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Numerical study on mixed convection in an inclined lid-driven cavity with discrete heating ☆

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ABSTRACT

The present investigation deals with numerical analysis on mixed convection in an inclined square cavity with different sizes and locations of the heater. The left wall is heated fully or partially with higher temperature whereas the opposite wall is kept with lower temperature. In the left wall, three different sizes and locations of the heater are considered. The governing transformed equations are solved numerically using the finite volume method. Simulations are performed on different Richardson numbers, different sizes and locations of the heater and the cavity inclination angles. It is observed that the high heat transfer is found at cavity inclination angle of $\gamma = 30°$ in the buoyancy convection dominated regime when the heater is located at the middle of the cavity.

1. Introduction

The fundamental problem of mixed convection heat transfer in a closed cavity has received considerable attention from researchers in the past decades because of its applications in cooling of electronic systems, nuclear reactors, chemical processing equipment, lubricating grooves, coating and industrial processes, float glass manufacturing and solidification processes [1–4]. Mohammad and Viskanta [5] made an analysis on mixed convection in a square lid-driven cavity with a stable vertical temperature gradient. They found that the multi-cellular flow regimes and transition to three-dimensional flow occur in the cavity. Mixed convection in a square cavity in the presence of the magnetic field was studied by Chamkha [6]. The result showed that the average Nusselt number for aiding flow increases on increasing the Grashof number with a fixed value of the Reynolds number. Jeng and Tzeng [7] numerically investigated the heat transfer in a lid-driven cavity filled with water-saturated aluminum foam. Their result shows that the local heat transfer rate increases with the Reynolds number for constant Grashof number. Ghasemi and Aminossadati [8] studied mixed convection in a lid-driven triangular enclosure. They concluded that the downward sliding wall provides higher heat transfer rate compared to the upward sliding wall. Hydro-magnetic mixed convection heat transfer in a lid-driven cavity heated from below was numerically studied by Nasrin and Parvin [9]. They observed that the variation in the Reynolds number affects significantly the flow and thermal activities. Numerical simulation of mixed convection heat and mass transfer in a right triangular enclosure was investigated by Ching et al. [10]. They showed that multiple cells are formed for higher Richardson numbers when the lid moves downward.

Convection in a square cavity with partially heated walls has been a central topic due to its applications such as electronic equipment cooling, solar energy collectors and nuclear reactor [11]. Natural convection in an enclosure with partial heating and cooling was studied by Yucel and Turkoglu [12]. They observed that the mean Nusselt number decreases on increasing the heater size for a given cooler size. Oztop [13] made a numerical simulation on combined convection in a partially heated lid-driven enclosure. He found that the location of the heater is the most effective parameter on combined convection flow. The effect of location and size of the heater on mixed convection in a lid-driven cavity was investigated by Sivakumar et al. [14]. They observed that the heat transfer rate is enhanced on reducing the heater size along the left wall. Sankar and Do [15] performed a numerical investigation on the effect of discrete heating on free convection heat transfer in a vertical cylindrical annulus. They exhibited that the maximum heat transfer takes place at the bottom heater. Numerical study on natural convection in a rectangular porous enclosure with five different locations of heating and cooling zones was investigated by Bhuvaneswari et al. [16]. They found that the heat transfer rate is increased on increasing the Grashof number.

The effects due to the inclination of the cavity play an important role in the cooling of electronic equipments especially inside the laptop computers. These types of computers work at different inclination angles on humans' hands and in different environmental conditions. Natural convection in an inclined enclosure was numerically studied by Rasoul and Prinos [17]. They observed that the mean heat transfer

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Nomenclature

c_p	specific heat (J/(kg.K))
g	gravitational acceleration (m/s^2)
Gr	Grashof number (−)
k	thermal conductivity (W/(m·K))
L	cavity length (m)
L_H	heater length (m)
Nu	local Nusselt number (−)
$\overline{Nu}$	average Nusselt number (−)
p	pressure (N/m^2)
Pr	Prandtl number (−)
Re	Reynolds number (−)
Ri	Richardson number (−)
T	dimensionless temperature
t	time (s)
u, v	velocity components (m/s)
U, V	dimensionless velocities
U_0	lid velocity (m/s)
x, y	Cartesian coordinates (m)
X, Y	dimensionless coordinates

Greek symbols

α	thermal diffusivity (m^2/s)
β	coefficient of thermal expansion (K^{-1})
ε_1	center of the heater from bottom (m)
ζ	dimensionless vorticity
θ	temperature (K)
μ	dynamic viscosity (Pa.s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
τ	dimensionless time
ψ	stream function (m^2/s)
Ψ	dimensionless stream function
Ω	vorticity (1/s)

