COMBINED LAMINAR NATURAL CONVECTION
AND SURFACE RADIATION IN TOP OPEN CAVITIES
WITH RIGHT SIDE OPENING

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Abstract

The numerical analysis of combined laminar natural convection and surface radiation in the two-dimensional top open cavities with left side isothermal wall and different right side wall openings is presented. The cavity is filled with natural air (Pr = 0.70) as the fluid medium. The governing equations, i.e., Navier-Stokes Equation in the stream function-vorticity form and Energy Equation are solved for a constant property fluid under Boussinesq approximation. For discretization of these equations, a finite volume technique is used. For the radiation calculations, radiosity-irradiation formulation is used and the shape factors is calculated by using Hottel’s crossed-string method. The effects of geometrical parameters like aspect ratio (A = 1-2), side vent ratio and side port ratio (W2 = 0.25-1) as well as the other prime governing parameters like Rayleigh number (Ra = 10⁴-10⁷) and the surface emissivity of the cavity walls (ε = 0.05-0.85) are analysed. The cooling of left side hot wall in the cavity with different openings in the right side wall under natural convection and surface radiation is analysed. Correlations are developed for the average convection Nusselt number and average radiation Nusselt number at the left side isothermal hot wall.

Keywords: Natural convection, Optimization, Surface radiation, Thermal management, Top and right side open cavities.
1. Introduction

Natural convection heat transfer in the closed and open cavities has witnessed a recent renewed interest by the engineering researchers and industrial scientists. This included the well-elaborated experimental studies and the numerical investigations with several different approaches. The reasons attributed for it underlies in its important role in several engineering and practical science problems and applications like the aerospace engineering, cryogenics, air conditioning and refrigeration, power plant engineering, food processing engineering, chemical and metallurgical engineering, fire control engineering, chemical and biological warfare, solar energy, meteorological predictions, ocean engineering, safe design of nuclear reactors, etc. It has wide applications in steam generators, effective cooling of electrical generators and internal combustion engines also.

Despite the lower values of the convective heat transfer coefficient, the natural convection finds its wide application in numerous electrical, electronic and other industrial cooling applications. Cooling by the unaided natural convection is one of the most important considerations for the safe thermal design of the electrical and electronic systems. Efficient thermal design for enhanced cooling by natural convection is always a prime criterion for a successful thermal design. It is a popular and widely used cooling process due to its inherent simplicity and reliability. Natural convection cooling process is noiseless, energy saving and economical also. One of the interesting features of natural convection is that it is inherently present in a thermal system in some form.

In the systems operating under natural convection and surface radiation, the heat transfer by surface radiation is significant in comparison to the convective heat transfer even at the moderate temperatures differences. Thus, the radiative heat transfer must be accounted in for calculating the total heat transfer or the overall thermal performance of a thermal system. Natural convection and surface radiation interaction affect the airflow pattern inside the cavity as well as the heat dissipation from the heat sources like electronic components, etc.

In an open cavity, there is a continuous intake of fresh air from the ambient or atmosphere and exhaust of heated air back to the ambient reducing the stratification observed in closed cavities. The stratification effect can be further reduced by providing the additional sidewall ports or openings. In the present study, the airflow patterns and its effect on a temperature field inside the right side vented top open cavity are analysed. The present study provides a good simulation of such a cooling process like the cooling of heat generating electronic components, etc.

Natural convection in cavities finds a consistent and prominent place in the literature of heat and mass transfer since a few decades back. The motivation behind the present work is rich and diverse theoretical and experimental literature of computational heat and mass transfer available ranging over the last six decades. Sparrow and Gregg [1] explained that one of the earliest work of modern times is from which, presented the experimental results of natural convection on a vertical plate with the uniform surface heat flux about six decades ago. Patankar and Spalding [2] presented a calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows. Abib and Jaluria [3] made a numerical simulation of the buoyancy-induced flow in a partially open enclosure. Angirasa and Mahajan [4] studied the natural convection from L-shaped corners with the adiabatic and cold isothermal horizontal walls. Balaji and Venkateshan [5]


Recent time has witnessed the numerous research in the computational fluid dynamics to enhance the heat transfer by the different approaches in the different practical situations. Belazizia et al. [20] studied conjugate natural convection in a two-dimensional enclosure with a top heated vertical wall. Menni et al. [21] made a numerical analysis of turbulent forced convection flow in a channel with staggered L-shaped baffles. Singh and Singh [22] made a numerical analysis of combined free convection and surface radiation in the tilted open cavity. Karatas and Derbentli [23] studied the three-dimensional natural convection and radiation in a rectangular cavity with one active vertical wall. Menni and Azzi [24] studied the effect of fin spacing on turbulent heat transfer in a channel with cascaded rectangular-triangular fins. Menni et al. [25] suggested the use of waisted triangular-shaped baffles to enhance heat transfer in a constant temperature surfaced rectangular channel. Chen et al. [26] studied conjugate natural convection heat transfer in an open-ended square cavity partially filled with porous media.

Feng et al. [27] analysed natural convection in a cross fin heat sink. Lugarini et al. [28] studied natural convection and surface radiation in a heated wall, C-shaped fracture. Menni and Azzi [29] designed the air solar channels with the diverse baffle structures and evaluated its performance. Menni and Azzi [30] made a numerical analysis of thermal and aerodynamic fields in a channel with the cascaded baffles. Menni et al. [31] made enhancement of convective heat transfer in smooth air channels with wall mounted obstacles in the flow path. Menni et al. [32] made a computational fluid dynamical analysis of new obstacle design and its impact on heat transfer enhancement in a specific type of air flow geometry. Miroshnichenko and Sheremet [33] analysed turbulent natural convection combined with thermal surface
radiation inside an inclined cavity having a local heater. Prasad et al. [34] presented a systematic approach for optimal positioning of heated side walls in a side vented open cavity under natural convection and surface radiation. Prasad et al. [35] studied coupled laminar natural convection and surface radiation in partially right side open cavities. The above research works show the importance of natural convection and surface radiation heat transfer in different practical situations.

On the basis of the above literature survey, it is found that there are several research papers discussing the natural convection and surface radiation in open cavities having sidewall openings. However, the effect of changing the side wall opening sizes and positions on the combined natural convection and surface radiation in the top open cavities has been less studied so far. Studying the effect of changing side opening sizes and positions on the combined natural convection and surface radiation in the top open cavities is useful for optimizing the cooling of heat sources inside these cavities.

