A Numerical Investigation of the Free Flow in a Square Porous Cavity with Non-Uniform Heating on the Lower Wall

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Abstract-Natural convection in a steady state of incompressible air inside a cavity's porous with a heated low wall of a sinusoidal profile is investigated numerically in this paper. The upper horizontal wall is kept cold while the two sides are thermally insulated. The proposed physical model was developed and studied with two-dimensional conditions, using the finite element method and adapting the Darcy-Brinkman model. This paper examines the laminar natural convection in a square porous cavity for different Rayleigh numbers ($10 \leq Ra \leq 10^4$), aspect ratios ($0.25 \leq AR \leq 1.0$), and sinusoidal temperature amplitude ($0.25 \leq \lambda \leq 1.0$). Moreover, the variation effect of Ra, AR, and λ on isotherms, streamlines, and the mean and local Nusselt numbers has been presented and analyzed. The results showed that an increase in the sinusoidal thermal amplitude, mean Nusselt number, and AR reduced somewhat the Rayleigh number. This provided a solution in which the mean Nusselt number increased by increasing the sinusoidal thermal amplitude and the Rayleigh number. On the other hand, it decreases slightly by increasing the AR. In addition, the convection transfer mechanism is the main mode when the Rayleigh number is high. Thus, it was found that the Darcy number also has an effect on heat transmission. The obtained results were compared with those found in the literature and were found to be in good accordance.

I. INTRODUCTION

It is observed that the heat exchange can be improved by using porous media. The term "porous medium" is generally referred to the solid complex shape which contains cavities or interstitial voids accessible to a fluid. In a porous medium multiple physicochemical and transport phenomena take place. It was often considered as a homogeneous and isotropic medium, nevertheless, it was considered anisotropic in its mechanical and thermal properties in various practical applications. Indeed, the field of porous environments is a part of a very spacious and quite complex research area which requires detailed and reliable understanding and knowledge. For a scientific survey on porous media, see [1-9]. The interstitial fluid flow in a porous medium is caused by convective heat transfer. This type of convection can be caused by different natural forces like the gravity force and the variation in density caused by the temperature difference. Low-temperature gradients affect the mass and heat transport that occurs in a porous medium. This type of convection is particularly important in a wide range of industrial and natural

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processes. It is present in the exchange of moisture between the soil and the atmosphere. As a result of the daily and seasonal temperature variation of the surface, it is also present in certain technical fields, but we can find it as well in some biological and biotechnological systems [10]. Recently, authors in [11] proposed a three-dimensional model to study the effect of external vibrations on the drying process of a porous material.

In order to contribute to a more scientifically satisfactory study of the mechanism of heat transmission in a porous medium, numerical studies using the finite element method where it uniformly heats one of the side walls have been reviewed in [12-13]. Other researchers [14-16] applied heat both uniformly and non-uniformly to the bottom wall, whereas the sides were kept at a constant temperature and the top wall was considered adiabatic. Their interest was to see the impact of thermal boundary conditions that are both continuous and discontinuous. The Rayleigh number and the Darcy number have a wide range of values. It was discovered that when the Rayleigh number grew larger, the convective transfer mechanism took over as the major mode of transfer and that the Darcy number has an effect on heat transmission as well. In a similar model, authors in [17] heated both sidewalls with sinusoidal temperature while making use of the Darcy-Brinkman-Forchheimer model. Another shape was tested by the authors in [18] who studied the impact of inclined and sinusoidally heated wall in the trapezoidal cavity. The flow and heat transfer also were affected by the inclination of the walls. Authors in [19] investigated the convection of an inclined porous cavity at various inclination angles. The laminar, monophasic, and steady flows of water and air through cracks with penetrable walls were investigated in [20]. In order to describe heat and mass flows in porous media the authors in [21] employed a CVFEM technique. Others heated the side wall differentially [22-26] and the results were quite similar. Authors in [27] used heat generation. They verified the impact of porosity and heat generation on the streamlines and isotherms. According to the authors in [28], the ratio of the cavity's length to its width greatly influences streamlines and isotherms. Authors in [29] numerically analyzed the influence of the aspect ratios with Rayleigh numbers smaller than 10^4. They discovered that the heat transfer rate rises with the raising of the aspect of the cavity. However, the authors in [30] measured the partially heated porous enclosure and they observed that raising the aspect ratio causes the flow function to increase and the heat transfer rate to cut back on.

