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A Numerical Simulation on Multimode Heat Transfer Characteristics of a Diversely Heated Closed Cavity for Variable Aspect Ratio

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Abstract - *The current endeavor is about excitement ponders on multimode heat move from a rectangular depression with numerous non indistinguishable discrete hotness sources in its left divider for various viewpoint proportions (l/d) and air is believed to be a cooling medium. Administering conditions for temperature dispersion along the dividers of the depression are acquired by making an energy balance among the hotness produced, led, convected and emanated. Calculations connecting with radiation are finished by utilizing "Hottel's cross string technique". The resulting nonlinear incomplete differential conditions are changed over into mathematical structure utilizing limited distinction procedure and are tackled all the while by Gauss-Siedel iterative strategy with a rigid assembly model. The effects of various boundaries, for instance, surface emissivity, convection heat move coefficient, heat conductivity of the divider material and perspective proportion on divider temperature circulation and pinnacle temperature of the pit are analyzed exhaustively. For the above purpose a code is written in C.*

Key Words: Multi-Mode Heat transfer, Cavity, Discrete heat sources, Aspect Ratio, Convection, Conjugate convection

1. INTRODUCTION

Cavities are like utilization of an insulting material. On the off chance that an air space is left between two layers making a wall or rooftop in any building, the air caught between two layers being poor conductor of heat goes about as a hindrance to heat transfer. Heat is exchanged over an air space by a blend of conduction, convection and radiation. Cavity wall insulation is utilized to diminish heat loss through a cavity wall by filling the air space with material that restrains heat transfer. Heat exchange through the cavities is utilized for expelling the heat from the greater part of the electronic gadgets like processers, chips, diodes, and so on. General cavity shapes considered are rectangular, circular, and tapered.

Many researchers were being carried out experimental or theoretical analysis on different ways to improve the performance of high rate of heat transfer through cavities. Examination was done utilizing the basics of Computational Fluid Dynamics (CFD). Majority of researchers used code in C language to simulate the peak wall temperature. Boundary conditions were taken similar to that of actual working environment to carry out the numerical simulations.

Experiments were done by Hakan et al. [2017] on six rectangular closed cavities filled with air each having following aspect ratios 1, 2.09, 3, 4, 5 and 6 having height of 340 mm and width of 210 mm. and detected that the temperature changes slightly in the central region and isotherms are focused near the vertical active wall. Natural convection heat transfers in a square enclosure with left hot wall comprising roughness elements in sine wave shape were experimentally studied by Usman et al. [2015]. Numerical examination was finished with an extent of the Prandtl number 1.0 and Rayleigh number from 10^3 to 10^6 for a Newtonian fluid. Average heat transfer increased and accomplished by keeping number of components equivalent equal to 10 and Rayleigh number as 10^6.Numerical investigation of arbitrary regimes of the natural convection on rectangular enclosure consisting of extreme heat source and heat-conducting walls with constrained width were performed by Sheremet et al. [2011]. The examination was progressed under the situation of the radiative-convective heat interchange with the surrounding medium on one of the outside wall surfaces. Silva et al. [2011] had done numerical analysis on a steady buoyancy-driven flow square two dimensional cavity with internal heat source and consistent temperatures on left wall, opening on the right wall and insulated top and bottom walls for low Rayleigh number the temperature distributions came as parabolic shape. The numerical outcomes show a generous effect of the opening on the heat transfer in the cavity. Effect of discrete heating on free convection heat transfer in a rectangular enclosure with left surface is contained with discrete heat sources and the right wall is isothermally cooled at a lower temperature is contemplated by Sankar et al. [2010]. The average heat exchange rate is seen to be progressed than on account of base heater while appeared differently with the top heater. Roy et al. [2009] completed investigation on both natural and forced convection in a top driven porous square cavity with unvarying heated base and side walls and the rate of heat transfer and Nusselt number are studied. Convective-radiative heat moves in a closed enclosure with a heat making source on the walls is numerically contemplated by Sheremet et al. [2009]. Kandaswamy et al. [2008] have done numerical exploration on unsteady laminar natural convection in a closed cavity furnished with inside heat generation and the four walls of the enclosure are thermally dynamic on side surfaces. Rao et al. [2007] numerically investigated the combination of surface radiation with conduction and convection from a vertical channel furnished with three flush mounted different heat sources in the left wall. It is observed that the impact of surface radiation in issues of this class can't be disregarded in any administration of convection.

