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On the importance of thermal boundary conditions in heat transfer and entropy generation for natural convection inside a porous enclosure

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Abstract

The aim of the present paper is to analyze the importance of thermal boundary conditions of the heated/cooled walls in heat transfer and entropy generation characteristics inside a porous enclosure, heated from below. Both the heating and the cooling are carried out uniformly and non-uniformly and the results are compared. The laminar, steady, natural convection heat transfer is calculated by solving numerically the mass, momentum, and energy conservation equations whilst viscous dissipation and the work of pressure forces are included in the energy equation. Moreover, the generation of entropy is calculated taking into account both heat transfer irreversibility and fluid friction irreversibility. As the thermal boundary conditions, sinusoidal temperature distributions are invoked for the non-uniformly heated/cooled walls. Comparison between the results of the present numerical model with the previously published works provides excellent agreement. Results are presented in terms of streamlines, isothermal lines, iso-entropy generation lines, and iso-Bejan lines. Additionally, variations of average Nusselt number, global entropy generation rate, and global Bejan number are analyzed over a wide range of Darcy-modified Rayleigh number ($10 < Ra < 1000$). Inspection of the results indicates that thermal boundary conditions are of profound influences on the induced flow as well as heat transfer characteristics and possess prominent consequences on entropy generation rates. It is demonstrated that, the optimum case with respect to heat transfer as well as entropy generation could be achieved by non-uniform heating.

Keywords: Porous media; Entropy generation; Natural convection; Enclosure; Boundary condition

1. Introduction

Buoyancy driven flows inside porous media occur in diverse applications such as geothermal energy systems, solar collectors, insulation systems, and cooling of radioactive waste containers. Going on to the literature, one can find many works concerning natural convection inside enclosures filled with fluid-saturated porous media with differentially heated vertical walls and insulated horizontal walls (e.g., Moya et al. [1]; Baytas and Pop [2]; Saeid and Pop [3]; Misirlioglu et al. [4]; Badruddin et al. [5]). Nevertheless, there is a noticeable dearth of works concerning natural convection inside porous enclosures under other thermal boundary conditions, especially with walls under non-uniform temperatures.

Yoo [6] and Yoo and Schultz [7] have obtained analytical solution in the context of the natural convection problem in a fluid-saturated porous media between two infinite vertical and horizontal walls, respectively. They invoked sinusoidal temperature distribution for the walls and investigated the influences of wave number and phase indifference on the induced flow as well as heat transfer characteristics. Saeid [8] has analyzed natural convection inside a porous enclosure with partially-heated bottom wall, cooled top wall and adiabatic vertical walls. He demonstrated that average Nusselt number enhances due to the length of heat source and due to the amplitude of temperature variation. Basak et al. [9] have investigated a porous enclosure with uniformly and non-uniformly heated bottom wall, and adiabatic top wall, maintaining constant temperature for the cold vertical walls. Concerning their results, they concluded that, non-uniform heating of the bottom wall provides higher heat transfer rates at the central region of the bottom wall whereas average Nusselt number is lower for non-uniform heating. Re-

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Nomenclature

Be	Bejan number	
Be_{global}	global Bejan number	
c_p	constant pressure specific heat	J/kg K
Ec	Eckert number	
FFI	fluid friction irreversibility	
g	gravitational acceleration	m/s^2
HTI	heat transfer irreversibility	
K	permeability	m^2
L	enclosure height	m
N	local entropy generation rate	
N_{global}	global entropy generation rate	
Nu	local Nusselt number	
$\overline{Nu}$	average Nusselt number	
p	pressure	Pa
Ra	Darcy-modified Rayleigh Number	
T	temperature at any point	K
T_C	temperature of the cold wall	K
T_H	temperature of the hot wall	K
ΔT	temperature difference, $\Delta T = T_H - T_C$	K
u, v	velocity components in x and y directions	m/s
x, y	Cartesian coordinates	m
X, Y	dimensionless coordinates	

Greek symbols

α	effective thermal diffusivity	m^2/s
β	volumetric expansion coefficient	K^{-1}
ν	kinematic viscosity	m^2/s
ρ	density	kg/m^3
ψ	streamfunction	m^2/s
Ψ	dimensionless streamfunction	
Θ	dimensionless temperature	
τ	dimensionless temperature difference, $\Delta T/T_C$	

cently, Oztop [10] has discussed the problem of partially cooled and inclined porous enclosure with one side wall at constant hot temperature, one adjacent wall being partially cooled, and the remaining ones adiabatic. He found that inclination angle is the dominant parameter on heat transfer and fluid flow.

All of the aforementioned studies are the First-Law (of thermodynamics) analyzes of non-uniformly heated porous enclosures. Recently, Second-Law based investigations have been utilized to study entropy generation. This could gauge the significance of irreversibility related to heat transfer, fluid friction, and other non-ideal processes within thermal systems and enables us to find thermal boundary conditions where by minimum irreversibility is produced. Consequently, cares needs to be taken to the problem of entropy generation in porous enclosures under non-uniform wall temperatures. Nevertheless, the literature survey carried out by the author reveals the subject has not been investigated thus far. The available works in entropy generation associated with natural convection heat transfer inside porous enclosures go back to the investigations conducted by Baytas [11] and Mahmud and Fraser [12]. Baytas [11] has examined an inclined enclosure with differentially heated vertical walls and insulated horizontal walls and concluded that as Darcy-modified Rayleigh number decreases, heat transfer irreversibility begins to dominate fluid friction irreversibility. Mahmud and Fraser [12] have analyzed the problem of entropy generation in the magnetohydrodynamic natural convective flow inside a porous enclosure with thermal boundary conditions similar to Baytas [11]. They found that as the magnetic force is introduced, entropy generation rate decreases.

