentum equations (3) and (4) compared with the Darcy and Brinkman terms. The porosity of the porous layer, the ratios between effective viscosity and fluid viscosity, and the ratios of the thermophysical properties of the porous medium and of the fluid, all have been implicitly set to unity [5, 6, 27].

The thermal Rayleigh number Ra is usual when analyzing the single natural convection heat transfer in porous enclosure. In addition, the buoyancy ratio N and Lewis number Le arise when analyzing the double-diffusive natural convection problems. The buoyancy ratio N denotes the relative strengths of the thermal and solutal buoyancy forces, and it can be either positive or negative, its sign depending on the ratio

of  $\beta_s$  and  $\beta_t$ . Concerning the left vertical wall, the thermal and solute buoyancy effects are augmented when *N* is positive, and they are opposed otherwise. Further, Buoyancy ratio is zero for no species diffusion, infinite for no thermal diffusion, positive for both effects combining to drive the flow, and negative for the effects opposed. In the present work, for the thermal-driven flow limit (N = 0), the ensuing flow is driven solely by the buoyancy effect associated with the temperature gradients. As a result, the mass species can be advected by the flow field that is driven by thermal buoyancy forces. If N<<-1, the solutal buoyancy forces can mostly determine the flow. If N>>1, the thermal buoyancy forces play a key role in fluid flow and heat and moisture transfer. Combined global heat and solute flows and negative values of parameter *N* can lead to multiple solutions or oscillatory solutions [10], such situations are intentionally avoided in the present work.

#### **3. Boundary conditions**

Generally, the boundary conditions of the system shown in Fig. 1 are U=V=0,  $\partial T/\partial n = \partial S/\partial n = 0$  on all solid surfaces, while constant temperature and concentration are maintained along the left wall,

$$X = -AR/2 \text{ and } -1/2 \le Y \le +1/2, T = 1, S = 1$$
 (10)

The solution of elliptic partial differential equations of natural convection flow in open cavities is found to be highly sensitive to the boundary conditions at the entry and exit [11-17]. In the present work, the derivative of tangential velocity is set at zero and

the longitudinal velocity is obtained by the mass balance at the boundary cells [12]. The mass conservation across this enclosure has been got around by the efficient solution of pressure correction equations. The boundary derivatives of tangential and longitudinal components should be carefully coped with finite volume method [31]. Mathematically, the present model accounts for the momentum boundary conditions by imposing the following conditions on the openings,

$$Y = 1/2 \text{ and } -D_{\mathrm{T}}/2 \le X \le +D_{\mathrm{T}}/2, \quad \frac{\partial U}{\partial Y} = 0, \quad \frac{\partial V}{\partial Y} = -\frac{\partial U}{\partial X}$$
 (11)

$$Y = -1/2 \text{ and } -D_{\rm B}/2 \le X \le +D_{\rm B}/2, \quad \frac{\partial U}{\partial Y} = 0, \quad \frac{\partial V}{\partial Y} = -\frac{\partial U}{\partial X}$$
 (12)

$$X = AR/2 \text{ and } -D_R/2 \le Y \le +D_R/2, \quad \frac{\partial V}{\partial X} = 0, \quad \frac{\partial U}{\partial X} = -\frac{\partial V}{\partial Y}$$
 (13)

The simulated results have been compared with those of extended domain method, which demonstrates that the present boundary assumptions are valid.

#### 4. Convective transport evaluation and visualization

The overall heat and mass transfer rates across the system are important in engineering applications. It is appropriate at this stage to define the Nusselt and Sherwood numbers on the surface of heat and mass sources can be written respectively as,

$$Nu_{L} = \int_{1/2}^{t/2} -\frac{\partial T}{\partial X}\Big|_{X=-AR/2} dY, \quad Sh_{L} = \int_{1/2}^{t/2} -\frac{\partial S}{\partial X}\Big|_{X=-AR/2} dY$$
(14)