Subscripts

0	reference state
c	cold wall
H	heater

rate increases on increasing the Rayleigh number for all the cavity inclination angles. Cianfrini et al. [18] numerically studied natural convection in a tilted square enclosure with two adjacent walls heated and the two opposite walls cooled. They revealed that the heat transfer across the enclosure occurs as pure conduction for inclination angle 225°. Sharif [19] numerically studied the combined convection in an inclined cavity. It is observed that the average Nusselt number is increased with cavity inclination. Alinia et al. [20] made a numerical study on mixed convection in an inclined cavity. They found that inclination angle is more pronounced at high Richardson numbers due to domination of natural convection.

In most of the studies found in the literature, main attention has been given in detail to the problems of the convective flow in either the vertical cavities or the tilted cavities with isothermal boundaries. To the best knowledge of the authors, since the problem of mixed convection in an inclined cavity subjected to different lengths and locations of the heater has not yet been dealt in the literature, we investigate the effects of the different sizes and locations of the heater on mixed convection in an inclined square cavity.

2. Mathematical formulation

Fig. 1(a) shows the physical model of the present study schematically. The system is considered to be unsteady, laminar, incompressible mixed convective flow and heat transfer in a two dimensional square cavity of length L filled with air. The top horizontal wall (lid) of the cavity moves rightwards with a constant speed U_0. The right wall is maintained at a lower temperature T_c while the left wall is either fully or partially heated to higher temperature T_H such that $T_H > T_c$. Three different sizes L_H of the heater and three different locations of it are used along the left wall. The remaining portions of the left wall, top and bottom walls of the cavity are taken to be insulated. The cavity inclination angle (γ) varies from 0° to 90°. The gravity acts in the negative y-direction and changes according to the inclination of the enclosure. The thermo-physical properties of the fluid are taken to be constant, except the density in the buoyancy term. The density varies linearly with temperature as $\rho = \rho_0[1 - \beta(T - T_c)]$, where β is being the coefficient of thermal expansion and subscript 0 denotes the reference state. The Boussinesq approximation is valid. In this investigation, the viscous dissipation is neglected. With above mentioned assumptions, the unsteady governing equations can be written as

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho_0}\frac{\partial p}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) + g\beta(T - T_c)\sin\gamma \quad (2)$$

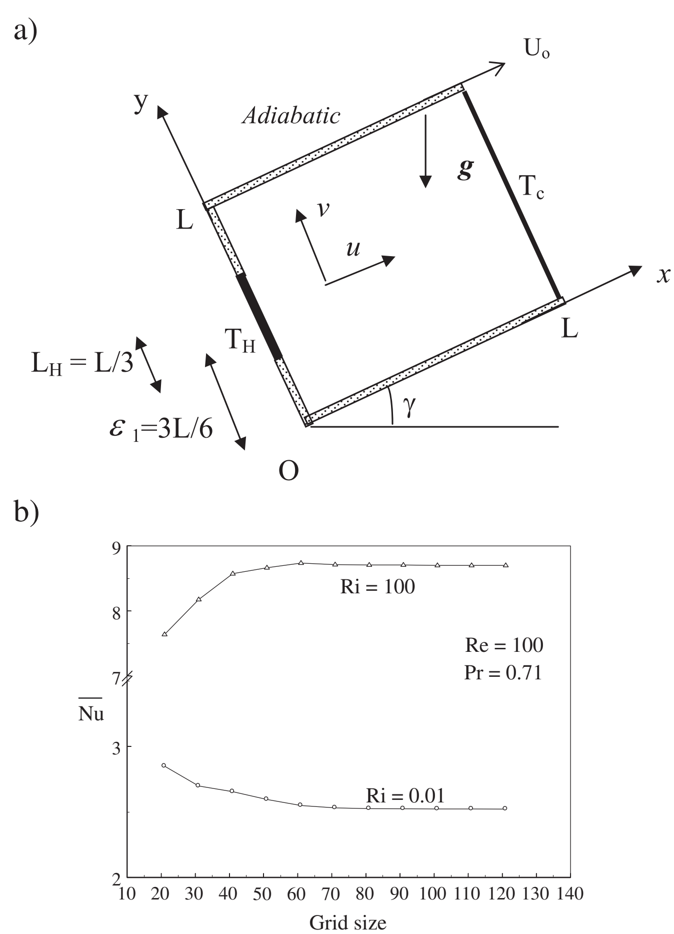

Fig. 1. (a) Schematic diagram of physical configuration and coordinate systems. (b) Grid independence test.