In both of the aforementioned investigations, viscous dissipation term was taken into account in the equation of entropy generation whereas it was not included in the energy conservation equation. On a similar tack, in all of the previous works concerning viscous dissipation term in the energy equation, addition of the term has made the enclosure as a heat multiplier which violates the First Law of Thermodynamics [5,13]. Recently, Costa [14] has proposed that in order to have an energy conservation formulation compatible with the First Law, the work of pressure forces must also be taken into account if the viscous dissipation term is going to be included in the energy equation. This occurs since in natural convective flows, viscous dissipation results from fluid motion which is due to the work of pressure forces involved in the expansion–contraction cycle experienced by the fluid.

In the view of the above, this study investigates a porous enclosure in which the bottom wall is heated and the top wall is thermally insulated; whereas the two vertical walls are cooled. Attention will be focused in the present paper to analyze the importance of thermal boundary conditions of the heated/cooled walls in the development of flow, heat transfer, and entropy generation. Both the heating and the cooling are carried out uniformly and non-uniformly and the results are compared. The governing flow and energy equations are solved numerically whilst viscous dissipation and the work of pressure forces are included in the energy conservation equation. Moreover, the rate of entropy generation is calculated taking into account both heat transfer irreversibility and fluid friction irreversibility.

2. Mathematical formulation

Natural convective flow inside an enclosure filled with fluid-saturated porous media is concerned as depicted in Fig. 1. Here, the vertical walls are cooled uniformly or non-uniformly. The top wall is maintained adiabatic. In addition, uniform or non-uniform heating is concerned for the bottom wall. In order to facilitate the solution of the governing equations, several assumptions are adopted. These assumptions include

- The fluid is ideal gas.
- The developed flow is laminar.
- The porous media is isotropic and possesses homogeneous permeability.

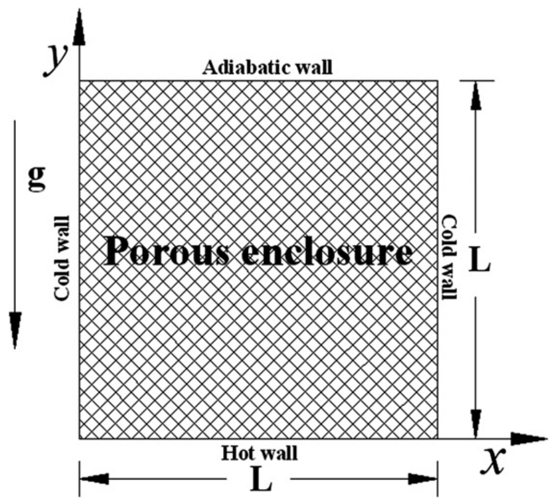

Fig. 1. Physical model of the 2D porous enclosure.

- There is local thermal equilibrium between the medium and the fluid.
- Fluid flow inside the fluid-saturated porous medium is governed by the Darcy's law.
- The fluid physical properties are constant except the density in the body force term in the momentum equation for which the Oberbeck–Boussinesq approximation is invoked.

Under these assumptions and by using the averaged temperature of the cold walls as the reference temperature, the equations of continuity, momentum, and energy are reduced to the following dimensionless form in terms of streamfunction (Ψ) and temperature (Θ)

$$\frac{\partial^2\Psi}{\partial X^2}+\frac{\partial^2\Psi}{\partial Y^2}=-Ra\frac{\partial\Theta}{\partial X} \quad (1)$$

$$\frac{\partial\Psi}{\partial Y}\frac{\partial\Theta}{\partial X}-\frac{\partial\Psi}{\partial X}\frac{\partial\Theta}{\partial Y}=\frac{\partial^2\Theta}{\partial X^2}+\frac{\partial^2\Theta}{\partial Y^2}+Ra\frac{Ec\,Pr}{Da}\frac{1}{\beta\Delta T}\frac{\partial\Psi}{\partial X} \quad (2)$$

Here, the last term in Eq. (2) represents viscous dissipation and the work of pressure forces combined to each other by the assumption of ideal gas. The corresponding dimensionless variables are

$$(X,Y)=\left(\frac{x}{L},\frac{y}{L}\right),\quad (U,V)=\left(\frac{u}{\alpha/L},\frac{v}{\alpha/L}\right),\quad Pr=\frac{\nu}{\alpha}$$
$$Da=\frac{K}{L^2},\quad Ra=\frac{g\beta\Delta T K L}{\nu\alpha},\quad Ec=\frac{(\alpha/L)^2}{c_p\Delta T},\quad \Psi=\frac{\psi}{\alpha}$$
$$\Theta=\frac{T-T_C}{T_H-T_C},\quad \tau=\frac{\Delta T}{T_C} \quad (3)$$

Moreover, dimensionless streamfunction, Ψ, is defined by

$$\frac{\partial\Psi}{\partial y}=U \quad\text{and}\quad \frac{\partial\Psi}{\partial x}=-V \quad (4)$$

In the cases under consideration, Eqs. (1) and (2) are subjected to the following boundary conditions