However, for the heat and mass transfer rates across the free vents, the spatial Nusselt and Sherwood numbers should be defined,

$$Nu_{T} = \int_{D_{T}/2}^{D_{T}/2} (\sqrt{\Pr Ra}VT - \frac{\partial T}{\partial Y}) \Big|_{Y=1/2} dX , \quad Sh_{T} = \int_{D_{T}/2}^{D_{T}/2} (Le\sqrt{\Pr Ra}VS - \frac{\partial S}{\partial Y}) \Big|_{Y=1/2} dX \quad (15)$$

$$Nu_{B} = \int_{D_{B}/2}^{D_{B}/2} (\sqrt{\Pr Ra}VT - \frac{\partial T}{\partial Y}) \Big|_{Y=-1/2} dX , \quad Sh_{B} = \int_{D_{B}/2}^{D_{B}/2} (Le\sqrt{\Pr Ra}VS - \frac{\partial S}{\partial Y}) \Big|_{Y=-1/2} dX \quad (16)$$

$$Nu_{R} = \int_{D_{R}/2}^{D_{R}/2} (\sqrt{\Pr Ra}UT - \frac{\partial T}{\partial X}) \Big|_{X=\frac{AR}{2}} dY , \quad Sh_{R} = \int_{D_{R}/2}^{D_{R}/2} (Le\sqrt{\Pr Ra}US - \frac{\partial S}{\partial X}) \Big|_{X=\frac{AR}{2}} dY \quad (17)$$

Streamfunction and streamlines are routinely the best way to visualize the convective fluid flow. The dimensionless streamfunction  $\Psi$  is defined such that,

$$\frac{\partial \Psi}{\partial Y} = U, \quad -\frac{\partial \Psi}{\partial X} = V \tag{18}$$

The heat and mass transport processes are analyzed through the heatlines and masslines, respectively [9, 10, 27-30]. The heatfunction and massfunction can be made dimensionless respectively as,

$$\xi = \xi^* / [k\Delta t], \ \eta = \eta^* / [\rho D\Delta s]$$
<sup>(19)</sup>

The dimensionless first order derivatives of heatfunction and massfunction equations can be obtained as follows,

$$\frac{\partial\xi}{\partial Y} = \sqrt{Ra}\Pr UT - \frac{\partial T}{\partial X}, \quad -\frac{\partial\xi}{\partial X} = \sqrt{Ra}\Pr VT - \frac{\partial T}{\partial Y}$$
(20)

$$\frac{\partial \eta}{\partial Y} = Le\sqrt{Ra\Pr}US - \frac{\partial S}{\partial X}, \quad -\frac{\partial \eta}{\partial X} = Le\sqrt{Ra\Pr}VS - \frac{\partial S}{\partial Y}$$
(21)

The  $\xi$  and  $\eta$  fields are defined through its first order derivatives, being thus important only differences in its values but not its level, which is similar to the flow field defined by Eq. (18). Thus, we have the freedom to state that,

$$\Psi(-AR/2, -1/2) = \xi(-AR/2, -1/2) = \eta(-AR/2, -1/2) = 0$$
(22)

The heat and mass functions are evaluated for visualization purposes, once known the flow, temperature and concentration fields, they can be obtained using the solution

method for conduction-type problems [28, 32].

#### 5. Numerical technique and code validation

A FVM (Finite Volume method) was used to obtain numerical solutions of the complete governing Eqs. (2) - (6) on a staggered grid system [31, 33]. In the course of discretization, the third-order deferred correction QUICK scheme [34] and a second-order central difference scheme are respectively implemented for the convection and diffusion terms. The SIMPLE algorithm was chosen to numerically solve the governing differential equations in their primitive form. The pressure correction equation is derived from the continuity equation to enforce the local mass balance [29, 31]. To obtain better convergence properties, the unsteady terms in these equations are implicitly treated and hence approximated by backward differencing. For each time step, the discretized equations are solved using a line-by-line procedure, combining the tri-diagonal matrix algorithm (TDMA) and the successive over-relaxation (SOR) iteration. The permanent solution has been obtained by this false transient procedure. The iterative procedure is repeated until the following condition is satisfied,

$$\frac{\sum_{i} \sum_{j} \left| \Phi_{i,j}^{new} - \Phi_{i,j}^{old} \right|}{\sum_{i} \sum_{j} \left| \Phi_{i,j}^{new} \right|} \le 10^{-4}$$
(23)

Where  $\Phi$  stands for *U*, *V*, *T* and *S*. The subscript *i* and *j* indices denote grid locations in the (*X*, *Y*) plane. A further decrease of the convergence criteria 10<sup>-5</sup> does not cause any significant change in the final results.

Simultaneously, reliable numerical results are obtained by performing an energy and

solute balance at each time step over the physical domain [28, 29]. As the walls are impermeable and adiabatic, the thermal and solute conservations of the two-dimensional natural convective system illustrated in Fig. 1 can be written in mathematical forms respectively as follows,

$$Nu_{\rm L} + Nu_{\rm R} + Nu_{\rm B} + Nu_{\rm T} = 0$$

 $Sh_{\rm L} + Sh_{\rm R} + Sh_{\rm B} + Sh_{\rm T} = 0$ 

For most of the results reported here, the energy and mass balances (aforementioned two formulations), were satisfied to within 0.1%.

