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Numerical simulation of opposing mixed convection in differentially heated square enclosure with partition

S.K. Mahapatra [a,*], Anjan Sarkar [b], A. Sarkar [b]

[a] National Institute of Technology, Rourkela, Orissa 769008, India
[b] Jadavpur University, Kolkata-32, West Bengal, India

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Abstract

In the present study, opposing mixed convection in a differentially heated partitioned enclosure is investigated. The Richardson number (*Ri*) has been varied to simulate mixed convection ($Ri = 1$) and forced convection dominated flow ($Ri = 0.1$). Comparison is also made with natural convection in order to establish clarity in understanding the physics. The phenomenon inside the enclosure with centrally located partitions and offset partitions are also analyzed through isotherm pattern and streamline pattern and shielding effect of such partitions has been critically examined. It is found that when height of centrally located partition increases beyond a certain height of $0.3H$, the heat transfer in case of opposing mixed convection is found to be more than that of natural convection for a partitioned enclosure.

Keywords: Mixed convection; Partitioned enclosure; Rayleigh number; Richardson number

1. Introduction

Mixed convection problem has got its extensive applications in the field of engineering, for example cooling of electronic devices, furnaces, lubrication technologies, chemical processing equipment, drying technologies etc. Analysis of above phenomena incorporating the partitions, extends its usability to various other practical situations such as wooden partitions in an air-conditioned room, the plastic coated partitions in a refrigerator, a card or any projections on a motherboard of a computer etc. Also, various instruments provided for intrusive measurements inside practical systems can be simulated as insulated partitions. In this paper the shielding effect of such projections in the form of partitions along with movement of isothermal walls on energy transfer has been thoroughly examined and discussed.

Mixed convection within enclosures can be categorized as buoyancy aiding and buoyancy opposing flow. In the first case, it can easily be conceived that the natural convection loop becomes stronger forming a single loop inside the enclosure while in the later case, shear flow induced by the motion of wall opposes the buoyancy driven flow and makes the phenomenon more complex. Considering the complexity of phenomenon inside a partitioned enclosure with moving walls, a brief review of the relevant literature has been hereby presented.

Arpaci and Larsen [1] have presented an analytical treatment of the mixed convection heat transfer in tall cavities having one moving isothermal vertical wall at a temperature different from the other wall while horizontal walls are kept adiabatic. They showed that in this particular case, the forced and buoyancy driven parts of the problem could be solved separately and combined to obtain the general mixed convection problem. Aydin [2] studied numerically and revealed the mechanisms of aiding and opposing forces in a combined shear and buoyancy driven cavity. The phenomenon inside square enclosure was analyzed with moving vertical hot wall, either upwards or downwards while keeping the opposite cold wall fixed. In their work, a parametric study was carried out by varying Gr/Re^2 from 0.01 to 100 with $Pr = 0.71$ and three kinds of heat transfer regimes were identified. Further it was pointed out that the range of Gr/Re^2 for opposing mixed convection was

* Corresponding author.
E-mail addresses: swarupkumar_m@yahoo.com (S.K. Mahapatra), anjansirkar@yahoo.com (A. Sarkar).

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Nomenclature

AR	Aspect Ratio	
d	distance of partition from left wall	m
g	acceleration due to gravity	$\mathrm{m\,s^{-2}}$
Gr	Grashof number, $\beta g(T_h - T_c)H^3/\nu^2$	
h	partition height	m
H	enclosure height	m
K	thermal conductivity of air at mean temperature	$\mathrm{W\,m^{-1}\,K^{-1}}$
Nu	Nusselt number	
n	direction of normal vector	
p	pressure	$\mathrm{N\,m^{-2}}$
Pr	Prandtl number	
q_c	conductive heat flux on boundaries	$\mathrm{W\,m^{-2}}$
Ra	Rayleigh number, $\beta g(T_h - T_c)H^3/\nu\alpha$	
Re	Reynold number	
Ri	Richardson number, Gr/Re^2	
T	temperature	K
T_h, T_c	temperature of hot wall and cold wall	K
T_f	mean temperature of air, $(T_h + T_c)/2$	K
u, v	x-component and y-component of velocity	$\mathrm{m\,s^{-1}}$
Vp	velocity of isothermal walls	$\mathrm{m\,s^{-1}}$
w	width of the partition	m
x, y	rectangular coordinates	m

Greek symbols

ψ	stream function	$\mathrm{kg\,s^{-1}}$
φ^k	numerical value of any variable at kth iteration	
ρ	density	$\mathrm{kg\,m^{-3}}$
ν	kinematic viscosity	$\mathrm{m^2\,s^{-1}}$
β	volumetric expansion co-efficient	$\mathrm{K^{-1}}$

