NUMERICAL STUDY OF THE NATURAL CONVECTION IN A TWO-DIMENSIONAL PARTIALLY OPEN TILTED CAVITY

J.F. HINOJOSA PALAFOX

Dto. de Ing. Química y Metalurgia, Univ. de Sonora, 83000 Hermosillo, SON, MEXICO.
fhinojosa@iq.uson.mx

***Abstract*— In this paper the numerical results of temperature profiles and flow patterns for natural convection in a two-dimensional partially open tilted cavity are studied. The cavity holds the opposite wall to the aperture at a constant temperature, while the remaining walls are thermally insulated. The most important assumptions in the mathematical formulation are: the flow is laminar, two-dimensional and the Boussinesq approximation is valid. The results are obtained for Rayleigh numbers from 10^6 to 10^8 and inclination angles from 90° to 150°, for the steady state. The comparison between the Nusselt numbers of the fully open cavity and partially open cavity, indicates that the greatest difference is 55.4% (Ra=10^8 and 135°) and the smallest difference is 4.6% (Ra=10^6 and 90°), with an average percentage difference of 29%.**

***Keywords*— natural convection, partially open cavity.**

I. INTRODUCTION

Heat transfer in open cavities is relevant in several thermal engineering applications, for example in the cooling of electronic devices and the design of solar concentrator receivers, among others. In the thermosolar system of parabolic dish with a Stirling thermal engine used to produce electricity, the solar concentrator has a tracking system to maintain its optical axis pointing directly towards the sun. During the tracking, the receiver (open cavity) located at the focal point of the concentrator operates with different inclination angles. The rotation modifies the air motion pattern and the thermal field, and as a consequence the heat losses of the open cavity. A better understanding of how heat is transferred from the receiver will help to optimize the thermal design and improve the thermal performance of the thermosolar system.

In the literature a large number of numerical studies have been reported to describe the heat transfer in open cavities. Some studies have analyzed the heat transfer by natural convection, considering the effects of the Rayleigh number, aspect ratio, and the cavity inclination angle on the flow patterns, temperature fields, and heat transfer behavior inside the open cavity (Le Quere *et al.*, 1981; Penot, 1982; Chan and Tien, 1985a; Chan and Tien, 1985b; Humphrey and to, 1986; Miyamoto *et al.*, 1989; Vafai and Ettefagh, 1990a; Angirasa *et al.*, 1992; Mohamad, 1995; Angirasa *et al.*, 1995; Abib and Jaluria, 1995; Polat and Bilgen, 2002; Bilgen and Oztop, 2005). Other authors have focused on the analysis of flow instabilities (Penot, 1982, Vafai and Ettefagh, 1990b; Angirasa *et al.*, 1995; Hinojosa *et al.*, 2005b), the definition of approximated boundary conditions for the aperture plane (Chan and Tien, 1985b; Angirasa *et al.*, 1992, Khanafer and Vafai, 2000; Khanafer and Vafai , 2002), the mixed convection (Humphrey and To, 1986; Khanafer *et al.*, 2002), the presence of conjugate heat transfer (Polat and Bilgen, 2003; Koca, 2008), three dimensional flow systems (Sezai and Mohamad, 1998; Hinojosa *et al.*, 2006; Hinojosa and Cervantes, 2010) and combined natural convection and radiative heat transfer in the cavity (Lage *et al.*, 1992; Balaji and Venkateshan, 1994; Balaji and Venkateshan, 1995; Singh and Venkateshan, 2004; Hinojosa *et al.*, 2005a; Hinojosa *et al.* 2005b; Nouaneguea *et al.*, 2008).

The investigations of the heat transfer by natural convection in partially open cavities are briefly presented next. Bilgen and Oztop (2005) made a numerical study on inclined partially open square cavities, which are formed by adiabatic walls and a partial opening. A parametric study was carried out using following parameters: Rayleigh number from 10^3 to 10^6, dimensionless aperture size from 0.25 to 0.75, aperture position at high, center and low, and inclination of the opening from 0° (facing upward) to 120° (facing 30° downward). It was found that the Nusselt number is an increasing function of Rayleigh number, aperture size and generally aperture position. Koka (2008) solved the laminar natural convection and conduction in partially open square cavity with a vertical heat source. The cavity has an opening on the top with several lengths and three different positions. The heat source was located on the bottom wall of cavity. The results were reported for various governing parameters such as Rayleigh number. It was found that ventilation position has a significant effect on heat transfer.

However the Rayleigh numbers studied by Bilgen and Oztop (2005) correspond to smaller size cavities than those commonly used as receivers in the dish-Stirling thermosolar systems; furthermore the considered inclination angles do not cover the full rotation of the thermal receiver during its operation. Considering above this work is focused to analyze numerically the fluid motion pattern, the temperature field and heat transfer in a partially open tilted cavity, extending the

Rayleigh number range to 10^6-10^8 and the inclination angle range to 90^o-150^o.

