COMPUTATIONAL STUDY OF NATURAL CONVECTION AND ENTROPY GENERATION IN THREE-DIMENSIONAL CAVITY WITH ACTIVE LATERAL WALLS

Abdulwahab A. Alnaqi\textsuperscript{1}, Ahmed Kadhim HUSSEIN\textsuperscript{2}, Lioua KOLSI\textsuperscript{3,4*}, Abdullah A.A.A AL-RASHED\textsuperscript{5}, Dong LI\textsuperscript{6}, Hafiz Muhammad ALI\textsuperscript{6}

\textsuperscript{1} Dept. of Automotive and Marine Engineering Technology, College of Technological Studies, the Public Authority for Applied Education and Training, Kuwait,

\textsuperscript{2} College of Engineering - Mechanical Engineering Department University of Babylon - Babylon City – Hilla– Iraq,

\textsuperscript{3} College of Engineering, Mechanical Engineering Department, Haïl University, Haïl City, Saudi Arabia

\textsuperscript{4} Unité de Métrologie et des Systèmes Énergétiques, École Nationale d’Ingénieurs 5000, Monastir, University of Monastir, Tunisia,

\textsuperscript{5} School of Architecture and Civil Engineering , Northeast Petroleum University, Fazhan Lu Street , Daqing 163318 , China,

\textsuperscript{6} Mechanical Engineering Department, University of Engineering and Technology, Taxila, 47050, Pakistan,

\* Corresponding author; E-mail: lioua_enim@yahoo.fr

Numerical simulation of the natural convection and entropy generation in an air-filled cubical cavity with active lateral walls is performed in this work. Both the lateral front and right sidewalls are maintained at an isothermal cold temperature. While an isothermal hot temperature is applied for both the lateral back and left sidewalls. The upper and lower walls are kept adiabatic. Entropy generation rates due to the fluid friction and the heat transfer are simulated by using the second law of thermodynamics. Results are illustrated for Rayleigh numbers varied from \((10^{3} \leq Ra \leq 10^{5})\). It was shown that the increase in the Rayleigh number leads to increase the average Nusselt number and to decrease the Bejan number. Also, it was found that both \((S_{th})\) and \((S_{tot})\) increase slightly with the increase in \((Ra)\) until they reach \([Ra = 10^{5}]\) and then begin to jump after this value. After \([Ra = 10^{5}]\), the increase in both \((S_{tot})\) and \((S_{fr})\) is greater than \((S_{th})\). Moreover, it was observed that iso-surfaces of \((S_{tot})\) are similar to \((S_{th})\) at \((10^{3} \leq Ra \leq 10^{5})\), while they are similar to \((S_{fr})\) at high Rayleigh number.

Keywords: Natural convection; Three-dimensional flow; Entropy generation; Active lateral walls

**NOMENCLATURE**

- \(Be\) Bejan number
- \(g\) Acceleration due to gravity (m/s\(^2\))
- \(k\) Thermal conductivity (W/m.K)
1. INTRODUCTION

Buoyancy driven flow or the natural convection phenomena in a cubical cavity has long been investigated because of its relevance to many practical industrial applications. These applications include solar collector receivers, aircraft cabin insulation, cooling of electronic equipment, nuclear reactor systems, atmospheric science applications, semi-conductor production, double pane windows, thermal storage systems and so on [1].

The natural convection was generated by density gradients, which are in most due to imposed external heat source [2]. In the available literature, various research works have been investigated both numerically and experimentally the natural convection in a two-dimensional cavities under different boundary conditions. Samples of these investigations include Aktas and Farouk [3], Hussain and Hussein [4], Corvaro et al. [5], Hussein et al. [6] and Mâatki et al. [7].