In order to resolve the boundary layers along the surfaces of heat and moisture sources, non-uniform grids in *X* and *Y* direction were used for all computations and the grid was clustered toward the sidewalls, especially for the case of large *Le* and *Ra*. A non-uniform systematic grid independence study was conducted, and  $31\times31$ ,  $41\times41$ ,  $51\times51$  and  $61\times61$  was adopted separately. The final grid resolution of  $51\times51$  was selected at the balance between the calculation accuracy and the speed for *AR* = 1. The density of grid was the constant of 1.08.

The current numerical technique has been very successfully used in a series of recent papers, including single-component natural convection [32], conjugate natural convection [33], and double diffusive natural convection [10, 27-29] in gaseous/porous enclosures. To further validate the present numerical code, natural convection in square open cavities has been numerically analyzed. One of the initial numerical studies on single component natural convection in an open enclosure was performed by Chan and

(24)

(25)

Tien [11, 12]; Angirasa et al. [13]; Hinojosa et al. [16]. For this comparison, solutions presented in Table 1 are obtained for  $Ra = 10^3 - 10^7$ , Pr = 1.0 in a square cavity where the vertical wall facing to the opening is maintained at uniform and higher temperature, while the horizontal walls are adiabatic. The Prandtl number in the benchmark case was 1.0, not the same as the one in the present work, 0.7. Therefore, in the validation, Pr = 1.0 was adopted in order to compare the results with other researchers. Additionally, the combined heat and moisture convective flow is independent of Prandtl numbers if Pr is of O(1), which has been investigated in our past studies [10, 28]. Observing from Table 1, the agreement between the present study and other published results is fairly good.

#### 6. Results and discussion

The foregoing analysis indicates that there are nine parameters, including Ra, N, Le, Da, Pr, AR,  $D_T$ ,  $D_B$  and  $D_R$ , that could be varied in this study. Because of the abundance of parameters, a full-blown parametric investigation of the problem is unrealistic. The study is limited to a cavity with an aspect ratio of unity, i.e., a square enclosure (AR = 1). The sizes of three openings are maintained constant and same,  $D_T = D_B = D_R = 1/5$ . In the actual computations, Pr is set equal to 0.70, Le is fixed at 2.0, which could not only simulate the gaseous fluid at room temperature, and also distinguish species (moisture) from general fluid.

The buoyancy ratio N is in the range -5 to +5, covering the spectrum from solute-driven opposing flows (N << -1), to pure heat-driven flows (N = 0) and to

solute-driven aiding flows (N >> 1) [1]. The brinkman extended Darcy model has been used through the study: in the first step, the Darcy number is fixed at 10<sup>-5</sup>, and then its value is fixed at 10<sup>2</sup>. The coordinates are chosen in the present work such that counterclockwise (or clockwise) movement will be associated with positive (or negative) streamfunctions. Due to the thermal and solutal boundary conditions considered here, the left side wall has a higher temperature and higher concentration than other sides. As a result, the direction of the thermal flow is generally clockwise, whereas the direction of the solutal flow depends upon the sign of the concentration expansion coefficient  $\beta_s$ in Eq. (9). Thus the direction of the solutal flow is clockwise for  $\beta_s(N) > 0$  and counterclockwise for  $\beta_s(N) < 0$ .

The computed streamlines, heatlines and masslines are plotted in following figures for several combinations of Darcy number, buoyancy ratio and thermal Rayleigh number. The flow directions of fluid, heat and species could be identified by the tangential arrows presented in contours.

#### 6.1. Porous medium of low Darcy number (10<sup>-5</sup>)

As the Darcy number is low, typically  $10^{-5}$ , the boundary frictional resistance becomes progressively significant and adds to the bulk frictional drag induced by the solid matrix to slow the convection motion. This is expected since, in the limit of *Da* approaching 0, the Brinkman model reduces to Darcy Law.