Subscript

0	reference state
h	hot wall
c	cold wall

Superscript

$*$	non-dimensional form

wider than that of aiding buoyancy case, although any quantitative information regarding these regimes was not presented. Prasad and Koseff [3] studied experimentally the combined forced and natural convection heat transfer process within a recirculating flow in an insulated lid-driven cavity of rectangular cross-section (150 mm × 450 mm) and depth varying between 150 mm and 600 mm. They established a correlation in their study varying Gr/Re^2 in the range 0.1–1000 by suitably changing the lid speed, the vertical temperature differential and the depth. Bhoite et al. [4] investigated numerically the problem of mixed convection flow and heat transfer in a shallow enclosure with a series of block like heat generating components for a range of Re and Gr and block-to-fluid thermal conductivity ratios. At higher values of Reynolds number ($Re > 600$), recirculation region is established in the core and the effect of buoyancy becomes insignificant. Guo and Sharif [5] studied numerically mixed convection heat transfer in a 2-D rectangular cavity with constant heat flux from partially heated bottom wall with vertically moving isothermal walls. In their work they have considered several different values of heat source length, the aspect ratio (AR) of the cavity, as well as symmetric and asymmetric placement of the heat source. They found that for asymmetric placement of heat source, the maximum temperature decreases, and the average Nusselt number increases as the source is moved more and more towards the sidewall. Pantokratoras [6] obtained numerical solution for the steady laminar boundary layer flow along upward moving vertical isothermal plates and cylinders maintained at a temperature lower than the surrounding. He has presented that the velocity boundary layer thickness increases as the buoyancy parameter decreases. Alleborn et al. [7] studied two-sided lid-driven shallow cavities. Vorticity-stream function formulation was employed in the work. They found that length of the cavity and velocity were the parameters affecting the mass transfer and established two flow configurations in which heat and mass transfer were minimized. Khanafer [8] studied the effect of mixed convection numerically in 2-D open ended enclosure for three different flow angles of attack. Their result shows that the thermal insulation can be achieved through the high horizontal velocity of flow. Oztop and Dagtekin [9] have investigated numerically the effect of mixed convection in two-sided lid-driven differentially heated square cavity. The work constitutes of both aiding and opposing mixed-convection. Ri emerges as a measure of relative importance of natural and forced convection modes of heat transfer. The work on mixed convection conducted by Mahapatra et al. [10] reveals that radiation has strong bearing on natural convection, which causes flow reversal in the middle of the cavity. Authors hereby feel to outline few earlier research works delineating the phenomenon in partitioned enclosure. The phenomenon in a two-dimensional, air-filled square enclosure with a vertical partition of finite thickness and varying height was investigated numerically in the laminar regime by Mezrhab and Bchir [11]. The horizontal end walls are assumed to be adiabatic, and the vertical walls are at different temperatures. Calculations are made by using a finite volume method and an efficient numerical procedure is introduced for calculating the view factors including the shadow effects. The results indicate that the partition does not significantly modify the heat transfer rate through the cavity, especially at high Rayleigh numbers, provided that its height is less than 90% of the cavity height. Kangni et al. [12] studied laminar natural convection and conduction in enclosures filled with air having multiple vertical partitions with finite thickness and conductivity using finite difference method as numerical technique. Varying Rayleigh number in the range 10^3–10^7 and aspect ratio in the range 5–20, the study was conducted. Apart from different spacing of the partitions, the ratio of thermal conductivity of the partition for the fluids was varied from 1 to 10^4 and the ratio of the thickness of the partition to the width of the enclosure was varied from 0.01 to 0.1. The number of partitions were varied from one to five. However no correlation was reported in the study. Ho and Chang [13] presented a numerical study concerning conjugate heat transfer across a vertical rectangular water-filled enclosure divided by multiple horizontal fins. The objective of the study was mainly to explore the effects of the conducting horizontal fins on heat transfer. Dagtekin and Oztop [14] studied numerically natural convection heat transfer and fluid flow of two heated partitions within an enclosure. The right side wall and the bottom wall of the enclosure were insulated perfectly while the left side wall and the top wall were maintained at the same uniform temperature. The partitions were placed on the bottom of the enclosure and their temperatures were kept higher than the non-isolated walls. It has been pointed out that the positions of partitions' have more effects on fluid flow than that of heat transfer.

The objective of the present study is to investigate the shielding effect of partitions on energy transfer in a square enclosure. The square enclosure is considered as the computational domain for the problem with moving vertical isothermal walls and the horizontal insulated walls. The Richardson number (Ri) has been varied to simulate natural convection, opposing mixed convection and forced convection dominated flow. The size and the location of the partitions are varied. In the present work the partitions are assumed to be adiabatic keeping in view their applications in many practical situations.

2. Analysis

The physical model of the problem along with its boundary condition is shown in (Fig. 1). It is assumed that the flow is two-dimensional, steady, laminar and incompressible. The thermo-physical properties of the fluid at a reference temperature is taken and kept constant except in the buoyancy term of the momentum equation where the Boussinesq approximation is used. In the light of assumption mentioned above, the non-dimensional continuity, momentum and energy equations can be written as follows:

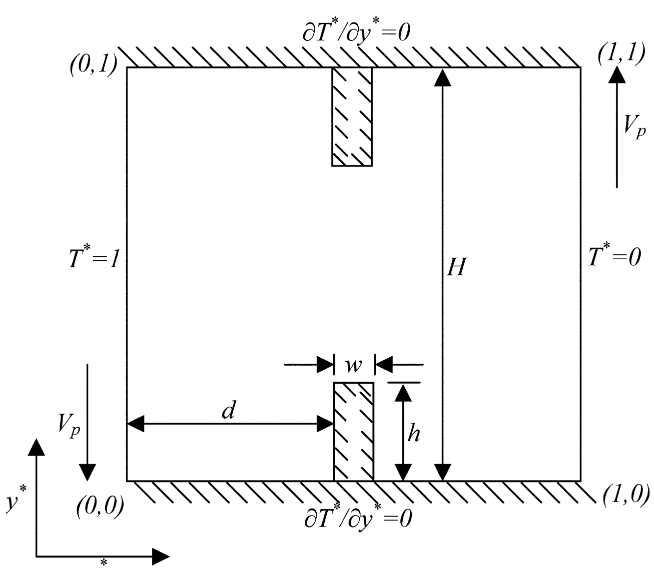


Fig. 1. Computational test domain.

Continuity equation

$$\frac{\partial u^*}{\partial x^*}+\frac{\partial v^*}{\partial y^*}=0 \tag{1}$$

The momentum equation

$$u^*\frac{\partial u^*}{\partial x^*}+v^*\frac{\partial u^*}{\partial y^*}=-\frac{\partial p^*}{\partial x^*}+\frac{1}{Re}\left(\frac{\partial^2 u^*}{\partial x^{*2}}+\frac{\partial^2 u^*}{\partial y^{*2}}\right) \tag{2}$$

$$u^*\frac{\partial v^*}{\partial x^*}+v^*\frac{\partial v^*}{\partial y^*} =-\frac{\partial p^*}{\partial y^*}+\frac{1}{Re}\left(\frac{\partial^2 v^*}{\partial x^{*2}}+\frac{\partial^2 v^*}{\partial y^{*2}}\right)+\frac{Ra}{Pr}(T^*) \tag{3}$$

Energy equation can be expressed as

$$u^*\frac{\partial T^*}{\partial x^*}+v^*\frac{\partial T^*}{\partial y^*}=\frac{1}{Re\,Pr}\left(\frac{\partial^2 T^*}{\partial x^{*2}}+\frac{\partial^2 T^*}{\partial y^{*2}}\right) \tag{4}$$

The following dimensionless variables are used to obtain above equations in non-dimensional form

$$x^*=x/H, \qquad y^*=y/H$$
$$u^*=u/v_p, \qquad v^*=v/v_p$$
$$p^*=p/\rho v_p^2, \qquad T^*=(T-T_c)/(T_h-T_c)$$

Vertical moving walls are considered to be isothermal while horizontal walls are assumed to be adiabatic. The relevant boundary conditions in dimensionless form are given as follows:

$$T^*=1 \quad x^*=0,\ u^*=0,\ v^*=-1,\ 0\leqslant y^*\leqslant 1$$
$$T^*=0 \quad x^*=1,\ u^*=0,\ v^*=1,\ 0\leqslant y^*\leqslant 1$$
$$q_c=0 \quad y^*=0,\ u^*=0,\ v^*=0,\ 0\leqslant x^*\leqslant 1$$
$$q_c=0 \quad y^*=1,\ u^*=0,\ v^*=0,\ 0\leqslant x^*\leqslant 1$$

$q_c=\frac{\partial T^*}{\partial n}=0$, for surfaces on partitions

The dimensionless velocities (u^*, v^*) appear as $u^*=\frac{\partial\psi^*}{\partial y^*}$, $v^*=-\frac{\partial\psi^*}{\partial x^*}$.