Although sinusoidal heating in porous surfaces has important implications for research and engineering, there are still gaps in this area. The purpose of the current study is to improve the thermal performance by using a new temperature function (4). It is much more logical, rational and realistic, it can be found in different natural and industrial processes, such as solar energy and cooling of electronic components, where thermal heating takes the form of a cosine function, that is to say, the heating is concentrated in the middle and then distributed to the periphery. We will examine bottom-heated heat transfer by natural convection. The cavity is cooled from above, while the side walls are adiabatic. The main objective is to test this function on heat transfer and streamlines as a function of the Rayleigh number, the magnitude of the temperature profile, and the aspect ratio. Our results are compared with the ones in [16].

II. MATHEMATICAL FORMULATION

The finite element method was utilized in laminar, stationary and two-dimensional conditions. The natural incompressible air circulation within a cavity of height H and length L (see Figure 1), is simulated numerically. The cavity with impermeable walls is filled by an isotropic porous medium, homogenous and thermodynamically balanced with the saturating fluid, where the high temperature of the cavity is maintained cold (Tc). The low wall is sinusoidally heated (Th), whereas the two vertical walls are considered adiabatic.
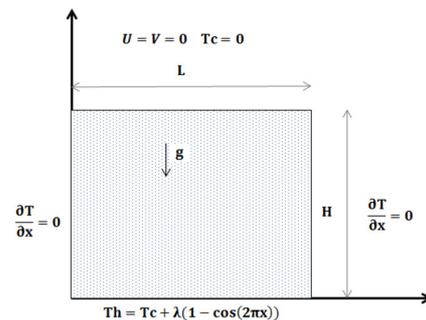

Fig. 1.   Boundary conditions of cavity geometry.

In the present investigation, the interactions between mass and heat transfer (Soret and Dufour effects), as well as heat transfer by radiation, are neglected. However, the thermal and fluid's physical characteristics are expected to remain constant. Boussinesq's approximation stands. Using these hypotheses and the Darcy–Brinkman model, the dimensionless equations governing this problem are [27]:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (1)$$

$$\frac{\partial U\Omega}{\partial X} + \frac{\partial V\Omega}{\partial Y} = \left(\frac{\partial^2}{\partial X^2} + \frac{\partial^2}{\partial Y^2} - \gamma \right)\Omega + \frac{\partial \theta}{\partial X} \frac{Ra}{Pr} \quad (2)$$

$$\frac{\partial U\theta}{\partial X} + \frac{\partial V\theta}{\partial Y} = \frac{1}{Pr}\left(\frac{\partial^2}{\partial X^2} + \frac{\partial^2}{\partial Y^2} \right)\theta \quad (3)$$

where, θ, γ, and Ω are the dimensionless temperature, the porosity parameter, and dimensionless vortices, respectively. The other physical parameters are mentioned in the nomenclature. The control parameters of this problem AR, Pr, Ra, and Da are defined as follows:

$$AR = \frac{H}{L} , \ Pr = \frac{\upsilon}{\alpha} , \ Ra = \frac{g \, K \beta \, H \, \Delta T}{\upsilon \, \alpha} = Gr.Pr ,$$

$$\text{and } Da = \frac{K}{H^2} .$$

The stream function is defined as:

$$U = \frac{\partial \psi}{\partial Y} \ \text{and} \ V = -\frac{\partial \psi}{\partial X} .$$

Dimensionless variables are now included in the dimensionless equations:

$$Y = y / H \,,\ X = x / L \,,\ V = v / V_0 \,,\ U = u / U_0 \,,$$

$$P = \frac{p}{\rho U_0^2} \,,\ \text{and}\ \theta = \frac{T - T_c}{T_H - T_c} \,.$$

In order to improve the performance of heat transfer within the porous medium, we have studied a new function totally absent in the literature, and yet widely encountered in natural and industrial processes, where thermal heating takes the form of the cosine function, i.e. the heating is concentrated in the middle and then distributed to the periphery. The boundary conditions of the current research are:

For the vertical walls:

$$X = 0 \text{ and } 0 \le Y \le H \,,\ U = V = 0 \,, \frac{\partial \theta}{\partial X} = 0$$

$$X = L \text{ and } 0 \le Y \le H \,,\ U = V = 0 \,, \frac{\partial \theta}{\partial X} = 0$$

For the low wall:

$$Y = 0 \ \text{and} \ 0 \le X \le L \,, U = V = 0 \,,$$

$$\theta = \lambda \big(1 - \cos(2\pi x) \big)$$

For the high wall:

$$Y = H \ \text{and} \ 0 \le X \le L \,,\ U = V = 0 \,, \theta = 0$$

The low wall is heated as the sinusoidal formula in (4).

