Two Dimensional Natural Convection Heat Losses from Square Solar Cavity Receiver

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Abstract
This research presents the quantitative outcomes of the convective heat loss through natural convection in a two dimensional (2D) open cavity. The experimental study presented here marks considerable value in solving numerous problems in the area of thermal engineering e.g. from designing of semiconductor electronic devices to thermal receivers specifically for solar concentrators. The numerical analysis is done on two-dimensional cavity models with Computational Fluid Dynamics (CFD) FLUENT Software Tool. The variation in temperature of the contrary wall referring to the aperture in the cavity was observed in the range from $10\,°K$ to $40\,°K$, while the temperature was maintained as $300\,°K$ at the surrounding fluid relating with the aperture. The other walls were kept insulated. The results are acquired for Rayleigh ranging from 9.41×10^5 to 3.76×10^6 and tilt angle φ of 0°, 15°, 30°, 45°, 60°, 75° and 90° in steady state.

Keywords:
Computational Fluid Dynamics; Convective heat loss; Nusselt number; Solar cavity; Solar concentrators;

1. Introduction

The "Cavities" are found and categorized in two forms: (1) Closed cavity and (2) Open cavity. Whereas if the aperture is open to the atmosphere, such one is known to be an "Open cavity". There are several applications of open cavity heat transfer in numerous areas of practical interest, for example, Solar concentrator systems, Solar passive architecture, Refrigeration, electronic equipment cooling, and fire research. Therefore this has been widely focused and analyzed [1] in research.

The Buoyancy-driven heat transfer imparts major role in heat transfer appliances concerning to the open cavities. For example, the geometrical shape of the cavity, the inclination angle, the wall boundary conditions, the aspect ratio and the opening ratio; thus making it difficult to estimate [1]. The literature on convection heat transfer in open cavities is mainly explored such as cubical, rectangular and square shaped cavities [2]. Cylindrical i.e. cylindrical with a conical frustum, spherical and hemispherical shaped cavities that are used for specific applications like solar thermal receivers were also studied. Also, convective losses from cubical and rectangular open cavities have been extensively studied [3] [4] [5] [6]. Prakash in [7] had investigated a cylindrical receiver having the ratio of "Aperture diameter" to the "Cavity diameter" greater than one as shown in $Eq. (1)$.

$$\frac{Aperture\ Diameter}{Cavity\ Diameter} > 1 \qquad \cdots (1)$$

He determined and reported the influence of the parameters such as receiver inclination, the fluid inlet temperature and the external wind on the convective losses. Thereafter he carried out computation on correlations of Nusselt number (Nu) concerning to the receiver's aperture diameter; and proposed model for convective losses under no-wind conditions with three dimensional (3D) numerical analysis [7] using the Software Package FLUENT [8] for Computational Fluid Dynamics CFD.

A numerical study of combined i.e. convection heat, as well as radiation heat loss from a reformed cavity receiver of solar parabolic dish collector, was also carried out using Asymptotic Computational Fluid Dynamics technique (ACFD) by N. Sendhil Kumar [9]. He determined and concluded that the convection heat loss is influenced by radiation heat loss [9].

2. Cavity Investigated

The cavity in a square shape with an open ratio $\frac{a}{H} = 1$ with three different temperature ranges is analyzed. The open cavity structure is shown in Fig.1, where (H) is the height, and (L) is the length or depth of the cavity. The effect of Rayleigh number (Ra) and the inclination angle (φ) of the cavity on the convective loss Q_C and Nusselt number (Nu) are studied extensively. The variation of temperature of the opposite wall to the aperture in the cavity was observed in the range from $10°K$ to $40°K$ with step of $10°K$. While the other walls were found adiabatic [10] [11] [12] [13].