First, the heat-driven flow limit (N = 0) depicted in Fig. 2(a) is discussed. In this configuration, the solute buoyancy force is not presented, but mass transfer is induced

by the thermally driven flow, i.e., heat-transfer-driven flows. This means that the temperature field is coupled to the flow field and not coupled to the concentration field. Entrained ambient fluid at the bottom port splits, most of it approaching the heated wall, while a small portion proceeding upward along clockwise recirculation and exhausting at the right port. Heatlines show that the heat along the heated wall mostly transports clockwisely into the bottom vent. Masslines show that the species has similar transport structure to the heat. As a result, the overall heat and mass transfer rates along the bottom vent,  $Nu_B$  and  $Sh_B$ , are far higher than those of other venting sides (illustrated in Fig. 6(a)).

Figures 2(b)-(c) exemplify typical features of aiding double-diffusive flow (N>0). When the buoyancy ratio is increased above zero the flow near the heating wall is driven vertically upward, and meanwhile the low concentration at the left-hand wall causes the fluid near it to sink. As expected, both thermal and solutal buoyancy effects are augmenting each other and thus they simultaneously accelerate the flow clockwise. Due to the great frictional drag imposed by the porous matrix, the flow intensity could not be enhanced greatly, which is demonstrated by the similar flow patterns shown in Fig. 2(b) and 2(c). Correspondingly, the heatlines and masslines also show similar characteristics to the heat transfer driven flow presented in Fig. 2(a). Consequently, the overall heat and mass transfer rates vary little with the buoyancy ratio as shown in Fig. 6(a) and 6(c). Particularly, the transport rates  $Nu_R$  and  $Sh_R$  almost increases little with the buoyancy ratio. However, closely observing the contours shown in Fig. 2, with the

increasing of buoyancy ratio, one could find that more heatlines and masslines effusing from the heated wall cluster to the top vent, while the function lines toward the bottom vent become sparse and bended. These transports respectively contribute to the enhancement of transport rates along the top vent and degradation of transport rates along the bottom vent, with increasing buoyancy ratio.

The typical feature of opposing double-diffusive flow (N < 0) is discussed herein. For the buoyancy ratio  $N \sim (-1)$ , the thermal and solutal buoyancy forces of almost equal intensity and counteract, the resulting convective flow tends to be unstable, the results have been intentionally avoided in the present work. Fig. 3 provides exemplary results for some buoyancy ratios (N = -1.5, -3.0, -5.0). The main contribution for buoyancy is due to the solutal one, and the fluid near the sink would be driven downward. The direction of the fluid circulation has been completely reversed and the flow pattern consists of a secondary cell moving counter-clockwise in the cavity. The top vent and side vent entrain ambient fluid to sweep the left wall and exhaust at the bottom port. Due to the entrained fluid is of lower temperature and concentration, the heat and solute are visualized respectively by heatlines and masslines that they are transported directly from the left wall to top and side vents. Transport paths from the left wall to top vent are the shortest and the  $Nu_{T}$  and  $Sh_{T}$  also are a bit higher than that on other ports, as shown in Fig. 6(a) and 6(c). With the decreasing buoyancy ratios (increasing of absolute values of buoyancy ratio), the boundary layer adjacent to the left wall becomes thinner, which strengthens the heat and mass transfer rates simultaneously. The convective transport

rates on the plane of bottom vent,  $Nu_B$  and  $Sh_B$ , increase absolutely due to the strengthened counterclockwise fluid circulation transports more heat and solute to the bottom vent. On the contrary, the transport rates on the top and side vents decrease with the decreasing buoyancy ratios.

#### **6.2.** Porous medium of high Darcy number (10<sup>2</sup>)

As Darcy number increases from  $10^{-5}$  to  $10^2$ , the effect of the viscous forces accounted for in the Brinkman term on the flow velocity becomes significant. As the permeability of the porous medium *Da* is increased, the boundary frictional resistance becomes gradually less important and the fluid circulation within the enclosure is progressively enhanced. Indeed, increasing the Brinkman term implies that the balance between the Darcy term and the buoyancy force in the boundary layer is progressively replaced by the balance between a viscous force and the buoyancy term. The viscous force enhances the velocity at high Darcy numbers.

For the typical heat transfer driven flow (N = 0) shown in Fig. 4(a), a multicellular flow structure is observed in the lower half of the cavity. As the buoyancy ratio increases positively, the aforementioned trends are aggravated, demonstrating from Figs. 4(b) and 4(c). The streamlines show that the dynamic boundary layers are thinner for the higher value of Da, and consequently the temperature and concentration gradients at the walls are larger when the Brinkman term becomes significant. Consequently, observing from Fig. 6, the heat and mass transfer rates of  $Da = 10^2$  are greatly higher than that of  $Da = 10^{-5}$ . Additionally, the stratifications of thermal and solute in the center

of the cavity blockage the heat and solute transports from the source to the side vent and bottom vent, thus it contributes to the significant increase of heat and mass transfer on the top vent, which also can be demonstrated by the fact that most of heatlines and masslines flow towards the top vent.