Based on the above literature review and the other literature available, it is found that pure natural convection, as well as natural convection combined with surface radiation, has been studied and analysed in different cavities having different geometries.

This includes the open cavities having different openings in the side walls also. Studying the effect of changing the size as well as the position of openings in detail is needed for the optimization of combined natural convection and surface radiation cooling process. This includes studying the effect of central openings as well as the eccentric openings in the right side wall. The effect of size and position of openings for a wide range of other parameters are analysed and the correlations are developed for the average convection Nusselt number ($\overline{\text{Nu}}_{c}$) and the average radiation Nusselt number ($\overline{\text{Nu}}_{R}$) at the left side isothermal hot wall.

The present analysis finds its wide practical application in electronic cooling, solar energy, air conditioning and refrigeration. The prime interest of the present study is to analyse and optimize the cooling of heat sources in the top open cavity by finding the optimal positioning of the port or opening in the right side wall. The results of the present study are useful in optimizing the cooling of heat sources in such cavities.

2. Mathematical Formulation

This section presents the mathematical formulation used for natural convection and surface radiation as well as the boundary conditions used for the different closed and open boundaries.

2.1. Formulation for natural convection

The two-dimensional steady incompressible laminar natural convection heat transfer is analysed in a right side vented top open cavity having height ‘$H$’, horizontal width ‘$d$’, right bottom and right top vent wall heights ‘$w_1$’ and ‘$w_3$’ respectively and height of right port ‘$w_2$’. From the geometry of the cavity, $H = w_1 + w_2 + w_3$. Here, the cartesian coordinate system is used as shown in Fig. 1.
The governing Eqs. (1), (2) and (3) in stream function ($\psi$)-vorticity ($\omega$) form, for a constant property fluid under the Boussinesq approximation, in the non-dimensional form are:

$$
\frac{U}{X} \frac{\partial \omega}{\partial X} + \frac{V}{Y} \frac{\partial \omega}{\partial Y} = Pr \left[ \frac{\partial^2 \omega}{\partial X^2} + \frac{\partial^2 \omega}{\partial Y^2} \right] - Ra \frac{\partial \theta}{\partial Y} \\
\frac{\partial^2 \psi}{\partial X^2} + \frac{\partial^2 \psi}{\partial Y^2} = - Pr \omega \\
\frac{U}{X} \frac{\partial \theta}{\partial X} + \frac{V}{Y} \frac{\partial \theta}{\partial Y} = \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}
$$

where $U = \frac{\partial \psi}{\partial Y}$, $V = - \frac{\partial \psi}{\partial X}$ and $\omega = \frac{\partial V}{\partial X} - \frac{\partial U}{\partial Y}$.

The variables normalized with their normalized value are shown in Eq. (4).

$$
X = \frac{x}{d}, \ Y = \frac{y}{d}, \ U = \frac{u d}{\alpha}, \ V = \frac{v d}{\alpha}, \ \Psi = \frac{\psi d}{\alpha}, \ \omega = \frac{\omega d^2}{\nu}, \ \theta = \frac{T - T_\infty}{T_h - T_\infty}
$$

### 2.2. Formulation for surface radiation

According to Balaji and Venkateshan [5], the radiosity-irradiation formulation is used to describe the surface radiation. For an elemental area on the boundary of the cavity, the non-dimensional radiosity is given by the Eq. (5).

$$
J_i = \varepsilon_i \left( \frac{T_i}{T_h} \right)^4 + \left( 1 - \varepsilon_i \right) \sum_{j=1}^{2(m+n-2)} F_{ij} J_j \quad \text{where } i = 1, 2(m+n-2)
$$

The cavity walls are assumed to be diffuse and grey, i.e., independent of direction and wavelength. Here the emissivity of all the cavity walls is considered to be the same and equal to $\varepsilon$.

At the bottom, right side bottom and right side top adiabatic walls, the convection and radiation energy transfer balance each other.

Hence, for the bottom adiabatic wall in Eq. (6):
\[-\frac{\partial \theta}{\partial X} = N_r c (J-G) \quad (6)\]

and for the right side bottom and right side top adiabatic wall in Eq. (7):
\[\frac{\partial \theta}{\partial Y} = N_r c (J-G) \quad (7)\]

Here the shape factors \(F_{ij}\) are calculated by using the Hottel’s crossed string method.

2.3. Boundary conditions

The boundary conditions used for the computational domain enclosed by the cavity are specified in the Eqs. (8) to (13). Based on several research papers by Balaji and Venkateshan [5], Rao et al. [6] and Singh and Venkateshan [8], the stream function and vorticity boundary conditions are specified on each of the boundaries in this section. The stream function and vorticity boundary conditions specified on each of the boundaries in this section are the clarity and simplicity.

For left isothermal wall AF:
\[0 < X < A, \quad Y = 0, \quad U = 0, \quad V = 0 \text{ or } \nu = 0, \quad \omega = \frac{1}{P_r} \frac{\partial^2 \psi}{\partial Y^2}, \quad \theta = 1 \quad (8)\]

For bottom adiabatic wall AB:
\[X = 0, \quad 0 < Y < 1, \quad U = 0, \quad V = 0 \text{ or } \nu = 0, \quad \omega = \frac{1}{P_r} \frac{\partial^2 \psi}{\partial X^2} \quad (9)\]

For right side bottom adiabatic vent wall BC:
\[0 < X < W_1, \quad Y = 1, \quad U = 0, \quad V = 0 \text{ or } \nu = 0, \quad \omega = \frac{1}{P_r} \frac{\partial^2 \psi}{\partial Y^2}, \quad \frac{\partial \theta}{\partial X} = N_r c (J-G) \quad (10)\]

For right side top adiabatic vent wall DE:
\[W_2 < X < A, \quad Y = 1, \quad U = 0, \quad V = 0 \text{ or } \nu = c \text{ (an unknown constant)}, \quad \omega = \frac{1}{P_r} \frac{\partial^2 \psi}{\partial Y^2}, \quad \frac{\partial \theta}{\partial Y} = N_r c (J-G) \quad (11)\]

Along the right open boundary CD:
\[Y = 1, \quad W_1 < X < W_2, \quad \omega = 0, \quad \frac{\partial U}{\partial X} = \frac{\partial V}{\partial Y} = \frac{\partial^2 \psi}{\partial X \partial Y} = 0 \quad (12)\]

Here in this case neither the vertical velocity \((U)\) nor the horizontal velocity \((V)\) is assumed zero. This is a mixed boundary condition providing the smooth variation of the two velocity components. From the definition of stream function, the equation of continuity is satisfied everywhere. However, this smoothness boundary condition makes the continuity equation satisfied everywhere as well as makes both of its derivative terms identically zero along the opening in the right side wall.
Rao et al. [6] have implemented this boundary condition in studying the problem of laminar mixed convection from a heated vertical plate and found this boundary condition the most appropriate. Singh and Venkateshan [8] compared this boundary condition with the other possible boundary conditions for studying the natural convection and surface radiation inside vented open cavities and found it to be the most appropriate.