$X=0$ and $0<Y<1$:

$$\Psi=0 \text{ and } \Theta=0 \text{ or } \Theta=1-\sin(\pi Y) \quad (5a)$$

$X=1$ and $0<Y<1$:

$$\Psi=0 \text{ and } \Theta=0 \text{ or } \Theta=1-\sin(\pi Y) \quad (5b)$$

$0<X<1$ and $Y=0$:

$$\Psi=0 \text{ and } \Theta=1 \text{ or } \Theta=\sin(\pi X) \quad (5c)$$

$$0<X<1 \text{ and } Y=1:\quad \Psi=0 \text{ and } \partial\Theta/\partial Y=0 \quad (5d)$$

Here, the sinusoidal thermal boundary conditions correspond to non-uniform heating or cooling cases. The quantities of physical significance are the local and average Nusselt numbers at the non-adiabatic walls. These quantities are calculated as

$$Nu=-\left[\frac{\partial\Theta}{\partial Y}\right]_{Y=0} \quad\text{and}\quad \overline{Nu}=\int_0^1 Nu\,\mathrm{d}X \quad (6)$$

for the bottom wall and as

$$Nu=-\left[\frac{\partial\Theta}{\partial X}\right]_{X=0,1} \quad\text{and}\quad \overline{Nu}=\int_0^1 Nu\,\mathrm{d}Y \quad (7)$$

for the vertical walls.

3. Entropy generation

Following a similar non-dimensionalization procedure, dimensionless form of the entropy generation equation becomes

$$N(X,Y)=\frac{\tau}{(\Theta\tau+1)^2}\left[\left(\frac{\partial\Theta}{\partial X}\right)^2+\left(\frac{\partial\Theta}{\partial Y}\right)^2\right]+\frac{Ec\,Pr}{Da}\frac{1}{\Theta\tau+1}\left[\left(\frac{\partial\Psi}{\partial X}\right)^2+\left(\frac{\partial\Psi}{\partial Y}\right)^2\right] \quad (8)$$

Here, the first term on the right-hand side is due to the transfer of heat in the direction of finite temperature gradients and is generally termed as heat transfer irreversibility (*HTI*) whilst the second term is the contribution of fluid friction irreversibility (*FFI*).

As the distribution of volumetric entropy generation rate inside the enclosure is obtained, it would be integrated over the whole domain to yield global entropy generation rate

$$N_{\mathrm{global}}=\iint_{\forall} N\,\mathrm{d}\forall=\int_0^1\int_0^1 N(X,Y)\,\mathrm{d}X\,\mathrm{d}Y \quad (9)$$

An alternative irreversibility distribution parameter is Bejan number (*Be*), which is the ratio of heat transfer irreversibility to global entropy generation rate [15]

$$Be(X,Y)=\frac{HTI}{HTI+FFI} \quad (10)$$

$Be \gg 0.5$ is the limit at which, heat transfer irreversibility dominates; $Be \ll 0.5$ is the opposite limit at which, irreversibility is dominated by fluid friction effects; and $Be \sim 0.5$ is the case wherein, *HTI* and *FFI* are of equal importance. Here also, integrating Bejan number yields global Bejan number

$$Be_{\mathrm{global}}=\iint_{\forall} Be\,\mathrm{d}\forall=\int_0^1\int_0^1 Be(X,Y)\,\mathrm{d}X\,\mathrm{d}Y \quad (11)$$

4. Solution procedure

4.1. Numerical scheme

The resulting dimensionless coupled partial differential equations (Eqs. (1) and (2)) are solved simultaneously along with the corresponding boundary conditions (Eq. (5)). The governing equations are converted into a system of algebraic equations through integration over each control volume using the formulation outlined by Patankar [16]. This method assures that conservation laws are obeyed over each control volume. A non-staggered and non-uniform grid is used, with control volume faces lying on the boundaries. The grid expands from the walls to the center with a geometric expansion factor equal to 1.05. The algebraic equations are solved by a line-by-line iterative procedure. The method sweeps the domain of integration along the x and y axes and uses Tri-Diagonal Matrix Algorithm (TDMA) iteration to solve the system of equations. The convergence criterion employed is the following condition

$$\frac{\sum_{i,j} |\phi_{i,j}^{n} - \phi_{i,j}^{n-1}|}{\sum_{i,j} |\phi_{i,j}^{n}|} \leqslant 10^{-6} \tag{12}$$

where ϕ stands for either Θ or Ψ and n denotes the iteration. During this study, the following parameters are retained constant.

$$Ec\,Pr/Da = 0.001; \quad \Delta T = 10\ \text{K}; \quad T_C = 300\ \text{K}$$
$$\beta_C = 1/T_C\ \text{K}^{-1}$$

4.2. Computational grid

To obtain a grid suitable for this study, a grid independence test is performed. The results in terms of overall Nusselt number at the heated bottom wall of the enclosure are listed in Table 1. Here, the uniform heating/cooling case is concerned. Inspection of the results indicates that, a 100×100 grid provides acceptable results for the whole range of Darcy-modified Rayleigh number. Further investigations also verify the suitability of such a grid under non-uniform heating and cooling cases.

5. Results and discussion

Initially, a code validation study is undertaken. In order to resemble the previous investigations, such a study is carried out in a porous enclosure with differentially-heated vertical walls and insulated horizontal walls. Numerical values of the overall Nusselt number in the vertical walls are compared with the previously published works in Table 2. Inspection of the results demonstrates that, results of the present model bear a strong resemblance to the previously published works. This provides confidence to the developed mathematical model and the solution procedure for further studies. Consequently, in the foregoing section, it is used to analyze the development of flow and thermal fields as well as heat transfer characteristics and entropy generation in the natural convective flow inside a porous enclosure heated from below. The importance of thermal boundary conditions in heat transfer and entropy generation is then discussed.