As buoyancy ratio becomes negative (N = -1.5, Fig. 5(a)), solute-dominated opposing flow occupies the whole enclosure, it comprises a primary counterclockwise vortex, with two secondary counterclockwise eddies near two vertical sides. With the decreasing buoyancy ratios, the left secondary eddy gradually dwindle and could not sustain as N<-2.5; on the other hand, the right secondary eddy gradually expand and shift toward right side. This is due to the fluid draft from right side vent promotes with increasing of absolute values of buoyancy ratio. Correspondingly, transport rates along the right vent increase till N<-3.0. Below that, the flow patterns vary little. Continuously decreasing buoyancy ratio, the primary flow increases in flow intensity while not in quantities. Thus, the convective heat and mass transfer rates on the bottom vent increase greatly contrasting with those on other vent ports.

#### **6.3. Effect of thermal Rayleigh numbers**

In order to investigate the effect of thermal Rayleigh number on the coupled heat and mass transports, the buoyancy ratio is maintained at 2.0, while the Darcy number of porous medium could be varied as  $10^{-5}$  and  $10^{2}$ , just simulating the limiting situations.

When Da is low (10<sup>-5</sup>), as illustrated in Fig. 7(a), the fluid flow is of relatively low intensity, and entrained fluid from bottom vent almost paralleling flows toward the top

vent, little portion of it turns right to the side vent. Simultaneously, heat and solute from the left wall diffusion-dominated flow toward the bottom vent, which are respectively demonstrated by the heatlines and masslines. Actually, this flow chart is similar to that shown in Fig. 2(a). Introducing the porous thermal Rayleigh number  $R_t$ , defined as  $R_t =$ RaDa, and using the scale analysis of double diffusive natural convection boundary flow, heat and mass transfer potentials are respectively proportional to  $R_tN$  and  $R_t(1+N)$ with constant Lewis number [28]. As expected, the flow charts presented in Fig. 2 correspond to lower porous thermal Rayleigh number and lower combined thermosolutal buoyancy force. With the promotion of Ra, shown in Fig. 7(b), the fluid flow tends to be convection-dominated, where primary flow eddy forms in the center of the enclosure, with fluid entrained from bottom vent being heated and polluted by the right wall, then exhausted toward top and side vents. Particularly, as thermal Rayleigh number continuously increases, shown in Figure 7(c), a flow circulation in the clockwise direction gradually oriented along the diagonal joining the left lower and right upper corners has been observed in this limiting Darcy regime.

Generally, an increase in *Ra* would result in monotonously increase in both *Nu* and *Sh* due to the increased flow rate. Fig. 9(a) also has demonstrated that point. Here should be noted that, for the fluid with relatively high Lewis number (Le = 2 > 1), the thermal boundary layer is expected to be a bitter thicker than the solute boundary layer. Consequently, the Sherwood number seems a bitter higher than that of heat transfer rate, Nusselt number. However, on the bottom vent, *Nu*<sub>B</sub> is unexpectedly higher than *Sh*<sub>B</sub>.

Vividly, masslines in Fig. 7 show that solute from left wall tracks longer toward the bottom vent, comparing that heat transport structures. As *R*a increases, the difference between solute and heat transports enlarges, with  $Nu_{\rm B}$  differing from  $Sh_{\rm B}$  shown in Fig. 9(a).

Fig. 8 presents the results of high Darcy number  $(10^2)$ , fluid flow undergoes less resistance from the boundary friction, and buoyancy induced flow eirculates in the enclosure, even with low thermal Rayleigh number ( $Ra = 10^5$ , Fig. 8(a)). With the increase of thermal Rayleigh numbers, thermal and solutal boundary layers develop greatly as the thermal Rayleigh number promotes to higher ones, typically shown in Figs. 8(b) and 8(c). Significant thermal and solutal stratifications form in the enclosure, and heat/solute from left wall transport toward top and side vents with a shortcut, heat and mass transfer rates on the top and side vents are also boosted (shown in Fig. 9(b)). However, transport rates on the bottom vent varies nonlinearly with the thermal Rayleigh number, firstly increasing and peaking respectively at  $8 \times 10^5$  and  $3.5 \times 10^5$ , subsequently decreasing in absolute values. This could be explained by the fact that the secondary flow eddy just situated in the bottom right corner gradually expands and deflates, which changes the route of convective transports toward bottom vent.