Table 1
Grid independent test for enclosure without partition ($T_h=323$, $T_c=303$, $Ra=10^4$)

Mesh size	q_c (hot wall)	q_c (cold wall)	Nu_h	Nu_c
31 × 31	1.4182	−1.4205	2.2727	−2.2764
41 × 41	1.4105	−1.4119	2.2604	−2.2626
51 × 51	1.4069	−1.4079	2.2546	−2.2562
61 × 61	1.4047	−1.4051	2.2511	−2.2517
71 × 71	1.4045	−1.4047	2.2508	−2.2511
81 × 81	1.4044	−1.4045	2.2506	−2.2508
101 × 101	1.4044	−1.4045	2.2506	−2.2507

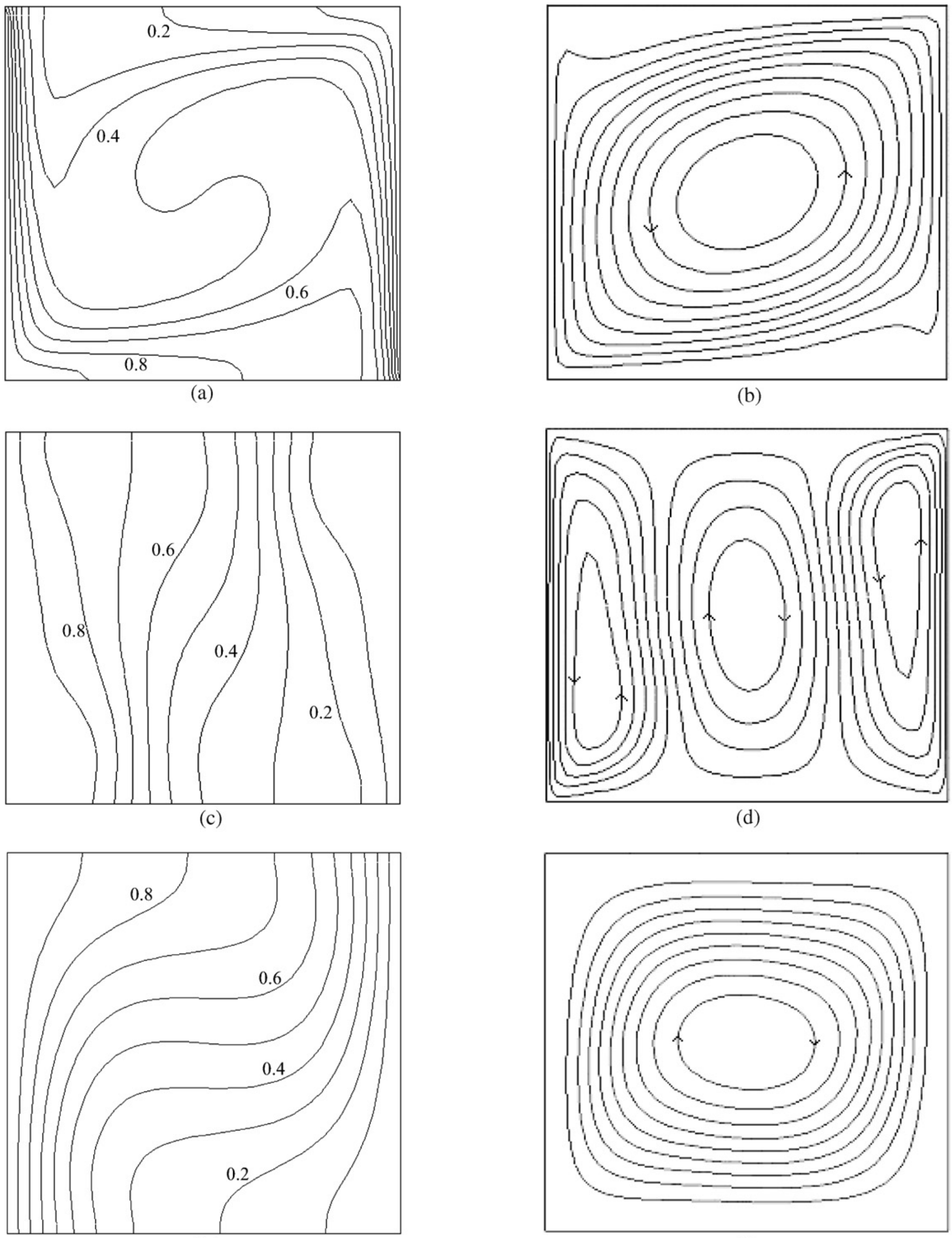


Fig. 2. Isotherm and streamline patterns, (a, b) $Ri = 0.1$, (c, d) $Ri = 1$ and (e, f) natural convection when $Ra = 10^4$.

Average Nusselt numbers on both the walls are calculated as per the given formula below:

$$Nu = \frac{q_c}{K(T_h - T_c)/H}$$

where $q_c = K\frac{\partial T^*}{\partial n}|_{x^*=0,1}$ is the conductive heat flux from or to the active walls.

3. Numerical procedure

All the governing equations are discretized by finite volume method provided in FLUENT 6.5 software. Three different grid sizes (101×101, 81×81 and 61×61) have been used and the mesh size of 81×81 has been considered for computational purpose in the present work after conducting grid-independent test (Table 1). The N-S equation formulated is solved by primitive variable approach following SIMPLE algorithm. While solving momentum and energy equation, the second order upwind scheme is chosen from options of the used software for obtaining convergence. In the present solution, under-relaxation parameter of 0.7 and 0.3 are used for momentum equation and pressure equation respectively for obtaining convergence.