II. PHYSICAL PROBLEM AND MATHEMATICAL MODEL

The heat transfer in a square partially open tilted cavity is considered in this paper as shown in Fig. 1. The opposite wall to the aperture was kept to a constant temperature T_H, while the surrounding fluid interacting with the aperture was fixed to an ambient temperature T_∞. The two remaining walls were insulated. The opening was reduced to half and placed in the center. The fluid was air and the fluid flow was laminar. The properties of the fluid were considered constant except for the density in the buoyant force term in the momentum equations, according to the Boussinesq approximation. The cavity is rotated with respect to the horizontal, so that when the angle is equal to 90°, the isothermal wall is vertical. The inclination angles greater than 90°, corresponding to a cavity where the opening is below the horizontal.

From the above considerations, the transient state dimensionless 2D conservation equations governing the transport of mass, momentum, and energy in primitive variables are expressed as

Continuity:

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

X-momentum:

$$\frac{\partial U}{\partial \tau}+\frac{\partial (U^2)}{\partial X}+\frac{\partial (UV)}{\partial Y}=-\frac{\partial P}{\partial X}+\sqrt{\frac{\Pr}{Ra}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)+\theta\cos\phi \qquad (2)$$

Y-momentum:

$$\frac{\partial V}{\partial \tau}+\frac{\partial (UV)}{\partial X}+\frac{\partial (V^2)}{\partial Y}=-\frac{\partial P}{\partial Y}+\sqrt{\frac{\Pr}{Ra}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\theta\, sen\phi \qquad (3)$$

Energy:

$$\frac{\partial \theta}{\partial \tau}+\frac{\partial (U\theta)}{\partial X}+\frac{\partial (V\theta)}{\partial Y}=\frac{1}{\sqrt{\Pr Ra}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \qquad (4)$$

where Ra is the Rayleigh number and Pr is the Prandtl number. The dimensionless variables were defined as

$$X=\frac{x}{L} \qquad Y=\frac{y}{L} \qquad \tau=\frac{U_0 t}{L}$$

$$U=\frac{u}{U_0} \quad V=\frac{v}{U_0} \quad P=\frac{p-p_\infty}{\rho U_0^2} \quad \theta=\frac{T-T_\infty}{T_H-T_\infty} \qquad (5)$$

$$\Pr=\frac{\nu}{\alpha} \qquad Ra=\frac{g\beta L^3(T_H-T_\infty)}{\alpha\nu}$$

the reference velocity U_o is related to the buoyancy force term and was defined as:

$$U_0=\sqrt{g\beta L(T_H-T_\infty)} \qquad (6)$$

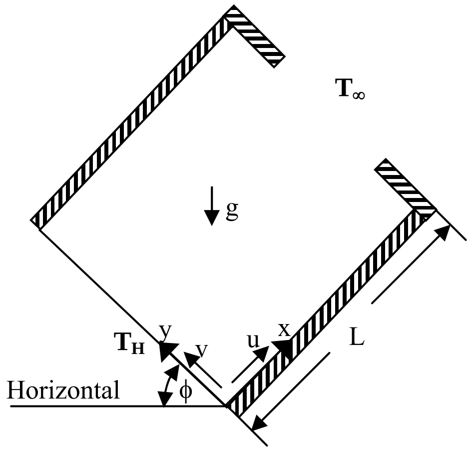

Figure 1. Scheme of the partially open tilted cavity.

the initial and boundary conditions were taken as follow:

for τ=0 y 0<X<1, 0<Y<1

$$U(X,Y,0)=V(X,Y,0)=P(X,Y,0)=\theta(X,Y,0)=0 \qquad (7)$$

In the hydrodynamic boundary conditions, non-slip behavior was assumed at the solid walls, then $U(X,Y,\tau)=V(X,Y,\tau)=0$.

The thermal boundary condition for the isothermal wall is given by:

$$\theta(0,Y,\tau)=1. \qquad (8)$$

In the adiabatic walls the temperature gradient in the normal direction is equal to zero. In non-dimensional form:

$$\left(\frac{\partial\theta}{\partial n}\right)=0. \qquad (9)$$

In the aperture plane it was supposed that no velocity gradients and thus no momentum transfer occurs at this location, also was considered that the incoming fluid enters to the cavity at ambient temperature, while for the fluid leaving the cavity the thermal conduction is negligible (Chan and Tien, 1985b; Mohammad, 1995):

$$\left(\frac{\partial U}{\partial X}\right)_{X=1}=\left(\frac{\partial V}{\partial X}\right)_{X=1}=0 \qquad (10)$$

$$\theta=0 \text{ for } U<0 \text{ or } \left(\frac{\partial\theta}{\partial X}\right)_{X=1}=0 \text{ for } U>0$$