From the another side, the number of papers related with the same problem but in a 3D cavities were still much less than the corresponding works performed in a 2D one. Fusegi et al. [8] investigated numerically the natural convection in a cubical cavity. The vertical sidewalls of it were maintained at different temperatures. The other walls were kept insulated. It was found that the computed temperature and velocity profiles in the symmetry planes were greatly matched with the experimental investigations. Fusegi et al. [9] considered the same geometry and boundary conditions in Fusegi et al. [8] and they deduced that the Nusselt number for 3D cavity was smaller than that for 2D one at $\text{Ra}<10^5$. Hiller et al. [10] performed a numerical and experimental study on the natural convection in a 3D cavity with two isothermal walls maintained at a prescribed temperature. The experiments were performed at ($\text{Ra} = 1.66 \times 10^5$ and $\text{Pr} = 1109$). Frederick [11] performed a numerical study about the natural convection in a cubical cavity with two active sectors on one vertical wall. His results indicated that Nusselt numbers became more than that computed for a side heated cavity at low values of (Ra). Frederick and Berbakow [12] numerically studied the natural convection in a cubical
enclosure with a hot source centered on a vertical wall and with an adjacent, fully cooled vertical wall. They concluded that the heat transfer had a weak dependent on the Rayleigh number. While it was greatly affected by the hot source mounted on the vertical wall. Oosthuizen et al. [13] studied numerically the natural convection in a 3D rectangular enclosure. A hot rectangular element was inserted on the centre of one vertical walls of the enclosure, while the horizontal top surface of it was cooled to a uniform temperature. All another surfaces were insulated. The results showed that as Rayleigh number decreased, the relative change in the average Nusselt number with decreasing dimensionless width of the plate increased. Lo and Leu [14] investigated numerically by using a velocity-vorticity formulation the natural convection in an inclined cubic cavity for various values of Rayleigh number and inclination angles. Bocu and Altac [15] examined numerically the natural convection in 3D rectangular enclosure, with pin arrays attached to its hot wall. The enclosure was heated from a lateral wall and cooled from the opposite one, while the other walls of it were adiabatic. The results explained that the Nusselt number ratio with respect to the enclosure without pins increased with pin length and its number. Onyango et al. [16] examined numerically by using a finite difference method, the natural convection in a three -dimensional square enclosure with a localized heating from below. The top and the two opposite vertical walls were maintained at a constant cold temperature. A heat source was fixed at the middle of the bottom wall. The other two vertical walls together with the non-heated part of the bottom wall were considered adiabatic. The results showed that the temperature was decreased as the fluid moved from the heat source towards the cold walls. Lee et al. [17] performed a numerical study of the natural convection in a differentially heated cubical enclosure filled with air. The right vertical sidewall was maintained at a hot temperature, while all another walls were kept cold. The flow and thermal characteristics of the natural convection were presented and discussed. Ahmadi [18] performed a numerical study about the laminar natural convection in a 3D rectangular enclosure. The left and right vertical sidewalls were maintained at hot and cold temperatures respectively. The other walls were kept adiabatic. They investigated the effects of (Ra), size and position of hot and cold sources on the flow and thermal characteristics inside the enclosure. It was found that the average Nusselt number together with the heat transfer rate from hot and cold sources were increased with increasing (Ra). Additional useful researches related with the subject can be found in [19- 33].

From the above literature survey, it can be seen that no research available up to date studies the natural convection phenomena and the entropy generation in an air-filled cubical cavity with active lateral walls. In summary, this is the first original work which studied this problem in more details.

2. Geometry description

The 3D natural convection problem inside an air filled (Pr = 0.71) cubical cavity is investigated numerically in the present paper. Both the lateral front and right sidewalls are maintained at cold temperature (Tc). An isothermal hot temperature (Ts) is applied for both the lateral back and left sidewalls. The upper and lower walls are kept adiabatic as shown in Fig.1. The flow inside the considered cavity is assumed unsteady, incompressible, laminar, three-dimensional and Newtonian. The thermo-physical properties of the fluid (except the density) are considered constant. The thermal and the flow fields are computed for (10^3 ≤ Ra ≤ 10^5) and the irreversibility coefficient is considered constant as (ϕ= 10^-4). The latter represents the relative importance of the fluid friction and heat transfer on the entropy generation rate.

