The heat transfer by mixed convection coupled to the radiation in a rectangular cavity continues to be a fertile area of research, due to the interest of the phenomenon in many technological processes, such as the design of solar collectors, thermal design of buildings, air conditioning, and recently the cooling of electronic circuit boards, electronic enclosures, industrial furnaces. The basic nature the problem interaction between the forced external air stream and the buoyancy-driven flow by the heat source.

The effect of height partition on combined mixed convection and surface radiation in a vented rectangular cavity is study by Bahlaoui et al., [1]. They concluded that the relative height of the partition, contributes to increase/decrease the radiative/convective heat transfer component at the level of the heated wall. The radiation effect leads to a reduction of the convective Nusselt number component but the Reynolds number, supports both radiative and convective heat transfer modes. Effect of baffle number on mixed convection within a ventilated cavity, this study presented by Belmiloud et al., [2].

Their results indicated the two provisions of the outlet of the cavity (BB and BT configurations) have a significant influence on the number of average Nusselt whose maximum value is obtained for the BB configuration due to streamlines that are
clamped to the hot wall. Gravity has no influence on heat transfer and whatever the number of baffles used. The influence of gravity is observed only for the BT configuration and particularly in the case of a single baffle.

Chang et al., [3] studied the effect of the baffle position on the heat transfer by mixed convection of pulsed flow in a vertical channel. They concluded that for the channel with both flow pulsation and a baffle will generate a improved heat transfer. A non-pulsating flow with a baffle has a best heat transfer. The heat transfer enhancement increases with Re.

Cheng et al., [4] study the heat transfer enhancements of back-ward-facing step flow in a two-dimensional channel through the installation of solid and slotted baffles on to the channel wall. They pointed out that the pressure drop for the situation with slotted baffle is substantially smaller than that with solid baffle. With a slotted baffle installed on to the channel wall, the higher value of average Nusselt number obtained in the cases the solid baffle compared slotted baffle.

How et al., [5] transient laminar mixed convection in a to-dimensional cavity partitioned by a baffle. The results show that the higher values of Re or lower the values of Gr/Re² delay the attainment of steady time. Increasing both the values of Re and Gr/Re² give rise to an increase of the heat transfer coefficient. The numerical study of laminar mixed convection in a rectangular cavity is study by Hsu et al., [6].

Their results indicated that, the average Nusselt number is increased as Re is increased at fixed Gr/Re². Similarly, a gradual increase in heat transfer rate is found with increasing Gr/Re² at constant value of Re. The mixed convection with conduction and surface radiation from a vertical channel with discrete heating study by Londhe et al., [7]. Their results show a good emitting surface is observed to decrease the local left wall temperature and increase the local right wall temperature.

Singh et al., [8] studied the Conjugate laminar free convection with surface radiation in a two-dimensional open top cavity. The results indicate that the surface radiation changes the basic flow physics and enhances the radiative heat transfer as a result of which heat transfer by convection decreases. Thermal conductance and volumetric heat generation decrease the non-dimensional maximum temperature. Volumetric heat generation rate enhances the convective and radiative heat transfer.

Tsay et al., [9] showed that the variation of the baffle position on the top wall, relative to the entrance, leads to an increase of the average Nusselt number. The results show that the higher value of average Nusselt number obtained in the cases with baffle compared without baffle. The effects of baffle width on heat transfer are in significant.

2. MATHEMATICAL MODEL

The numerical simulation depends on the variation of the emissivity walls ε, the length of the baffle L_b and the Reynolds number Re between 50 and 500. The walls of the geometry under consideration are gray, the left vertical wall of the cavity is heated to a constant heat flux, the other walls are adiabatic. The input of the cavity is at the bottom of the left vertical wall and the outlet at the upper part of the opposite wall. Three baffle lengths are used (L_b =0.3, 0.5, 0.7), the aspect of the cavity report A = 2 and the fluid temperature is T_in = 288 K.