The absolute convergence criterion $(\varphi^k - \varphi^{k-1})/\varphi^k < 10^{-9}$ for u^*, v^* and T^* is considered, where 'k' and '$k-1$' represent subsequent level of iteration. The calculated Nusselt numbers for both hot wall and cold wall are found to be almost equal for all type of configurations ensuring energy conservation.

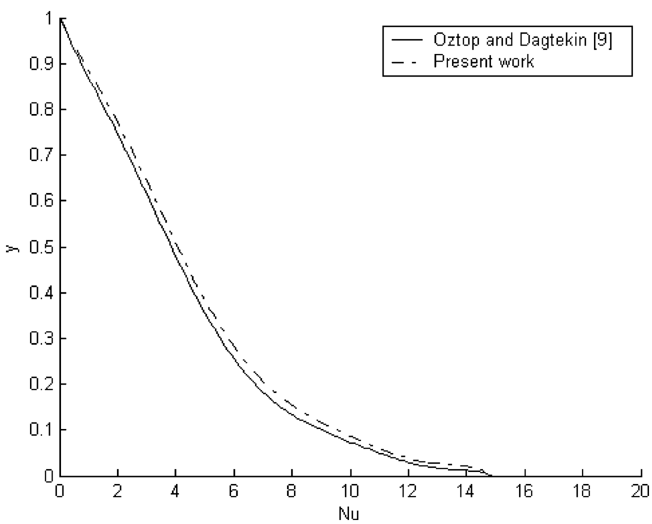


Fig. 3. Local Nu distribution along the cold wall and its validation against Oztop and Dagtekin [9].

4. Results and discussions

The mixed convection phenomenon inside a differentially heated partitioned enclosure is influenced by Ri, Ra, partition height (h) and its location (d). Rayleigh number is kept fixed at 10^4. The width of the partition is kept constant (i.e. $w = 0.1H$). Analysis of the results is made through obtained isotherm pattern and streamline pattern. Isotherm pattern is represented in the non-dimensional range of '0' (cold wall) to '1' (hot wall). Buoyant force caused due to temperature difference between hot wall and cold wall, generates a clockwise motion of the fluid. The mixed convection, i.e. coupling effect of natural convection and forced convection can be either aiding or opposing depending upon the direction of movement of the walls. Upward motion of hot wall and downward motion of cold wall induce a clockwise rotation of fluid inside the cavity, which obviously strengthens the natural convection. This is termed as buoyancy aiding mixed convection. Where as the opposing mixed convection is resulted when shear force induced by wall movement opposes the flow caused due to natural convection. The present work is focused only on buoyancy opposed mixed convection. The isotherm pattern and streamline pattern are shown in Fig. 2 for different values of $Ri(0.1, 1)$ and natural convection. At $Ri = 1$, when both the buoyant force and

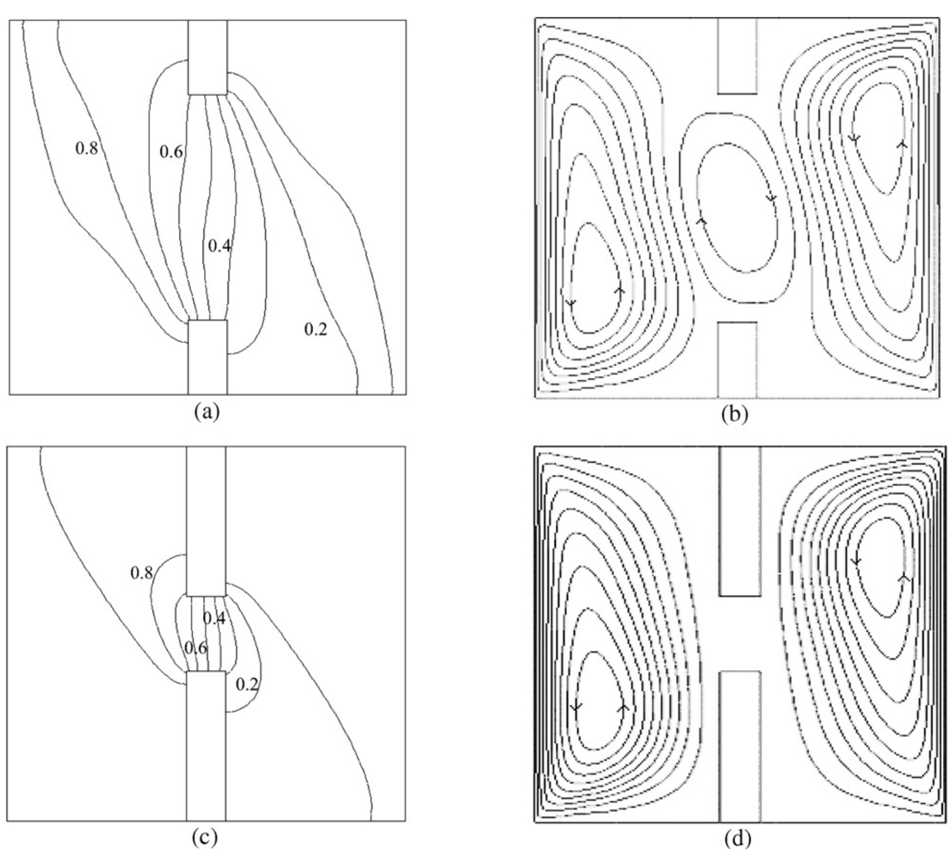


Fig. 4. Variation in isotherm and streamline patterns with mid-partition height, (a, b) $h = 0.2H$ and (c, d) $h = 0.4H$, when $Ra = 10^4$, $Ri = 1$.

the shear force are of the same order of magnitude, two shear cells are formed adjacent to the moving walls and a vortex cell caused due to buoyant force is seen in the middle. The isotherm pattern reflects a conductive pattern of energy transfer. It results a decrease in heat transfer compared to both pure natural convection and forced convection dominated flow ($Ri = 0.1$). $Ri = 0.1$ implies the dominance of shear induced flow. This is clearly reflected in streamline and isotherm pattern. The streamline pattern becomes unicellular in a counterclockwise direction (Fig. 2(b)), which is opposite to that caused due to natural convection (Fig. 2(f)). The isotherms get clustered at the top of the hot wall and bottom of the cold wall, where as for natural convection the packing of isotherms are found near the bottom of the hot wall and top of the cold wall. At $Ri = 0.1$, due to dominance of forced convection over natural convection, energy transfer is found to be higher.