**Along the top open boundary EF:**

According to the Balaji and Venkateshan [5], the boundary condition in such a case is specified as:

\[
X = A, \quad 0 < Y < 1, \quad \frac{\partial \omega}{\partial X} = 0, \quad V = -\frac{\partial \psi}{\partial X} = 0.
\]

If \( U = \frac{\partial \psi}{\partial Y} > 0 \), \( \frac{\partial \theta}{\partial X} = 0 \), or, if \( U = \frac{\partial \psi}{\partial Y} < 0 \), then \( \theta = 0 \) \hspace{1cm} (13)

**3. Method of Solution**

The governing Eqs. (1) to (3) are transformed into finite difference equations using the finite volume based finite difference method. Then the Gauss-Seidel iterative procedure is used to solve the algebraic equations obtained. The set of discretized equations obtained are solved by using a line-by-line procedure of the Tri-Diagonal Matrix Algorithm (TDMA) or the Thomas algorithm.

Singh and Venkateshan [8] explained that a suitable grid size of 41 \( \times \) 51 is selected for the computational domain on the basis of grid sensitivity analysis presented later in Section 4.1 A semi-cosine and a cosine function have been chosen to generate the grids along \( X \) and \( Y \) directions respectively in the computational domain of the cavity. These semi-cosine and cosine grids are very fine near the solid boundaries, where the gradients are very steep, while they are relatively coarser in the remaining part of the domain as shown in Fig. 2.

Derivative boundary conditions are implemented by three-point formulae using the Lagrangian polynomial. The integration required in calculations is performed by using the Simpson’s one-third rule for the non-uniform step size. Upwinding has been used for representing the advection terms to ensure the stable and convergent solutions.

Under-relaxation with the relaxation parameter, 0.1 is used for all the equations except for the radiosity equations, where the relaxation parameter 0.5 is used. A convergence criterion (\( \delta \)) in the percentage form has been defined as in Eq. (14).

\[
\delta = \left| \frac{\zeta_{\text{new}} - \zeta_{\text{old}}}{\zeta_{\text{new}}} \right| \times 100,
\]

where \( \zeta \) is any dependent variable like \( \psi, \omega, \theta, J \) and \( G \), over which the convergence test is to be applied. Here the subscripts “old” and “new” refers to the present and previous values of \( \zeta \) calculated in any two successive iterations.

A convergence criterion of 0.1 \% or \( 10^{-3} \) has been used for stream function, vorticity and temperature, whereas the convergence criterion of 0.01 \% or \( 10^{-4} \) has been used for the radiosity.
4. Results and Discussion

This section presents the streamlines and isotherms inside the cavity and the effect of aspect ratio, port size and Rayleigh number on it. The effect of emissivity on the bottom wall temperature, non-dimensional temperature and non-dimensional vertical velocity at the different horizontal cross-sections of the cavity are discussed. The variation of local \( \text{Nu}_C \) and local \( \text{Nu}_R \) at the left isothermal hot wall with the height at different emissivity and the variation of \( \text{Nu}_C \) and \( \text{Nu}_R \) at the left isothermal hot wall with the eccentricity of port or with the different position of the opening are also analysed. The effect of size and position of openings for a wide range of other parameters are analysed and the correlations are developed for the average convection Nusselt number (\( \text{Nu}_C \)) and the average radiation Nusselt number (\( \text{Nu}_R \)) at the left side isothermal hot wall.

The range of parameters for the present study is listed as in Table 1.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rayleigh number, ( R_{AH} )</td>
<td>( 10^4-10^7 )</td>
</tr>
<tr>
<td>Conduction-Radiation Parameter, ( N_{rc} )</td>
<td>10-50</td>
</tr>
<tr>
<td>Emissivity, ( \varepsilon )</td>
<td>0.05-0.85</td>
</tr>
<tr>
<td>Temperature ratio, ( T_R )</td>
<td>0.8-0.9</td>
</tr>
<tr>
<td>Aspect ratio, ( A )</td>
<td>1.0-2.0</td>
</tr>
<tr>
<td>Port ratio, ( W_2 )</td>
<td>0.25-1.0</td>
</tr>
</tbody>
</table>

4.1. Grid sensitivity study

As suggested by Singh and Venkateshan [8], a grid sensitivity or grid independence study is performed to find the optimum grid size. The present problem involves both the natural convection and surface radiation. These two grid sizes ‘m’ and ‘n’ affect \( \text{Nu}_C \) and \( \text{Nu}_R \) differently, where \( m \) and \( n \) are the total number of grid points in the horizontal \( Y \) direction and the vertical \( X \) direction respectively. Hence, it is necessary to analyse the variations in the two Nusselt numbers in order to obtain the optimum grid size. Here the grid sensitivity analysis is done into two parts. In the first part, ‘n’ is fixed and in the second part ‘m’ is fixed at a moderate value of 31.
Tables 2 and 3 shows the effect of horizontal and vertical grid sizes, i.e., m and n on \( \overline{Nu_c} \), \( \overline{Nu_R} \) and \( \overline{Nu_T} \) for a typical case with \( A = 1.0, N_{Re} = 45.2507, Pr = 0.70, Ra_H = 1.075 \times 10^6, T_R = 0.872, W_2 = 0.50, \varepsilon = 0.85 \).