The average convective Nusselt number was calculated integrating the temperature gradient over the isothermal wall as:

$$Nu=\int_0^1 -\frac{\partial\theta}{\partial X}dY \qquad (11)$$

III. NUMERICAL METHOD OF SOLUTION

The Eqs. (1)-(4) were discretized using staggered and uniform control volumes. For the discretization of the convective terms was used the SMART scheme (Gaskell and Lau, 1988) and for the diffusive terms the central differencing scheme. The fully implicit scheme was used for the time discretization. The SIMPLEC algorithm (Van Doormaal and Raithby, 1984) was applied to couple continuity and momentum equations. The resulting system of linear algebraic equations was solved iteratively by the SIS method (Lee, 1989). After an independence mesh study, the results were obtained with a 120x120 grid mesh. The dimensionless time step used for the calculations was $1x10^{-3}$.

IV. RESULTS

In order to validate the numerical code, the natural convection with Pr=1, in a square open cavity with side facing was solved, and the results were compared with the ones obtained with an extended computational domain. The comparison between the average Nusselt numbers is presented in Table 1. It can be observed that the higher percentage difference was 19.6% for Ra=10^3 and the lowest was 0.3% for Ra=10^7, with an average percentage difference of 5.2%. However the natural convection in square open cavity was solved with Pr=0.71 (air), varying the Rayleigh number in the range of 10^4 to 10^7 and the inclination angle between 10° and 90°. The calculated Nusselt numbers are compared with those reported by Mohamad (1995) in the Table 2. The highest percentage difference (the values with oscillations were not considered) was 4 % for Ra=10^5 and ϕ=30° and the lowest was 0% for Ra=10^4 and ϕ=90°, while the average percentage difference was 1.2%. Table 3 presents the results for a fully open three dimensional tilted cavity reported by Hinojosa *et al.* (2006) and those obtained with the present numerical code. The absolute percentage difference are between 6.71% (Ra=10^6 and ϕ=135°) and 1.25% (Ra=10^6 and ϕ=135°), indicating a good agreement. Based on above results the present numerical code was considered as validated.

In order to carry out the parametric study, the values of the Rayleigh number for different dimensions and operating temperatures of the open cavity were computed (assuming an ambient temperature of 300 K and evaluating the air physical properties at average temperature between the isothermal wall and the ambient). It is noted from values of Table 4 that most of the Rayleigh numbers are between 10^6 and 10^8, therefore the parametric study will focus to cover this range.

Figure 2 shows the results for Ra=10^6 and an inclination angle of 90°. It is observed in the isotherms graph the formation of a thermal boundary layer near the hot wall and a thermal stratification of the fluid for about half of the top of the cavity; the graph of the streamlines indicates that the main flow of fluid occurs in the clockwise direction, with the cold fluid entering at the bottom of the opening, reaches the hot wall and rises impelled by the buoyancy force and finally goes out by the top of the opening. However in the lower right corner is observed the formation of a vortex that rotates counter clockwise.

The results for Ra=10^6 and an inclination angle of 120° are presented in Figure 3, the isotherms graph shows again the formation of thermal stratification with the fluid temperature layers oblique to the coordinate axes. In the streamlines graph is observed that the vortex in the lower right corner for 90° has disappeared. The fluid entering by the center of the opening returns to the aperture without reaching the isothermal wall. However the values of the streamlines are much lower indicating the fluid circulates with a slower speed inside the cavity.

Table 1. Comparison of the Nusselt number results for the square open cavity with Pr=1.

Ra	This work	Chan and Tien (1985a)	Difference (%)
10^3	1.28	1.07	19.6
10^4	3.57	3.41	4.7
10^5	7.75	7.69	0.8
10^6	15.11	15.0	0.7
10^7	28.70	28.6	0.3

Table 2. Comparison of the Nusselt number results for the natural convection in a tilted open cavity.

Mohamad (1995)				
ϕ	Ra			
	10^4	10^5	10^6	10^7
10°	2.57	6.87	16.21	----
30°	3.34	6.17	12.08	27.3±5.0
60°	3.7	7.36	13.72	29.2±5.0
90°	3.44	7.41	14.36	28.6±2.5
This work				
ϕ	Ra			
	10^4	10^5	10^6	10^7
10°	2.54	6.54±0.93	11.6±1.59	23.47±3.51
30°	3.34	6.42	12.25	23.24±0.28
60°	3.69	7.4	14.15	26.94±0.17
90°	3.44	7.44	14.51	27.58

Table 3. Comparison with three-dimensional Nusselt numbers results for the natural convection in a tilted fully open cavity.