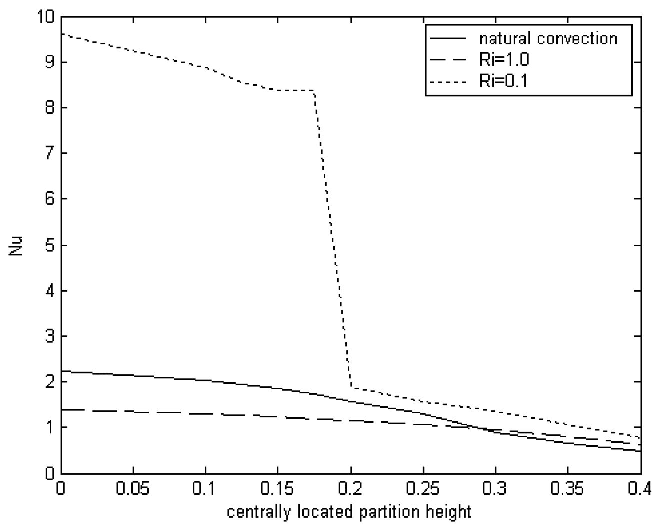


Fig. 5. Nusselt number of natural convection, $Ri = 1.0$ and $Ri = 0.1$ with respect to increasing centrally located partition height when $Ra = 10^4$.

The work has been validated against the work of Oztop and Dagtekin [9] for opposed mixed convection and presented in the Fig. 3. It is seen from Fig. 3 that a small deviation (<2%) is noticed only towards the lower portion of the cold wall and for the rest of the wall, the result is in good agreement. In the enclosure with insulated partition, the isotherm and streamline pattern for partition height of $0.2H$ and $0.4H$ are presented in Fig. 4. With increase in partition height, heat transfer decreases as partitions provide obstruction to the flow and consequently to the heat transfer. At $h = 0.4H$, the fluid flow because of wall motion becomes confined to either side of the partition. The middle vortex cell due to buoyant force vanishes. Therefore, the energy transfer through fluid movement is highly obstructed. The isotherm pattern reflects a conduction band between the partitions. On the either side of the partition, the size of the isothermal pool substantially increases, implying reduced rate of heat transfer. The reduction in rate of heat transfer becomes prominent in case of partition height $h = 0.4H$, where as, same is less with partition height $h = 0.2H$, as seen from Fig. 5. Comparing the 'Nu' as given in Fig. 5, it is observed that heat transfer in forced

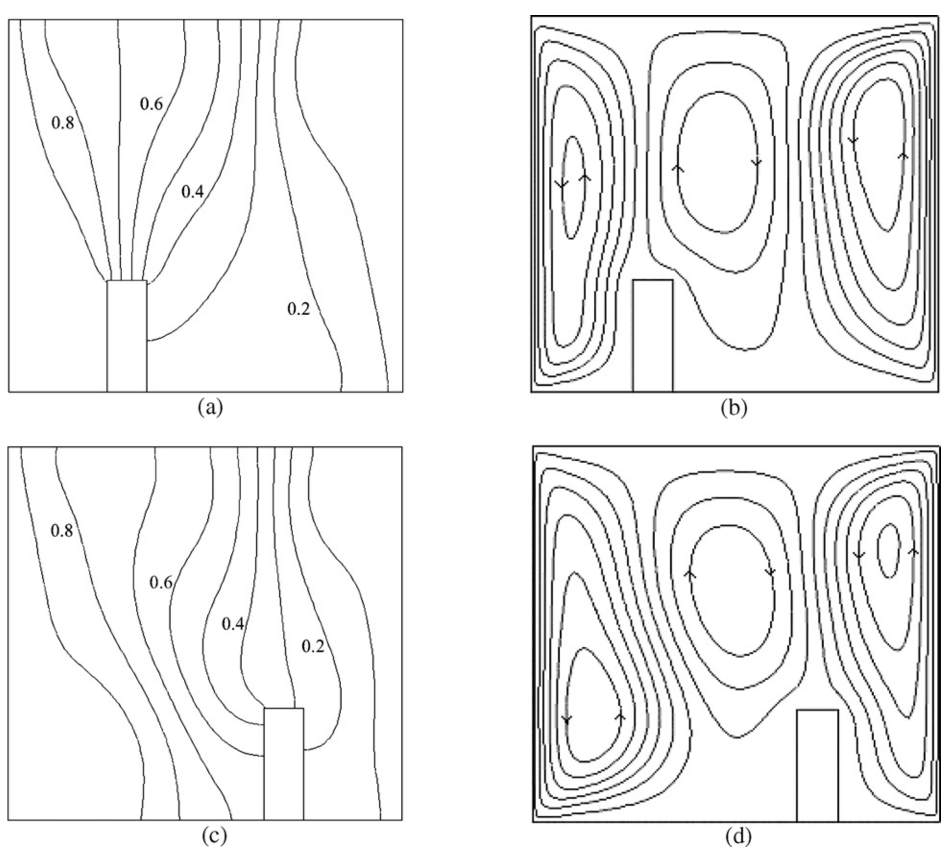


Fig. 6. Variation in isotherm and streamline patterns with location of partitions at base, (a, b) $d = 0.3H$ and (c, d) $d = 0.7H$, when $h = 0.3H$, $Ri = 1$, $Ra = 10^4$.