The general equations of conservation of mass, momentum and energy along the axes x and y, are:
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]  
(1)

\[
\begin{aligned}
&u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{\partial P}{\partial x} + \nu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) + g\beta (T - T_{in}) \\
&u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{\partial P}{\partial y} + \nu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \\
&u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)
\end{aligned}
\]  
(2, 3, 4)

The general equation of the radiosity is given by,
\[
J_i = \varepsilon_i \cdot \sigma \cdot T_i^4 + (1 - \varepsilon_i) \sum_{j=1}^{N} F_{ij} \cdot J_j
\]  
(5)

\[
F_{ij} = \frac{1}{A_i} \int_{A_i} \int_{A_j} \frac{\cos \theta_i \cdot \cos \theta_j}{\pi r^2} \delta_{ij} \, dA_i \, dA_j
\]  
(6)

The general equation of the net flux is given by,
\[
Q_{nd} = \varepsilon_i \cdot \sigma \cdot T_i^4 - \sum_{j=1}^{N} F_{ij} \cdot J_j
\]  
(7)

In the above equations, the dimensionless variables are defined by,
\[
T_{in}^* = \frac{\lambda \cdot T_{in}}{Q \cdot H} \quad \text{and} \quad T^* = \frac{\lambda \cdot (T - T_{in})}{Q \cdot H}
\]  
(8)

The Grashof number \( \text{Gr} \), the Reynolds number \( \text{Re} \), the Richardson number \( \text{Ri} \) and the Prandtl number \( \text{Pr} \) are given by:
\[
\text{Gr} = \frac{g \beta Q H^4}{\lambda v^2}; \quad \text{Ri} = \frac{Gr}{Re^2}; \quad \text{Re} = \frac{u_{in} H}{v}; \quad \text{Pr} = \frac{v}{\alpha}
\]  

(9)

To solve the equations (2) - (4) considering the conditions on these dimensionless limits:

At input of the cavity: \(T_{in} = 288 \text{ K} \); \(u_{in} \neq 0 \); \(v = 0\)

(10)

Adiabatic walls: \(u = v = 0 \); \(Q = 0 \text{ W/m}^2\)

(11)

In hot vertical wall: \(u = v = 0 \); \(Q = 100 \text{ W/m}^2\)

(12)

To determine the characteristics of heat transfer in the steady state, we must take into consideration the contribution of convection and radiation. In this study, the total average Nusselt number is defined as:

\[
\text{Nu}_T = \text{Nu}_{cv} + \text{Nu}_{rd} = \frac{(Q_{cv} + Q_{rd}) \cdot H}{\lambda (T_H - T_{in})}
\]  

(13)

3. NUMERICAL ANALYSIS

The mass, momentum, and energy equations have been solved by a finite difference algorithm called the semi-implicit method for pressure linked equations (SIMPLE). Details of this method are described by Patankar [10].

The differential equations are discretized over a control volume. The power law difference scheme (PLDS) has been employed for the calculation of scalar variables and the quadratic upstream-weighted interpolation for convective kinematics (QUICK) scheme for Hayase et al., [11] vector variables. The relative tolerance for the error criteria is considered to be:

\[
\left| \frac{\varphi_n - \varphi_{n-1}}{\varphi_n} \right|_{\text{max}} < 10^{-9}
\]  

(14)

Table 1 show the dimensionless values of \(T_{av}^*\) and \(\text{Nu}_{av}\) determined in the case length \(L_b = 0.5\) and wall emissivity \(\varepsilon = 0.15\). Note that the relative difference between the obtained meshes (30 × 80) and (40 × 101) is 1.36 %, for \(\text{Nu}_{av}\) the relative difference of 1.48 %. For meshes (34 × 93) and (40 × 101) the relative difference is 1.22 % and for \(\text{Nu}_{av}\) is 1.47 %. The mesh used in all subsequent calculations is (40 × 101).