Here in Table 2, first ‘n’ is kept fixed at 31, whereas ‘m’ is varied from 21 to 61. In this table, it is observed that the percentage change in \( \overline{Nu_c} \), \( \overline{Nu_R} \) and \( \overline{Nu_T} \) is less than 1.5 % at the grid size 41 \( \times \) 31. There is no significant change in \( \overline{Nu_R} \) for the grid size 41 \( \times \) 31 and above. There the percentage change in \( \overline{Nu_c} \) and \( \overline{Nu_T} \) is less than 3.5 % for the grid sizes 41 \( \times \) 31 and above. Thus, ‘m’ may be fixed at 41.

Here in Table 3, ‘m’ is kept fixed at 31, whereas ‘n’ is varied from 21 to 61. Here the percentage change in \( \overline{Nu_c} \), \( \overline{Nu_R} \) and \( \overline{Nu_T} \) is found to be less than 0.50 % for the grid size 31 \( \times \) 51 and above. Hence, ‘n’ may be fixed at 51.

Thus, the grid size 41 \( \times \) 51 may be selected as an optimal grid size for the present study. The change in various parameters like the bottom wall temperature and the vertical air velocity at the different horizontal sections is not significant with the further increase in the grid size. The validity of optimum grid size with some other different relevant parameters is also analysed by analysing streamlines and isotherms at different grid sizes, which are not presented here. Any further increase in the grid size increases the computational work manifolds without many significant improvement in the accuracy of results.

### Table 2. Grid independence study n = 31, m varied. (for \( A = 1.0, N_{Re} = 45.2507, Pr = 0.70, Ra_H = 1.075 \times 10^6, T_R = 0.872, W_2 = 0.50, \varepsilon = 0.85 \)).

<table>
<thead>
<tr>
<th>m ( \times ) n</th>
<th>( \overline{Nu_c} )</th>
<th>( \overline{Nu_R} )</th>
<th>( \overline{Nu_T} )</th>
<th>% Change in ( \overline{Nu_c} )</th>
<th>% Change in ( \overline{Nu_R} )</th>
<th>% Change in ( \overline{Nu_T} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>21 ( \times ) 31</td>
<td>19.01</td>
<td>13.86</td>
<td>32.87</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>31 ( \times ) 31</td>
<td>16.73</td>
<td>13.84</td>
<td>30.57</td>
<td>11.99</td>
<td>0.14</td>
<td>7.00</td>
</tr>
<tr>
<td>41 ( \times ) 31</td>
<td>16.49</td>
<td>13.85</td>
<td>30.34</td>
<td>1.43</td>
<td>0.07</td>
<td>0.75</td>
</tr>
<tr>
<td>51 ( \times ) 31</td>
<td>15.92</td>
<td>13.82</td>
<td>29.74</td>
<td>3.46</td>
<td>0.22</td>
<td>1.98</td>
</tr>
<tr>
<td>61 ( \times ) 31</td>
<td>15.74</td>
<td>13.82</td>
<td>29.56</td>
<td>1.13</td>
<td>0</td>
<td>0.60</td>
</tr>
</tbody>
</table>

### Table 3. Grid independence study m = 31, n varied (for \( A = 1.0, N_{Re} = 45.2507, Pr = 0.70, Ra_H = 1.075 \times 10^6, T_R = 0.872, W_2 = 0.50, \varepsilon = 0.85 \)).

<table>
<thead>
<tr>
<th>m ( \times ) n</th>
<th>( \overline{Nu_c} )</th>
<th>( \overline{Nu_R} )</th>
<th>( \overline{Nu_T} )</th>
<th>% Change in ( \overline{Nu_c} )</th>
<th>% Change in ( \overline{Nu_R} )</th>
<th>% Change in ( \overline{Nu_T} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>31 ( \times ) 21</td>
<td>16.46</td>
<td>13.70</td>
<td>30.16</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>31 ( \times ) 31</td>
<td>16.73</td>
<td>13.84</td>
<td>30.57</td>
<td>1.64</td>
<td>1.02</td>
<td>1.36</td>
</tr>
<tr>
<td>31 ( \times ) 41</td>
<td>16.96</td>
<td>14.04</td>
<td>31.00</td>
<td>1.38</td>
<td>1.44</td>
<td>1.41</td>
</tr>
<tr>
<td>31 ( \times ) 51</td>
<td>17.04</td>
<td>14.04</td>
<td>31.08</td>
<td>0.47</td>
<td>0</td>
<td>0.26</td>
</tr>
<tr>
<td>31 ( \times ) 61</td>
<td>17.10</td>
<td>14.06</td>
<td>31.16</td>
<td>0.35</td>
<td>0.14</td>
<td>0.26</td>
</tr>
</tbody>
</table>

### 4.2. Validation

Ramesh and Merzkirch [7] reported that the program code is validated by comparing the results of present numerical study for the case \( W_1 = 0.50, W_2 = 0.50 \) and \( W_3 = 0 \) with the available results of experimental work. Ramesh and Merzkirch [7] have presented the experimental results for a very narrow range of Rayleigh numbers \( 10^5 \text{--} 10^6 \), \( A = 2.0, W_2 = 0.50 \) with \( \varepsilon = 0.05 \) (called low emissivity) and \( \varepsilon = 0.85 \) (called high emissivity). A typical case considered in the paper corresponds to
the following set of parameters, i.e., $H = 0.070$ m, $d = 0.035$ m, $T_c = 293$ K, $\Delta T = 43$ K, $e = 0.05$ and 0.85.

The total Nusselt number ($N_u$) corresponding to the low emissivity case for the present study and the study by Ramesh-Merzkirch are 14.97 and 17.00 respectively. The total Nusselt number corresponding to the case of high emissivity for the present study and study by Ramesh-Merzkirch are 27.51 and 28.70 respectively. These values show a good agreement between the present study and the referred experimental work.

The present problem involves the interaction between the natural convection and surface radiation in the right side vented, top open cavity. Percentage of heat transferred by natural convection and surface radiation is an important parameter to be compared between the present work and the experimental work of Ramesh-Merzkirch at the different Rayleigh numbers.

The percentage of heat transferred by natural convection and surface radiation at different Rayleigh number and emissivity with the same parameters are also very close to their experimental values as shown in Fig. 3. There is an excellent agreement between the present numerical results and referred experimental work for the different Rayleigh numbers and emissivity.

4.3. Effect of port size on streamlines and isotherms

In Fig. 4, streamlines are shown for the aspect ratio, $A = 1.0$ and the port ratio $W_2 = 0.25, 0.50, 0.75$ and 1.00. In Fig. 6, streamlines are shown for the aspect ratio, $A = 2.0$ and the port ratio $W_2 = 0.50, 0.60, 0.75$ and 1.00.