Contour plots of dimensionless streamfunction and temperature for the cases of uniform heating/cooling, non-uniform heating, and non-uniform cooling at $Ra = 100$ are depicted in Fig. 2. Comparison between the results of the three cases of heating/cooling indicates that, thermal boundary conditions are of profound influence on the induced flow as well as thermal fields. As can be observed from the streamfunction contours, due to the cold vertical walls, fluids ascend from the middle portion of the bottom wall and flow down along the two vertical walls forming two symmetric rolls with clockwise and anticlockwise rotations inside the enclosure. Under non-uniform cooling, two smaller roles also exist in the upper part of the enclosure. As shown in Fig. 2, under uniform heating/cooling, contour lines of streamfunction and temperature are very concentrated in the vicinity of the edges of the bottom wall. This is mainly due to high temperature gradients at the edges of the bottom wall which may result in high intensity of fluid flow, heat transfer, and entropy generation there. By contrast, the cases of non-uniform heating and non-uniform cooling have much more uniform streamfunction and temperature distributions over the whole domain. Consequently, these cases have the potential to produce lower heat transfer and entropy generation than the uniform one.

Distributions of local Nusselt number along the heated bottom wall under the three cases of heating/cooling are presented in Fig. 3. Clearly, under uniform heating/cooling, due to high temperature gradients, the value of local Nusselt number is very high at the edges of the wall whilst the heat transfer rate reduces toward the center with nearly uniform values at the central region. By contrast, the cases with non-uniform heating and non-uniform cooling provide much more uniform results. Here, the heat transfer rates increases toward the center, with nearly uniform values at the central region. Inspection of Fig. 3 indicates that at the central region, the greatest heat transfer rates occur under non-uniform heating. This occurs since

Table 1
Grid independence test

Grid size	$Ra = 10$	$Ra = 100$	$Ra = 1000$
200×200	5.732	6.840	12.256
100×100	5.735	6.842	12.250
50×50	5.599	6.736	12.239
25×25	5.183	6.589	12.107

Table 2
Comparison of the average Nusselt number with previously published works

Author	$Ra = 10$	$Ra = 100$	$Ra = 1000$
Moya et al. [1]	1.065	2.801	–
Baytas and Pop [2]	1.079	3.16	14.06
Saeid and Pop [3]	–	3.002	13.726
Misirlioglu et al. [4]	1.119	3.05	13.15
Badruddin et al. [5]	1.0798	3.2005	–
Present simulation	1.067	3.026	13.134

