EFFECT OF SURFACE RADIATION ON CONJUGATE CONVECTION IN A CLOSED AND DISCRETELY HEATED RECTANGULAR CAVITY

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ABSTRACT
The present paper reports results of simulation studies on combined conduction-convection-radiation from a rectangular cavity equipped with a discrete heat source in each vertical wall. The discrete heat sources are traversable along the respective vertical walls. Air, a radiatively transparent medium, is considered to be the cooling medium. The exterior surfaces of the walls of the cavity are assumed to be adiabatic. The governing equations for temperature distribution along the walls of the cavity are obtained by appropriate energy balance between heat generated, conducted, convected and radiated. Calculations pertaining to radiation are performed using enclosure analysis, while the view factors required therein are computed using the crossed-string method of Hottel. The resulting nonlinear partial differential equations are converted into algebraic form using finite difference formulation and are subsequently solved through Gauss-Seidel iterative solver. The effects of various pertinent parameters, viz., heat source position, surface emissivity, convection heat transfer coefficient, thermal conductivity and aspect ratio, on the results of the present problem are studied in detail.

KEYWORDS
Surface radiation, Conduction, Convection, Rectangular cavity, Discrete heat sources.

1. INTRODUCTION
Studies on multi-mode heat transfer (two or more of the three possible modes, namely, conduction, convection and radiation) have been numerous in the recent past owing to several applications including cooling of electronic equipment and devices, solar collectors and gas-cooled nuclear reactors. One of the earliest in this category is credited to Zinnes [1], who probed into laminar natural convection heat transfer from a vertical, heat-conducting flat plate of finite thickness with an arbitrary heating distribution along its surface. Lee and Yovanovich [2] proposed a quasi-analytical conjugate heat transfer model for a two-dimensional vertical flat plate provided with arbitrarily sized discrete heat sources dissipating heat by free convection. Tewari and Jaluria [3] conducted an experimental study on the fundamental aspects of conjugate mixed convection from two heat sources of finite width and negligible thickness located on a flat plate. Gorski and Plumb [4] numerically investigated conjugate laminar forced convection from a flat plate with a flush-mounted discrete heat source. Cole [5] reported solution to the problem of conjugate heat transfer from a small heated strip. Mendez and Trevino [6] numerically dealt with the problem of conjugate free convection from a thin vertical strip with non-uniform internal heat generation. Gururaja Rao et al. [7] numerically investigated the problem of two dimensional, steady, incompressible, conjugate, laminar mixed convection with surface radiation from a vertical plate with a flush-mounted discrete heat source. They solved the governing fluid flow and energy equations without boundary layer approximations using a finite volume method. Gururaja Rao [8] investigated buoyancy-aided conjugate mixed convection with surface radiation from a vertical electronic board equipped with a traversable flush-mounted discrete heat source. He performed the study by varying the position of the heat source from the leading to the trailing edge of the board and inferred that the leading edge is the best possible location if one were to use a single heat source in problems of this class in any regime of mixed convection and for any given board surface emissivity. Very recently, Gururaja Rao et al. [9] performed simulation studies on an open cavity containing a flush-mounted discrete heat source in its left wall. They provided results for the coupled problem of conduction, convection and radiation assuming the heat source to be traversable along the wall of the cavity.

The foregoing review of the literature concerning multi-mode heat transfer from geometries typically encountered in electronic cooling applications brings out the following points. Firstly, studies involving rectangular cavity with discrete heating along its
walls and with varying aspect ratio are very scarce. Coupled to this, interactive effect of radiation on conjugate convection in this kind of geometry has not been explored satisfactorily. In view of the above, an attempt is made to numerically probe into the problem of multi-mode heat transfer from a closed rectangular cavity with a traversable discrete heat source provided in each of its two vertical walls. The problem is solved using a computer code specifically written and various parametric studies are carried out.