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3. Mathematical Formulation

The simulation of flow, as well as heat transfer, is depending on the formulation of a simultaneous system of equations expressing the parameters of energy, momentum, and mass conservation that can be described for an incompressible fluid [14] as shown in $Eq.$ (2), (3), (4), and (5):

$$\frac{\partial U}{\partial x}+\frac{\partial V}{\partial y}=0 \qquad \cdots (2)$$

$$U\frac{\partial U}{\partial x}+V\frac{\partial U}{\partial y}=-\frac{1}{\rho}\frac{\partial(p-p_0)}{\partial x}+V\left(\frac{\partial^2 U}{\partial x^2}+\frac{\partial^2 U}{\partial y^2}\right) \qquad \cdots (3)$$

$$U\frac{\partial V}{\partial x}+V\frac{\partial V}{\partial y}=-\frac{1}{\rho}\frac{\partial(p-p_0)}{\partial y}+V\left(\frac{\partial^2 V}{\partial x^2}+\frac{\partial^2 V}{\partial y^2}\right)+g\beta(T-T_0) \qquad \cdots (4)$$

$$U\frac{\partial T}{\partial x}+V\frac{\partial T}{\partial y}=\frac{\kappa}{\rho\, C_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \qquad \cdots (5)$$

Where U , V are Fluid velocities; x , y are Cartesian Coordinates; g is Acceleration due to gravity in m/S^2; κ is Thermal conductivity coefficient in Watts per meter Kelvin $W\,/\,(m\cdot K)$; p is Pressure in Pascal (Pa) ; T is Temperature in kelvin K; C_p is Specific heat of air in $kJ/(kg\cdot K)$ Kilo Joule per Kg Kelvin; β is Volumetric coefficient of thermal expansion $1/K$; ρ is Fluid density in kg/m^3 [15].

4. Numerical Approach and Validation

A two dimensional (2D) numerical model of the square cavity is created. The simulation is performed using the Gambit tool/ CFD Package facilitated in FLUENT® Software Tool [8] [14] [1]. The surroundings of the cavity are set ambient. The receiver is covered with unbounded atmosphere with an ambient air temperature of $T_a = 300°K$. In order to set this phenomenon, the area of the flow is selected so that the receiver is centrally positioned in an outsized square-shaped enclosed field, formulating the numerical model. Care has been taken to assure that the air flow remains unaffected within the cavity [14].

On the contrary wall to the aperture cavity, the isothermal boundary condition was applied with the caution to keep the receiver's outer walls adiabatic, whereas, on the outer domain, the pressure inlet boundary condition was applied. The Boussinesq Approximation criteria are validated for the current work since the temperature range is set as $10°K$ to $40°K$ with the reason that the product of the thermal expansion coefficient of air and temperature difference between the cavity wall and air is determined to

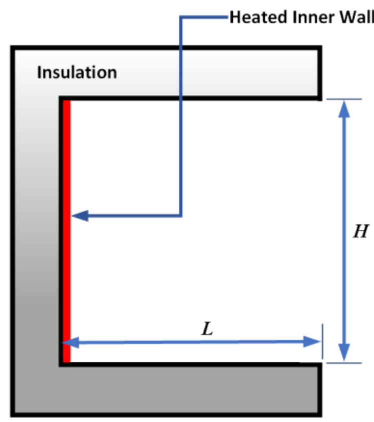


Fig.1. Open Cavity Structure.

be in the range of 0.033 and 0.133. The solutions are determined by solving simultaneous equations of energy, momentum and continuity [14]. The FLUENT® software with SIMPLE algorithm scheme is used to couple continuity and momentum equations [10] [16]. The first order upwind type controls are taken for energy and momentum solution. The convergence criteria for the residuals of the equations of continuity and the velocity are of the order of 10^{-3} and 10^{-4} while for the energy equation, it is 10^{-6} [7] [14]. The solutions are obtained as soon as the criteria for convergence are fulfilled. The study of the outer domain for Grid Independent Sensitivity was conceded for each square opening ratio [14]. The results are recorded in Table-1.

The sensitivity study has been carried out for the outer domain by picking the sample example of a square cavity with wall temperature $T_W = 40°K$. This value is taken to perform the simulations since the initial boundary condition at the cavity aperture is unknown [1]. The variations in domain size is held from 3 to 15 times that of cavity height (H). The results are tabulated in Table-1. It is found that the percentage change in convective loss Q_C for $15H$ domain revealed very small i.e. 0.40% in ratio with of $3H$. With this reason, a square domain having the height (H) and length (L) equal to $15H$ is chosen [14] [17] [18] [19] [20].