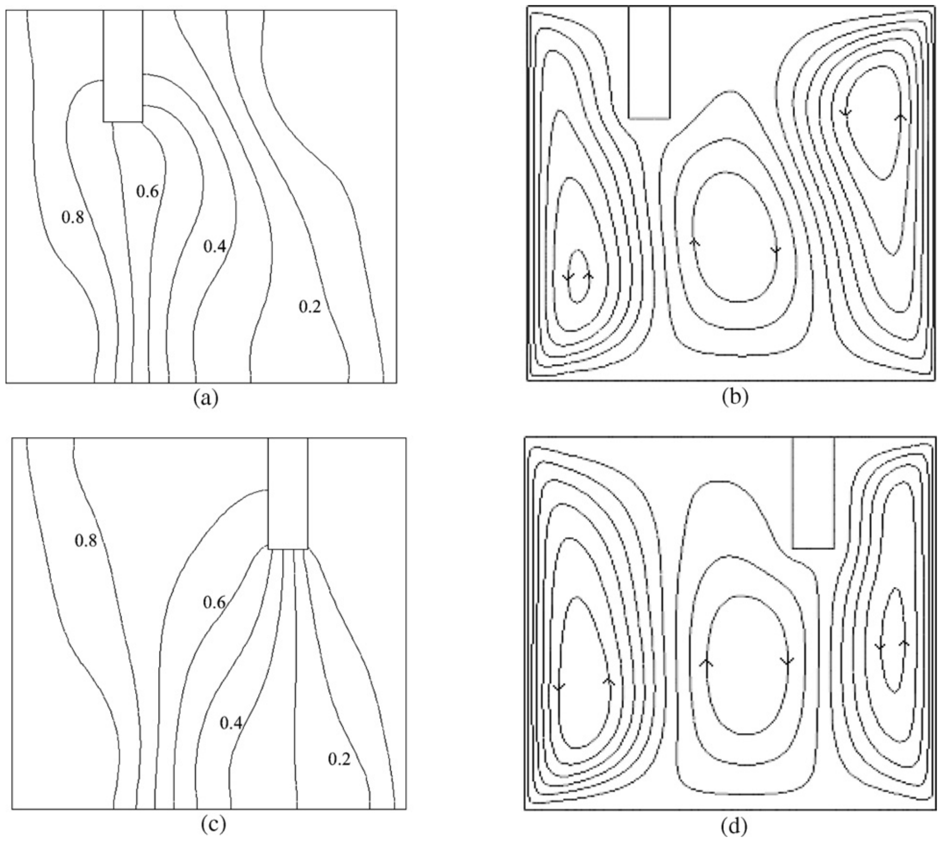


Fig. 7. Variation in isotherm and streamline patterns with location of partitions at ceiling, (a, b) $d = 0.3H$ and (c, d) $d = 0.7H$, when $h = 0.3H$, $Ri = 1$, $Ra = 10^4$.

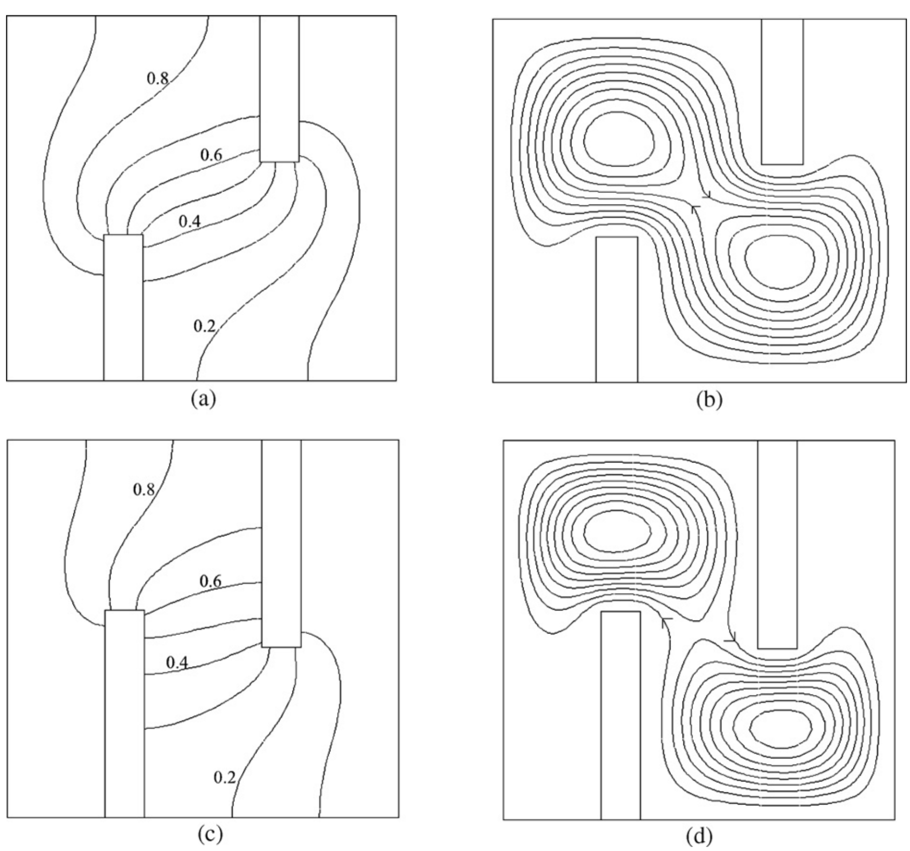


Fig. 8. Variation in isotherm and streamline patterns of natural convection with offset partition height, (a, b) $h = 0.4H$, (c, d) $h = 0.55H$, when $Ra = 10^4$.

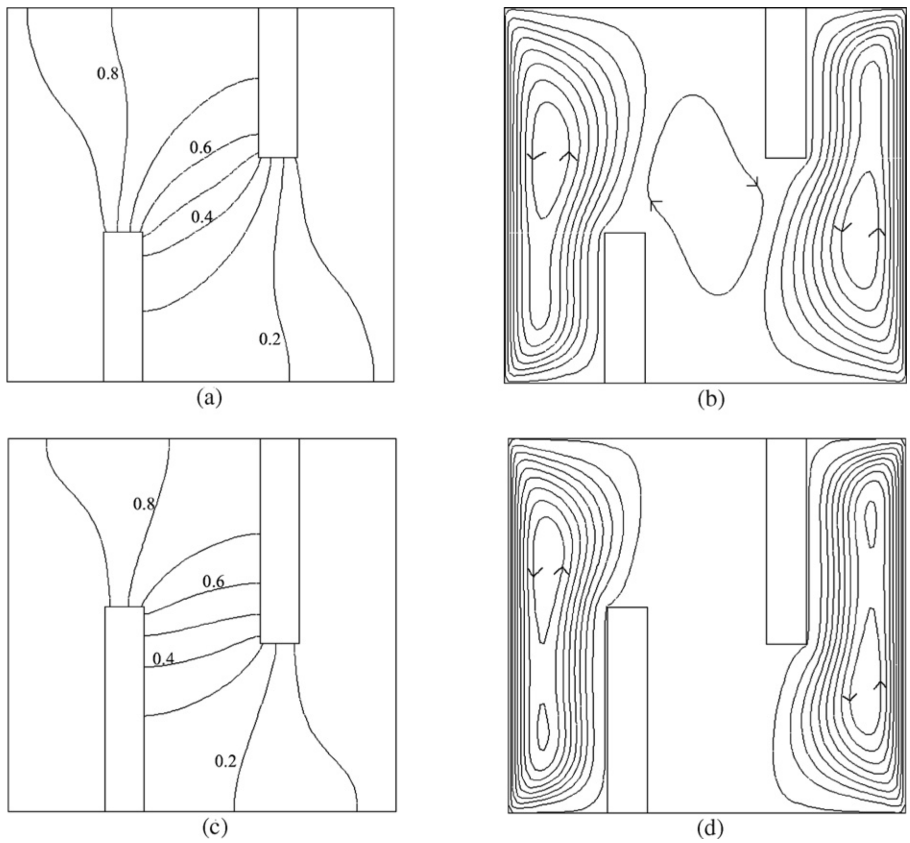


Fig. 9. Variation in isotherm and streamline patterns of mixed convection ($Ri = 1$) with offset partition height, (a, b) $h = 0.4H$, (c, d) $h = 0.55H$, when $Ra = 10^4$.