$$\theta = \lambda \big(1 - \cos(2\pi x) \big) \quad (4)$$

In the level of this wall, the average and local Nusselt number are:

$$\overline{Nu} = \int_0^1 Nu(X) \, dX$$

$$\overline{Nu} = \int_0^1 Nu(X) \, dX \ \text{and} \ Nu(X) = -\frac{\partial \theta}{\partial Y} \bigg|$$

III. RESULTS AND DISCUSSION

In the experimental stage, we created an appropriate grid for a numerical model of the natural convection of a square cavity filled with porous material and saturated with incompressible air. Four varieties of uniform grids were designed, namely 30×30, 40×40, 50×50, and 60×60. The values AR=1.0, λ=0.5, and Ra=1000 were fixed as well as the tested function introduced in [16]: $Th = Tc + \lambda \sin(2\pi x)$. We found that the evenly spaced 60×60 mesh grid is fine enough to be used throughout the simulation. Our own result of the mean Nusselt number was 12.03, however, the result found in [16] was 13.56, giving a good agreement between the two. Then, we calculated the Nusselt number using our function in (4), and we found 18.91, which is it sufficiently larger than the one found in [16]. The isotherms, the streamlines and the local Nusselt number with fixed values of λ=0.5, AR=1.0 and varying values

of the Rayleigh number $500 \le Ra \le 1000$ can be seen in Figures 2-4. We have obtained the same results as in [16].

TABLE I.   NUSSELT NUMBER COMPARISON

Reference	[13]	[31]	[16]	This study
$\overline{Nux}$	15.80	14.06	13.56	12.03

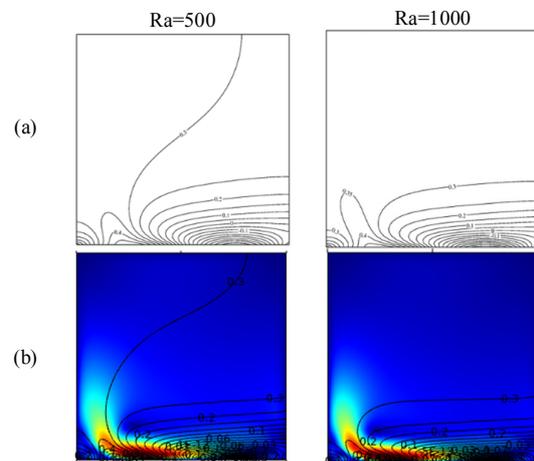

Fig. 2.   Isotherms for AR=1.0, λ=0.5 and $500 \le Ra \le 1000$: (a) [16], (b) current study.
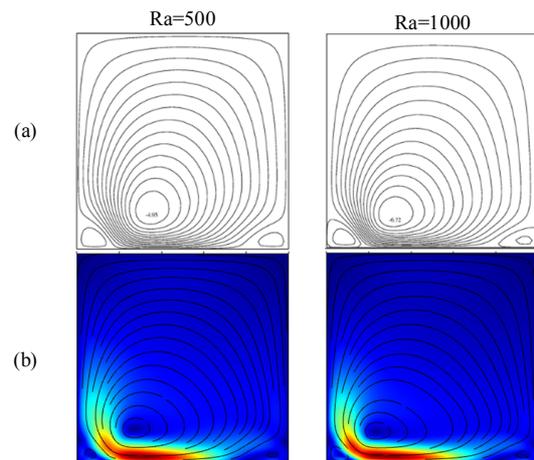

Fig. 3.   Streamline isotherms for AR=1.0, λ=0.5 and $500 \le Ra \le 1000$: $500 \le Ra \le 1000$: (a) [16], (b) current study.

The numerical results obtained from the development of laminar natural convection in steady-state when an isotropic porous medium fills the square cavity, assuming that the lower wall is heated in a non-uniform way whereas the upper wall is cooled to a consistent temperature and the two vertical walls are thermally insulated. As mentioned above, we fixed the Prandtl number, porosity and permeability values, while we tested various Rayleigh values, aspect ratios, and amplitudes of the sinusoidal temperature. The temperature distribution and streamlines due to the buoyancy force effect are analyzed numerically, for different shape ratios $0.25 \le AR \le 1.0$. In each cavity shape ratio, the influence of the temperature amplitude $0.25 \le \lambda \le 1$ and the Rayleigh number $10 \le Ra \le 10000$ were studied regarding heat transfer and streamlines. The current