Table-1. Sensitivity Study of Outer Domain of Cavity Receiver

Size of Cavity Outer Domain			Q_C (Watts)	Change (%)
3 times H	$3\times H$	$3H$	2.25	--
5 times H	$5\times H$	$5H$	2.20	2.22
10 times H	$10\times H$	$10H$	2.19	0.51
15 times H	$15\times H$	$15H$	2.18	0.40

Table-2. Grid Independent Study

Elements in Mesh	Q_C (W)	Change (%)	Nu_C	Change (%)
80×80	2.33	--	22.03	--
100×100	2.18	6.40	20.66	6.20
120×120	2.16	0.88	20.51	0.73
130×130	2.15	0.23	20.47	0.20
140×140	2.15	0.09	20.46	0.06

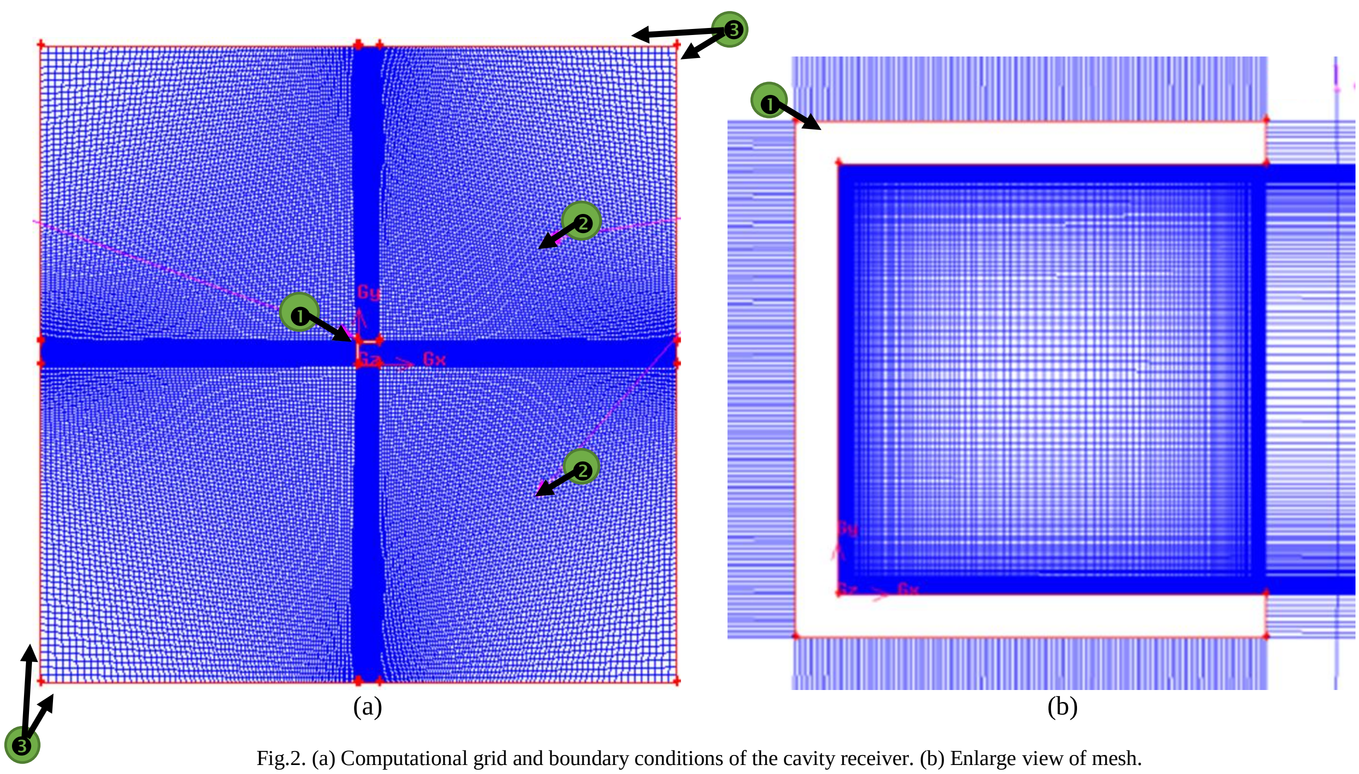
Table-3. Comparative Analysis of Average Nusselt Numbers $\overline{Nu}$ of the Open Cavity on Hot Wall for Different Rayleigh Numbers with $Pr = 0.71$