Hinojosa *et al.* (2006)		
ϕ	Ra	
	10^6	10^7
90°	14.33	27.21
120°	9.55	18.67
135°	4.32	6.05
150°	1.86	2.08
This work		
ϕ	Ra	
	10^6	10^7
90°	14.51	27.58
120°	9.37	17.54
135°	4.03	5.96
150°	1.75	2.01

Table 4. The Rayleigh numbers for the open cavity (different dimensions and operating temperatures).

T_H	Length of the isothermal wall (m)		
(K)	0.1	0.2	0.30
400	$4.49x10^6$	$3.60x10^7$	$1.21x10^8$
500	$4.85x10^6$	$3.88x10^7$	$1.31x10^8$
600	$4.28x10^6$	$3.42x10^7$	$1.16x10^8$
700	$3.57x10^6$	$2.86x10^7$	$9.64x10^7$
800	$2.94x10^6$	$2.35x10^7$	$7.93x10^7$
900	$2.42x10^6$	$1.94x10^7$	$6.54x10^7$
1000	$2.01x10^6$	$1.61x10^7$	$5.43x10^7$
1100	$1.68x10^6$	$1.34x10^7$	$4.54x10^7$

For a tilt angle of 135° and Ra=10^6, the results are shown in Fig. 4. The isotherms graph indicates that the temperature field is similar to that described for 120°, but the thickness of the thermal boundary layer is wider and the temperature distribution on the adiabatic walls has changed. In the flow pattern presented in the streamlines graph is observed a big vortex in the half of the cavity near the isothermal wall, which reduces considerably the amount of incoming cold fluid reaching the isothermal wall.

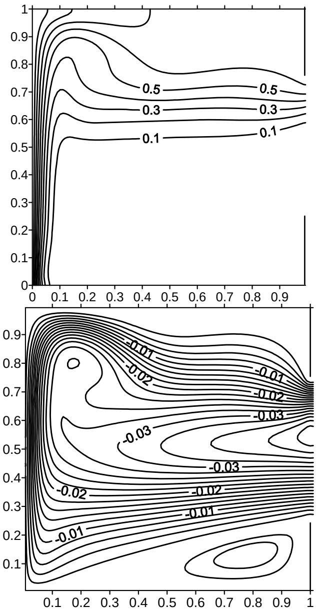

Figure 2. Isotherms (top) and streamlines (bottom) for Ra=10^6 and ϕ=90°.

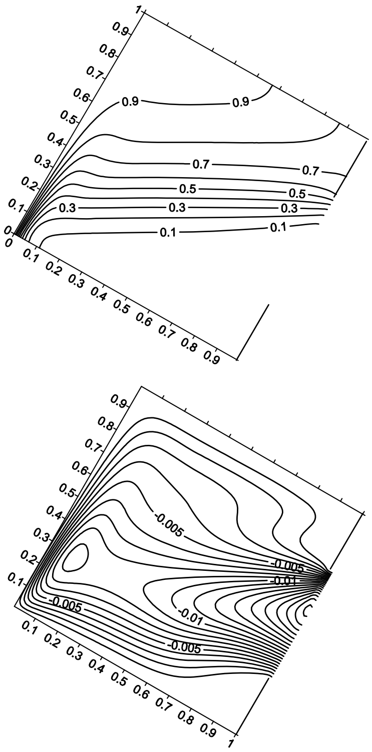

Figure 3. Isotherms (top) and streamlines (bottom) for Ra=10^6 and ϕ=120°.

The Fig. 5 presents the numerical results for a tilt angle of 150° and Ra=10^6. It can be observed in the isotherms graph, that the thermal boundary layer is vanishing, tending to a conductive regime. In the streamlines graph is seen a big vortex in the half of the cavity near the isothermal wall and a small vortex adjacent to the right top corner. The incoming air collides with the big vortex and returns to the aperture without touching the isothermal wall.

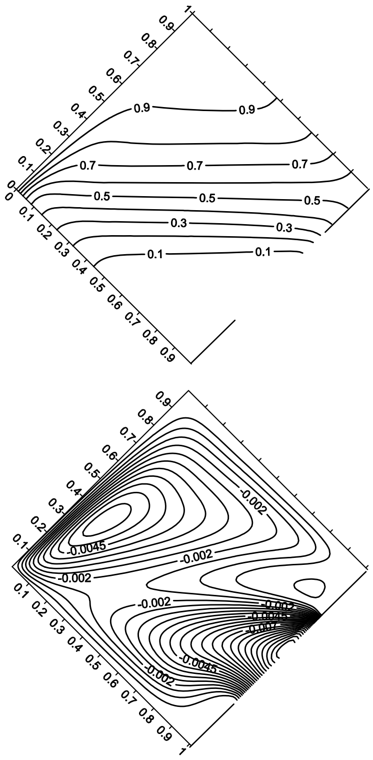

Figure 4. Isotherms (top) and streamlines (bottom) for Ra=10^6 and ϕ=135°.