line results are shown in Figures 5-7 and the isotherms in Figures 8-10. It should be noted that the streamline contours are defined by two symmetrical vortices that occupy the whole cavity and rotate clockwise and counterclockwise for all the cases of aspect ratio. The hot fluid is driven from down along the bottom heated wall, under the influence of buoyancy. Buoyancy leads to circulation inside the cavity. However, close to the cavity's center, the circulation is more important in relation to the boundary of the enclosure due to the condition of non-slip.
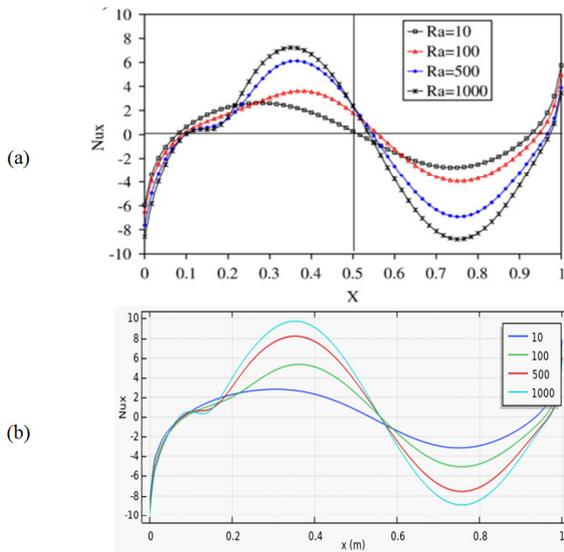
(a)

(b)

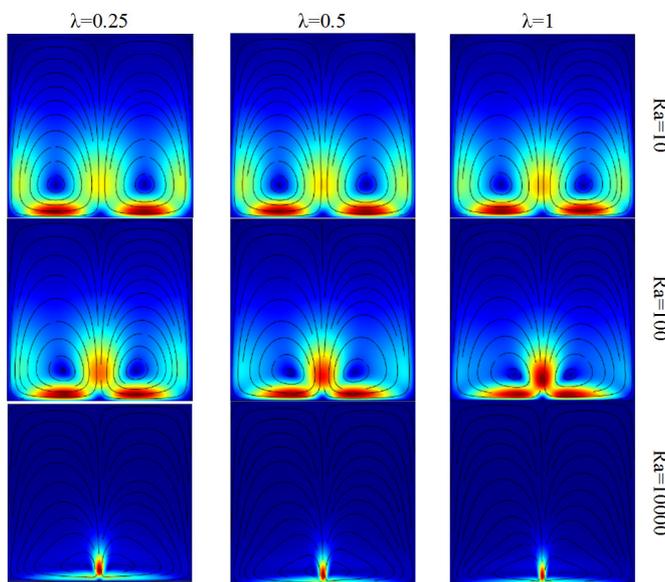Fig. 4.   Local Nusselt number. (a) [16], (b) this study.

λ=0.25   λ=0.5   λ=1

Ra=10

Ra=100

Ra=10000

Fig. 5.   Streamlines for $10 \leq Ra \leq 10000$, $0.25 \leq \lambda \leq 1$, and *AR*=1.
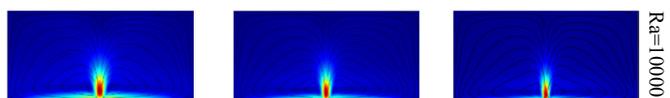

Ra=10000

Fig. 6.   Streamlines for $Ra = 10000$, $0.25 \leq \lambda \leq 1$, and *AR*=0.5.
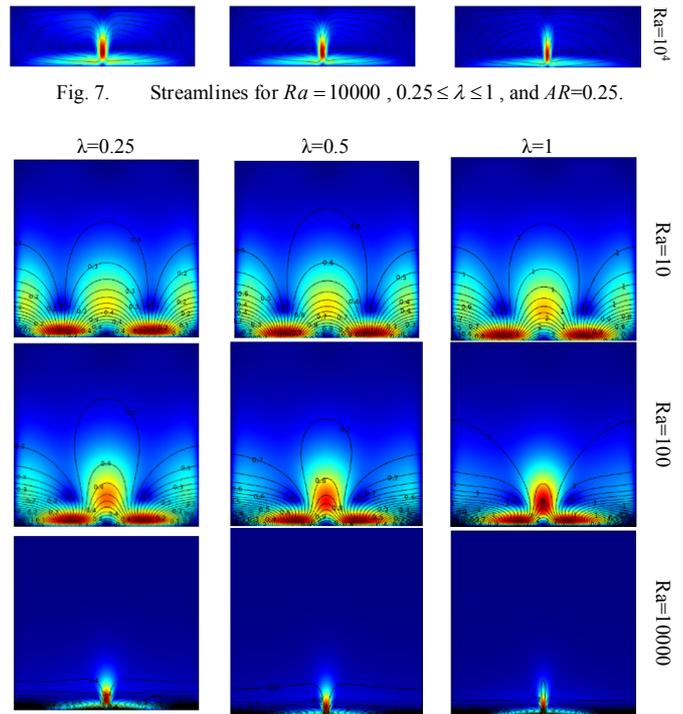
Ra=10⁴

Fig. 7.   Streamlines for $Ra = 10000$, $0.25 \leq \lambda \leq 1$, and *AR*=0.25.