In Fig. 5, isotherms are shown for the aspect ratio, $A = 1.0$ and the port ratio $W_2 = 0.25, 0.50, 0.75$ and 1.00. In Fig. 7, isotherms are shown for the aspect ratio, $A = 2.0$ and the port ratio $W_2 = 0.50, 0.60, 0.75$ and 1.00.
This streamline pattern indicates that the air enters from the top opening as well as the right side port and leaves primarily through the top opening. It is observed that two circulation loops are formed in for $A = 1.0$ in all cases and for $A = 2.0$ in cases $W_2 = 0.75$ and $W_2 = 1.00$. Here three circulation loops are formed for $A = 2.0$ in cases $W_2 = 0.50$ and $W_2 = 0.60$. These loops are formed as a result of thermal boundary layers’ formation on the different walls and their interactions.

The ambient air enters in the cavity from the top opening and the right side port. After gaining heat from the hot walls, the air rises upwards. Isotherm contours show the formation of thermal boundary layers along the left isothermal hot wall as well as the bottom adiabatic wall, the right bottom and the right top adiabatic vent walls due to heating by the radiative heat transfer.

Development of thermal boundary layers along the highly emissive adiabatic walls is strong evidence of radiative interactions between the walls of the cavity. Streamlines cut the openings as shown as the dashed lines in Fig. 1 at different angles as shown in Figs. 4 and 6, which shows that the air streams enter into the cavity through the openings at different angles. This is one of the evidence showing the implementation of correct boundary conditions at the port.

Effect of port size on streamlines ($A = 1.0$)

![Streamlines for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for $A = 1.0$, $N_{rc} = 45.2507$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $\varepsilon = 0.85$.](image)

Fig. 4. Streamlines for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for $A = 1.0$, $N_{rc} = 45.2507$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $\varepsilon = 0.85$.

Effect of port size on isotherms ($A = 1.0$)

![Isotherms for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for $A = 1.0$, $N_{rc} = 45.2507$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $\varepsilon = 0.85$.](image)

Fig. 5. Isotherms for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for $A = 1.0$, $N_{rc} = 45.2507$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $\varepsilon = 0.85$. 

Effect of port size on streamlines \((A = 2.0)\)

![Streamlines for different port sizes](image)

Fig. 6. Streamlines for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for \(A = 2.0\), \(N_r = 22.6253\), \(Pr = 0.70\), \(Ra_H = 1.075 \times 10^6\), \(T_R = 0.872\), \(\varepsilon = 0.85\).

Effect of port size on isotherms \((A = 2.0)\)

![Isotherms for different port sizes](image)

Fig. 7. Isotherms for combined natural convection and surface radiation in cavity at different port sizes in the right side wall for \(A = 2.0\), \(N_r = 22.6253\), \(Pr = 0.70\), \(Ra_H = 1.075 \times 10^6\), \(T_R = 0.872\), \(\varepsilon = 0.85\).

4.4. Effect of rayleigh number on streamlines and isotherms

Figure 8 shows the streamlines for the aspect ratio, \(A = 1.0\) at the Rayleigh numbers \(1.0 \times 10^3\), \(1.0 \times 10^4\), \(1.0 \times 10^5\) and \(1.0 \times 10^6\) respectively. Figure 9 shows the isotherms for the aspect ratio, \(A = 1.0\) at the Rayleigh numbers \(1.0 \times 10^3\), \(1.0 \times 10^4\), \(1.0 \times 10^5\) and \(1.0 \times 10^6\) respectively.

Here at the lower Rayleigh numbers \((Ra_H = 1.0 \times 10^3\) and \(1.0 \times 10^4\)), the small values of stream function indicate weak circulation of air inside the cavity. Air enters from the right part of the cavity, reaches to the bottom wall and rises along the left isothermal hot wall. At the higher Rayleigh numbers \((Ra_H = 1.0 \times 10^5\) and \(1.0 \times 10^6\)), air enters from the central part of the cavity, reaches to the bottom and
rises along both the left isothermal hot wall and right adiabatic vent wall. A small amount of air enters through the opening or port in the right side vent wall. At the higher Rayleigh numbers, the higher the values of stream function indicate the higher air circulation in the cavity.

Here at the lower Rayleigh numbers \( (Ra_H = 1.0 \times 10^3 \text{ and } 1.0 \times 10^4) \), the isotherms are rarer showing the weak cooling of left isothermal hot wall. The conduction and radiation are the prominent modes of heat transfer at the lower Rayleigh numbers. At the higher Rayleigh numbers \( (Ra_H = 1.0 \times 10^5 \text{ and } 1.0 \times 10^6) \), the isotherms become denser indicating the increased natural convection cooling of the left isothermal hot wall. The thermal boundary layers formed at the bottom adiabatic wall and right adiabatic vent walls indicate the radiative heat transfer between the walls of the cavity. The left isothermal hot wall loses heat to the bottom adiabatic wall, the right bottom and the right top adiabatic vent walls by the surface radiation, which in turn loses heat to air by the natural convection.

**Effect of Rayleigh number on streamlines**

![Streamlines](image)

Fig. 8. Streamlines for combined natural convection and surface radiation in cavity at different Rayleigh numbers for \( A = 1.0, N_{rc} = 45.2507, Pr = 0.70, W2 = 0.50, \varepsilon = 0.85 \).

**Effect of Rayleigh number on isotherms**

![Isotherms](image)

Fig. 9. Isotherms for combined natural convection and surface radiation in cavity at different Rayleigh numbers for \( A = 1.0, N_{rc} = 45.2507, Pr = 0.70, W2 = 0.50, \varepsilon = 0.85 \).

4.5. Variation of bottom wall temperature with emissivity

Figure 10 shows the variation of the non-dimensional bottom wall temperature with the non-dimensional horizontal distance along the width of the cavity at three different emissivities \( (\varepsilon = 0.05, 0.45 \text{ and } 0.85) \). It can be observed that at the lower emissivity \( (\varepsilon = 0.05) \), the non-dimensional temperature of the bottom wall is very
close to zero just after a very small distance from the left isothermal hot wall. At the higher emissivity ($\epsilon = 0.45$ and $\epsilon = 0.85$), the temperature of the bottom wall of the cavity is significantly increased. At the moderate emissivity ($\epsilon = 0.45$) and at the very high emissivity ($\epsilon = 0.85$) of the cavity walls, the temperature profile of bottom horizontal adiabatic wall shows the evidence of radiative interactions between the walls of the cavity. It is also evidence of increased radiative interactions with the increased emissivity of the cavity walls.