2. MATHEMATICAL FORMULATION AND METHOD OF SOLUTION

Figure 1 shows the schematic of the problem geometry chosen for the study. It consists of a rectangular cavity of height L and width W, while \( t \) (\( << L \) and \( W \)) refers to wall thickness. The width \( W \) of the cavity may be varied for a given height \( L \). Thus, a geometric parameter called aspect ratio (A) is defined as \( \frac{L}{W} \). Obviously, a smaller value of \( A \) means a wider cavity and a larger values of \( A \) implies a narrower cavity. There are two identical discrete heat sources, one in the left wall and the other in the right wall. As can be seen, the heat sources are flush-mounted and are of height \( L_h \) and thickness \( t \). The heat source in each wall can take up any position along the wall. The volumetric heat generation in each heat source is \( q_v \) W/m\(^3\). The thermal conductivity of the cavity as well as the heat source is taken to be \( k_s \), while \( \varepsilon \) is the emissivity of the surface of the cavity. Air, a radiatively transparent medium, is assumed to be filled in the cavity with \( T_\infty \) indicating its mean temperature. The exterior surface of the cavity is adiabatic and thus the heat generated in the two heat sources is conducted along the four walls of the cavity before subsequently getting dissipated to air by the combined modes of convection and radiation. Thus, one has a multi-mode heat transfer configuration, with each of the modes exhibiting its role in deciding the temperature distribution along the cavity.

Fig. 1 Schematic of the geometry chosen for the present study along with system of coordinates.
For example, the governing equation for temperature distribution along the portion of the left wall of the cavity possessing the heat source can be obtained by making an energy balance on a typical element pertaining to that portion as:

\[ q_{x,\text{cond, in}} + q_v \Delta x \frac{q_{x,\text{cond, out}} + q_{\text{conv}} + q_{\text{rad}}}{\Delta x} \]

(1)

The above, however, excludes the interfaces between heat source and non-heat source portions of the wall, which are to be tackled separately. After substituting for various terms in the above equation and simplifying, the governing partial differential equation for the above portion turns out to be,

\[
k_s t \frac{\partial^2 T}{\partial x^2} + h(T - T_{\infty}) - \frac{e}{1 - e} \sigma T^4 = 0
\]

(2)

Here, the Stefan-Boltzmann constant (\(\sigma\)) is taken equal to \(5.6697 \times 10^{-8}\) W/m² K⁴. Further, \(J\) is the radiosity (W/m²) of the element under consideration with all other terms having their meaning spelt out already. The following equation is used for computing the radiosity \[J(i)\] of a given element “i” of an enclosure comprising “N” number of elements:

\[
J(i) = e\sigma T^4(i) + (1 - e) \sum_{i=1}^{N} F_{ik}J_k
\]

(3)

In the above equation, \(F_{ik}\) refers to the view factor of any given element “i” of the enclosure with reference to each of the “N” elements of the enclosure including itself. For the non-heat source portion of the left wall, the energy balance would be the same as Eq. (1) with the second term on the left side pertaining to heat generation absent. The concerned partial differential equation comes out as:

\[
k_s t \frac{\partial^2 T}{\partial x^2} + h(T - T_{\infty}) - \frac{e}{1 - e} \sigma T^4 = 0
\]

(4)

Proceeding in a similar manner, the governing partial differential equation for the temperature at the interface between heat source and non-heat source portions of the left wall will be:

\[
\frac{q_v}{2} + k_s t \frac{\partial^2 T}{\partial x^2} + h(T - T_{\infty}) - \frac{e}{1 - e} \sigma T^4 = 0
\]

(5)

For the top and bottom walls of the cavity excluding the corners, the governing equation will be:

\[
q_{y,\text{cond, in}} = q_{y,\text{cond, out}} + q_{\text{conv}} + q_{\text{rad}}
\]

(6)

The corners of the cavity are the unique points of the computational domain as each of them will be shared by two of the walls of the cavity. For example, the temperature of the left bottom corner of the cavity will be given by:

\[
k_s t \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) - h(\frac{\Delta x + \Delta y}{2})(T - T_{\infty}) - \frac{e}{1 - e} \sigma T^4 = 0
\]