λ=0.25   λ=0.5   λ=1

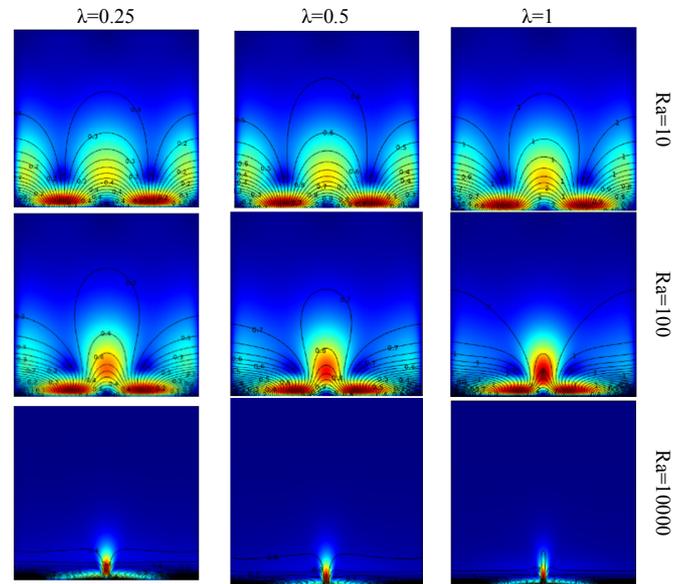Ra=10

Ra=100

Ra=10000

Fig. 8.   Isotherms for $10 \leq Ra \leq 10000$, $0.25 \leq \lambda \leq 1$, and *AR*=1.
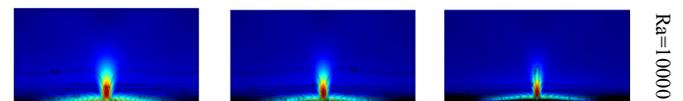
Ra=10000

Fig. 9.   Isotherms for $Ra = 10000$, $0.25 \leq \lambda \leq 1$, $0.25 \leq \lambda \leq 1$, and *AR*=0.5.
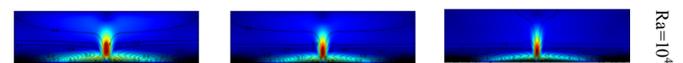
Ra=10⁴

Fig. 10.   Isotherms for $Ra = 10000$, $0.25 \leq \lambda \leq 1$, $0.25 \leq \lambda \leq 1$, and *AR*=0.25

The center of the two cells is almost in the middle. The distance between the center and the hot wall is less than that between the center and the cold wall. As the Rayleigh number rises, the center of the cells shifts to the middle of the bottom wall. As a result of the conduction mode's dominance, the circulation force is very weak. The impact of the Rayleigh number on natural convection flow is a critical parameter. The flow convection is insignificant when the Rayleigh number is low. Conduction is the primary mode of heat transmission in the cavity. However, we notice a slight modification on the streamlines when we change the value of the sinusoidal temperatures amplitude. The flow forces as well as the values of the function rise when the Rayleigh number is high, which explains why the flow rate also increases. The current lines are shown in Figures 5-7 for different shapes ratios. It is obvious that the flow force becomes powerful and that the convection heat transfer is the most dominant to conduction. The increase in the value of the aspect ratio generates an increase in the flow force. We observe that when the shape ratio is decreased, the two cells become depressed. When the aspect ratio is high, the flow velocity of the fluid is higher. As a result, when the

distance between the two cold and hot walls is short, the heat exchange rate in the cavity is higher. At low Rayleigh number values, the temperature distribution is in the form of three sets of undulations symmetrical with relation to the vertical median of the cavity. Its amplitude in the middle is higher with a higher temperature gradient by compared to the other two next to it. On the other hand, we see a slight upward stretch of the ripples as we increase the value of the sinusoidal temperature distribution amplitude. The flow force increases as a result of an increase in the Rayleigh number and the heat transfer by natural convection becomes more dominant than the conduction, which is greatly influenced by the decrease in the boundary layer at the level of the wall hot. We notice that the isotherms become denser near the hot wall and more uniform in the central areas due to the enhanced free convection effect with the increase in the Rayleigh number so that the heat transfer is intensified and the temperature contour thins and condenses at the bottom wall approximation while the ripple turns into plumes.