**Fig. 10.** Variation of non-dimensional bottom wall temperature with non-dimensional horizontal distance along the width of cavity at different emissivity for $A = 1.0$, $N_r = 45.2507$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $W_2 = 0.50$.

**4.6. Variation of non-dimensional temperature along the width at different horizontal cross-sections**

Figure 11 shows the temperature distribution along the width of cavity at the three different horizontal cross-sections ($X = 0.0$, $1.0$ and $2.0$). Here it can be observed that the bottom adiabatic wall (at $X = 0$) is substantially heated by the thermal radiation heat transfer at the high emissivity ($\epsilon = 0.85$). However, the circulating air in the major portion of the cavity remains at the ambient temperature, except in the boundary layer regions near the left isothermal hot wall, the bottom adiabatic wall, the right bottom and the right top adiabatic vent walls.

**Fig. 11.** Temperature profiles along the width of cavity at three horizontal cross-sections for $A = 2.0$, $N_r = 22.6253$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $W_2 = 0.50$, $\epsilon = 0.85$. 

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February 2019, Vol. 14(1)
4.7. Variation of non-dimensional vertical velocity $U$ along the width at different horizontal cross-sections

Figure 12 shows the variation of non-dimensional vertical velocity $U$ along the width of cavity at the four different horizontal cross-sections ($X = 0.5, 1.0, 1.5$ and $2.0$).

It is observed that the high non-dimensional vertical velocity exists near the left wall and it is highest at the top cross-section. The velocity profile has two peaks located adjacent to the left isothermal hot wall and right adiabatic wall. The higher peak is located close to the left isothermal hot wall. The lower peak is located close to the right adiabatic vent wall. The radiation heat transfer from the left isothermal hot wall heats the bottom adiabatic wall, the right bottom and the right top adiabatic vent walls. These two walls lose heat to the air by the natural convection heat transfer.

![Figure 12. Variation of vertical velocity $U$ along the width of cavity at four horizontal cross-sections for $A = 2.0$, $N_c = 22.6253$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $W_2 = 0.50$, $\varepsilon = 0.85$.]

4.8. Variation of Local $Nu_C$ and Local $Nu_R$ at the left isothermal hot wall with the height at different emissivity

Figure 13 shows the variation of local $Nu_C$ and local $Nu_R$ at the left isothermal hot wall with the height at different emissivity.

At both the lower and higher emissivity, the local $Nu_C$ decreases with the height. The incoming air stream interacts with the bottom of the wall, picks some heat and rises along the left isothermal hot wall. Thus, the temperature difference between the left hot wall and hot air stream decreases. This reduces the temperature gradient in the perpendicular direction to the left isothermal hot wall. This is exhibited by decreased cooling of the left hot wall with height and decreased local $Nu_C$.

At the lower emissivity of cavity walls, a small radiative heat transfer is observed. Thus, at the small emissivity, the local $Nu_R$ is small and it is almost constant with the height. At the higher emissivity of the cavity walls, the radiative heat transfer is comparable to the convective heat transfer. Thus, at the higher emissivity, the local $Nu_R$ is comparable to the local $Nu_C$. The local $Nu_R$ increases with the height at the points closer to the open end. At these points, more thermal
Radiations are escaped without being irradiated back by the bottom adiabatic walls, the right bottom and the right top adiabatic vent walls.

**Fig. 13. Variation of Local NuC (NUC) and Local NuR (NUR) at the left isothermal hot wall with the height at different emissivity for A = 1.0, Nuc = 22.6253, Pr = 0.70, RaH = 1.075 \times 10^6, T_R = 0.872, W_2 = 0.50.**

4.9. Variation of Nu_C and Nu_R with eccentricity or with different position of opening or port

Figure 14 shows the variation of Nu_C and Nu_R at the left isothermal hot wall with the eccentricity of port or with the different position of the opening. The eccentricity of the port in the present problem may be defined as the amount of its vertical displacement from its central position.

When the port is at its central position, W_1 = W_3. However, when the port is at an asymmetric position or it is eccentric, then W_1 \neq W_3.

Thus, the eccentricity of the port in the right adiabatic wall

\[
\frac{H}{2} - (W_1 + \frac{W_2}{2})
\]

\[
= \left| \frac{W_1 + W_2 + W_3}{2} - (W_1 + \frac{W_2}{2}) \right|
\]

\[
= \left| \frac{W_1 - W_3}{2} \right| 
\]  
(Here, W_1 \neq W_3) \quad (15)

Due to the non-symmetrical geometry of the problem, it is quite reasonable to predict that the effects of changing the position of the port in the upward direction and in the downward direction with respect to its central position will be quite different.
Combined Laminar Natural Convection and Surface Radiation in Top . . . . .

Fig. 14. Variation of $\overline{Nu}_C$ (ANUC) and $\overline{Nu}_R$ (ANUR) with the bottom vent ratio $W_1$ for $A = 2.0$, $N_rc = 22.6253$, $Pr = 0.70$, $Ra_H = 1.075 \times 10^6$, $T_R = 0.872$, $W_2 = 0.50$, $\epsilon = 0.85$.

Since $H = W_1 + W_2 + W_3$ is fixed for a given aspect ratio of the cavity from the problem geometry and $W_2$ is fixed in a particular case. Hence, from Eq. (15) it may be concluded that the eccentricity of a port depends on $W_1$ only in a particular case having $W_2$ fixed. Here a case with the moderate size of the port in the right adiabatic wall is selected. In this case, the size of the port is fixed at a moderate value $W_2 = 0.50$. Hence, in this case, the eccentricity of the port equals to $H/2 - (W_1 + 0.25)$. Therefore, it is interesting to analyse the effect of bottom vent ratio $W_1$ or the eccentricity of the port on the $\overline{Nu}_C$ and the $\overline{Nu}_R$ at the left isothermal hot wall.