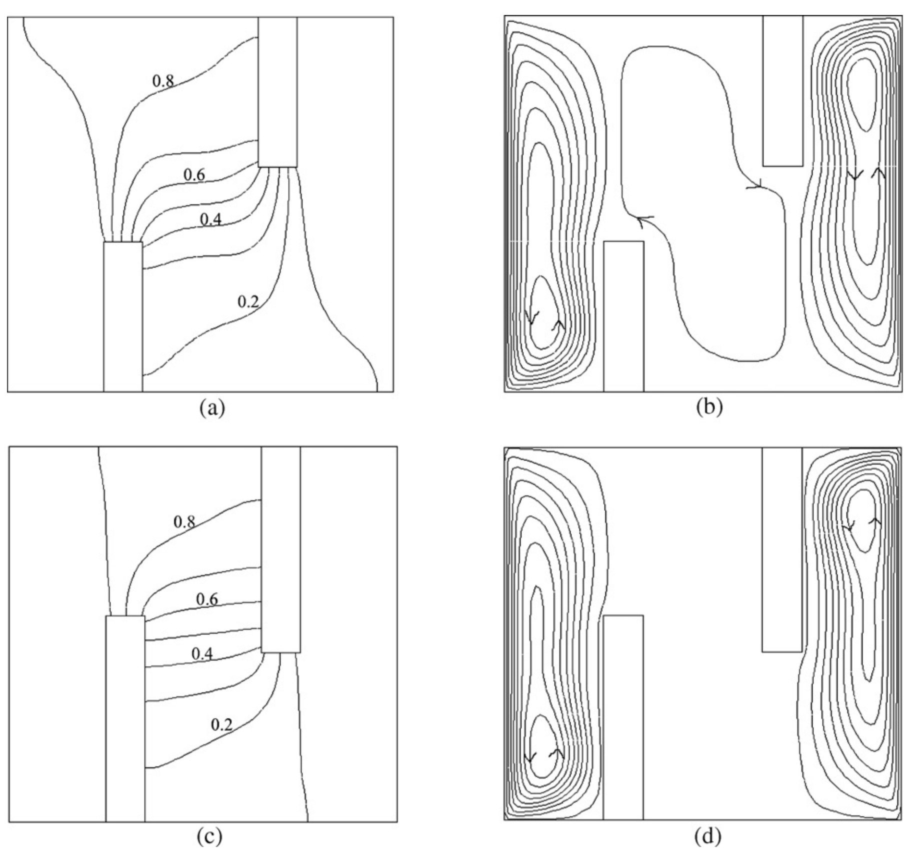


Fig. 10. Variation in isotherm and streamline patterns of forced convection dominated flow ($Ri = 0.1$) with offset partition height, (a, b) $h = 0.4H$, (c, d) $h = 0.55H$, when $Ra = 10^4$.

Table 2
Variation of Nu with the location of partition at base for natural convection, $Ri = 1.0$, $Ri = 0.1$ when $Ra = 10^4$

d	Natural convection		$Ri = 1.0$		$Ri = 0.1$	
	Nu_h	Nu_c	Nu_h	Nu_c	Nu_h	Nu_c
$0.3H$	1.8043	−1.8038	1.1474	−1.1545	6.3104	−6.2920
$0.5H$	1.6587	−1.6623	1.1845	−1.1917	5.7067	−5.7181
$0.7H$	1.5638	−1.5697	1.1909	−1.1984	8.4960	−8.4808

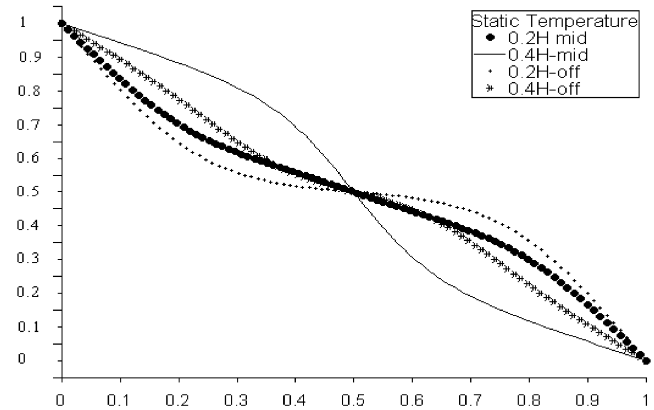
Table 3
Variation of Nu with the location of partition at ceiling for natural convection, $Ri = 1.0$, $Ri = 0.1$ when $Ra = 10^4$

d	Natural convection		$Ri = 1.0$		$Ri = 0.1$	
	Nu_h	Nu_c	Nu_h	Nu_c	Nu_h	Nu_c
$0.5H$	1.6609	−1.6604	1.1910	−1.1853	3.8079	−3.8053
$0.7H$	1.8024	−1.8059	1.1535	−1.1479	4.8997	−4.9170

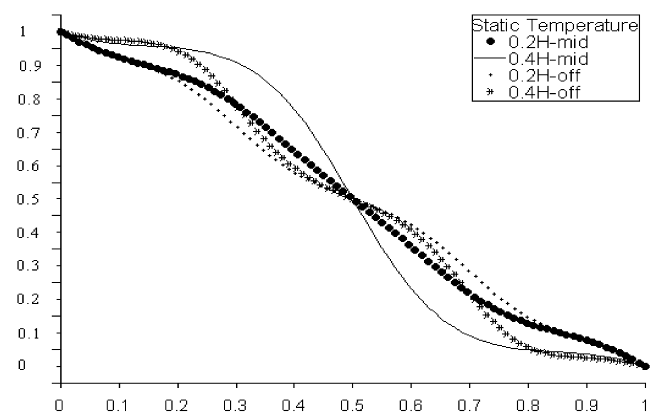
convection dominated flow is affected severely and reduced by almost 90%, where as, in natural convection it is reduced by almost 75%. The reduction in energy transfer due to the partition is found to be minimum in mixed convection phenomenon. An interesting feature may be noticed with reference to energy transfer when the height of centrally located partition increases. Beyond a certain height ($0.3H$), the heat transfer in case of opposing mixed convection is found to be more than that of natural convection for a partitioned enclosure. The presence of partition suppresses the fluid motion caused due to natural convection and may be contributing to a relatively quiescent zone near the active walls. However, in case of mixed convection ($Ri = 1$), the flow due to motion of the wall plays an important role in transferring energy. In this case, weak buoyancy flow (weak due to the partition) near the active walls results in a shear dominated flow leading to higher energy transfer. Hence partitions do not have the shielding effect of same order of magnitude in opposed mixed convection as seen in natural convection.