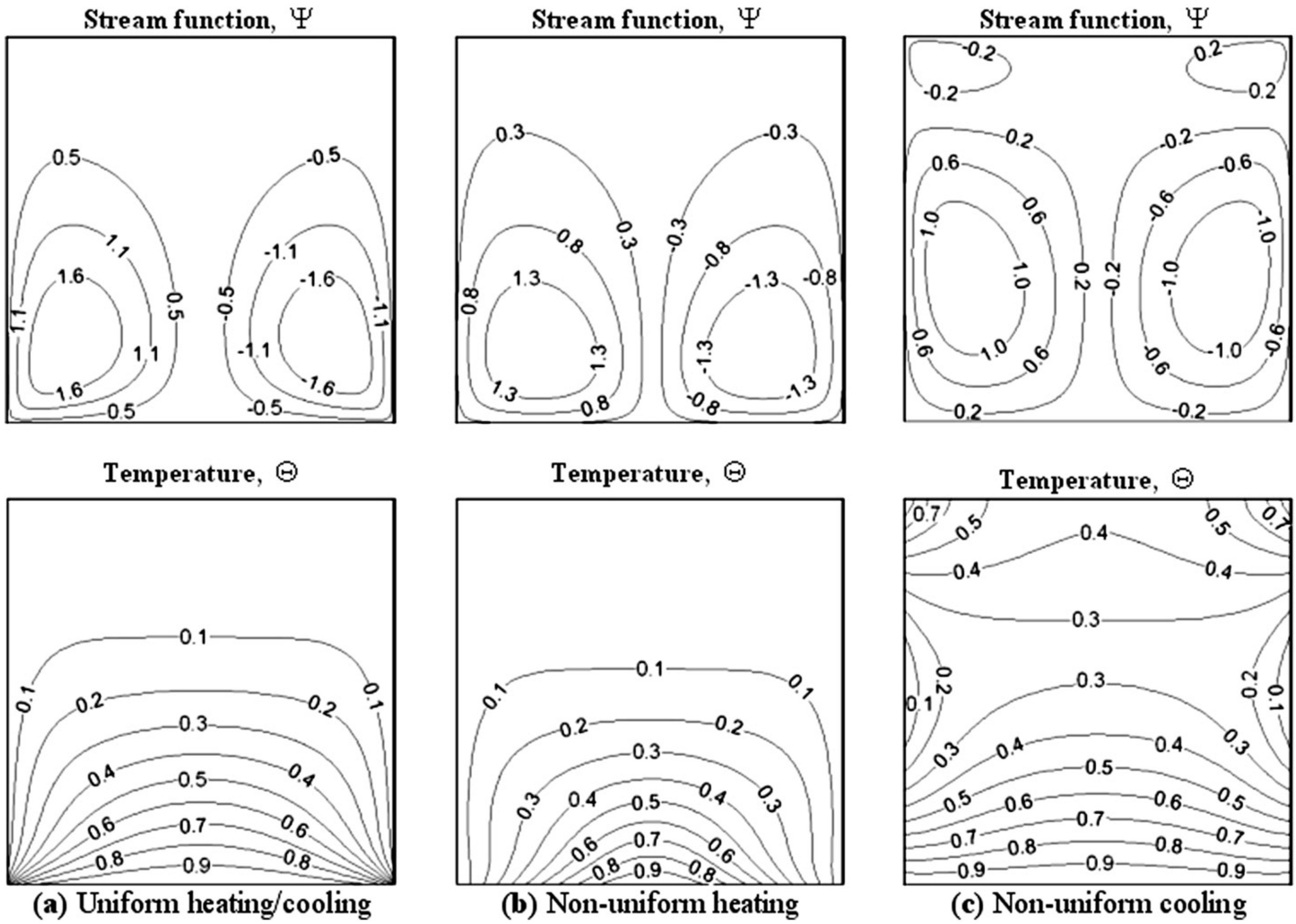

Fig. 2. Distributions of streamlines and isothermal lines at $Ra = 100$.

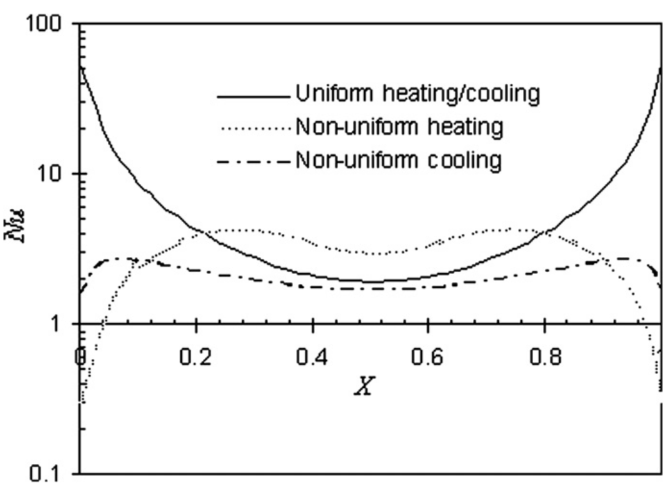

Fig. 3. Distributions of local Nusselt number along the heated bottom wall at $Ra = 100$.

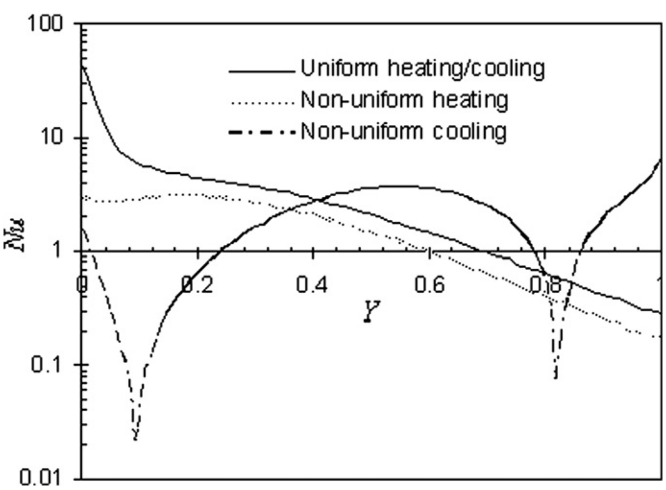

Fig. 4. Distributions of local Nusselt number along the cooled vertical walls at $Ra = 100$.

non-uniform heating establishes the highest temperature gradients there which can be observed in Fig. 2. Such a result confirms that of Basak et al. [9] who compared heat transfer rates under the two cases of uniform heating and non-uniform heating.

Fig. 4 displays the heat transfer rates along the cooled walls. Under uniform heating/cooling, the local Nusselt number is found to be decreasing as we proceed along the height of the vertical walls. Here also, a very high heat transfer rate exists at the bottom edges of the walls. The physical reasoning for this behavior is the reduction of temperature difference between the vertical walls and their adjacent fluid in the higher heights. Similar decreasing trends can also be observed under non-uniform heating case for $Y > 0.25$. Variation of local Nusselt number is completely different under the non-uniform cooling case. Here, temperature difference between the walls and their adjacent fluid is high in the central and the upper parts of the vertical walls which enhance heat transfer rates, there.

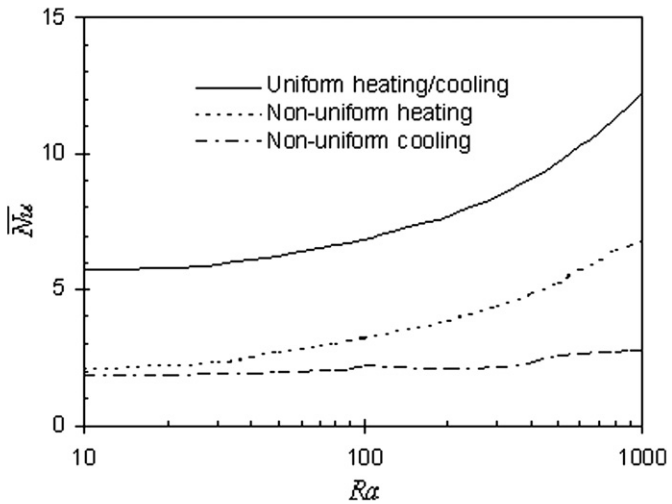

Fig. 5. Variations of average Nusselt number according to Darcy-modified Rayleigh number.