#### 7. Conclusions

The problem of double-diffusive convective flow of a binary mixture inside an enclosure with three vented ports is numerically studied. Finite volume method is

employed on non-uniform grids for the solution of the present problem. The obtained heatlines and masslines, for the combined heat and moisture natural convection in porous medium are shown to be a very effective way to visualize the paths followed by heat and solute through this porous partial enclosure.

Broad range of buoyancy ratios, covering solute-dominated opposing flow, thermal-dominated flow and solute-dominated aiding flow, is examined for several different Darcy numbers. As the permeability of the porous medium is decreased, fluid flow, heat and mass transfer tend to be diffusion dominated. The main contributions of decreasing the Darcy number are predicted to be a flow retardation effect and a suppression of the overall heat and mass transfer in the enclosure.

As Darcy number is low (limiting to Darcy flow), heat and solute are transported to ambient medium through the bottom vent when thermal-dominated flow and solute-dominated aiding flow subsist, otherwise, they are mainly transported toward top and side vents. Whereas, as Darcy number is high (limiting to fluid flow), heat and solute are transported to ambient medium through the top vent when thermal-dominated flow and solute-dominated aiding flow subsist, otherwise, they are mainly transported toward bottom vent. These trends could be intensified as buoyancy ratio increases positively or decreases negatively.

Promoting thermal Rayleigh number could enhance fluid flow and corresponding heat and mass transfer. As the porous thermal Rayleigh number exceeds 10, natural convective fluid flow is enhanced greatly, simultaneously increasing the heat and mass

transfer rates on top and side vents and decreasing those on bottom vent. Due to the multi-cellular flow structures develop in the high Darcy number porous enclosure, heat and mass transfer rates on the bottom vent vary nonlinearly with the thermal Rayleigh numbers.

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#### Table(s)

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Table 1

#### Table 1

Comparison of results with the benchmark solutions for partial cavity natural convection (Pr=1.0)

			Ra					
		$10^{3}$	$10^{4}$	$10^{5}$	$10^{6}$	$10^{7}$		
Chan and Tien (1985a)	Nu	1.07	3.41	7.69	15.00	28.60		
	М	1.95	8.02	21.10	47.30	96.00		
Chan and Tion (1085h)	λ/	1 22			15.00			
Chan and Then (19850)	INU	1.55			13.00	_		
	М	2.65	_	—	47.40	_		
Angirasa et al. (1992)	Nu	—	—	—	14.39	—		
	М	_	—	—	—	—		
Hinojosa et al. (2005a)	Nu	1.28	3.57	7.75	15.11	28.70		
	М	_	_	_	_	_		
Present work	Nu	1.177	3.326	7.562	15.507	31.020		
	Μ	1.664	6.683	17.200	40.364	73.680		

Figure 1



Fig.1 Physical model and Cartesian ordinates of the present study





**Fig.2** Plots of streamlines (top), isotherms (top-1), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0,  $Ra = 10^5$ , and  $Da = 10^{-5}$ .

Figure 3



**Fig.3** Plots of streamlines (top), isotherms (top-1), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0,  $Ra = 10^5$ , and  $Da = 10^{-5}$ .

#### Figure 4



**Fig.4** Plots of streamlines (top), isotherms (top-1), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0,  $Ra = 10^5$ , and  $Da = 10^2$ .

Figure 5



**Fig.5** Plots of streamlines (top), isotherms (top-1), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0,  $Ra = 10^5$ , and  $Da = 10^2$ .

Figure 6



**Fig.6** Overall Nusselt and Sherwood numbers along each port plane as functions of buoyancy ratio with Le = 2.0,  $Ra = 10^5$ , and two limiting Darcy numbers,  $Da = 10^{-5}$  and  $Da = 10^2$  respectively.





**Fig.7** Plots of streamlines (top), isotherms (top-1), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0, N = 2.0, and  $Da = 10^{-5}$ .

#### Figure 8



**Fig.8** Plots of streamlines (top), iso-concentrations (top-2), heatlines (top-3) and masslines (bottom) for double diffusive natural convection with Le = 2.0, N = 2.0, and  $Da = 10^2$ .





**Fig.9** Overall Nusselt and Sherwood numbers along each port plane as functions of thermal Rayleigh numbers with Le = 2.0, N = +2, and two limiting Darcy numbers,  $Da = 10^{-5}$  and  $Da = 10^{2}$  respectively.