3. Mathematical model, initial and boundary conditions and the numerical solution.
In order to begin the mathematical modeling of the present problem and to cancel the pressure term in a 3D cavity, the vector potential-vorticity formula \((\mathbf{\psi} - \mathbf{\omega})\) is adopted and is written using the following:

\[
\mathbf{\omega'} = \mathbf{\nabla} \times \mathbf{\psi'} \quad \text{and} \quad \mathbf{\psi'} = \mathbf{\nabla} \times \mathbf{\omega'}
\]  

The dimensionless governing equations can be constructed as follows [23]:

\[
-\mathbf{\omega} = \mathbf{\nabla}^2 \mathbf{\psi}
\]  

\[
\frac{\partial \mathbf{\omega}}{\partial t} + (\mathbf{V} \cdot \nabla) \mathbf{\omega} - (\mathbf{\omega} \cdot \nabla) \mathbf{V} = \Delta \mathbf{\omega} + RaPr \left[ \frac{\partial T}{\partial z} + \frac{\partial T}{\partial x} \right]
\]  

\[
\frac{\partial T}{\partial t} + \mathbf{V} \cdot \nabla T = \Delta T
\]  

Where \( Pr = \frac{v}{\alpha} \) and \( Ra = \frac{g \beta \Delta T L^3}{\nu \alpha} \)

The velocity \( \mathbf{V}' \), time \( t' \), vector potential \( \mathbf{\psi}' \) and vorticity \( \mathbf{\omega}' \) are written in the dimensionless forms by \( \alpha/\ell, \ell/\alpha \), \( \alpha \) and \( \ell/\alpha \).

The dimensionless temperature is written as:

\[
T = \frac{T(t') - T_c}{T_s - T_c}
\]  

The initial conditions for the present geometry are:

\[
T = 0, \quad V_x = V_y = V_z = 0 \quad \text{and} \quad \omega_x = \omega_y = \omega_z = 0 \quad \text{at} \quad t = 0
\]

The boundary conditions are defined by:

**Temperature:**

\[
T = 1 \quad \text{at} \quad x = 0 \quad \text{and} \quad z = 1; \quad T = 0 \quad \text{at} \quad x = 1 \quad \text{and} \quad z = 0; \quad \frac{\partial T}{\partial n} = 0 \quad \text{at} \quad y = 0 \quad \text{and} \quad y = 1
\]

**Vector potential:**

\[
\frac{\partial \mathbf{\psi}_x}{\partial x} = \mathbf{\psi}_y = \mathbf{\psi}_z = 0 \quad \text{at} \quad x = 0 \quad \text{and} \quad 1; \quad \mathbf{\psi}_x = \frac{\partial \mathbf{\psi}_y}{\partial y} = \mathbf{\psi}_z = 0 \quad \text{at} \quad y = 0 \quad \text{and} \quad 1; \quad \mathbf{\psi}_x = \mathbf{\psi}_y = \frac{\partial \mathbf{\psi}_z}{\partial z} = 0 \quad \text{at} \quad z = 0 \quad \text{and} \quad 1
\]

**Velocity:**

\[
V_x = V_y = V_z = 0 \quad \text{on all cavity walls}
\]

**Vorticity:**

\[
\omega_x = 0, \quad \omega_y = \frac{-\partial V_y}{\partial x}, \quad \omega_z = \frac{-\partial V_z}{\partial x} \quad \text{at} \quad x = 0 \quad \text{and} \quad 1; \quad \omega_x = \frac{\partial V_y}{\partial y}, \quad \omega_y = 0, \quad \omega_z = \frac{-\partial V_x}{\partial y} \quad \text{at} \quad y = 0 \quad \text{and} \quad 1;
\]

\[
\omega_x = \frac{-\partial V_y}{\partial z}, \quad \omega_y = \frac{\partial V_z}{\partial z}, \quad \omega_z = 0 \quad \text{at} \quad z = 0 \quad \text{and} \quad 1
\]

Fig. 1. Schematic diagram of the present problem.