The overall effects of the three cases upon heat transfer rates are compared in Fig. 5. Here, variations of the average Nusselt number at the heated wall is plotted against Darcy-modified Rayleigh number under the three cases of heating/cooling. As expected, the uniform heating/cooling achieves the highest heat transfer rates. It is also observed that, due to the enhanced convection regime, increase in *Ra* raises the heat transfer rates both for uniform heating/cooling and non-uniform heating. In spite of this, under non-uniform cooling, the value of average Nusselt number remains nearly constant in the whole range of Darcy-modified Rayleigh number.

In what concerns fluid friction irreversibility and heat transfer irreversibility, the results at $Ra = 100$ are presented in Fig. 6. The view of the observer is directed towards the lower half of the vertical walls, as well as, the corner sides of the horizontal wall. These locations act as strong concentrators of fluid friction irreversibility and heat transfer irreversibility due to higher values of near wall velocity components and temperature gradients, respectively. However, a significant portion of the enclosure acts as an ideal region for entropy generation wherein both *FFI* and *HTI* are zero or negligible.

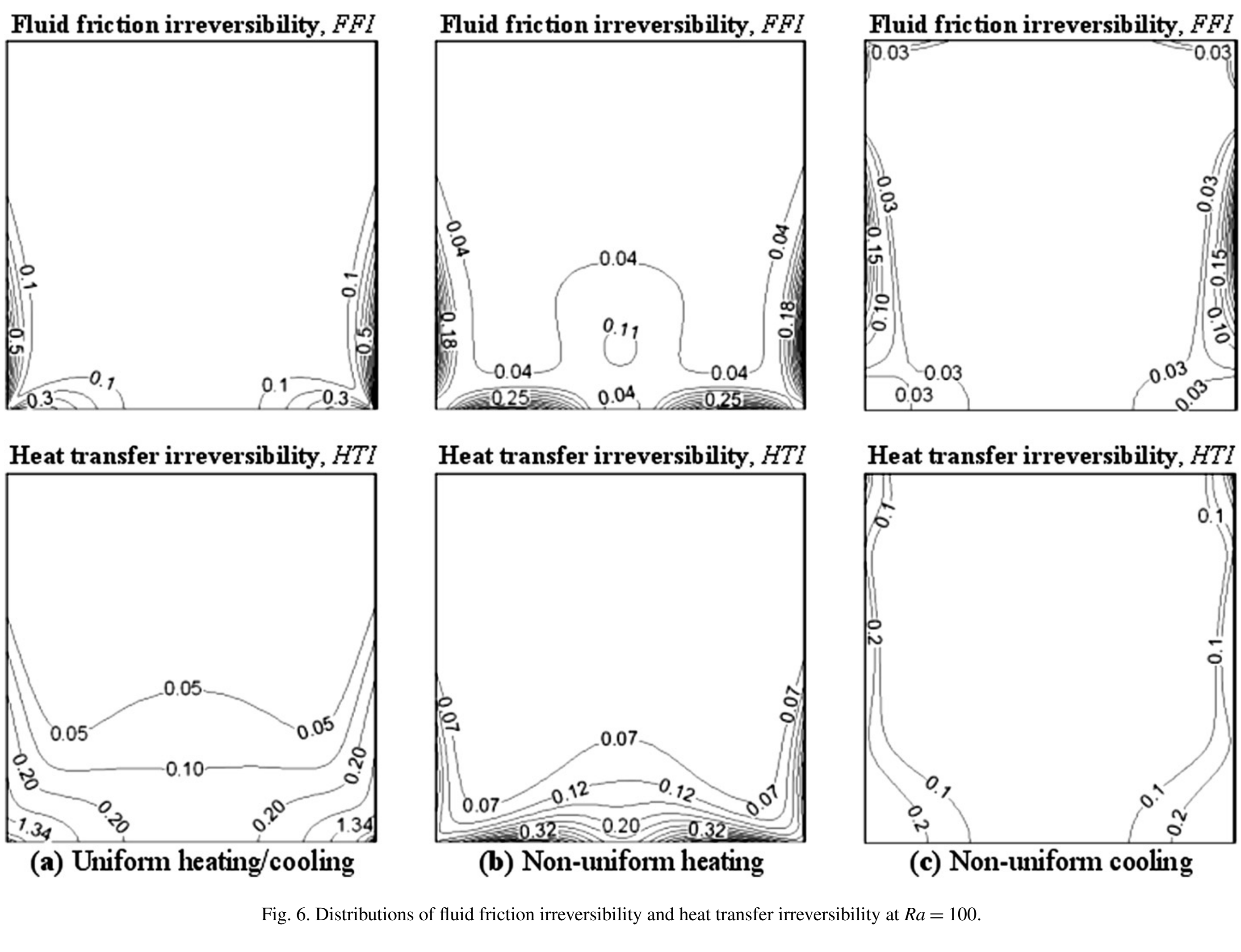

Fig. 6. Distributions of fluid friction irreversibility and heat transfer irreversibility at $Ra = 100$.