The literature comprises of huge experimental and numerical examinations on heat transfer in different cavities by altering aspect ratios and heat sources. A careful in sight reveals that the studies accessible on a discretely heated cavity are less. likewise, in most part of the examination the impact of radiation is not considered. So by keeping the above actualities, the present problem, unique in its kind, has been considered to perform a numerical multi-mode heat transfer simulation from a cavity equipped with multiple non-identical discrete heat sources in one of its wall. The mathematical formulation of the present problem involves acquiring the governing equations by appropriate energy balance by changing over the governing equations into algebraic form by FDM and to illuminate the algebraic equations by writing a code in C language. It is further decided to study the impact of different parameters on wall temperature distribution and peak wall temperature for various aspect ratios by considering radiation into account

2. PROBLEM DEFINITION AND SOLUTION METHODOLOGY

A. Problem Definition

Figure 1 shows the geometry of the problem chosen for study along with coordinate system. It consists of a closed cavity of height *l* and width *d*. Thickness of each wall is t. Unit thickness is considered in z-direction for all the walls. The cavity is installed with three non-uniform discrete heat sources organized in the descending order of their heights from bottom to top of its left wall. The heights of the three heat sources are, individually, H1, H2 and H3. The first, second and third heat sources are placed, respectively, at the bottom end, geometric center and top end of left wall. The heights of the two non-heat sources are, respectively, NH1 and NH2 of equal size. The first non-heat source is halfway between the first and second heat sources, while the second one is halfway between the second and third heat sources. The remaining three which are right, top and bottom walls act on heat sinks. The thermal conductivity of the wall material is k W/m-K, while ε is its surface emissivity. In spite of the fact that the volumetric heat generation in every one of the heat source is q_v W/m^3, the net rate of heat generation in these heat source is non-identical attributable to distinction in their heights.

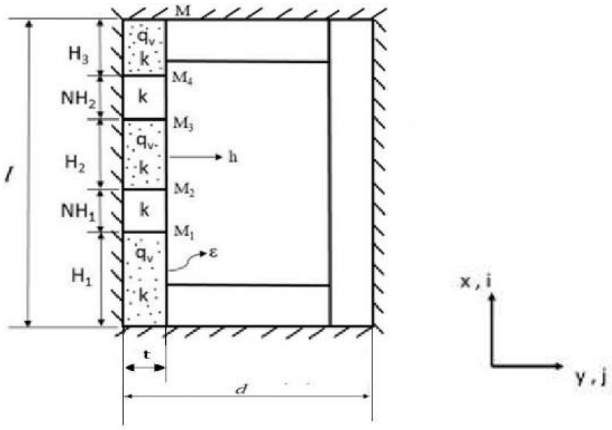

Figure 1: A rectangular closed cavity with multiple non-identical discrete heat sources along with its coordinate system

The heat generated in the three heat sources is conducted along the wall and is in this way dispersed through each of the three modes for heat transfer to be specific conduction, convection and radiation. Multimode heat exchange propagation is an examination in which every one of the three conceivable modes of heat exchanges will be considered however it is reliable practice to disregard the radiation impact. Air, radiatively transparent, is the cooling medium.

B. Solution Methodology:

Governing equations:

In the present problem the governing equations are obtained by traditional energy balance between heat generated, conducted, convected and radiated. For instance, the energy interactions on a typical interior element relating to the heat source on left side of cavity are shown in the Figure 2. After applying energy balance

$$q_g + q_{x\,in.} = q_{x\,out.} + q_{conv.} + q_{rad.} \tag{1}$$

After substituting for appropriate terms the above equations turn out to be

$$q_x + q_v * \Delta x(t) = q_x - \frac{\partial}{\partial x} kt \frac{\partial T}{\partial x}(\Delta x) + h * \Delta x(T_{ij} - T_\infty) + q_{rad.}$$

$$kt \frac{\partial^2 T}{\partial x^2}(\Delta x) - h * \Delta x\left(T_{ij} - T_\infty\right) - q_{rad.} + q_v * \Delta x(t) = 0 \tag{2}$$

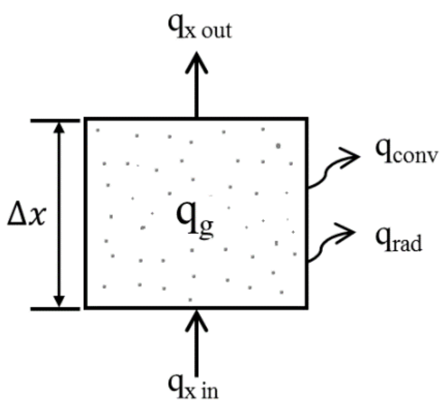

Figure 2: Energy interactions at the interior heat source elements of left wall of the cavity

Upon derivation of appropriate expressions for different terms, normalization, and ensuing simplification, the former non-linear partial differential equation obtained is

$$\frac{\partial^2 T}{\partial x^2} - \frac{h}{kt}(T_{ij} - T_\infty) - \frac{q_{rad}}{kt\Delta x} + \frac{q_v}{k} = 0 \qquad (3)$$

A similar procedure is adopted to obtain governing equations for various elements of the cavity which includes non-heat source elements, interface elements and corner elements.