The dimensionless local generated entropy \( (s_N) \) is expressed as [23]:

\[
\]
\[ N_v = \left[ \left( \frac{\partial T}{\partial x} \right)^2 + \left( \frac{\partial T}{\partial y} \right)^2 + \left( \frac{\partial T}{\partial z} \right)^2 \right] \]

\[ + \phi \cdot \left[ 2 \left( \frac{\partial V_y}{\partial x} \right)^2 + \left( \frac{\partial V_y}{\partial y} \right)^2 + \left( \frac{\partial V_y}{\partial z} \right)^2 \right] \left[ \left( \frac{\partial V_x}{\partial x} + \frac{\partial V_x}{\partial y} \right)^2 + \left( \frac{\partial V_x}{\partial y} + \frac{\partial V_x}{\partial z} \right)^2 + \left( \frac{\partial V_x}{\partial z} + \frac{\partial V_x}{\partial x} \right)^2 \right] \]

\text{(5)}

with \( \phi \) is the irreversibility coefficient defined by \( \phi = \frac{\mu a^2 T_0}{l^2 k \Delta T^2} \)

The first part in Eq.5 refers to the local generated entropy due to the temperature gradients and noted as \((N_{v,th})\). The second part of it refers to the local generated entropy due to the fluid friction and noted as \((N_{v,fr})\). Furthermore, the total dimensionless generated entropy \((S_{tot})\) is written as:

\[ S_{tot} = \int N_v dv = \int (N_{v,th} + N_{v,fr}) dv = S_{th} + S_{fr} \text{ (6)} \]

The ratio of the thermal entropy to the total entropy is defined by the Bejan number (Be):

\[ Be = \frac{S_{th}}{S_{th} + S_{fr}} \text{ (7)} \]

The local and average Nusselt numbers are given by:

\[ Nu = \frac{\partial T}{\partial x} \text{ and } Nu_{av} = \frac{\int Nuds}{S} \text{ (8)} \]

1. Numerical approach and code verification

The solution of the governing equations (2-5) is performed numerically using the Control Volume Technique. The Alternating Direction Implicit (ADI) scheme is used to discretize the temporal derivatives, while the central-difference approach is used to deal with the convection parts in the governing equations. The solution is considered fair when the following convergence criterion is satisfied for each step of time:

\[ \sum_{i}^{1,2,3} \frac{\max |\psi^n_i - \psi^{n-1}_i|}{\max |\psi^n_i|} + \max |T_i^n - T_i^{n-1}| \leq 10^{-5} \text{ (9)} \]

In order to verify the accuracy of our computer program, the natural convection problems in a differentially heated cubical cavity which are studied previously by Wakashima and Saitoh \[34\] and Fusegi et al. \[35\] are re-solved again by our computer program. Table 1 shows a comparison between these results for various values of \((Ra)\). The results of the comparison indicate a very good matching between the results computed by our computer program with their corresponding results given by Wakashima and Saitoh \[34\] and Fusegi et al. \[35\] respectively. This verification gives a good confidence in the present computer program to investigate with the current problem.

**Table 1.** Comparison of present results with the 3D results of (Wakashima and Saitoh \[34\]) and (Fusegi et al. \[35\]) for differentially heated cubic closed space and \(Pr = 0.71\).