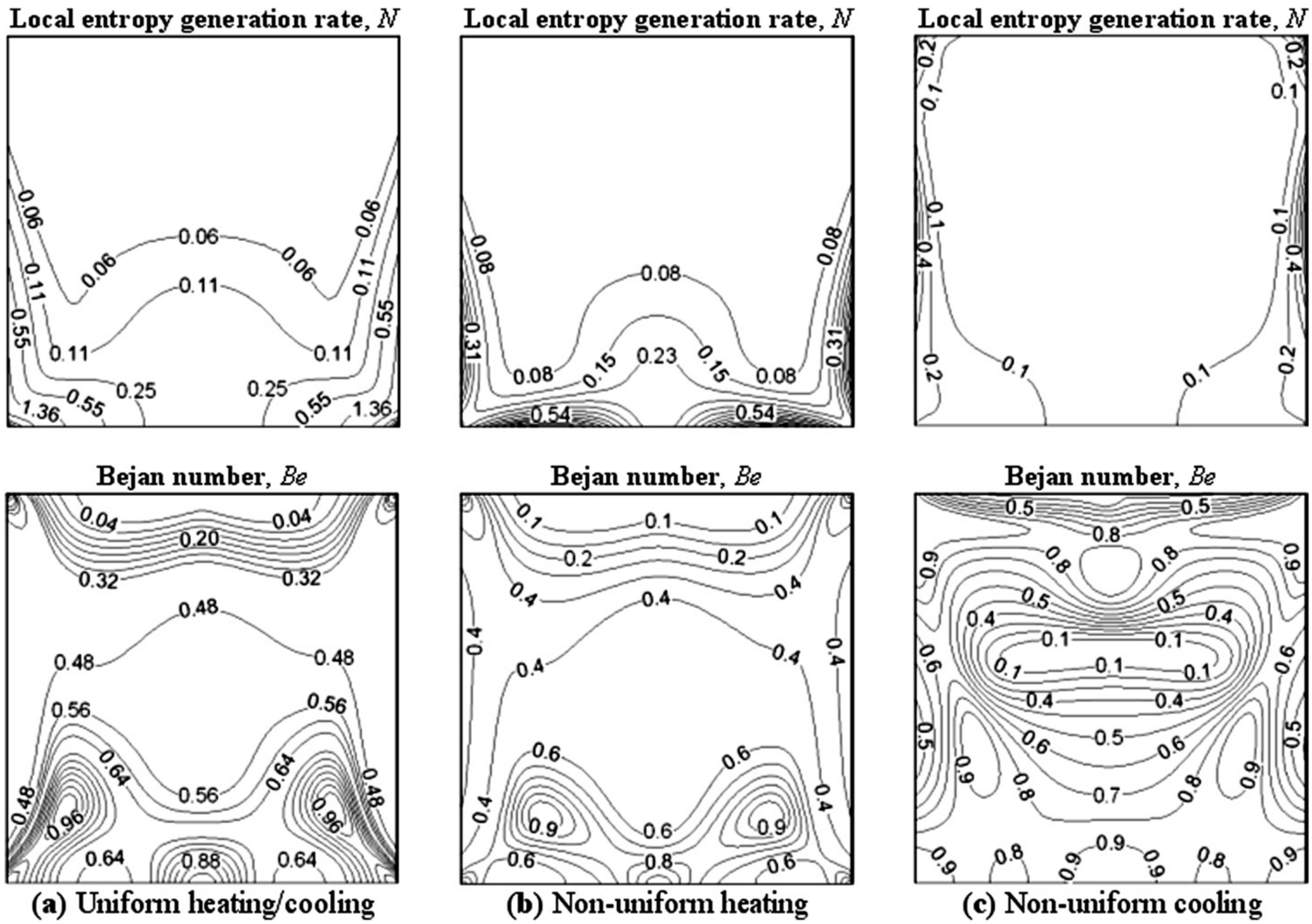

Fig. 7. Distributions of iso-entropy generation lines and iso-Bejan lines at $Ra = 100$.

Such variations for fluid friction irreversibility and heat transfer irreversibility make the distributions of local entropy generation rate (iso-entropy generation lines) as well as local Bejan number (iso-Bejan lines) inside the enclosure as depicted in Fig. 7. Comparison between the results of the three cases of heating/cooling indicates that, thermal boundary conditions possess prominent consequences on entropy generation rates. As can be illustrated with the figure, under all cases, lower walls act as strong concentrators of irreversibility whilst Bejan number spreads over the whole domain.

Fig. 8 shows the distributions of the global entropy generation rate as a function of Darcy-modified Rayleigh number under the three cases of heating/cooling. In a general way, notice that uniform heating/cooling suffers from the highest values of entropy generation rates whereas the lowest rates of entropy generation are achieved by non-uniform heating. It is also observed that, due to the enhanced induced flow and heat transfer rate, increase in Ra always raises the global entropy generation rates.