The standardized governing equations, concerning heat transfer are nonlinear partial differential equations. They are changed into algebraic equations using the finite -difference method with 2nd degree of accuracy.

The obtained algebraic equations are solved simultaneously by Gauss-Siedel iterative technique with stringent convergence criteria. A code in C has been written for above purpose.

3. RESULTS AND DISCUSSION

The present work is proposed to carry a multi-mode heat transfer analysis from a discretely heated closed cavity and to study the variation in the local temperature distribution and maximum temperatures. The computational domain with coordinate of system is shown in Figure 3.

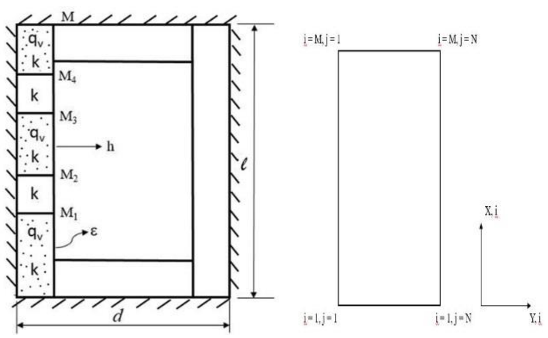

Figure 3: Computational domain of the problem geometry along with node representation

Range of input parameters

The scopes of different free boundaries concerning the issue were picked after some underlying examination. The length (l) and the thickness (t) of the plate are taken to be 20 cm (the normal kind of printed circuit sheets) and 1.5 cm, independently. The statures of the three hotness sources are H1 =5l/20 (5 cm), H2 =3l/20 (3 cm), H3 = 2l/20 (2 cm) and consequently the statures of the two non-heat sources are NH1 = l/4 (5 cm), and NH2 = l/4 (5 cm). The reach chose for the warm conductivity (k) of the hole is 0.25 to 1 W/m-K attributable to the way that the electronic sheets are consistently made of materials of warm conductivity of the request for solidarity (e.g., Mylar-covered epoxy glass has warm conductivity comparable to 0.26 W/m-K). The review has been performed for the scope of h=5 W/m2K, from asymptotic free convection limit to h=100 W/m2K free convection limit. For surface emissivity (ε) of the pit, the reach considered is 0.05-0.85, with ε = 0.05 addresses an unfortunate producer (or great reflector) while ε =0.85 is a decent producer (or unfortunate reflector). Since the current places of business form convection with radiation, there would be multi-mode heat move peculiarities present. Every one of the computations have been performed considering about air as the cooling medium, accepting for a moment that its encompassing temperature (to be equivalent to 310K.

Grid Convergence test:

Preceding taking up the parametric examinations, it is required to solidify the best possible grid system to be utilized during grid generation by performing out a proper grid convergence test. The test has been carried out by varying the number of grids both in x and y directions simultaneously and to study its effect on maximum temperature of cavity. Figure 4 demonstrates that T_{max} reaches its asymptotic limit for the elemental size Δx = 2.5 mm and Δy = 2.5 mm. The results are summarized in Table 1. So the optimum size of the grid for the present work is 81 × 41.

Table 1. Results of the grid convergence study

Δx	Δy	Nodes							T_{max}
		M_1	M_2	M_3	M_4	M_5	N	M× N	
0.01	0.01	6	11	14	19	21	11	231	971.18
0.005	0.005	11	21	27	37	41	21	861	965.22

0.0025	0.0025	21	41	53	73	81	41	3321	961.92
0.002	0.002	26	51	66	91	101	51	5151	961.14

