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Effect of active wall location in a partially heated enclosure ☆

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ABSTRACT

Natural convection in a partially heated enclosure has been investigated in order to identify the optimum location of the active wall for better heat transfer, in consideration with entropy generation. To achieve it, the effect of active wall positions on heat transfer and entropy generation has been studied exhaustively considering six different configurations with enclosure aspect ratios 1.5, 2 and 4. The study shows total entropy generation sets the criteria for better heat transfer. To find the suitability of present geometry and condition in other relevant applications (like heating, drying or mixing), the thermal mixing and temperature uniformity are also analyzed in this work. From the numerically simulated results, different sets of active wall locations have been identified for better heat transfer, thermal mixing and temperature uniformity for Rayleigh number 10^3–10^6 and Prandtl number 0.71.

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1. Introduction

Natural convection due to its inherent simplicity and adaptability finds widespread applications in many branches of sciences and engineering applications. In recent years [1–3], the awareness in natural convection increases significantly in the direction of entropy generation and heat transfer enhancement in context of the development of faster and high performance equipment like electronic devices or solar heating systems.

In the present study, the effects of different locations of active walls (hot and cold) on the thermal aspects are investigated in consideration with entropy generation and thermal mixing. The existing works available in literatures [4–11] related to shallow, square or tall differentially heated enclosures with partial heating and cooling, mainly emphasized in analyzing heat transfer and fluid-flow pattern in terms of Nusselt number and streamfunction. However, none of these works relates the heat transfer with the corresponding entropy analysis and tries to find out the criteria for better heat transfer. Moreover, the quality of thermal mixing or temperature uniformity in a partially heated rectangular enclosure has also remained unaddressed so far.

Kuhn and Oosthuizen [4] investigated the effect of heater location in a rectangular enclosure with a partially heated vertical wall keeping the other vertical wall fully cooled. They found that the Nusselt number was increased to a maximum and then decreased as the heater location was changed from the top to the bottom. Valancia and Fredrick [5] numerically analyzed the square cavity with partial (50%) active side walls at different heating locations, and observed that the heat transfer rate was enhanced when the heater was placed at the middle in this configuration. Ho and Chang [6] numerically and experimentally studied the effect of aspect ratio in a vertical rectangular enclosure. Numerical simulation was conducted for aspect ratio varying from 1 to 10 with a given relative heater size and location. The partially heated and partially cooled square cavities with unequal lengths of heater and cooler were numerically analyzed by Yucel and Turkoglu [7]. For a given cooler size, the mean Nusselt number was found to decrease with the increase in heater size. On the other hand, for a given heater size, the mean Nusselt number was observed to be increased with the cooler size. The effect of heater and cooler location on heat transfer was analyzed by Turkoglu and Yucel [8]. Frederick [9], using differentially heated rectangular cavities, found the aspect ratio for which the overall Nusselt number reached a maximum for several Rayleigh numbers. The study was performed from a shallow cavity regime to a slender one but the entire left wall was maintained as heated for all the cases. Nithyadevi et al. [10] investigated the effect of aspect ratio of rectangular cavity with 50% active side walls and different relative positions of the active zones and the result revealed that the heat transfer rate increased with aspect ratio and was observed to be high for the bottom–top thermally active location. More recently a comprehensive numerical investigation on the natural convection was presented by Alam et al. [11] in a rectangular enclosure with partial heating at the lower half of the left vertical wall and partial cooling at the upper half of the right vertical wall.

Andreozzi et al. [12] numerically predicted the local and global entropy generation rates in a natural convection in air in a vertical channel symmetrically heated at uniform heat flux. Within a vented enclosure with a heat generating solid body Shuja et al. [13] estimated entropy

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Nomenclature

A	aspect ratio (H/L)
Ec	Eckert number
g	acceleration due to gravity
L	heater length
N	number of scalar grid points
Nu_{avg}	average Nusselt number
NS	dimensionless total entropy generation
NS_{loc}	dimensionless local entropy generation
p	effective pressure
P	dimensionless pressure
Pr	Prandtl number (ν/α)
Ra	Rayleigh number ($g\beta(T_H - T_C)L^3 \Pr/\nu^2$)
RMSD	Root Mean Square Deviation
T	temperature
T_C	cold wall temperature
T_H	hot wall temperature
T_{ref}	dimensionless reference temperature
u, v	dimensional velocities
U, V	dimensionless velocities
x, y	Cartesian coordinates
X, Y	dimensionless Cartesian coordinates