(7)

Here, \(\Delta x\) and \(\Delta y\) indicate the lengths of the elements of the cavity in vertical (x) and horizontal (y) directions. The governing equations for temperature distribution along the right wall and the remaining three corners of the cavity are also obtained in a similar manner. The various governing equations for the temperature distribution along the rectangular cavity obtained as above are nonlinear partial differential equations. They are first converted into algebraic form using finite difference formulation ensuring second order accuracy. The resulting equations are solved simultaneously using Gauss-Seidel iterative solver. Full relaxation (relaxation parameter = 1) is used on both radiosity (J) and temperature (T) during the iterations. A strict convergence criterion of \(10^{-8}\) is used to terminate the iterations. A computer code in C++ is written specifically for solving the problem.

3. RESULTS AND DISCUSSION

Before taking up the parametric study, it is customary to arrive at an optimum grid system for discretising the computational domain.

3.1 Grid sensitivity analysis

Grid sensitivity analysis is carried out for fixed set of input parameters, viz., \(q_v = 5 \times 10^3\) W/m³, \(h = 5\) W/m² K, \(k_s = 0.25\) W/m K, \(e = 0.45\), \(L = 20\) cm, \(W = 10\) cm (or A equal to 2) and \(T_{\infty} = 25\)°C. Further it is assumed that the heat sources of both the left and right walls of the cavity are at the respective wall centers. Grid sensitivity is tested in two stages. In Stage 1, the number of grids (N) in the horizontal direction (width of the cavity) is held fixed at 51 arbitrarily, while the grid number (M) in the vertical direction (left or right wall of the cavity) is varied. The variation of the maximum cavity temperature \(T_{\text{max}}\) with reference to varying M is studied.
Simultaneously, an energy balance check has also been made with varying M. Table 1 shows the results of this stage of the study. It may be noticed that $T_{\text{max}}$ changes by 0.0386% as M increases from 97 to 113, by 0.0299% as M further increases to 129. A further increase in M to 145 changes $T_{\text{max}}$ only by 0.0241%, indicating a satisfactory convergence. Even the check for energy balance, as noticeable from the same table, substantiates the fact that a value of M = 129 would suffice. Subsequently, Stage 2 of the grid study is performed keeping the value of M equal to 129 frozen as above and varying N. Table 2 summarizes results of this study. It may be noticed that $T_{\text{max}}$ changes by 0.0723% as N increases from 51 to 61, while it changes by 0.0502% with a subsequent increase in N from 61 to 71. The table also shows a satisfactory energy balance check for N = 61 with net heat generated in the cavity deviating from net heat dissipated only by 0.3768% for that value of N. Thus, the value of N is fixed at 61. In conclusion, for all the subsequent studies, the values of M and N are taken to be 129 and 61 respectively.

### Table 1: Stage 1 of grid sensitivity analysis.

<table>
<thead>
<tr>
<th>S. No</th>
<th>M</th>
<th>$T_{\text{max}}$ [°C]</th>
<th>Percentage change (abs)</th>
<th>Energy balance check (%) (abs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>65</td>
<td>103.729</td>
<td>-</td>
<td>0.913</td>
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<tr>
<td>2</td>
<td>81</td>
<td>103.659</td>
<td>0.0675</td>
<td>0.73</td>
</tr>
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<td>3</td>
<td>97</td>
<td>103.602</td>
<td>0.055</td>
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<td>0.0386</td>
<td>0.523</td>
</tr>
<tr>
<td>5</td>
<td>129</td>
<td>103.531</td>
<td>0.0299</td>
<td>0.46</td>
</tr>
<tr>
<td>6</td>
<td>145</td>
<td>103.506</td>
<td>0.0241</td>
<td>0.41</td>
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</table>