Fig. 6 shows the effect of partition in the base on isotherm pattern and streamline pattern. Referring to the Table 2, the heat transfer for mixed convection ($Ri = 1$) is found to be marginally less when the partition is placed near to the hot wall. This is opposite to the trend as observed in pure natural convection. It is to be noted that variation in heat transfer (Nu) with location of partition at base in mixed convection is significantly less compared to that in pure natural convection. For forced convection dominated flow, however, when location of partition is shifted from middle to either side causes an enhancement in the 'Nu' as observed from Table 2. When the partition is near the cold wall, it yields higher rate of heat transfer compared to its placement near to the hot wall. Fig. 7 shows the effect of partition provided on the ceiling. From Table 3, it is observed that the trend of energy transfer gets reversed as compared to that of partition at the base. Change in heat transfer is found to be less for mixed convection compared to that of natural convection.

The effect of offset partitions on the flow has also been studied and compared with that of centrally located partitions. The isotherm patterns and streamline patterns for natural convection, mixed convection ($Ri = 1$) and forced convection dominated flow ($Ri = 0.1$) are shown in Figs. 8–10 respectively for partition heights of $0.4H$ and $0.55H$. For offset partitioned enclosure, it is observed from the streamline patterns that even though the partition height is $0.4H$, there is no isolation of flow as is observed in case of centrally located partition. At partition height of $0.55H$, it is seen that a stagnant zone is resulted near the bottom of hot wall and at the top of cold wall. However, for $Ri = 0.1$ and $Ri = 1$, complete movement of fluid is seen within the active wall and partition because of wall motion. But they



(a)



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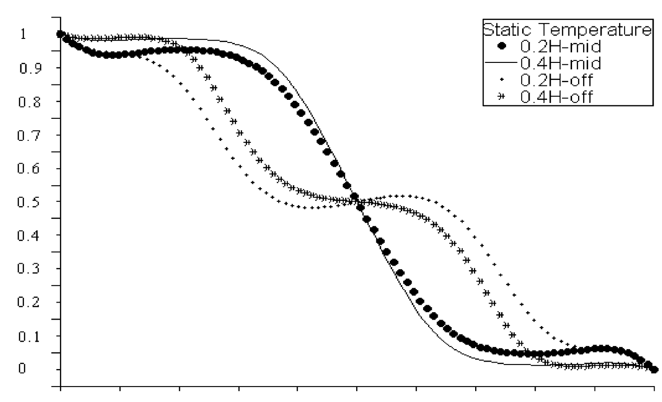

(c)

Fig. 11. Variation in mid-plane temperature with partition height of centrally located partitioned enclosure and offset partitioned enclosure for (a) natural convection, (b) mixed convection and (c) forced convection dominated flow.

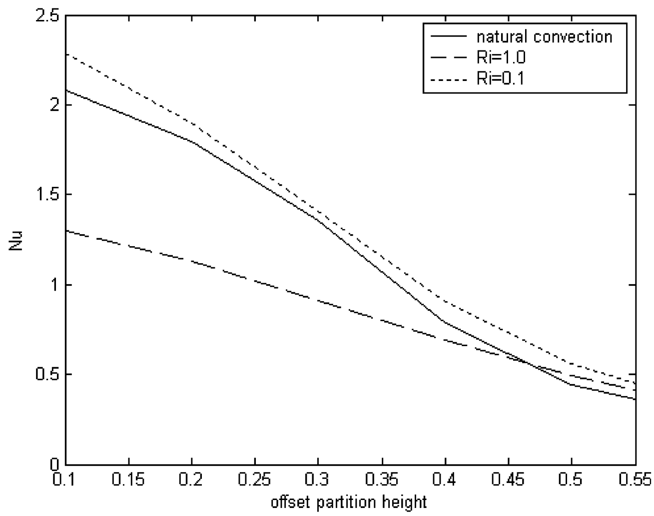

Fig. 12. Nusselt number of natural convection, $Ri = 1.0$ and $Ri = 0.1$ with respect to increasing offset partition height when $Ra = 10^4$.

get delinked at the middle of the cavity. The appearance of the conduction band in isotherm pattern in centrally partitioned enclosure somewhat disappears in offset partitioned enclosure. In order to establish the above findings the mid plane temperature variations for natural convection, mixed convection ($Ri = 1$) and forced convection dominated flow ($Ri = 0.1$) are presented in Fig. 11. From Fig. 12, it is seen that energy transfer is obstructed less in case of offset partitioned enclosure than that of centrally partitioned enclosure. This effect is quite marginal in case of mixed convection ($Ri = 1$). However, in natural convection, this change cannot be neglected. It is to be mentioned herewith that energy transfer in presence of offset partition is abruptly lower than the same in centrally partitioned enclosure, in case of $Ri = 0.1$, for partition height less than $0.2H$. But for other heights of partition, shielding effect in offset partition is less than the same in presence of central partition, in case of $Ri = 0.1$.

5. Conclusions

- For natural convection, higher heat transfer occurs in the enclosure when the partition at the base is near to the hot wall compared to the same when partition is located near the cold wall. This trend is reversed for $Ri = 1.0$ and 0.1. The effect of the partition placed at ceiling is found to be opposite in characteristics to the phenomenon in the enclosure having partition at the base.
- The effect of the location of partition on heat transfer is marginal for $Ri = 1$ and more pronounced for $Ri = 0.1$.
- When partition height exceeds $0.3H$, the heat transfer in case of opposing mixed convection is more than that of natural convection in centrally partitioned enclosure.
- Centrally located partition provides better insulating effect compared to offset partition for partition height exceeding $0.2H$ and this effect is less in case of mixed convection.

Acknowledgements

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