Greek symbols

α	thermal diffusivity
β	thermal expansion coefficient
θ	dimensionless temperature
θ_{cup}	dimensionless cup-mixing temperature
ν	kinematic viscosity
ρ	fluid density
ψ	dimensionless streamfunction

generation during the natural convection. In transient laminar natural convection the entropy generation due to heat transfer and friction was predicted by Magherbi et al. [14] and found that entropy generation becomes the maximum at the onset of transient state. Whereas, Erbay et al. [15] investigated the entropy generation in a partially heated square enclosure during the transient laminar natural convection. Entropy generation during natural convection in a square cavity for different boundary conditions was studied by Basak et al. [16]. Mukhopadhyay [17] numerically analyzed the entropy generation due to the natural convection in square enclosures with multiple discrete heat sources. None of these works shows the effect of entropy generation in a partially heated enclosure and its relation on heat transfer.

In the present work heat transfer and entropy generation in a partially heated enclosure have been studied for Rayleigh number from 10^3 to 10^6, Prandtl number of 0.71 and enclosure aspect ratios of 1.5, 2 and 4. The present work is an extension of the previous works [10,11]. The objective of this work is to assess the heat transfer and entropy generation characteristics for different locations of active walls. It has been found that total entropy generation sets the criteria of maximum heat transfer. Distributed heating methodology gives better thermal mixing in material processing [18]. For many other industries, like drug manufacturing and chemical processing where mixing is required in small zone, the concept of distributed heating may not suit well. So, the effect of active wall locations on thermal mixing and temperature uniformity is also assessed in the present work. The investigation and analyses are presented in terms of streamlines, isotherms, average Nusselt number, entropy generation for different Ra and geometrical modifications.

2. Problem definition and numerical procedures

The schematic diagram of partially heated enclosure is depicted in Fig. 1. Except the portion of isothermal (hot and cold) walls, the rest of the walls of the enclosure are considered adiabatic. Six different configurations of active region arrangement are considered in this work — Case 1: bottom–bottom (BB), Case 2: middle–middle (MM), Case 3: top–top (TT), Case 4: middle–top (MT), Case 5: middle–bottom (MB), and Case 6: bottom–top (BT) as shown in Fig. 1. For each case, the hot region is located on the left vertical wall and the cold region on the right vertical wall. The length of both the active parts of the vertical walls is taken the same (L) and is equal to the width of the enclosure.

The numerical solution is based on the assumption that the temperature difference between the hot and cold parts of the walls follows Boussinesq approximation. The flow is considered to be steady, incompressible, laminar and two-dimensional. Since the temperature difference between two active regions is not so high, the transport properties of air (Pr = 0.71) are taken constant and the radiation from the walls is neglected. The dimensionless governing equations derived from the conservation principles of mass, momentum and energy become:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{1}$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \Pr\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \tag{2}$$

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \Pr\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) + \mathrm{Ra}\Pr\theta \tag{3}$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right). \tag{4}$$

The nondimensionalization is carried out on the basis of active wall length (L), thermal diffusivity (α) of fluid, and temperature difference ($T_H - T_C$) between the hot and cold parts of the walls as detailed below.

$$X = \frac{x}{L},\; Y = \frac{y}{L},\; U = \frac{uL}{\alpha},\; V = \frac{vL}{\alpha},\; P = \frac{pL^2}{\rho\alpha^2},\; \theta = \frac{T - T_C}{T_H - T_C} \tag{5}$$

The boundary conditions for Eqs. (1–4) are: U = V = 0 on all the walls of enclosure due to no-slip and no-penetration conditions; $\theta = 1$ and 0 at the hot and cold portions of the vertical walls, respectively, and zero normal gradient for adiabatic walls.

The solution of the above governing equations is obtained by means of finite volume method (FVM) of discretization. The evolved discretized equations are solved using SIMPLE algorithm [19] and alternating direction implicit (ADI) sweep on TDMA algorithm. For the convergence of solutions, the maximum value of mass defect is chosen to be less than 10^{-8} and the residual is below 10^{-6}. Uniform grid distribution of size 0.01×0.01 is considered for all Rayleigh numbers varying from 10^3 to 10^6.

Average Nusselt number represents the average heat flux across the active part of the wall. The local and average Nusselt numbers (Nu, Nu_{avg}) at the hot part of the wall are defined as,

$$Nu = -\left.\frac{\partial \theta}{\partial X}\right|_{X=0} \text{ and } Nu_{avg} = \int_{Y1}^{Y2} Nu\,dY \Bigg/ \int_{Y1}^{Y2} dY. \tag{6}$$

Here $Y1$ and $Y2$ are the locations of the lower edge and upper edge locations of the hot wall.