The emissivity of the cavity walls is a key factor influencing the cooling of the left isothermal hot wall by the natural convection and surface radiation. It is quite interesting to observe the variation of the $\overline{Nu}_C$ and the $\overline{Nu}_R$ at the left isothermal hot wall with bottom vent ratio $W_1$ in the case of low emissivity ($\epsilon = 0.05$) and in the case of high emissivity ($\epsilon = 0.85$).

In the case of low emissivity ($\epsilon = 0.05$), the $\overline{Nu}_C$ at the left isothermal hot wall is almost constant and its value is close to 14. In this case, the $\overline{Nu}_R$ at the left isothermal hot wall is almost constant and its value is close to 1. Thus, at the lower emissivity of the walls, the natural convection is the only dominant mode of heat transfer and there is a poor radiative cooling of left isothermal hot wall.

In the case of high emissivity ($\epsilon = 0.85$), the $\overline{Nu}_C$ at the left isothermal hot wall is almost constant at the value of 14 up to $W_1 = 0.20$ and afterwards, it decreases close to 11 and then again it increases back close to 14. It is quite obvious that the shifting of the opening or port in the upward direction from the bottom-most position may cause some stratification and formation of air circulation loops at the left and right bottom corners or the bottom of the cavity. Thus, it may be clearly interpreted that the presence of opening or port at the bottom of right adiabatic wall...
is the most effective in enhancing the natural convection cooling of the left isothermal hot wall.

In this case, similar observations are made for the $\overline{Nu}_R$ at the left isothermal hot wall also. In the case of high emissivity ($\epsilon = 0.85$), the $\overline{Nu}_R$ is almost constant at the value of 14 up to $WI = 0.20$ and afterwards, it decreases close to 13 and then increases back close to 14.

In the case of high emissivity ($\epsilon = 0.85$), the first important observation is made that the cooling of the left isothermal hot wall by natural convection and surface radiation are comparable. The second observation is made that when the port in the right adiabatic wall is close to the bottom wall, more surface radiation from the left isothermal hot wall is escaped through the port in the right adiabatic wall.

It results in the better radiative cooling of the left isothermal hot wall. However, the effect of shifting of the port in the upward direction from the bottom-most position is more on the $\overline{Nu}_C$ than on the $\overline{Nu}_R$ at the left isothermal hot wall. Thus, an important observation is made that the convective cooling of the left isothermal hot wall is significantly dependent on the size and position of the port in the right adiabatic wall.

Thus, considering the overall heat transfer by the combined natural convection and surface radiation from the left isothermal hot wall, the bottom-most position of the port is the most effective in comparison to its other upper positions. It may be reasonably concluded that the convective cooling of the left isothermal hot wall is more dependent on the size and position of the port in the right adiabatic wall in comparison with its radiative cooling.

4.10. Correlations

Correlations are developed for the average convection Nusselt number and average radiation Nusselt number at the left side isothermal hot wall with the central opening in the right side adiabatic wall ($WI = W3$).

4.10.1. Correlation for $\overline{Nu}_C$

Based on a large set of data (approximately 240 sets data) a correlation for the $\overline{Nu}_C$ is developed as given in Eq. (16).

$$\overline{Nu}_C = 0.3159 \left(\frac{Gr}{Pr}\right)^{0.2803} \left(\frac{Nr_C}{Pr + 1}\right)^{0.1941} (1 + \epsilon)^{-0.0152} A^{-0.2445} (1 + W2)^{-0.0464}$$

The Grashof number directly influences the convective heat transfer, hence it is used in a power law form. $Nr_C$ is a superfluous parameter for the case of study with a single fluid. In this correlation, the emissivity ‘$\epsilon$’ and the port ratio ‘$W2$’ is used as (1 + $\epsilon$) and (1 + $W2$) respectively, because when $\epsilon = 0$ or $W2 = 0$, even then the $\overline{Nu}_C$ is nonzero.

A high correlation coefficient of 0.9934 and a standard error of 0.5773 indicates the excellent goodness of the fit.

Here in the parity plot shown in Fig. 15, we can observe that the numerical data is distributed evenly around the parity line without any bias.
Fig. 15. Parity plot showing the excellent goodness of fit for \( \overline{\text{Nu}_C} \) versus numerical \( \overline{\text{Nu}_C} \).

4.10.2. Correlation for \( \overline{\text{Nu}_R} \)

Based on a large set of data (approximately 240 sets data) a correlation for the \( \overline{\text{Nu}_R} \) is derived as given in Eq. (17).

\[
\overline{\text{Nu}_R} = 8.7490 \text{Gr}^{0.0279} (1-T_R^4)^{0.2888} \text{Nu}_e^{0.0638} \epsilon^{0.9235} A^{0.0624} (1+W^2)^{0.4645}
\]  

(17)

With the increase of emissivity of the cavity wall \( \epsilon \), \( \overline{\text{Nu}_R} \) also, increases and hence \( \epsilon \) is being used in the power law form. The radiant heat flux is proportional to \( (T_R^4 - T_\infty^4) = T_R^4 (1 - T_\infty^4) \), where \( T_R = T_c/T_\infty \). Thus, the \( \overline{\text{Nu}_R} \) is correlated with \( (1 - T_\infty^4) \). \( \text{Nu}_e \) is a superfluous parameter for the case of study with a single fluid. In this correlation, the port ratio \( W^2 \) is used as \( (1 + W^2) \), because when \( W^2 = 0 \), even then the \( \overline{\text{Nu}_R} \) is nonzero.

A high correlation coefficient of 0.9975 and a standard error of 0.2802 indicates the excellent goodness of the fit.

Here in the parity plot shown in Fig. 16, we can observe that the numerical data is distributed evenly around the parity line without any bias.

Fig. 16. Parity plot showing the excellent goodness of fit for \( \overline{\text{Nu}_R} \) versus numerical \( \overline{\text{Nu}_R} \).
4.11. General discussion
From the present numerical investigation, the following observations can be made.