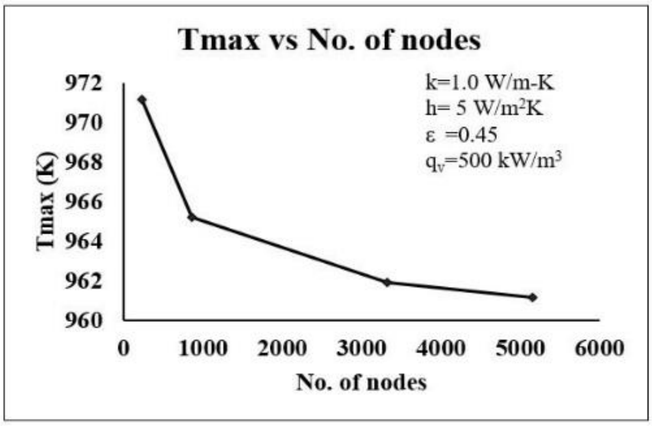

Figure 4: Variation of maximum temperature of the cavity with number of nodes

Notwithstanding the Grid Convergence, the outcomes of the current issue have also been tried for energy balance. This is done by discovering the net paces of hotness dispersal by blended convection and radiation by performing mathematical combination of their relating nearby qualities over the entire length of the hole. The net pace of hotness dissemination from the pit by joined strategies for conduction, convection and radiation is then acquired. This is contrasted and the total pace of hotness age in the hole that is identical to $q_v \times l \times t$.

***Validation*:**

The present work has been validated with the results obtained by (C. Gururaja Rao [2007]) for a heat transfer simulation over a channel with a single discrete heat source. For the above purpose, the pre geometry has been degenerated by removing the bottom and top walls of the cavity and by considering a single discrete heat source in the left wall. The comparison of results, as shown in Figure 5, shows a maximum deviation of about 2%. This acts as a good validation for the results of the present study.

A. *Variation of local temperature along the cavity in different regimes of convection*

Figure 6a shows the neighborhood divider temperature profiles for five unique upsides of convection heat move coefficient, h = 5, 10, 25, 50 and 100W/m2K, individually, to concentrate on the impact of systems of convection (free or constrained) on temperature dissemination. The worth of h = 5W/m2K infers free-convection-prevailing system, while h =100W/m2K means constrained convection-predominant system. It could be seen that temperature profiles relating to the left divider remains diminishes as one maneuver towards bigger upsides of convection heat move coefficient, clearly due to increment in the pace of hotness move by convection. Further, abatement of nearby left divider temperature is very huge between h = 5 and 25W/m2K, while that between h = 25 and 100W=m2K isn't quite as articulated as above. This is on the grounds that one is as of now in forced-convection system past h = 25W/m²K.

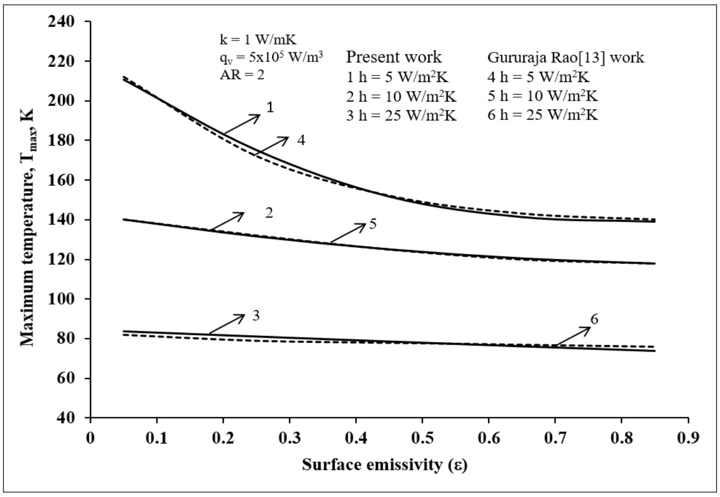

Figure 5: Validation of present work

The right, bottom, and top walls act as heat sinks those receive the heat from left wall through conduction and radiation. So the temperatures attained by right, bottom, and top walls due to the heat received from the left wall. In order to study the local temperatures along bottom, right and top walls the Figure 6b, 6c, 6d are drawn.

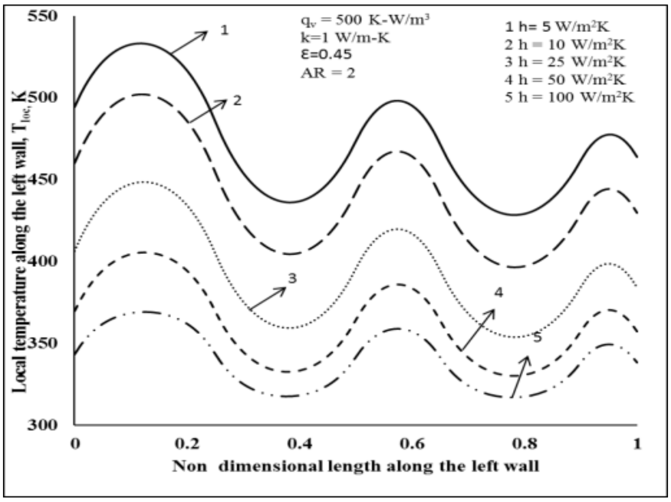

Figure 6a: Left wall

Figure 6b describes the variation in local temperature along the right wall for above input. The temperature along the right wall is first increased and reaches to maximum at nearer to the bottom of the wall and decreases further till the end. The local maximum is due to the high temperatures at the bottom of left wall from which it receives radiation.