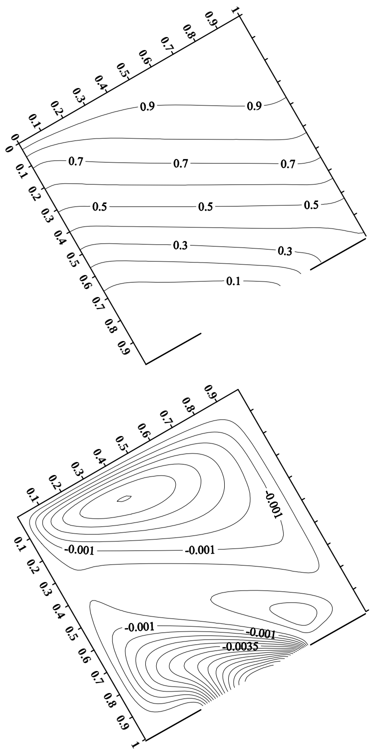

Figure 5. Isotherms (top) and streamlines (bottom) for Ra=10^6 and ϕ=150°.

The Fig. 6 shows the results for Ra=10^8 and an angle of inclination of 90°, the isotherms graph shows a thermal boundary layer adjacent to the hot wall with a thickness much smaller than that observed for Ra=10^6 for the same angle. The thermal stratification occupies a smaller area of the upper section of the cavity. In the

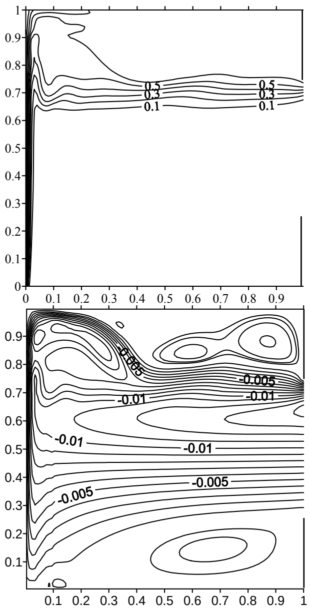

Figure 6. Isotherms (top) and streamlines (bottom) for Ra=10^8 and ϕ=90°.

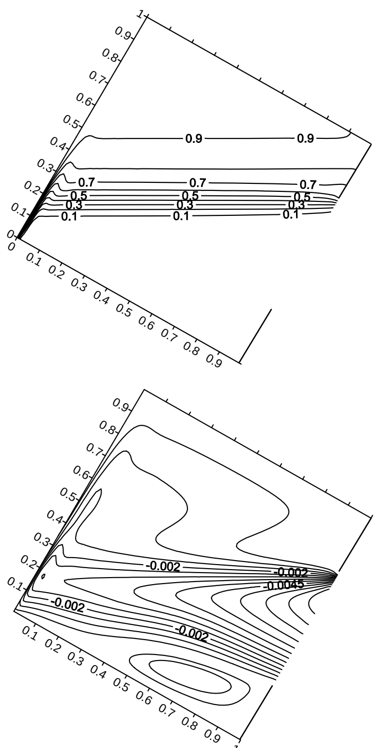

Figure 7. Isotherms (top) and streamlines (bottom) for Ra=10^8 and ϕ=120°.

streamlines graph is seen a complex flow pattern with multiple vortexes adjacent to top corners of the cavity. However a big recirculation zone is formed adjacent to the bottom adiabatic wall.

The results for a tilt angle of 120° and Ra=10^8 are shown in the Fig. 7. The isotherms graph indicates a thin thermal boundary layer adjacent to the lower half of the isothermal wall, but around to the upper left corner is observed an extended region of hot fluid in an oblique way. In the streamlines graph is seen a big vortex in the lower right corner. The cold fluid entering at the center of the opening returns without touching the isothermal wall.

The Fig. 8 shows the corresponding results for a tilt angle of 135° and Ra=10^8. In the isotherms graph is observed a thin thermal boundary layer adjacent to the lower quarter of the isothermal wall. However around to the upper left corner is observed a large region of hot fluid. In the streamlines graph can be seen a big recirculation zone adjacent to the isothermal wall, which extends from the lower left corner to almost the upper right corner. The entering cold air collides with the recirculation zone and returns to the aperture, leaving the cavity without reaching the isothermal.

The results for an inclination angle of 150° and Ra=10^8 are presented in the Fig. 9. The isotherms graph indicates a thermal boundary layer adjacent to the lower quarter of the isothermal wall, but around to the upper left corner is observed a region of hot fluid. In the streamlines graph can be seen a big recirculation zone adjacent to the isothermal wall, which extends from the lower left corner to the half upper adiabatic wall. Also is seen a small vortex near the upper right corner. The amount of incoming cold air that touches the isothermal wall is very small.