<table>
<thead>
<tr>
<th>Ra</th>
<th>Authors</th>
<th>( V_{x,\text{max}} ) (y)</th>
<th>( V_{x,\text{max}} ) (x)</th>
<th>( Nu_{av} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>10^4</td>
<td>Present work</td>
<td>0.199 (0.826)</td>
<td>0.221 (0.112)</td>
<td>2.062</td>
</tr>
<tr>
<td></td>
<td>Wakashima and Saitoh [34]</td>
<td>0.198 (0.825)</td>
<td>0.222 (0.117)</td>
<td>2.062</td>
</tr>
<tr>
<td></td>
<td>Fusegi et al. [35]</td>
<td>0.201 (0.817)</td>
<td>0.225 (0.117)</td>
<td>2.1</td>
</tr>
<tr>
<td>10^5</td>
<td>Present work</td>
<td>0.143 (0.847)</td>
<td>0.245 (0.064)</td>
<td>4.378</td>
</tr>
<tr>
<td></td>
<td>Wakashima and Saitoh [34]</td>
<td>0.147 (0.85)</td>
<td>0.246 (0.068)</td>
<td>4.366</td>
</tr>
<tr>
<td></td>
<td>Fusegi et al. [35]</td>
<td>0.147 (0.855)</td>
<td>0.247 (0.065)</td>
<td>4.361</td>
</tr>
</tbody>
</table>
The average Nusselt number on the hot walls was selected as a sensitive parameter. Results of the grid sensitivity analysis are shown in Table 2. The incremental increase in the percentage of $N_{\text{ave}}$, for grids of $81^3$ to $91^3$, is only 0.143%. Hence, considering the computational economy and accuracy, a spatial mesh size of $81^3$ and a time-step of $10^{-5}$ were opted for the present study. For all executions the solution is stored at $t=2$.

<table>
<thead>
<tr>
<th>Grid size</th>
<th>$N_{\text{ave}}$</th>
<th>Percentage Increase</th>
<th>Incremental Increase</th>
</tr>
</thead>
<tbody>
<tr>
<td>$61^3$</td>
<td>4.8401</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$71^3$</td>
<td>4.9205</td>
<td>1.6611</td>
<td></td>
</tr>
<tr>
<td>$81^3$</td>
<td>4.9639</td>
<td>2.5431</td>
<td>0.8820</td>
</tr>
<tr>
<td>$91^3$</td>
<td>4.971</td>
<td>2.6861</td>
<td>0.1430</td>
</tr>
</tbody>
</table>

### 2. RESULTS AND DISCUSSION

The laminar 3D natural convection and entropy generation in an air-filled cubical cavity with active lateral walls is simulated numerically in the present research. The effect of Rayleigh number (Ra) on the flow and thermal fields, heat transfer and entropy generation has been investigated. The results were illustrated by trajectory of particles, iso-surfaces of temperature, average Nusselt number, iso-surfaces of the entropy generations and the Bejan number. **Fig.2** shows the 3D trajectory of particles for different values of Rayleigh number. According to the buoyancy force effect, the convection currents begin from the hot lateral back and left sidewalls until they reach to the adiabatic upper wall. After that, they reverse their direction and move towards the cold lateral front and right sidewalls. Then, they reach again to the hot walls after passing the adiabatic lower wall.

This repeated rotation of the convection currents lead to produce the convection vortices which occupy the entire cavity as seen in **Fig.2**. For low Rayleigh number [$Ra = 10^3$] or when the effect of the buoyancy force which is due to the temperature difference inside the cavity is weak. The shapes of convection vortices are symmetrical and extend laterally between the hot and cold lateral walls. In this case, the natural convection role in the heat transfer process is slight, while the conduction is predominant. Now, as the Rayleigh number increases gradually, the intensity of the flow circulation begins to increase. In this case, the viscous forces become smaller than the buoyancy one. Therefore, a clear dissipation in the convection vortices towards all the cavity walls can be observed. For this reason, their shapes are greatly deformed as the Rayleigh number increases. When the natural convection becomes predominant or when [$Ra = 10^5$], the shape of vortices was highly deformed in comparison with their corresponding vortices which are seen at [$Ra = 10^3$]. Moreover, it can be noticed from the results of **Fig.2** that the flow circulation is stronger near the center of the cavity and slight at the lateral cold and hot walls due to non-slip boundary conditions.

From the other side, the effect of the Rayleigh number on the temperature field is apparent from the iso-surfaces of temperature as illustrated in **Fig.3**. When the effect of the natural convection is weak or when the Rayleigh number is low [$Ra = 10^3$], the iso-surfaces of temperature are uniform, symmetrical, smooth and parallel to the cavity sidewalls. It can be observed from **Fig.3**, that iso-
surfaces of temperature emanate from the lateral back and left sidewalls (i.e., hot walls) until they arrive to the cold lateral front and right sidewalls, indicating the path of the thermal field. Heat conduction is the dominant mode of the heat transfer inside the cubical cavity. Now, when the natural convection effect becomes dominant or when the Rayleigh number increases gradually from \([Ra=10^4]\) to \([Ra=10^6]\), the iso-surfaces of temperature become wavy, curved and clustered to the lateral hot back and left sidewalls as illustrated from Fig. 3. Moreover, it can be noticed a stagnant region from iso-surfaces of temperature at the upper part of the cubical lateral cavity and this region begins to decrease as the Rayleigh number increases.