Figure 6c, 6d depict, as the top wall and bottom wall act as extended surface for the left wall, the temperatures along left and right walls decreases monotonically along their lengths.

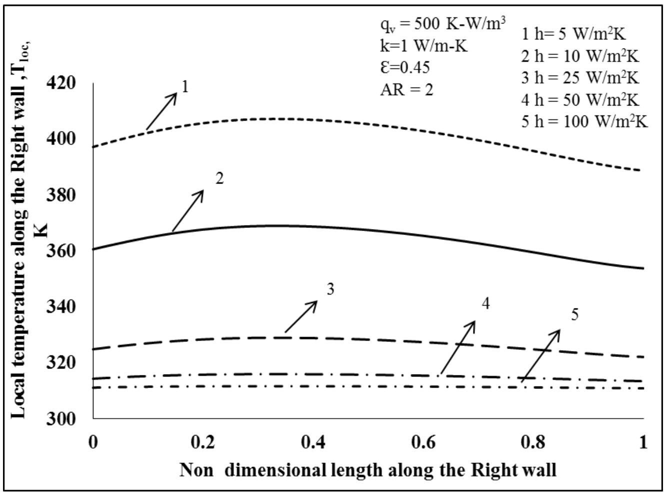

Figure 6b: Right wall

It is further observed from Figure 6a, 6b, 6c, 6d the temperatures attained by right bottom and top walls are almost equal to the ambient temperature. For h = 50 W/m^{2}K and h = 100 W/m^{2}K.Because, decreased source temperature (left wall temperatures) in forced convection regime.

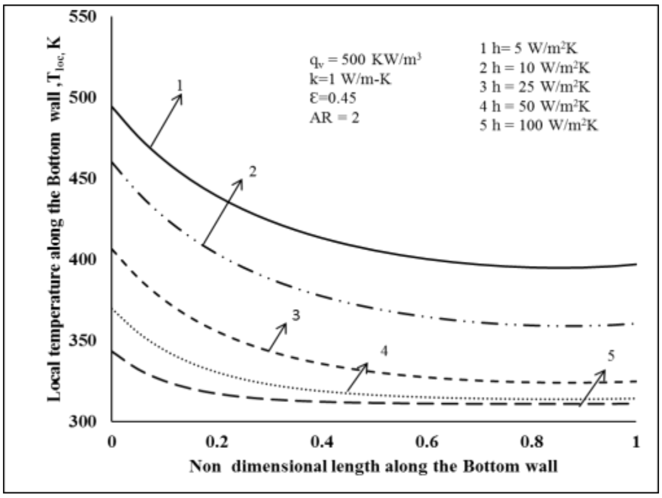

Figure 6c: Bottom wall

B. ***Variation in maximum temperature of cavity with emissivity in various regimes of convection***

The Figure 7 shows the variation in maximum temperature of the cavity (T_{max} with surface emissivity for a fixed Input of k = 1W/m-K. There is a monotonic decrease in T_{max} with increase in surface emissivity in all convection regime and with a more drop in maximum temperature in forced convection regime. The above is reasonable in light of the fact that the paces of hotness scattering by convection and radiation show an immediate proportionality with h and ε, separately. The diagram features the job of surface radiation in the free-convection-prevailing system. For instance, for h = 5 W/m2 K, Tmax goes through a 20.7% drop as (ε) increments from 0.05 to 0.85, while it diminishes by 1.135% a similar ascent in the constrained convection system (h = 100W/m^2 K).