### Table 2: Stage 2 of grid sensitivity analysis.

<table>
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<tr>
<th>S. No</th>
<th>N</th>
<th>$T_{\text{max}}$ [°C]</th>
<th>Percentage change (abs)</th>
<th>Energy balance check (%) (abs)</th>
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</thead>
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<td>61</td>
<td>103.456</td>
<td>0.0723</td>
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<tr>
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<td>71</td>
<td>103.404</td>
<td>0.0502</td>
<td>0.3189</td>
</tr>
<tr>
<td>6</td>
<td>81</td>
<td>103.366</td>
<td>0.0367</td>
<td>0.2762</td>
</tr>
</tbody>
</table>

### 3.2 Variation of local temperature distribution with other parameters

Fig. 2  Local wall temperature profiles for varying positions of the discrete heat source.

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In the present problem, since the discrete heat source in each of the two vertical walls of the cavity is traversable along the wall, an attempt is made to probe into the local wall temperature distribution for varying positions of the heat source. Figure 2 shows results of such a study for a fixed set of input parameters, viz., $q_v$, $h$, $k_s$, and $\varepsilon$, as shown. Since the aspect ratio ($A$) is varied in the present study between 2 and 14, a value of $A = 8$ is chosen for the current study. As many as five positions are chosen for the heat source starting from the bottommost position to the topmost position in each of the vertical walls. Even though the heat source in one vertical wall need not be placed at the same position as that in the other vertical wall, for the current study, both the heat source are assumed to be at an identical position in their respective walls. In view of the above, the figure shows five curves and the heat source position ($L_1$ or $L_2$) pertaining to each of the curves is also given in the figure. The location at which peak temperature is attained along the wall varies as the heat source position is varied. This is expected on account of the fact that major heat transfer activity would always be there along and in the neighborhood of the heat source, with the rest of the wall merely conducting the heat generated in the heat source. Further, though marginally, the maximum wall temperature varies with the position of the heat source. Of the five positions chosen here, case 3 that considers heat source at the center of the wall results in the least of all possible peak temperatures. Traversing the heat source from the central location in either direction along the wall increases the peak wall temperature with the maximum noticed when the heat source is positioned at the bottommost and topmost ends of the wall. In summary, the current study indicates that the best possible location for each of the heat sources is the wall center itself.

Figure 3 shows variation of the local wall temperature for three different surface emissivities for the case where both the vertical walls have centrally located heat sources. All the remaining parameters ($q_v$, $h$, $k_s$, and $A$) are held fixed as shown. It is to be noted that $\varepsilon = 0.05$ signifies a good reflector (poor emitter), while $\varepsilon = 0.85$ signifies a good emitter. The current study helps in understanding as to how the temperature distribution along the wall of the cavity can be controlled by just varying the surface emissivity of the wall with rest of the parameters held fixed. It may be noticed that there is a huge drop in the maximum wall temperature as $\varepsilon$ increases from 0.05 to 0.45. However, the drop in maximum wall temperature is getting diminished with a further increase in $\varepsilon$. In the present case, the peak wall temperature is decreasing by 23.52% as $\varepsilon$ increases from 0.05 to 0.45. In contrast, the decrease in the peak temperature is just by 15.71% due to a subsequent increase in $\varepsilon$ from 0.45 to 0.85. It may also be seen from the figure that the local temperature profiles are crossing each other beyond a certain length on either side of the center of the vertical wall. Further, beyond the point of crossover, the local temperature along the wall is increasing with increasing emissivity. This is due to the fact that, when $\varepsilon$ is smaller, the temperature of the heat source portion will be larger due to low levels of emission from that portion. Since the same applies to the opposing wall, the irradiation leaving the opposing wall and impinging on the non-heat source portions of the given vertical wall would be very low. Thus, with other parameters held fixed, the temperatures of these non-heat source portions will be far lower than that in the heat source portion. As $\varepsilon$ increases, due to increased emission, the temperature of the heat source portion decreases. Simultaneously, the irradiation incident on the non-heat source portion of the given wall from the opposing wall increases, which increases the temperature of this portion of the wall.