Table 1: Grid effect on the obtained results for \(L_b = 0.5\) and \(\varepsilon = 0.15\)

<table>
<thead>
<tr>
<th>Grid</th>
<th>30×80</th>
<th>34×93</th>
<th>40×101</th>
<th>30×80</th>
<th>34×93</th>
<th>40×101</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_{av}^*)</td>
<td>0.149</td>
<td>0.1488</td>
<td>0.147</td>
<td>1.36</td>
<td>1.22</td>
<td>-</td>
</tr>
<tr>
<td>(\text{Nu}_{av})</td>
<td>6.719</td>
<td>6.720</td>
<td>6.820</td>
<td>1.48</td>
<td>1.47</td>
<td>-</td>
</tr>
</tbody>
</table>

3.1 Validation of the code

Numerical simulation performed using the commercial code ANSYS (Fluent). Version 6.3.26. The numerical validation of computer code, we made two comparisons of the results obtained in this study. For the first comparison, use the results obtained by Saha et al. [12], for the second comparison, use the results of Bahlaoui et al. [13].

The table shows the values of the average Nusselt number \(\text{Nu}_{av}\) and determined the maximum temperature \(T_{max}^*\) at the hot wall to a Reynolds number \(Re = 100\) and the
Richardson number of $R_i = 10$. Note the values of $\text{Nu}_{av}$ and $T^*_\text{max}$ obtained in this study and those obtained by Saha et al. [12] are almost the same. The relative value is $1.80\%$ for $\text{Nu}_{av}$ and to $0.48\%$ for $T^*_\text{max}$.

| Table 2: Average Nusselt number and the maximal temperature evaluated in the heating wall |
|-----------------|----------------|----------------|
| $\text{Nu}_{av}$ | 2.235           | 2.276          |
| $T^*_\text{av}$       | 0.206           | 0.207          | % change in abs |
|                    |               | 1.80           |
|                    |               | 0.48           |

Table 3 shows the values of number average Nusselt $\text{Nu}_{av}$ determined at the hot wall for the two values of the Reynolds number $Re$ and for the emissivity $\varepsilon = 0.15$. Note the values are almost equal to those Bahlaoui et al. [13].

| Table 3: Average Nusselt number and the maximal evaluated in the heating wall for different $Re$ and $V$ |
|-----------------|----------------|----------------|
| $Re = 200$         | 2.235           | 2.276          |
| $Re = 300$         | 0.206           | 0.207          | % change in abs |
|                  |               | 1.80           |
|                  |               | 0.48           |

4. RESULTS AND DISCUSSION

4.1 Heat transfer

Figure 2 shows the influence of the emissivity $\varepsilon$ on the average Nusselt number convection $\text{Nu}_{cv}$, radiation $\text{Nu}_{rd}$ and total $\text{Nu}_T$ at the heated surface of the cavity as a function of Reynolds numbers and for $L_b = 0.5$. The Nusselt number $\text{Nu}_{cv}$, $\text{Nu}_{rd}$ and $\text{Nu}_T$, increase with the increase in emissivity $\varepsilon$.

Figure 2(a), shows that the $\text{Nu}_{cv}$ is increases with the Reynolds $Re$ regardless of the value of the emissivity $\varepsilon$, the valid result for the other length $L_b = 0.3$ and $L_b = 0.7$. The difference between the values obtained between $\varepsilon = 0$ and $\varepsilon = 0.15$ is of $21.05\%$. Knowing that the radiation effects reduced the number of convective Nusselt $\text{Nu}_{cv}$, the result does not agree with those found by Bahlaoui et al., [1].

Figure 2(b) show that the number of radiative Nusselt $\text{Nu}_{rd}$ decreases with increasing Reynolds number $Re$; However, increasing the emissivity $\varepsilon$ improved the number of radiative Nusselt $\text{Nu}_{rd}$. The result is in good agreement with those found by Bahlaoui et al., [1] this result holds for any length of baffle.

The result in figure 2 (c) shows that the value of the Nusselt number total $\text{Nu}_T$ increased when the emissivity $\varepsilon$ is increased. Heat transfer increases as the Reynolds number $Re$.

Figure 3 show the influence of the baffle length $L_b$ on the total Nusselt number $\text{Nu}_T$ as a function of the different values of the Reynolds number $Re$ for $\varepsilon = 0.15$ and $Gr = 10^4$. the total number of Nusselt $\text{Nu}_T$ decreases with decreasing. For $L_b = 0.3$, it
is observed that the Nusselt number almost constant when the number of Reynolds exceeded \( Re > 300 \).