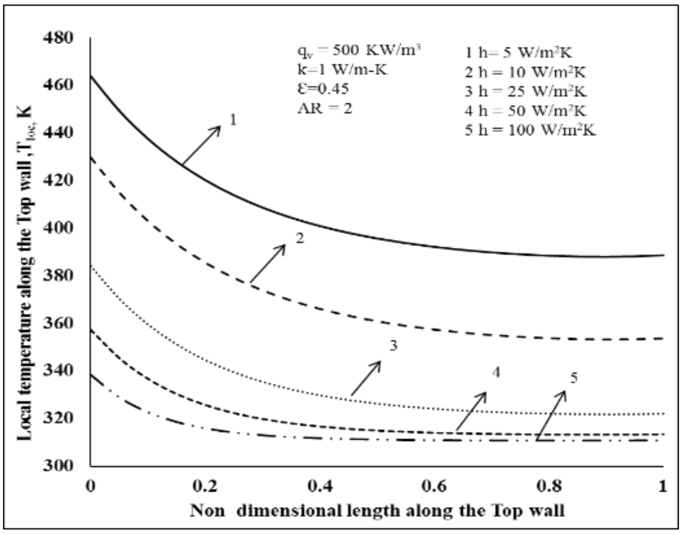

Figure 6d: Top wall

Figure 6: Local temperature profiles of cavity in different regimes of convection

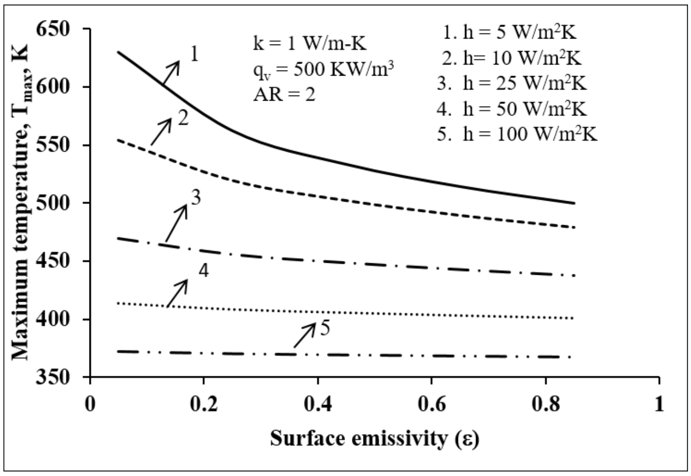

Figure 7: Variation in maximum temperature of cavity with emissivity in distinguished regimes of convection

C. ***Variation in peak temperature of cavity with convection heat transfer coefficient for different materials***

The family of curves in Figure 8 depicts the variation in maximum temperature of cavity in different cavity materials. Results are obtained for different values of thermal conductivity (k = 0.1, 0.25, 0.5, 0.75 and 1W/m-K) in five different regimes of convection for a fixed value of emissivity (ε =0.45).

Here h=5 W/m^2K implies an asymptotic limit of free convection, while h=100 W/m^2K is that of forced convection regime. For any material T_{max} decreases with transition in regime of convection from free to forced due to increased rate of convective heat dissipation. For example, for k = 0.1 W/m-K, T_{max} decreased by 33% with increase in h value from 5 to 100 W/m^2K. At higher values of h (i.e., beyond 80 W/m^2K) the effect of change of material on the maximum temperature is almost negligible.

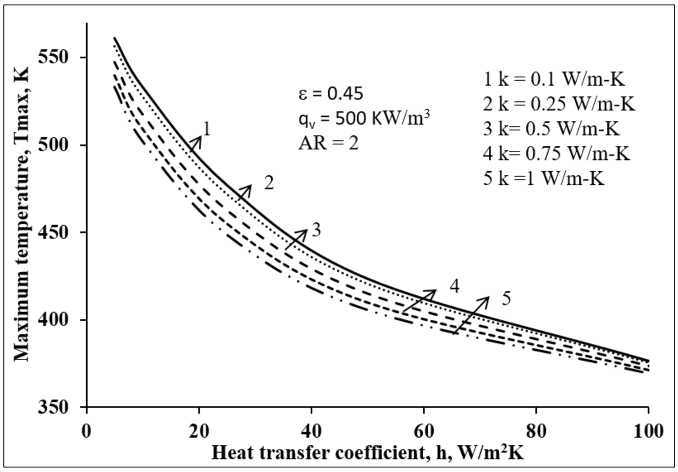

Figure 8: Variation in peak temperature of cavity with convection heat transfer coefficient for different materials of the cavity

D. *Variation in maximum temperature of cavity with thermal conductivity of materials*

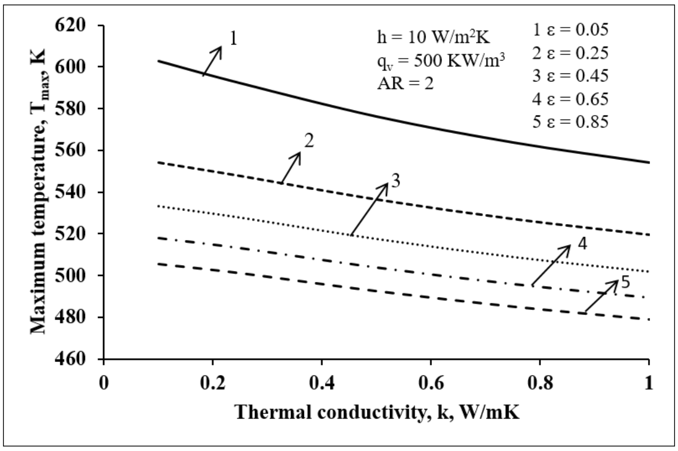

Figure 9: Variation in maximum temperature of cavity with thermal conductivity of cavity material