![Fig. 2. Particle trajectories for various Ra](image)

**Fig. 4** illustrates the behavior of the average Nusselt number \((Nu_{av})\) in the cubical lateral cavity for different values of the Rayleigh number. It can be seen that, when the Rayleigh number is low \([Ra = 10^3]\), the values of \((Nu_{av})\) is low also. This refers that the heat is transferred inside the lateral cavity by the conduction. After that, a clear increasing in the values of \((Nu_{av})\) can be observed as the Rayleigh number increases. This increasing can be returned to the high effect of the natural convection. At \([Ra = 10^6]\), the values of \((Nu_{av})\) reach their maximum range which reflect the strong effect of the natural convection inside the cavity.

**Fig.5** displays iso-surfaces of the entropy generations and the Bejan number in the cubical lateral cavity for different values of the Rayleigh number. For \((10^3 \leq Ra \leq 10^5)\), the \((S_{th})\) iso-surfaces are symmetrical and distributed vertically near the lateral cavity walls. But, for \((Ra = 10^6)\), it can be seen that \((S_{th})\) iso-surfaces extend also in the \((x-z)\) plane due to the increase in the heat transfer losses with increasing of \((Ra)\).
This leads to increase the \((S_{th})\). Furthermore, it can be observed that \((S_{th})\) iso-surfaces increase adjacent the lateral cavity walls due to the temperature difference. While, they absent at the adiabatic upper and lower walls. Moreover, it can be observed from Fig.5, that there is a space region in the center of the cavity. This is because of the stagnant or low velocity of the fluid. With respect to the iso-surfaces of \((S_{th})\), it can be seen that they are concentrated intensely adjacent the cavity walls. This is due to the high effect of the friction between the fluid inside the cavity and its walls. For iso-surfaces of \((S_{th})\), it is interesting to see that their behavior is similar to that seen for \((S_{th})\) at \((10^3 \leq Ra \leq 10^5)\). This refers to the dominance of \((S_{th})\) or in another words the domination of the irreversibility due to the heat transfer at this range of \((Ra)\). But, at \([Ra=10^6]\), the iso-surfaces of \((S_{tot})\) try to be similar to \((S_{th})\). This behavior can be returned to the dominance of the irreversibility due to the fluid friction at high Rayleigh number.

![Temperature contours](image1.png)

**Fig. 3.** Temperature contours for various values of Rayleigh number

![Variation on Nuav versus Ra](image2.png)

**Fig.4.** Variation on \(Nu_{av}\) versus \(Ra\)

**Fig.5** illustrates iso-surfaces of the Bejan number for various values of the Rayleigh number. The Bejan number is a dimensionless number which is used to measure the relative magnitude of the heat transfer and fluid friction irreversibilities; high values of \(Be\) indicate the dominance of thermal generated entropy and low values indicate the frictional one. When the Rayleigh number is low \((Ra = 10^3)\), the iso-surfaces of the Bejan number have a high intense and distribute uniformly on the lateral walls of the cavity. As Rayleigh number increases, they begin to move towards the center of the cavity. At \((Ra = 10^6)\), the intensity of the Bejan number iso-surfaces around the lateral walls of the
cavity begin to decrease in comparison with their corresponding iso-surfaces which are observed when the Rayleigh number is low.