Figure 11 depicts the effect of Ra and λ on the local Nusselt number Nux at the lower wall. In Figure 11(a) we considered $AR = 1$ and $\lambda = 1$, and in (b) we considered $AR = 1$ and $Ra = 1000$. The plot shows the acquired local Nusselt numbers as a function of Ra and λ. The curves are negative and divergent on the two edges and increase their value while going towards the center $X = [0,4-0, 6]$. Then they concave to the vertical mean where $X = 0.5$. The Nux is positive when the heat is supplied from the cavity to the environment and negative when heat is supplied from the enclosure to the environment. It is also observed that the effect of the height/width ratio is really negligible on Nux, The latter, on the other hand, is very changeable depending to Rayleigh and the amplitude of the sinusoidal function.
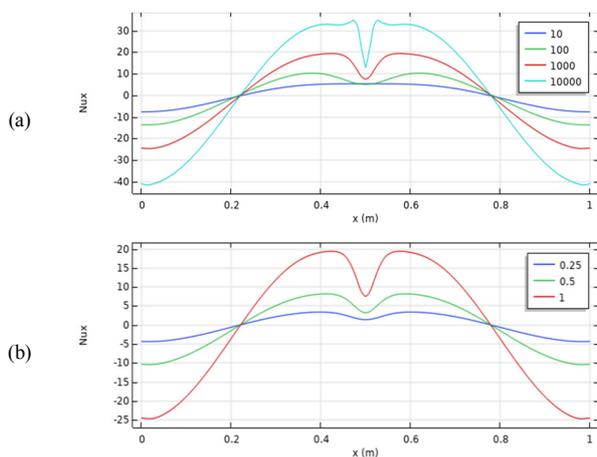

Fig. 11.   Local Nusselt number (a) for $10 \le Ra \le 10000$, $\lambda=1$, and $AR=1.0$, (b) for $0.25 \le \lambda \le 1$, $Ra=1000$, and $AR=1.0$.

Due to the effect of the heat transfer mode by convection, we can observe that as the Rayleigh number grows, so does the Nusselt number. Where we see that the curve takes a concave and in the middle it becomes sinusoidal and increases its amplitude when the value of Ra is increased. We can also see

this behavior when λ is increased, which is why λ has a significant impact on heat transmission at the level of the enclosure.

IV. CONCLUSION

The main objective of this paper was to study the impact of the sinusoidal temperature on the heat flow and the current lines in a porous cavity subjected to natural convection while varying the Rayleigh number, the amplitude sinusoidal temperature, and the cavity format ratio parameter. The two-dimensional finite element method permitted us to model the flow and heat transfer which allowed us to deduct the following:

- For all height/width ratio values, the current lines take the shape of two vortexes, hiding the entire cavity and the heat transfer occurs in the form of three sets of ripples.

- The sinusoidal thermal amplitude has a minor impact on the flow but has a significant impact on the heat transfer rate.

- When the Rayleigh number is high, the convection flow is dominated by natural convection.

- The flow and the heat transfer rate are intensified when the Rayleigh number increases.

- Increase in the Rayleigh number causes a decrease in the boundary layer.

- The relationship between the Nusselt and the Rayleigh numbers can be expressed as a power function.

- The Rayleigh number has a great influence on the flow and the heat transfer rate.

NOMENCLATURE

Cp	Thermal capacity	$J\,kg^{-1}\,K^{-1}$
g	Gravitational acceleration	$m\,s^{-2}$
H	Enclosure height	m
k	Thermal conductivity	$W\,m^{-1}\,K^{-1}$
K	Porous media's permeability	m^2
L	Enclosure length	m
p	Pressure	Pa
T	Temperature	K
$x\,,\,y$	Coordinates	m
α	Thermal diffusivity	$m\,s^{-2}$
β	Thermal dilatation coefficient	K^{-1}
θ	Dimensionless temperature, $(T-T_c)/(T_H-T_c)$	
ρ	Density	$Kg\,m^{-3}$
ν	Kinematic viscosity, $\mu\,\rho_0^{-1}$	$m^2\,s^{-1}$
ψ	Stream function	
ψ	Dimensionless stream function	
γ	Porosity parameter	
AR	Aspect ratio, H/L	
Da	Darcy number, K/H^2	
Gr	Grashof number, Ra/Pr	
Nux	Local Nusselt number	

$\overline{Nu}$	Average Nusselt number
P	Dimensionless pressure, $p / \rho U_0^2$
Pr	Prandtl number, υ / α
Ra	Rayleigh number, $g\,K\,\beta\,H\,\Delta T / \upsilon\,\alpha$
U	Dimensionless velocity components in horizontal direction
V	Dimensionless velocity components in vertical direction
$X\,,Y$	Dimensionless coordinates, $(x / L), (y / H)$

REFERENCES

[1] D. B. Ingham and I. Pop, *Transport Phenomena in Porous Media*. Amsterdam, Netherlands: Elsevier, 1998.