Figure 9 describes the variation of T_{max} with thermal conductivity of the cavity, for different emissivity value (i.e., ε□= 0.05, 0.25, 0.45, 0.65 and 0.85) for a fixed value of heat transfer coefficient (h=10 W/m^2K). It is obvious from the Fig. 11a that, for a given surface finish, the T_{max} is less for high thermal conductivity due to increased rate of diffusion and thus decreased temperature gradient. A similar trend is observed for all surface emissivities. For example, for ε = 0.05, T_{max} decreased by 8.02 % with increase in k value from 0.1 to 1 W/m-K.

E. *Exclusive effect of radiation in distinct regimes of convection on maximum temperature of cavity*

Figure 10 shows variation of maximum temperature with heat transfer coefficient for the emissivity value of ε = 0 and ε = 1. The maximum temperature is showing a decreasing trend with increase in the heat transfer coefficients. In the absence of radiation (i.e., ε = 0) the wall temperature is high in free convection regime. The effect of radiation on the maximum temperature at forced convection regimes is very less.

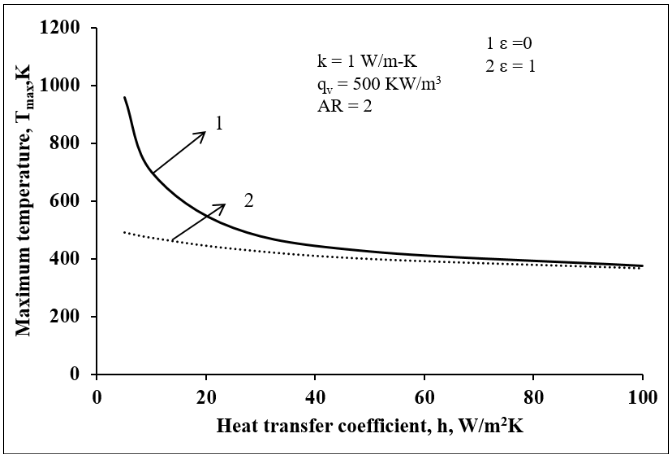

Figure 10: Role of radiation in distinct regimes of convection on maximum temperature of cavity

F. *Contribution of convection heat transfer and radiative heat transfer in total heat dissipation in various regimes of convection for different surface finish*

The present problem is multi-mode heat transfer simulation from a cavity in which the total heat generated is first conducted along the walls of cavity and subsequently dissipated through convection and radiation to the cooling medium.so, in order to know the contribution of convection and radiation. The following results are obtained for 5 different surface emissivities and five different regimes of convection as shown in Figure 11. It shows the relative contributions of convection and radiation to total heat dissipation, plotted against surface emissivity ε, for h = 5, 10, 25, 50 and 100W/m^2K, while the other parameters, namely, q_v =$5\times10^5 w/m^3$, k = 1 W/m-K, are held fixed. It may be seen that, for an increased surface emissivity in any regime of convection, the radiation contribution increases with a compensated drop in the convective heat dissipation.

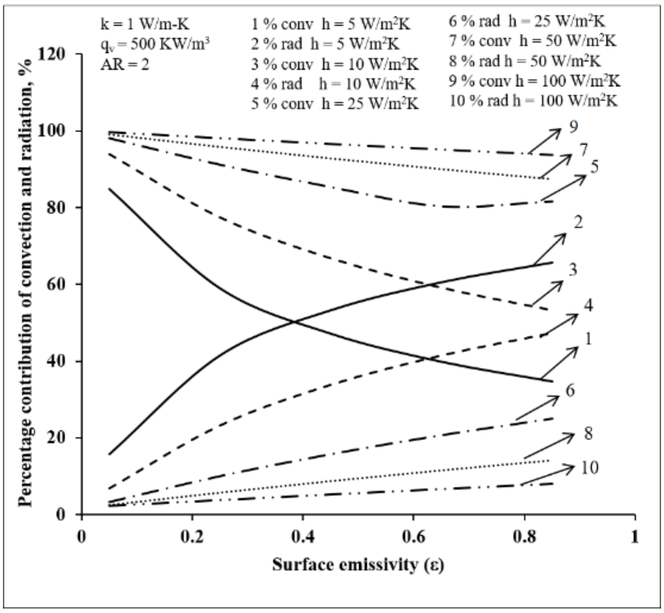

Figure 11: Relative contributions of convection and radiation to cavity heat dissipation for different surface emissivities in various regimes of convection