**Fig.6** demonstrates the variation of various entropies generation [$S_{th}$, $S_{fr}$ and $S_{tot}$] in the cubical lateral cavity for different values of the Rayleigh number. It can be observed that the entropy generation rate due to the fluid friction ($S_{fr}$) remains approximately constant as the Rayleigh number increases up to [Ra = $10^5$]. After that, it increases strongly with the increase in the Rayleigh number. From the opposite side, both the entropy generation due to the heat transfer ($S_{th}$) and the total entropy generation ($S_{tot}$) increase slightly with the increase in the Rayleigh number until they reach [Ra = $10^5$] and then begin to jump after this value. This jumping is strong for ($S_{th}$) and slight for ($S_{fr}$). The behavior noticed at ($10^3 \leq Ra \leq 10^5$) can be returned to the high increase in the heat transfer losses and weak increase in the friction between the fluid inside the cavity and its different walls. After [Ra = $10^5$], the increase in both ($S_{tot}$) and ($S_{fr}$) is greater than ($S_{th}$) which indicates a reduction in the heat transfer losses. For this reason, the peak value of the total entropy generation ($S_{tot}$) and entropy generation due to friction ($S_{fr}$) occurs at the maximum value of Rayleigh number [i.e., Ra = $10^6$].

![Fig. 5. Iso-surfaces of the entropy generations and Bejan number in the cubical lateral cavity for different values of Rayleigh number.](image)

Finally, **Fig.7** illustrates the behavior of the Bejan number (Be) for different values of the Rayleigh number. The results display that as the Rayleigh number increases, the values of the Bejan number begin to decrease. Since, for high values of (Ra) the increase in ($S_{tot}$) is much higher than the increase in ($S_{th}$) as explained previously in **Fig.6**. This leads of course to decrease the values of the Bejan number. Furthermore, an interesting behavior can be observed in **Fig.7**, where there is a linear variation between the Bejan number and the Rayleigh number for low range of (Ra). Therefore, it can
be concluded that for low values of (Ra) most of the total entropy generation ($S_{\text{tot}}$) is due to the heat transfer losses or ($S_{\text{th}}$) or in another words, the friction losses or ($S_{\text{fr}}$) are very weak. This leads to make the values of the Bejan number approximately unity for low range of (Ra) as displayed in Fig.7.

![Fig.6. Variations of various entropies generation in the cubical lateral cavity for different values of Rayleigh number.](image)

![Fig.7. Variation of Bejan number in the cubical lateral cavity for different values of Rayleigh number.](image)

3. CONCLUSION:

In this paper, natural convection and entropy generation in 3D cavity having active lateral wall is investigated numerically using the FVM. The following conclusions can be highlighted from the results of the present work:

- The shape of the flow vortices greatly affected by the variation of Rayleigh number.
  - Iso-surfaces of temperature are uniform, smooth and parallel to the cavity sidewalls when (Ra) is low. As (Ra) increases, they become wavy, curved and clustered to the lateral back and left sidewalls.
  - The values of the average Nusselt number increase as the Rayleigh number increases.
  - Entropy generation due to fluid friction ($S_{\text{fr}}$) remains approximately constant as (Ra) increases up to [Ra = $10^5$]. After that, it increases strongly with the increase in (Ra).
  - Both the entropy generation due to heat transfer ($S_{\text{th}}$) and total entropy generation ($S_{\text{tot}}$) increase slightly with the increase in (Ra) until they reach [Ra = $10^5$] and then begin to jump after this value. After [Ra = $10^5$], the increase in both ($S_{\text{tot}}$) and ($S_{\text{th}}$) is greater than ($S_{\text{fr}}$).
  - Bejan number increases as the Rayleigh number decreases.
  - Iso-surfaces of ($S_{\text{th}}$) are symmetrical and distributed vertically near the lateral cavity walls for ($10^3 \leq \text{Ra} \leq 10^5$). But, for (Ra = $10^6$), they try to extend also in the (x-z) plane.
  - Iso-surfaces of ($S_{\text{fr}}$) are high adjacent the cavity walls.
- Iso-surfaces of \( (S_{th}) \) are similar to \( (S_{hr}) \) at \( (10^3 \leq Ra \leq 10^5) \), while they are similar to \( (S_{ir}) \) at high Rayleigh number.

REFERENCES