[2] A. Bejan, I. Dincer, S. Lorente, A. Miguel, and H. Reis, *Porous and Complex Flow Structures in Modern Technologies*. New York, NY, USA: Springer, 2004.

[3] D. B. Ingham and I. Pop, *Transport Phenomena in Porous Media*, vol. III. Amsterdam, Netherlands: Elsevier, 2005.

[4] D. A. Nield and A. Bejan, *Convection in Porous Media*, 3rd ed. New York, NY, USA: Springer, 2006.

[5] A. Mojtabi, M. Prat, M. Quintard, and J. Taine, "Transferts de chaleur dans les milieux poreux - Conduction, convection, rayonnement," *Techniques de l'Ingénieur*, Mar. 2019, https://doi.org/10.51257/a-v2-be8250.

[6] P. Vadasz, *Emerging Topics in Heat and Mass Transfer in Porous Media: From Bioengineering and Microelectronics to Nanotechnology*. New York, NY, USA: Springer, 2008.

[7] M. Kaviany, *Principles of Heat Transfer in Porous Media*, 2nd ed. New York, NY, USA: Springer, 2012.

[8] J. J. Bear and Y. Bachmat, *Introduction to Modeling of Transport Phenomena in Porous Media*, vol. 4. New York, NY, USA: Springer, 2012.

[9] K. Vafai, *Handbook of Porous Media*, 3rd ed. Florida, USA: CRC Press, 2015.

[10] M. K. Das, P. P. Mukherjee, and K. Muralidhar, "Porous Media Applications: Biological Systems," in *Modeling Transport Phenomena in Porous Media with Applications*, New York, NY, USA: Springer, 2018, pp. 123–154.

[11] N. B. Khedher, R. Ramzi, and I. A. Alatawi, "Numerical Comparison of Triangular and Sinusoidal External Vibration Effects on the 3D Porous Drying Process," *Engineering, Technology & Applied Science Research*, vol. 10, no. 2, pp. 5554–5560, Apr. 2020, https://doi.org/10.48084/etasr.3486.

[12] T. Basak, S. Roy, T. Paul, and I. Pop, "Natural convection in a square cavity filled with a porous medium: Effects of various thermal boundary conditions," *International Journal of Heat and Mass Transfer*, vol. 49, no. 7, pp. 1430–1441, Apr. 2006, https://doi.org/10.1016/j.ijheatmasstransfer.2005.09.018.

[13] I. A. Badruddin and G. A. Quadir, "Heat and mass transfer in porous cavity: Assisting flow," *AIP Conference Proceedings*, vol. 1738, no. 1, Jun. 2016, Art. no. 480126, https://doi.org/10.1063/1.4952362..

[14] Y. Varol, H. F. Oztop, and I. Pop, "Natural convection in porous media-filled triangular enclosure with a conducting thin fin on the hot vertical wall," *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, vol. 222, no. 9, pp. 1735–1743, Sep. 2008, https://doi.org/10.1243/09544062JMES1031.

[15] T. Basak, S. Roy, D. Ramakrishna, and I. Pop, "Visualization of Heat Transport during Natural Convection Within Porous Triangular Cavities via Heatline Approach," *Numerical Heat Transfer, Part A: Applications*, vol. 57, no. 6, pp. 431–452, Feb. 2010, https://doi.org/10.1080/10407780903507866.

[16] Y. Varol, H. F. Oztop, and I. Pop, "Numerical analysis of natural convection for a porous rectangular enclosure with sinusoidally varying temperature profile on the bottom wall," *International Communications in Heat and Mass Transfer*, vol. 35, no. 1, pp. 56–64, Jan. 2008, https://doi.org/10.1016/j.icheatmasstransfer.2007.05.015.

[17] S. Sivasankaran and M. Bhuvaneswari, "Natural Convection in a Porous Cavity with Sinusoidal Heating on Both Sidewalls," *Numerical Heat Transfer, Part A: Applications*, vol. 63, no. 1, pp. 14–30, Jan. 2013, https://doi.org/10.1080/10407782.2012.715985.