- The ambient air enters in the cavity through the top opening and the ports in the right side adiabatic wall and exits primarily from the top opening.
- The mixed boundary condition is the most appropriate boundary condition for the opening at the right adiabatic wall.
- The left side hot wall loses heat by both of the natural convection and the surface radiation.
- The left side hot wall loses heat to other adiabatic walls by surface radiation, which in turn loses heat to the incoming fresh ambient air by natural convection.
- The emissivity of the cavity walls has a great effect on the streamlines, isotherms and the temperature profiles of the side walls.
- The emissivity of the cavity walls has a significant effect on both the convective and radiative heat transfer in the cavity.
- For a lower emissivity of the cavity walls, the natural convection is the only dominant mode of heat transfer.
- At the higher emissivity of the cavity walls, the heat transfer by the natural convection and the surface radiation are significant and comparable.
- At the higher emissivity of the cavity walls, the radiative heat transfer between the walls of the cavity has a significant effect on the natural convection flow pattern inside the cavity.
- Thermal and velocity boundary layers are formed at all the walls of the cavity. It results from the convective and radiative heat transfer inside the cavity and their interactions.
- Increasing the size of opening or port in the side wall can reduce the stratification and air circulation loops observed in closed cavities or in the cavities having a very small opening.
- The bottom-most position of the opening or the port is the most effective in cooling of the left isothermal hot wall.
- When the size of the port is small, the shifting of the port in the upward direction from the bottom-most position results in the significant decrease in combined convective and radiative cooling of the left isothermal hot wall.
- The convective cooling of the left isothermal hot wall is more dependent on the size and position of the port in the right adiabatic wall in comparison to its radiative cooling.

5. Conclusions
At the higher emissivity of the cavity walls of the cavity, the heat transfer by natural convection and surface radiation is comparable. Therefore, the heat transfer from the hot wall must be determined by taking both of the natural convection and the surface radiation into account. The openings at the lower positions of the right side adiabatic wall are the most effective for the cooling of left side hot wall by the natural convection and the surface radiation. When the size of the port is small, the
shifting of the port in an upward direction from the bottom-most position result in the significant decrease in combined convective and radiative cooling of the left isothermal hot wall. The bottom-most position of the opening or port is the most effective in cooling of the left isothermal hot wall. Studying the combined natural convection and surface radiation in the cavities having different geometries and openings is useful in optimizing the cooling of heat sources in these cavities.

**Nomenclatures**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Aspect ratio ($H/d$)</td>
</tr>
<tr>
<td>$d$</td>
<td>Spacing between left and right walls, m</td>
</tr>
<tr>
<td>$F_{ij}$</td>
<td>Shape factor or view factor between the elements $i$ and $j$</td>
</tr>
<tr>
<td>$G$</td>
<td>Elemental dimensionless irradiation</td>
</tr>
<tr>
<td>$Gr_H$</td>
<td>Grashof number based on $H = g\beta(T_h-T_\infty)H^3/\nu^2$</td>
</tr>
<tr>
<td>$g$</td>
<td>Acceleration due to gravity, 9.81 m/s$^2$</td>
</tr>
<tr>
<td>$H$</td>
<td>Height of the cavity, m</td>
</tr>
<tr>
<td>$J$</td>
<td>Elemental dimensionless radiosity</td>
</tr>
<tr>
<td>$m$</td>
<td>Total number of grid points in horizontal $Y$ direction in the computational domain</td>
</tr>
<tr>
<td>$N_r$</td>
<td>Radiation conduction parameter $= \sigma T_h^4/[k(T_h-T_\infty)/d]$</td>
</tr>
<tr>
<td>$Nu_C$</td>
<td>Convection Nusselt number</td>
</tr>
<tr>
<td>$\bar{Nu}_C$</td>
<td>Average convection Nusselt number</td>
</tr>
<tr>
<td>$Nu_R$</td>
<td>Radiation Nusselt number</td>
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<td>$\bar{Nu}_R$</td>
<td>Average radiation Nusselt number</td>
</tr>
<tr>
<td>$\bar{Nu}_T$</td>
<td>Sum of average convection Nusselt number and average radiation Nusselt number ($= \bar{Nu}_C + \bar{Nu}_R$)</td>
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<td>$n$</td>
<td>Total number of grid points in vertical $X$ direction in the computational domain</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$Ra_H$</td>
<td>Rayleigh number based on $H = Gr_H \cdot Pr$</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$T_h$</td>
<td>Temperature of the left hot wall of cavity</td>
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<tr>
<td>$T_\infty$</td>
<td>Temperature of the ambient</td>
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<td>$U$</td>
<td>Non-dimensional vertical velocity along $X$ axis</td>
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<tr>
<td>$V$</td>
<td>Non-dimensional horizontal velocity along $Y$ axis</td>
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<td>$W_1$</td>
<td>Bottom vent ratio ($w_1/H$)</td>
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<td>$W_2$</td>
<td>Port ratio ($w_2/H$)</td>
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<tr>
<td>$W_3$</td>
<td>Top vent ratio ($w_3/H$)</td>
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<tr>
<td>$w_I$</td>
<td>Dimension of right bottom vent wall</td>
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<td>$w_2$</td>
<td>Dimension of right middle port</td>
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<tr>
<td>$w_3$</td>
<td>Dimension of right top vent wall</td>
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<tr>
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<td>$Y$</td>
<td>Non-dimensional horizontal coordinate ($y/d$)</td>
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**Greek Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
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<tbody>
<tr>
<td>$\beta$</td>
<td>Isobaric co-efficient of volumetric thermal expansion of fluid</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Convergence parameter in percentage: $\left(\zeta_{\text{new}} - \zeta_{\text{old}}\right) / \zeta_{\text{new}} \times 100$</td>
</tr>
</tbody>
</table>
\( \varepsilon \) Emissivity of the walls

\( \zeta \) Symbol for the any dependent variable (\( \psi, \omega, \theta, J, G \)) over which, the convergence test is being applied

\( \theta \) Dimensionless temperature, \( (T - T_\infty)/(T_h - T_\infty) \)

\( \nu \) Kinematic viscosity of the fluid, m\(^2\)/s

\( \psi \) Dimensionless stream function

\( \omega \) Non-dimensional vorticity

**Subscripts**

\( \infty \) Ambient

\( C \) Convection

\( H \) Based on the height \( H \) of the left wall of the side vented open cavity

\( h \) Hot

\( i \) Any arbitrary elemental area of an enclosure in the horizontal direction

\( j \) Any arbitrary elemental area of an enclosure in the vertical direction

\( \text{new} \) Present value of any dependent variable (\( \psi, \omega, \theta, J, G \)) obtained in two successive iteration

\( \text{old} \) Previous value of any dependent variable (\( \psi, \omega, \theta, J, G \)) obtained in two successive iteration

\( R \) Radiation

\( rc \) Radiation-conduction

\( T \) Total

**References**