As observed from Figs. 5 and 8, although uniform heating/cooling achieves the highest heat transfer rates, it suffers from the highest rates of entropy generation. From the standpoints of the First Law and the Second Law (of thermodynamics), an efficient thermal boundary condition for the enclosure must lead to high heat transfer rates as well as low rates of entropy generation. Consequently, the performances of the three cases of heating/cooling can be evaluated in terms of $\overline{Nu}/N_{\mathrm{global}}$ which is the ratio of average Nusselt number to global entropy generation rate. Fig. 9 illustrates the variations of the parameter according to Darcy-modified Rayleigh number under the three

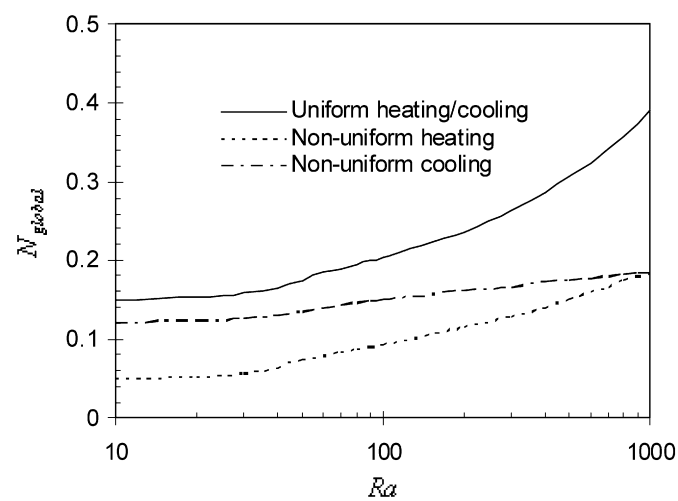

Fig. 8. Variations of global entropy generation rate according to Darcy-modified Rayleigh number.

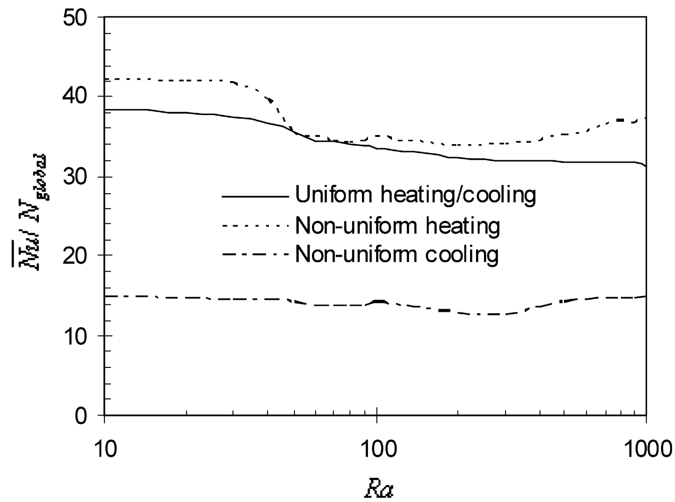

Fig. 9. Variations of the ratio of average Nusselt number to global entropy generation rate according to Darcy-modified Rayleigh number.

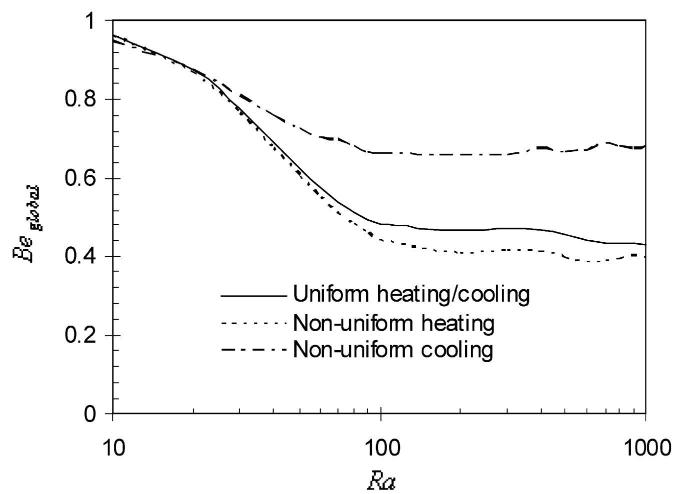

Fig. 10. Variations of global Bejan number according to Darcy-modified Rayleigh number.

cases of heating/cooling. Concerning this figure, it can be concluded that, non-uniform heating attains the highest values of $\overline{Nu}/N_{\text{global}}$ and achieves the optimum case with respect to heat transfer as well as entropy generation. Thermal performance of the uniform heating/cooling closely follows the non-uniform heating and provides satisfactory results. Nevertheless, the non-uniform cooling is found inappropriate and must be avoided both from the First Law and the Second Law points of view.

Finally, variations of global Bejan number corresponding to *Ra* are displayed in Fig. 10. At low Darcy-modified Rayleigh numbers, conduction mode dominates. Therefore, most of the contribution on the overall entropy generation comes from heat transfer irreversibility and Bejan numbers are almost equal to unity. As *Ra* increases, the induced flow accelerates and *FFI* tends to *HTI* asymptotically. Consequently, dramatic falls of Bejan numbers are observed up to $Ra \sim 100$ wherein the numbers approach to about 0.5. As can be witnessed, further increases in Darcy-modified Rayleigh number possess minor influences on Be_{global} and the number remain nearly constant for $Ra > 100$.

6. Concluding remarks

The importance of thermal boundary conditions of the heated/cooled walls in heat transfer as well as entropy generation characteristics inside a porous enclosure, heated from below was investigated here. Both the heating and the cooling were carried out uniformly and non-uniformly and the results were compared. Based on the presented results, the following conclusions may be drawn

- Thermal boundary conditions are of profound influence on the induced flow as well as heat transfer characteristics and possess prominent consequences on entropy generation rates.
- Uniform heating/cooling achieves the highest heat transfer rates whereas non-uniform cooling suffers from the lowest.
- Entropy generation rate is likely to be the highest for uniform heating/cooling and the lowest for non-uniform heating.
- The optimum case with respect to heat transfer and entropy generation could be achieved by non-uniform heating.

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