Rayleigh Numbers Ra	Domain Extended (This Work, Boussinesq)	Hinojosa et al. [23] Boussinesq Approximation	Mohamad [22] Boussinesq Approximation	Chakroun et al. [3] Boussinesq Approximation
1×10^4	3.41	3.44	3.44	-
1×10^5	7.44	7.44	7.41	-
1×10^6	14.50	14.51	14.36	-
6.3×10^6	23.89	-	-	24
1×10^7	27.13	27.58	28.6 ± 2.5	-

In order to conduct the grid independence reading, $Ra = 3.76 \times 10^6$ was set with $\Delta T = 40°$. The corresponding convective heat loss values Q_C (Watts) were obtained for the tested numerical grids and recorded in Table-2 [15]. The numerical results independency from the size of the grid was set with the condition having a difference of computation determined as less than 0.5% in between two consecutive grids. As per observed values presented in Table-2, a non-uniform grid of nodes 130×130 inner side of the cavity and 89860 nodes was selected for the entire enclosure [21]. The location of the nodes is selected with stretching function so that the node density in x and y direction is higher near the walls of the cavity. The same procedure is followed in the whole enclosure.

The computational grid and boundary conditions of the cavity receiver is expressed in Fig.2(a). and an enlarge view of the mesh is shown in Fig.2(b)., where label mark represent ❶ Cavity Receiver, ❷ Extended Domain, and ❸ Pressure Inlet [15].

A comparison between the average Nusselt numbers $\overline{Nu}$ of the open cavity on the Hot Wall for different Rayleigh numbers Ra with Prandtl Number $Pr = 0.71$ is determined and recorded in Table-3 [15]. The numbers are obtained for present work and compared with those which are previously reported by the original authors A.A. Mohamad [22], Chakroun et. al. [3], and Hinojosa et. al. [23] for the same configuration of the open cavity.



(a) (b)

Fig.2. (a) Computational grid and boundary conditions of the cavity receiver. (b) Enlarge view of mesh.
Labels: ❶ Cavity Receiver, ❷ Extended Domain, ❸ Pressure Inlet

Table-4. Values of the Average Convective Nusselt Number $\overline{Nu_C}$ and Convective Heat Loss Qc calculated under Rayleigh Numbers Ra for Inclination Angles for various Degrees

Rayleigh Number →	$Ra = 9.41 \times 10^5$		$Ra = 1.88 \times 10^6$		$Ra = 2.826 \times 10^6$		$Ra = 3.76 \times 10^6$	
Cavity Inclination Angle (φ)	$\overline{Nu_C}$	Q_C	$\overline{Nu_C}$	Q_C	$\overline{Nu_C}$	Q_C	$\overline{Nu_C}$	Q_C
0°	13.88	0.73	16.89	0.88	18.91	1.49	20.47	2.15
15°	12.44	0.33	15.18	0.80	17.03	1.34	18.45	1.94
30°	8.65	0.22	10.26	0.53	11.51	0.90	12.46	1.31
45°	3.64	0.10	4.10	0.22	4.40	0.35	4.62	0.49
60°	2.22	0.06	2.13	0.11	2.19	0.17	2.17	0.22
75°	1.07	0.03	1.10	0.06	1.12	0.09	1.14	0.12
90°	0.95	0.02	0.97	0.05	0.98	0.08	0.99	0.10

In these calculations, the governing equations were solved considering temperature-dependent fluid properties. Setting $\Delta T = 10°K$ to compare with the results drawn through Boussinesq Approximations approach. $\Delta T = 50°K$ is considered, in order to compare with the experimental results. Overall, a good agreement between the predictions of the present work and those reported [21] in the literature is observed in all cases [24] [25] [26] [27].

5. Results and Discussion

The values of the average Convective Nusselt Number $\overline{Nu_C}$ and Convective Heat Loss Q_C are calculated under Rayleigh Numbers Ra with distinct values 9.41×10^5, 1.88×10^6, 2.826×10^6, 3.76×10^6 for Inclination Angles φ for 0°, 15°, 30°, 45°, 60°, 75°, and 90° in order to understand the effect of inclination on Q_C and $\overline{Nu_C}$. The observations are recorded in Table-4.