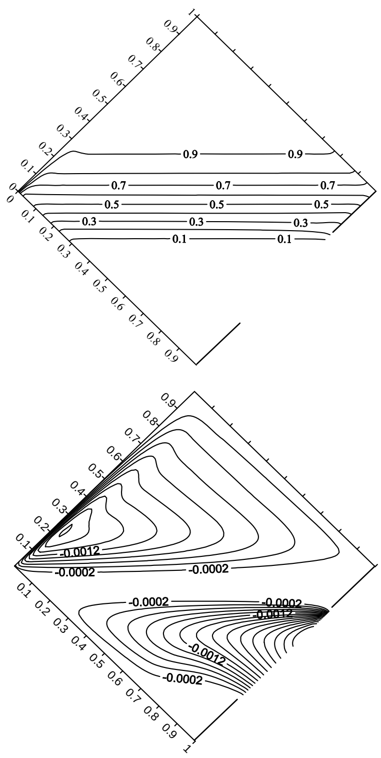

Figure 8. Isotherms (top) and streamlines (bottom) for Ra=10^8 and ϕ=135°.

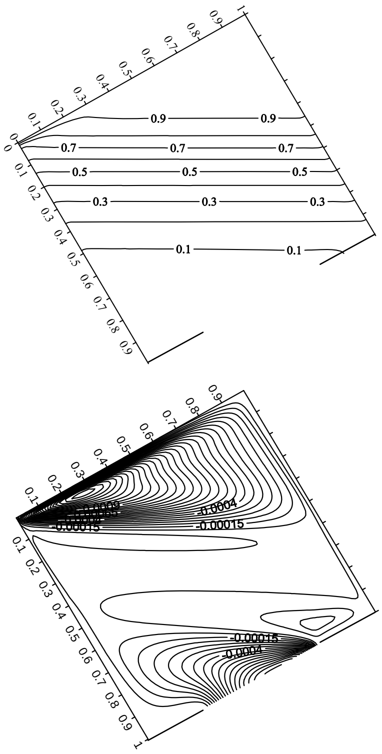

Figure 9. Isotherms (top) and streamlines (bottom) for Ra=10^8 and ϕ=150°.

The Table 5 presents the results of heat transfer (Nusselt number) for the partially open cavity. It can be observed that the Nusselt number increases with Rayleigh number for all angles studied. For a given value of Rayleigh number, the maximum Nusselt number occurs at 90° and decreases substantially with increasing tilt angle of the cavity as seen in the Figure 10. The largest decrease in the Nusselt number is presented for Ra=10^8, varying the angle of inclination of the cavity 90° (Nu=49.87) to 150° (Nu=1.86), representing a percentage decline of 96.2 %.

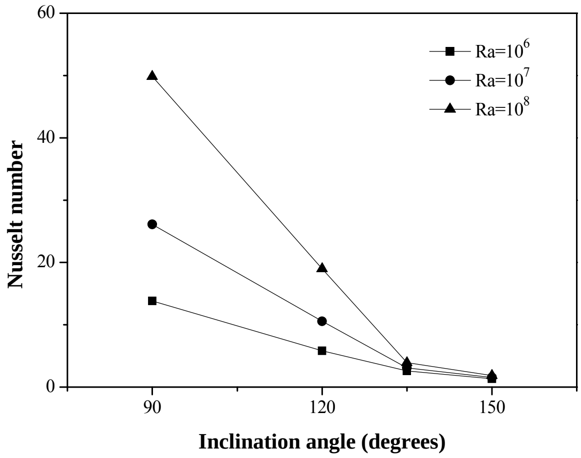

Figure 10. Nusselt number variation with respect to the angle of inclination of the cavity for different Rayleigh numbers.

Table 5. The Nusselt numbers for the partially open cavity.

ϕ	Ra=10^6	Ra=10^7	Ra=10^8
90°	13.84	26.13	49.87
120°	5.82	10.57	19.01
135°	2.59	3.09	3.93
150°	1.32	1.54	1.86

Table 6. The Nusselt numbers for the completely open cavity.

ϕ	Ra=10^6	Ra=10^7	Ra=10^8
90°	14.51	27.58	53.68
120°	9.37	17.54	33.25
135°	4.03	5.96	8.82
150°	1.75	2.01	2.43

Table 7 Convective heat loses (W/m^2) of the partially open cavity.