However, for \( L_b = 0.5 \) and \( L_b = 0.7 \) to increase the Nusselt number when the Reynolds number increase. The maximum different between the two curves corresponding to 39.26% for \( L_b = 0.3 \) and \( L_b = 0.5 \), most difference compared to the value obtained between \( L_b = 0.5 \) and \( L_b = 0.7 \).

The baffle length \( L_b \) contributes to the increase in the transfer of radiative and convective heat in the heated wall. This result does not agree with that obtained by Bahlaoui et al. [1] (increase / (decrease) in heat transfer radiative / (convective) at the heated wall). Because the problem Bahlaoui et al. [1] is a horizontal cavity mounted by a vertical baffle installed on the lower horizontal hot wall, However, our problems presented on figure 1. This meant the difference obtained on the results of convective heat transfer.

**Fig. 2.** Influence of emissivity number on the average Nusselt number for \( L_b = 0.5 \), \( Gr = 10^4 \) and different Reynolds number: a) \( Nu_{cv} \), b) \( Nu_{rd} \) and c) \( Nu_T \)

**Fig. 3:** Influence of length of baffle on the total Nusselt number for \( \varepsilon = 0.15 \), \( Gr = 10^4 \) and for different Reynolds number
4.2 Variation of dimensional temperature

The profile of the average temperature $T_{av}^*$ for the various values of the Reynolds number $Re$ is given by the figure 4.

Figure 4 (a) show that the average temperature profiles $T_{av}^*$ have the same shape and are parallel and whatever the number Reynolds $Re$, the maximum value is obtained in the case of an emissivity $\varepsilon = 0$. The scientific significance of the latter, due to the exchange surface the strongest, hence the presence of baffle causes the fluid to the hot wall. Figure 4 (b) shows the variation the average temperature $T_{av}^*$ as a function of the length $L_b$, it was observed that the average temperature decreases as the length increases, knowing that the $T_{av}^*$ has calculated the hot wall. This one verifying the results obtained on the number of total Nusselt (figure 3). Is found that the length and the baffle position influence the heat transfer.

![Fig. 4: Profile of $T_{av}^*$ depending on Reynolds number $Re$ obtained for](image)

a) $L_b = 0.5$ and b) $\varepsilon = 0.15$

5. CONCLUSION

The purpose of this study is to see the effect of emissivity $\varepsilon$, of the variation of the baffle length $L_b$ on mixed convection coupled to a radiation in a rectangular cavity. According to the results, we concluded the heat transfer increases when the emissivity $\varepsilon$. Knowing that the Nusselt number convective $\text{Nu}_{cv}$, radiation $\text{Nu}_{rd}$ and total $\text{Nu}_T$ increases with $\varepsilon$, for low values of Reynolds $Re$, radiative transfer is good relative large values.

The total Nusselt number $\text{Nu}_T$ increases as the length $L_b$ of the baffle increases. Knowing that, low values of $L_b$, the almost stable thermal transfer when exceeded the number of Reynolds $Re > 300$.

The maximum value of the average temperature $T_{av}^*$ is obtained in the case of the emissivity $\varepsilon = 0$ and for the length of the baffle $L_b = 0.3$. However the minimum values $\varepsilon = 0.15$ and $L_b = 0.7$.

One observation that, for improved heat transfer and cooled the hot plate must be used a gray body with a great emissivity and increased length.
NOMENCLATURE

A, aspect ratio of the cavity, \( A = H/L \)

\( F_{ij} \), View factor from \( S_i \) surface to \( S_j \)

\( G_r \), Grashof number

\( J_{i,j} \), Radiosity

\( N_r \), Convection-radiation interaction parameter

\( L \), Length of the cavity

\( N_u \), Average Nusselt number

\( \beta \), Thermal expansion coef. of fluid

\( \lambda \), Thermal conductivity of fluid

\( Q \), Heat flux

\( R_i \), Richardson number

\( T \), Fluid temperature

\( u_{in} \), The velocity inlet

\( w \), Height opening inlet and outlet of the cavity

\( \alpha \), Thermal diffusivity of fluid

\( \varepsilon \), Emissivity of the walls

\( \nu \), Kinematic viscosity

\( \sigma \), Stefan-Boltzmann constant