It is revealed that the values of the Convective Heat Loss Q_C and Average Convective Nusselt Number $\overline{Nu_C}$ varied significantly with respect to the inclination of the cavity [10]. It is further observed that, the maximum convection loss occurs at 0° at all temperatures. With an increase in inclination, the convection loss Q_C reduces to a minimum at 90°, as expected [14]. According to Clausing [28], a part of the cavity experiences stagnation condition.

The boundary separating this area from the remaining part of the cavity can be approximated as a horizontal plane called as "Stagnation Zone Boundary". The part of the cavity above this plane is called "Stagnation Zone" while the below area is called "Convective Zone" where strong convective

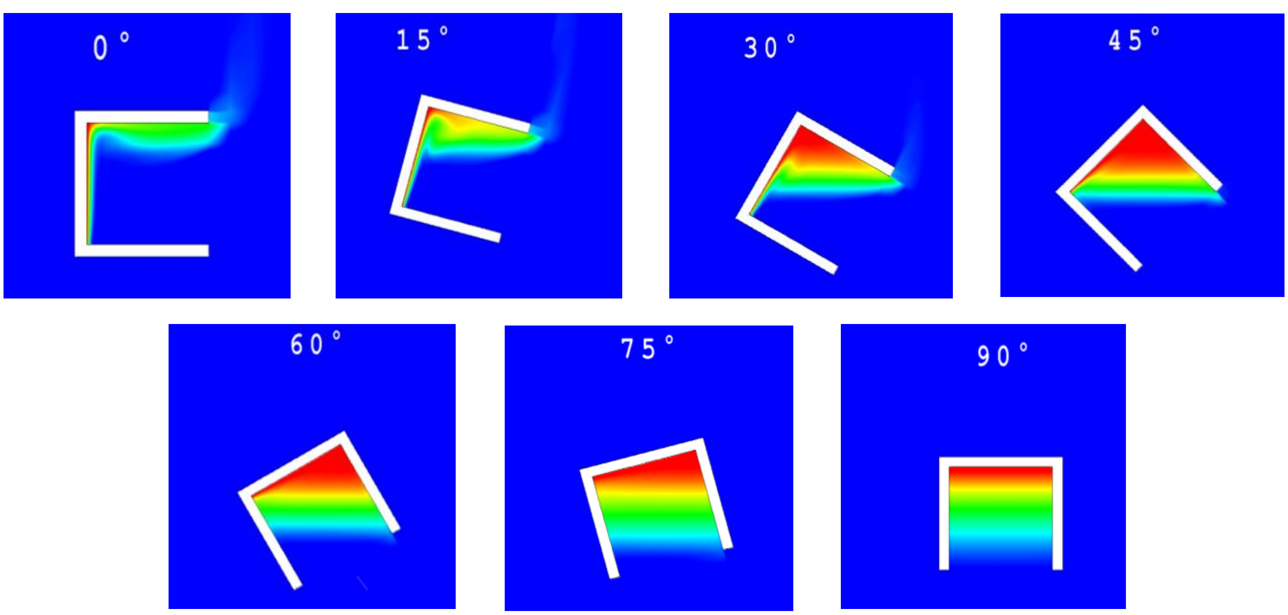


Fig. 3. Temperature contours of cavity at wall temperature $T_W = 340°K$ for various Degrees of Inclination.

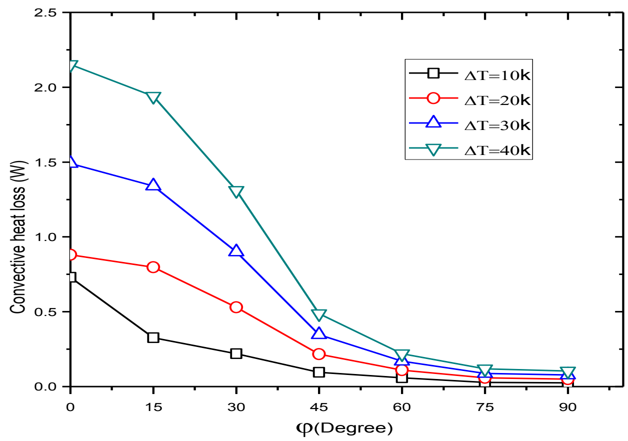


Fig. 4. Variation of the Convective heat loss Q_C with the cavity inclination angle φ