		Ra=10^6		
T_H (K)	**90°**	**120°**	**135°**	**150°**
400	726.26	305.41	135.91	69.27
500	1753.09	737.21	328.07	167.20
700	2896.90	1218.20	542.12	276.29
800	4106.37	1726.81	768.46	391.65
		Ra=10^7		
400	636.45	257.45	75.26	37.51
500	1536.29	621.45	181.67	90.54
700	2538.65	1026.93	300.21	149.62
800	3598.55	1455.67	425.55	212.08
		Ra=10^8		
400	563.81	214.92	44.43	21.03
500	1360.95	518.78	107.25	50.76
700	2248.90	857.26	177.22	83.88
800	3187.83	1215.17	251.22	118.90

The average convective heat losses (W/m^2) of the partially open cavity for different operating conditions are reported in the Table 7. It can be observed that the heat flux diminishes with the Rayleigh number (for a fixed inclination angle and temperature of the hot wall) and increases with the temperature of the hot wall (for a fixed Rayleigh number and inclination angle). However the average heat flux reduces significantly when the inclination angle is varied from 90° to 150°. The previous behavior of the convective heat losses has been observed experimentally by Leibfried and Ortjohann (1995) for a spherical a semispherical solar cavity receiver.

The results of heat transfer reported in Tables 5 and 7 and the knowledge of the behavior of the fluid flow for different operating conditions, help to optimize the size of the receiver of a dish-Stirling system and to evaluate its thermal performance.

V. CONCLUSIONS

In this paper numerical calculations for the heat transfer by natural convection in a partially open tilted cavity with a vertical isothermal wall and remaining adiabatic walls are presented. From the results we can conclude the following:

1. The temperature fields and flow patterns vary significantly with the Rayleigh number and the angle of inclination of the cavity.
2. When the cavity is tilted to 135° and 150°, is formed a fluid recirculation zone adjacent to the hot wall

and the upper adiabatic wall, which makes most of the fluid entering the cavity returns without reaching the isothermal wall.

3. The Nusselt number (used to determine the extent of the loss of heat by natural convection in the cavity) is a function of both the Rayleigh number and the angle of inclination of the cavity. The Nusselt number increases with Rayleigh number and decreases with the angle of inclination of the cavity.
4. The largest variation in the Nusselt number with respect to the angle of inclination of the cavity is presented for $Ra=10^8$ and angle change of 90° (Nu=49.87) to 150° (Nu=1.86).
5. The comparison between the Nusselt numbers of the fully open cavity and partially open cavity, indicates that the greatest difference is 55.4% ($Ra=10^8$ and 135°) and the smallest difference is 4.6% ($Ra=10^6$ and 90°), with an average percentage difference of 29%.

REFERENCES

Abib, A.H. and Y. Jaluria, "Penetrative convection in a stably stratified enclosure," *International Journal of Heat and Mass Transfer*, **38**, 2489-2500 (1995).

Angirasa, D., M.J. Pourquié and F.T. Nieuwstadt, "Numerical study of transient and steady laminar bouyancy-driven flows and heat transfer in a square open cavity," *Numerical Heat Transfer Part A*, **22**, 223-239 (1992).

Angirasa, D., J.G. Eggels and F.T. Nieuwstadt, "Numerical simulation of transient natural convection from an isothermal cavity open on a side," *Numerical Heat Transfer Part A*, **28**, 755-768 (1995).

Balaji, C. and S.P. Venkateshan, "Interaction of radiation with free convection in an open cavity," *International Journal of Heat and Fluid Flow*, **15**, 317-324 (1994).

Balaji, C. and S.P. Venkateshan, "Combined conduction, convection and radiation in a slot," *International Journal of Heat and Fluid Flow*, **16**, 139-144 (1995).

Bilgen, E. and H. Oztop, "Natural convection heat transfer in partially open inclined square cavities," *International Journal of Heat and Mass Transfer*, **48**, 1470-1479 (2005).

Chan, Y.L. and C.L. Tien, "A numerical study of two-dimensional natural convection in square open cavities," *Numerical Heat Transfer*, **8**, 65-80 (1985a).

Chan, Y.L. and C.L. Tien, "A numerical study of two-dimensional laminar natural convection in a shallow open cavity," *International Journal of Heat and Mass Transfer*, **28**, 603-612 (1985b).

Gaskell, P.H. and A.K.C. Lau, "Curvature-compensated convective transport: SMART, a new boundedness-preserving transport algorithm," *International Journal of Numerical Methods in Fluids*, **8**, 617-641 (1988).

Hinojosa, J.F., R.E. Cabanillas, G. Alvarez and C.A. Estrada, "Nusselt number for natural convection and surface thermal radiation in a square tilted open cavity," *International Communications for Heat and Mass Transfer*, **32**, 1184-1192 (2005a).

Hinojosa, J.F., R.E. Cabanillas, G. Alvarez and C.A. Estrada, "Numerical study of transient and steady-state natural convection and surface thermal radiation in a horizontal square open cavity," *Numerical Heat Transfer, Part A*, **48**: 179–196 (2005b).