In order to study the nature of variation of the local temperature distribution along the wall of the cavity in various regimes of convection for a given set of input, results are obtained as shown in Fig. 4. The fixed input chosen for the purpose is as shown and it may be noted that all the curves pertain to the case where the heat sources are at their respective wall centers. Based on some preliminary investigation, the range for convection heat transfer coefficient ($h$) is fixed to be $5 \leq h \leq 100$ W/m$^2$K, with $h = 5$ W/m$^2$K and $h = 100$ W/m$^2$K standing for asymptotic free and forced convection limits for the current problem. The figure reveals that the local wall temperature in
general and the peak temperature in particular decrease as the convection regime transits from free to forced convection dominance owing to increased convective heat dissipation from the surface for a fixed radiation activity. It may further be noticed that the above effect of $h$ is quite significant between $h = 5 \text{ W/m}^2 \text{K}$ and $h = 25 \text{ W/m}^2 \text{K}$, while it diminishes towards larger values of $h$. In the present example, the temperature at the center of either left or right wall is decreasing by 25.01% as $h$ increases from 5 to 10 $\text{W/m}^2 \text{K}$, by 33.92% for a subsequent increase of $h$ to 25 $\text{W/m}^2 \text{K}$ and by 36.93% for a further increase of $h$ from 25 to 100 $\text{W/m}^2 \text{K}$.

**Fig. 3** Variation of local temperature along the wall of the cavity for different surface emissivities.

**Fig. 4** Variation of local temperature along the wall of the cavity in different regimes of convection.
The effect of thermal conductivity \((k_s)\) of the wall material on the local temperature distribution is studied as shown in Fig. 5. The values of \(q_s, h, \varepsilon\) and \(A\) have been fixed as shown in the figure. Since electronic boards are typically made of materials of thermal conductivity of order unity, with glass epoxy \((k_e = 0.26 \text{ W/m K})\) being an example, the range for \(k_s\) is taken to be \(0.25 \leq k_s \leq 1 \text{ W/m K}\) for the study. For a given thermal conductivity, the maximum temperature is noticed at the wall center with a mirror image decrement in temperature taking place from there towards the ends of the wall. The peak wall temperature is diminishing with increasing \(k_s\). The decrease in local wall temperature with increasing \(k_s\) continues up to a little beyond the heat source portion on either side of the wall center. In the remaining portion of the wall on either side of the above section, one can notice an increase in the local temperature with increasing \(k_s\). The above trend may be attributed to the fact that for smaller values of \(k_s\), there is hardly any percolation of heat in the non-heat source portion of the wall. The above slightly improves with increasing \(k_s\). Thus the local wall temperature in the sections of the wall mentioned above increases with increasing \(k_s\), even though the maximum wall temperature still decreases. Since the present problem has the option of changing the aspect ratio \((A)\), an attempt is made to compare the temperature profiles pertaining to either of the vertical walls of the cavity for an identical heat source position in them with varying aspect ratios. Figure 6 shows the results of this study that considered heat sources at the respective wall centers. The aspect ratio is varied from 2 to 14 as shown with the remaining input parameters considered fixed. It is to be noted that an increasing aspect ratio implies a slender cavity with \(W \ll L\). The figure clearly shows that there is a large rise in the maximum cavity temperature with increasing aspect ratio. The above is because of the reduced radiative heat dissipation with convective heat dissipation too appropriately getting altered. With the two vertical walls getting closer, since both the vertical walls have heat sources at identical position, the radiative dissipation to the two horizontal walls of the cavity would obviously come down as their lengths get shortened gradually with increasing aspect ratio. In the present case, the peak cavity temperature is rising by 14.21% as \(A\) increases from 2 to 14. The figure further shows a crossover of the temperature profiles beyond a particular distance on either side of the heat source. Beyond the point of crossover, the local temperature of the vertical wall (left or right) is decreasing with increasing aspect ratio. The reason for this could be explained as follows. The volumetric heat generation and the net rate of heat dissipation from the wall is the same for all aspect ratios considered. Since the local temperature and thus the local temperature difference are increasing in the central portion of the wall with increasing \(A\), for a given \(T_s\), the local temperature towards the ends of the vertical wall should diminish with increasing \(A\). In summary, with the objective being control of the peak temperature assumed by the device, it is always wise to use cavities with smaller aspect ratios for a given rating of the heat sources.