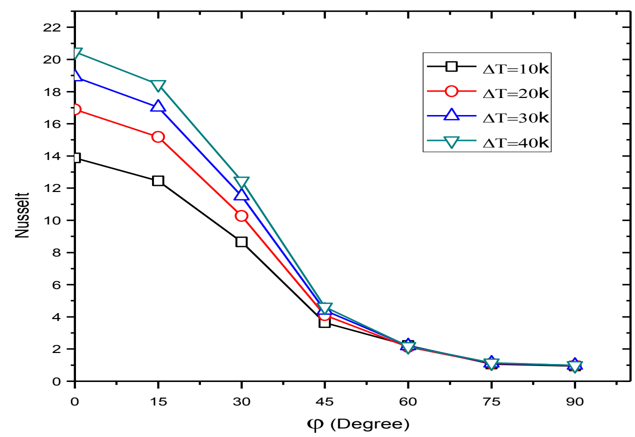


Fig. 5. Variation of the Average Convective Nusselt Number $\overline{Nu_C}$ with the cavity inclination angle φ

air currents are observed. The increase in the inclination results in the increase in stagnant zone volume within the cavity and a decrease in the convective zone [14]. Consequently, the internal wall area contributing to convective loss reduces, thereby reducing convective loss Q_C.

The air temperature profiles and temperature contours of the cavity for various inclinations are simulated for a typical temperature value of $T_W = 340°K$ as expressed in Fig.3. using the FLUENT CFD Software Tool. The convective air occupies almost the complete volume of the cavity at inclination $\varphi = 0°$. The area of high temperature stagnant air also increases as the cavity inclination increases and approaches gradually from 0° approaching to 90° with steps of 15°, the maximum being at 90°. Thus, the cavity area participating in convection decreases as the inclination increases and hence the convective heat loss Q_C decreases [14].

The values of the Convective Heat Loss Q_C and Average Convective Nusselt Number $\overline{Nu_C}$ that are reflected in Table-4 for different Rayleigh numbers Ra are graphically plotted in Fig.4. and Fig.5. It is noticed that the corresponding mean values were used at instances when the convective heat loss Q_C and Convective Nusselt number Nu_C did not reach the steady state [10] while elaborating these graphs. We can view that for all the studied Rayleigh numbers Ra the absolute maximums of convective heat loss and Nusselt number occurs at $\varphi = 0°$ inclination and the absolute minimum at $\varphi = 90°$.

This is due to the fact that the increase in the inclination results in the increase in stagnant zone area within the cavity and decrease in the convective zone. Consequently, the internal wall contributing to convective loss reduces, thereby reducing convective loss and Nusselt number.

6. Conclusions

In this paper the numerical calculations of convective heat loss Q_C and average Nusselt numbers $\overline{Nu_C}$ for the natural convection in a tilted open cavity were presented. After overall study and experimental observations, we reached to the following conclusions:

- The convective heat loss Q_C and average Nusselt number $\overline{Nu_C}$ change substantially with respect to the whole angle range of the tilted cavity.
- For a fixed angle φ, the convective heat loss Q_C increases with the Rayleigh number Ra.
- For a fixed angle φ, the average Nusselt numbers $\overline{Nu_C}$ increases with the Rayleigh number Ra except for angles of 60°, 75° and 90°, where the average convective Nusselt number stays almost constant.
- It can be seen that at 0° inclination, the convective zone covers the entire surface area of the receiver while the percentage of the convective zone is the smallest at 90° inclination.

The study of convective and radiative heat loss can help us in designing and optimizing the cavity heat receiver for both concentrating Solar Tower and Solar Parabolic Dish. There remains a lot of work on the effect of radiation and the effect of the opening ratio, the effect of wind condition on the convective heat loss, and determination of boundary separating stagnation and the convective zone.

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