[18] H. T. Cheong, S. Sivasankaran, and Z. Siri, "Effect of wall inclination on natural convection in a porous trapezoidal cavity," *AIP Conference Proceedings*, vol. 1605, no. 1, pp. 343–348, Jul. 2014, https://doi.org/10.1063/1.4887613.

[19] N. P. Mahmoud and A. Zabihi, "Numerical Simulation of a Single-Phase Flow Through Fractures with Permeable, Porous and Non-Ductile Walls," *Engineering, Technology & Applied Science Research*, vol. 7, no. 5, pp. 2041–2046, Oct. 2017, https://doi.org/10.48084/etasr.1448.

[20] N. B. Khedher, "Numerical Study of the Thermal Behavior of a Composite Phase Change Material (PCM) Room," *Engineering, Technology & Applied Science Research*, vol. 8, no. 2, pp. 2663–2667, Apr. 2018, https://doi.org/10.48084/etasr.1824.

[21] A. Latreche and M. Djezzar, "Numerical Study of Natural Convective Heat and Mass Transfer in an Inclined Porous Media," *Engineering, Technology & Applied Science Research*, vol. 8, no. 4, pp. 3223–3227, Aug. 2018, https://doi.org/10.48084/etasr.2179.

[22] K. L. Walker and G. M. Homsy, "Convection in a porous cavity," *Journal of Fluid Mechanics*, vol. 87, no. 3, pp. 449–474, Aug. 1978, https://doi.org/10.1017/S0022112078001718.

[23] S. Kimura and A. Bejan, "Natural convection in a differentially heated corner region," *The Physics of Fluids*, vol. 28, no. 10, pp. 2980–2989, Oct. 1985, https://doi.org/10.1063/1.865137.

[24] A. K. Singh, S. Roy, and T. Basak, "Visualization of Heat Transport during Natural Convection in a Tilted Square Cavity: Effect of Isothermal and Nonisothermal Heating," *Numerical Heat Transfer, Part A: Applications*, vol. 61, no. 6, pp. 417–441, Mar. 2012, https://doi.org/10.1080/10407782.2012.654678.

[25] S. Mekroussi, D. Nehari, M. Bouzit, and N.-E. S. Chemloul, "Analysis of mixed convection in an inclined lid-driven cavity with a wavy wall," *Journal of Mechanical Science and Technology*, vol. 27, no. 7, pp. 2181–2190, Jul. 2013, https://doi.org/10.1007/s12206-013-0533-9.

[26] S. Mekroussi, S. Kherris, B. Mebarki, and A. Benchattir, "Mixed convection in complicated cavity with non-uniform heating on both sidewalls," *International Journal of Heat and Technology*, vol. 35, no. 4, pp. 1023–1033, Dec. 2017, https://doi.org/10.18280/ijht.350439.

[27] M. A. Hossain and M. Wilson, "Natural convection flow in a fluid-saturated porous medium enclosed by non-isothermal walls with heat generation," *International Journal of Thermal Sciences*, vol. 41, no. 5, pp. 447–454, Apr. 2002, https://doi.org/10.1016/S1290-0729(02)01337-6.

[28] H. T. Cheong, S. Sivasankaran, and M. Bhuvaneswari, "Effect of Aspect Ratio on Natural Convection in a Porous Wavy Cavity," *Arabian Journal for Science and Engineering*, vol. 43, no. 3, pp. 1409–1421, Mar. 2018, https://doi.org/10.1007/s13369-017-2948-6.

[29] V. Prasad and F. A. Kulacki, "Convective Heat Transfer in a Rectangular Porous Cavity—Effect of Aspect Ratio on Flow Structure and Heat Transfer," *Journal of Heat Transfer*, vol. 106, no. 1, pp. 158–165, Feb. 1984, https://doi.org/10.1115/1.3246629.

[30] A. Abdulkadhim, A. M. Abed, and K. Al-Farhany, "Computational investigation of conjugate heat transfer in cavity filled with saturated porous media," *Frontiers in Heat and Mass Transfer*, vol. 11, May 2018, Art. no. 12, https://doi.org/10.5098/hmt.11.12.

[31] A. C. Baytas and I. Pop, "Free convection in a square porous cavity using a thermal nonequilibrium model," *International Journal of Thermal Sciences*, vol. 41, no. 9, pp. 861–870, Jul. 2002, https://doi.org/10.1016/S1290-0729(02)01379-0.