Hinojosa, J.F., G. Alvarez and C.A. Estrada, "Three-dimensional numerical simulation of the natural convection in an open tilted cubic cavity," *Revista Mexicana de Física*, **52**, 111-119 (2006).

Hinojosa, J.F. and J. Cervantes, "Numerical simulation of steady-state and transient natural convection in an isothermal open cubic cavity," *Heat and Mass Transfer*, **46,** 595–606 (2010).

Humphrey, J.A. and W.M. To, "Numerical simulation of buoyant, turbulent flow-II. Free and mixed convection in a heated cavity," *International Journal of Heat and Mass Transfer*, **29**, 593-610 (1986).

Khanafer, K. and K. Vafai, "Bouyancy-driven flow and heat transfer in open ended enclosures: elimination of extended boundaries," *International Journal of Heat and Mass Transfer*, **43**, 4087-4100 (2000).

Khanafer, K. and K. Vafai, "Effective boundary conditions for bouyancy-driven flows and heat transfer in a fully open two-dimensional enclosures," *International Journal of Heat and Mass Transfer*, **45**, 2527-2538 (2002).

Khanafer, K., K. Vafai and M. Lighstone, "Mixed convection heat transfer in two-dimensional open ended enclosures," *International Journal of Heat and Mass Transfer*, **45**, 5171-5190 (2002).

Koca, A.; "Numerical analysis of conjugate heat transfer in a partially open square cavity with a vertical heat source," *International Communications in Heat and Mass Transfer*, **35**, 1385-1395 (2008).

Lage, J.L, J.S. Lim and A. Bejan, "Natural convection with radiation in a cavity with open top end," *Journal of Heat Transfer*, **114**, 479-486 (1992).

Lee, S., "A strongly implicit solver for two-dimensional elliptic differential equations," *Numerical Heat Transfer Part B*, **16**, 161-178 (1989).

Le Quere, P., J.A. Humphrey and F.S. Sherman, "Numerical calculation of thermally driven two-dimensional unsteady laminar flow in cavities of rectangular cross section," *Numerical Heat Transfer*, **4**, 249-283 (1981).

Leibfried, U. and J. Ortjohann, "Convective heat loss from upward and downward-facing cavity solar receivers: measurements and calculations," *Journal of Solar Energy Engineering*, **117**, 75-84 (1995).

Miyamoto, M., T.H. Kuehn, R.J. Goldstein and Y. Katoh, "Two-dimensional laminar natural convection heat transfer from a fully or partially open square cavity," *Numerical Heat Transfer Part A*, **15**, 411-430 (1989).

Mohamad, A.A., “Natural convection in open cavities and slots,” *Numerical Heat Transfer Part A*, **27**, 705-716 (1995).

Nouaneguea, H., A. Muftuoglua and E. Bilgen, “Conjugate heat transfer by natural convection, conduction and radiation in open cavities,” *International Journal of Heat and Mass Transfer*, **51**, 6054-6062 (2008).

Penot, F., “Numerical calculation of two-dimensional natural convection in isothermal open cavities,” *Numerical Heat Transfer*, **5**, 421-437 (1982).

Polat, O. and E. Bilgen, “Laminar natural convection in inclined open shallow cavities,” *International Journal of Thermal Sciences*, **41,** 360-368 (2002).

Polat, O. and E. Bilgen, “Conjugate heat transfer in inclined open shallow cavities,” *International Journal of Heat and Mass Transfer*, **46**, 1563–1573 (2003).

Sezai, I. and A.A. Mohamad, “Three-dimensional simulation of natural convection in cavities with side opening,” *International Journal of Numerical Methods for Heat & Fluid Flow*, **8**, 800-813 (1998).

Singh, S.N. and S.P. Venkateshan, “Numerical study of natural convection with surface radiation in side-vented open cavities,” *International Journal of Thermal Sciences*, **43**, 865-876 (2004).

Vafai, K. and J. Ettefagh, “The effects of sharp corners on bouyany-driven flows with particular emphasis on outer boundaries,” *International Journal of Heat and Mass Transfer*, **33**, 2311-2328 (1990a).

Vafai K. and J. Ettefagh, “Thermal and fluid flow instabilities in bouyancy-driven flows in open-ended cavities”, International Journal of Heat and Mass Transfer, **33**, 2329-2344 (1990b).

Van Doormaal, J.P. and G.D. Raithby, “Enhancements of the SIMPLE method for predicting incompressible fluid flows,” *Numerical Heat Transfer*, **7**, 147-163 (1984).

Received: February 2, 2011
Accepted: October 30, 2011
Recommended by subject editor: Adrian Lew