### 3.3 Variation of maximum temperature in the cavity with other parameters

The prime task of a heat transfer engineer involved in the design of cooling systems for electronic equipment and devices would be control of peak temperature attained by the equipment or device. Figure 7 shows variation of maximum cavity temperature \((T_{max})\) plotted against surface emissivity \((\varepsilon)\) for three typical aspect ratios \((A)\). The fixed input considered for the study is also shown. The figure shows a very little effect of aspect ratio on \(T_{max}\) for the case where the cavity has poor surface emissivity \((\varepsilon = 0.05)\). However, towards larger values of \(\varepsilon\), there is a huge rise in \(T_{max}\) with increasing aspect ratio. It is to be noted that the heat sources are identically located at the concerned wall centers in this study. Thus, with increasing \(A\), the vertical walls get closer for a given height \(L\). This brings down the lengths of the horizontal walls receiving heat by radiation from the heat sources. Owing to this, the radiation activity from the vertical wall diminishes rising their temperatures. In summary, it would not be sufficient if only the surface emissivity \((\varepsilon)\) is increased in systems of this kind. Better results will be obtained if the above exercise is coupled with making cavities wider (or making \(A\) smaller). In the present example, for \(\varepsilon = 0.85\), the peak temperature of the cavity is
getting increased by 29.64% as A increases from 2 to 14. The interaction between convection and surface radiation in influencing the peak cavity temperature ($T_{\text{max}}$) has been looked into for a fixed set of input parameters comprising $q_s$, $k_t$, and $A$ as shown in Fig. 8. Since convective dissipation is quite small, while operating in free convection dominant regime ($h = 5 \text{ W/m}^2 \text{ K}$), there would expectedly be a greater role played by radiation (i.e., surface emissivity). This can be clearly seen in the figure where $T_{\text{max}}$ is dropping down quite significantly for the above value of $h$, a greater control of temperature in the cavity is possible by just altering surface of the cavity from poor emitter to good emitter (say from polished aluminum surface to black paint). However, increasing values of $h$ makes the flow forced convection dominant and cuts down the contribution from surface radiation. This is evident from much lower to insignificant drop in $T_{\text{max}}$ as the value of $h$ is increased first to $10\text{ W/m}^2 \text{ K}$ and subsequently to $25\text{ W/m}^2 \text{ K}$. In the present example, the drop in $T_{\text{max}}$ between $\varepsilon = 0.85$ is 31.3%, 19.68% and 7.99% for $h = 5, 10$ and $25\text{ W/m}^2 \text{ K}$, respectively.

Figure 9 summarizes the interactive influence of conduction within and convection from the surface of the cavity in influencing $T_{\text{max}}$. Here too, like what has been noticed in Fig. 8, the role of thermal conductivity in thermal control of the cavity is diminishing towards larger values of $h$. In other words, an increase in $k_t$ from 0.25 - 1 W/m K is bringing a very large drop in $T_{\text{max}}$ while operating the cavity in free convection ($h = 5\text{ W/m}^2 \text{ K}$). In contrast, if the cavity operates in forced convection environment, control of $T_{\text{max}}$ would hardly depend on the material chosen for the cavity as convection prevails here bringing down $T_{\text{max}}$ significantly for any given value of $k_t$. It is to be noted that the above trends are seen with cavities made of materials whose thermal conductivity is between 0.25 - 1 W/m K (i.e., materials like mylar-coated epoxy glass that is generally used in applications of this kind). In the present example, $T_{\text{max}}$ is coming down by 26.44% as $k_t$ increases from 0.25 - 1 W/m K for $h = 5\text{ W/m}^2 \text{ K}$. The same limits of $k_t$ for $h = 25\text{ W/m}^2 \text{ K}$ are bringing down $T_{\text{max}}$ only by 11.3%.