As the flow transits from free convection to forced convection the contribution of radiative heat dissipation decreases a lot and is almost negligible for h=100 W/m^{2}K.But, for h=5 W/m^{2}K for a surface emissivity greater than the heat dissipation through radiation is even more than that of convection. So one should take the radiation in to account while designing the cooling system. For example, the effect of radiation is more pronounced in the free-convection-dominant regime (h = 5 W/m^{2}K), the radiation contribution goes up to about 65.60 % from about 15.76 % as ε increases from 0.05 to 0.85.

G. *Variation of maximum temperature of the cavity with surface emissivity for different aspect ratios*

The heat generated in the cavity at the discrete locations is to be conducted first along the walls of the cavity is dissipated through convection and radiation to the cooling medium. The present geometry the left wall acts as heat source and all three other walls are heat sinks, where they receive heat from the left wall through conduction and radiation. It is a known fact that the radiative heat exchange is more if the objects are closer to each other. By keeping the above fact in mind the results are obtained to explore the effect of closeness of walls (aspect ratio) on maximum temperature of cavity for four different aspect ratios. The results are plotted for varying emissivity as shown in Figure 12. Due to increased radiative heat transfer from the left surface to other walls, the maximum temperature of the cavity is decreasing with increasing aspect ratio. For example, for emissivity = 0.05 as increase in aspect ratio from 1 to 8 T_{max} is decreased by 7.45%.

There is monotonic drop in maximum temperature with increase in aspect ratio for all values of emissivity due to increased radiation interaction with other walls. The drop in maximum temperature for ε = 0.05 is 6.48 % when changing the AR from 1 to 8 and for ε = 0.85 is 1.53 % for same change in AR.

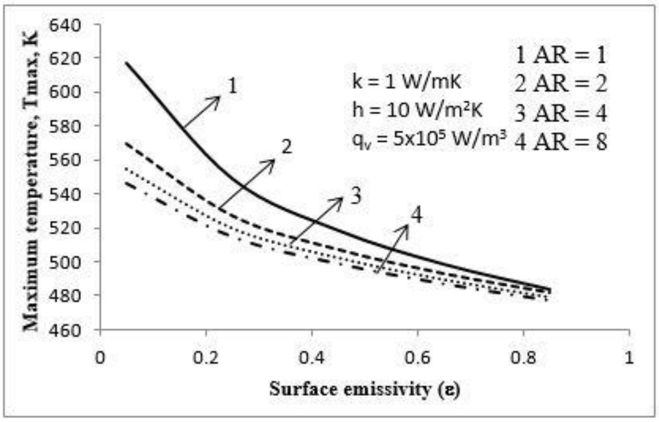

Figure12: Maximum temperature variation with aspect ratio for different surface emissivities

CONCLUSION

The problem of multimode heat transfer simulation on a rectangular closed cavity possessing three non-identical embedded discrete heat sources has been solved numerically by making use of finite difference method along with Gauss-Siedel iterative solution. A detailed study of the local temperature distribution of the cavity has been made by varying the input parameters like convection heat transfer coefficient. The effect of heat transfer coefficient describes that maximum temperature is decreased by 30.73% with increase in heat transfer coefficient from 5 W/m^{2}K to 100 W/m^{2}K for taken constant values of other parameters. A study on maximum temperature variation reveals that the increasing in emissivity values decreases the maximum temperature in all the regimes of convection, due to increased radiative heat dissipation to the cooling medium T_{max} for ε =0.05 to 0.85 at h=5 W/m^{2}K is 20.7%. The effect of thermal conductivity of cavity material on maximum temperature describes that a high thermal conductivity attains a lower values of peak temperature. It has observed 8.02% drop in T_{max} with change in cavity material possessing a 0.1 W/m-K thermal conductivity to 1 W/m-K for h=10 W/m^{2}K and ε = 0.05. A special study has been performed to elucidate the exclusive effect of radiation on maximum temperature. It shows that ignoring the radiation leads to an over estimation of maximum temperature of cavity and does the pumping power requirements. Quantitatively, in free convection regime, there is a huge drop of 48% is observed in T_{max} for the combined convection and radiation in comparison to that of pure convective heat dissipation.

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