3.4 Effect of varying heat source position in one vertical wall on temperature distribution in the opposing vertical wall

Since the present problem has the option of varying the position of the heat source along the wall concerned, an attempt is made to study the temperature profiles of one of the vertical walls with a fixed heat source position in it by varying the heat source position in the opposing wall. Figure 10 shows the results of such a study made for a fixed input of $q_s$, $h$, $k_t$, $\varepsilon$ and $A$ as shown. The heat source on the left vertical wall is kept fixed at the bottommost position of the wall. The heat source pertaining to the right vertical wall is varied from the bottommost position to the topmost position of the wall. As many as five positions are chosen for this heat source as described in the figure itself. Thus, the figure has five curves indicating the left wall temperature distribution for the five different cases. When the right wall heat source is at the bottommost position, which means identically positioned heat sources in both the walls, the left wall temperature profile resembles curve 1 of Fig. 2 described already. As the right wall heat source is traversed to its second position, viz., to quarter distance from the bottommost end, the peak temperature in the left wall comes down. However, the local temperature of the left wall increases compared to curve 1 as one moves along. When the right wall heat source occupies the central position in the wall, not only the peak temperature of the left wall comes down further but also a second local peak is noticed in the left wall temperature profile as shown in curve 3. Similarly, one can notice occurrence of second local peak in curves 4 and 5 too. The reasons for the above changes in the local left wall temperature profile may be attributed to the following. As the heat source of the right wall moves to the second position, the corresponding length of the left wall obviously receives increased irradiation from the right wall, which tends to increase the local temperature of the concerned left wall portion. Subsequently as the heat source keeps moving upwards along the right wall, the temperature in the corresponding left wall portion also tends to increase owing to the same reason as above. Hence, one can clearly see a progressively increasing local temperature at the topmost end of the
left wall as the heat source of the right wall is traversed accordingly. In summary, the average temperature of the left wall, expectedly, remains almost the same in all the cases. If one focuses on bringing down the peak temperature along, the best position for the right wall heat source is the topmost position in the concerned wall. In the present case, the peak temperature of the left wall is coming down from 117.82°C to 107.35°C as the right wall heat source moves from the bottommost to the topmost position holding the left wall heat source fixed at the bottommost position all the time.

![Graph showing variation of local temperature along the wall of the cavity for different materials considered.](image1)

**Fig. 5** Variation of local temperature along the wall of the cavity for different materials considered.

![Graph showing variation of local temperature along the wall of the cavity for different aspect ratios.](image2)

**Fig. 6** Variation of local temperature along the wall of the cavity for different aspect ratios.
Fig. 7 Variation of maximum temperature of the cavity with surface emissivity for different aspect ratios.

Fig. 8 Variation of maximum temperature of the cavity with surface emissivity in three typical convection regimes.
Fig. 9 Variation of maximum temperature of the cavity with thermal conductivity of the wall material in three regimes of convection.

Fig. 10 Local temperature profiles of the left wall of the cavity for a stationary heat source in it and traversing heat source in the right wall.
4. CONCLUDING REMARKS
The problem of combined conduction - convection - radiation from a rectangular cavity with a flush-mounted discrete heat source in each vertical wall has been numerically solved. A computer code making use of finite difference method coupled with Gauss-Seidel solver is written. A grid sensitivity analysis is made and the optimum grid system has been identified. A thorough study of the local temperature distribution and the maximum temperature attained by the cavity has been made by varying the input parameters like aspect ratio, heat source position, surface emissivity, convection heat transfer coefficient and thermal conductivity.

REFERENCES