Mixed Convection of a Bingham Fluid in Differentially Heated Square Enclosure with Partitions

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Mixed Convection of a Bingham Fluid in Differentially Heated Square Enclosure with Partitions[1]

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Abstract—This paper reports a two-dimensional numerical investigation of mixed convection inside a lid-driven square enclosure, completely filled with a non-Newtonian fluid obeying the Bingham model, having two rectangular adiabatic partitions mounted in different dispositions. Due to the problem' complexity, the latter is solved using finite volume method when the SIMPLER algorithm is adopted for the pressure-velocity coupling. In order to investigate the yield stress effects on flow field and heat transport, we maintain the Richardson number (Gr/Re^2) as 0.01, 1 and 10, respectively, which generates a good simulation of forced, mixed and natural convection dominated flow. The Prandtl and Grashof numbers are fixed at 50 and 10^4, respectively, while the Bingham number covers the range from 0 to 30. The validity of the computing code was ascertained by comparing our results with the numerical ones already available in the literature and that for both cases: Newtonian and non-Newtonian fluid. The phenomenon is analyzed through the streamlines, the isotherms and the Nusselt numbers with special attention to the partitions' arrangement and its size. It is found that all parameters related to the geometrical dimensions of the partitions play a crucial role on the temperature distribution, flow field and heat transfer enhancement. For all values of the Bingham number, the mean Nusselt number is found as an increasing function of the decrease Richardson number. Thus, the heat transfer improves.

Keywords: Bingham fluid, mixed convection, square enclosure, central located partitions, offset partitions, finite volume method

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INTRODUCTION

Mixed convection in a partitioned cavity has been widely investigated using numerical simulations and experiments owing to its interest and its importance in industrial applications; such as cooling of nuclear reactors, electronic devices, aeronautics, chemical processing equipment, coating and building engineering. As critical studies, extensive investigations have been made for mixed convection heat transfer in a lid-driven rectangular cavity [1–6]. Since the thermal coefficient improvement is critical, the non-dimensional parameters such Richardson and Prandtl numbers are carefully studied in specific ranges.

For instance, Sun et al. [7] indicated numerically that the flow and heat transfer can be controlled in a lid-driven square cavity using a short triangular conductive fin attached to the motionless walls. The vertical walls of the cavity were differentially heated; meanwhile, the bottom and the top lid one were kept adiabatic.

Bhave et al. [8] studied natural convection in a square enclosure including an adiabatic centred solid block. They assumed that the temperature difference driving the convection comes from the side walls. The heat transfer was examined using a wide range of Rayleigh number for three particular values of the Prandtl number. Summarizing their results, helpful correlations predicting the optimal block size for higher heat transfer have been established.

In terms of experimental study, Dogan and Sivrioglu [9] pointed the effects of fin spacing, fin height and heat flow magnitude on the mixed convection, when rectangular fin arrays heated from below in a horizontal channel. The study proved that the latter depends on the fin height and spacing, when the optimum fin spacing depends on the experimental governing parameters' values.

Frederick [10] explained that even the location of conducting fin attached vertically to the hot wall, play a crucial role on the natural convection enhancement, which confined inside a cubical air filled enclosure.

[1] The article is published in the original.

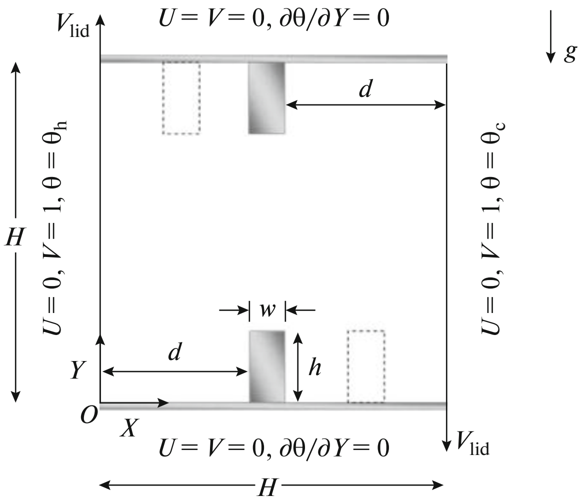

Fig. 1. Schematic of the simulation domain with its boundary conditions.

The thermal conductivity ratio was taken as 10, 1000 and 7000 for the vertical location case and 7000 for the horizontal one. For various values of the Rayleigh number, he proved that the enhancement caused by the presence of the vertical partition exceeds that one obtained with the horizontal location, by the fact that heat transfer is being better with 40% comparing both locations.

Regarding the case of non-Newtonian fluid, a limited number of works were archived [11–14]. Vradis et al. [15] studied the flow field and heat transfer of a Bingham fluid in the entrance region of a circular pipe. Meanwhile, Sayed-Ahmed [16] presented great results in studying the heat transfer of a laminar Harschel–Bulkley fluid type in a square duct.

From the above literature survey, there was no examination of mixed convection of non-Newtonian fluids into a partitioned square enclosure. Thus, the main purpose of our work is to examine the combined natural-forced convection flow field and temperature distribution in such configuration. The latter represents an industrial shear mixture, where we train the fluid on both sides to generate a mixture.

PROBLEM STATEMENT

The physical model of the problem along with its boundary condition is shown in Fig. 1. It is assumed that the temperature difference driving the convection comes from the side vertical moving walls, when both the bottom wall and the top one are kept adiabatic. The flow and heat transfer are two-dimensional, steady and laminar. The size and the location of the adiabatic partitions are varied, using centrally located and offset one. The Bingham fluid properties are assumed to be constant except for the density in the buoyancy term which follows the Boussinesq approximation.

MATHEMATICAL FORMULATION

The governing two-dimensional mass, momentum and energy conservation equations of our problem are given in non-dimensional form as follows:

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

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{\mathrm{Re}}\left[\frac{\partial}{\partial X}\left(2\eta_{\mathrm{eff}}\frac{\partial U}{\partial X}\right) + \frac{\partial}{\partial Y}\left(\eta_{\mathrm{eff}}\left(\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X}\right)\right)\right], \tag{2}$$

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{\mathrm{Re}}\left[\frac{\partial}{\partial X}\left(\eta_{\mathrm{eff}}\left(\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X}\right)\right) + \frac{\partial}{\partial Y}\left(2\eta_{\mathrm{eff}}\frac{\partial V}{\partial Y}\right) + \frac{\partial}{\partial Y}\left(2\eta_{\mathrm{eff}}\frac{\partial V}{\partial Y}\right)\right] + \frac{\mathrm{Gr}}{\mathrm{Re}^2}\theta, \tag{3}$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{\mathrm{Pr\,Re}}\left[\frac{\partial^2\theta}{\partial X^2} + \frac{\partial^2\theta}{\partial Y^2}\right]. \tag{4}$$

Equations (1)–(4) were normalized using the following dimensionless variables:

$$X = \frac{x}{H};\ Y = \frac{y}{H};\ U = \frac{u}{V_{\mathrm{lid}}};\ V = \frac{v}{V_{\mathrm{lid}}};\ P = \frac{p}{\rho V_{\mathrm{lid}}^2};\ \theta = \frac{2T - (T_{\mathrm{h}} + T_{\mathrm{c}})}{2\,(T_{\mathrm{h}} - T_{\mathrm{c}})}. \tag{5}$$

The following Bingham fluid constitutive equation, given by Papanastasiou [17] for an ideal Bingham fluid, is considered to be suitable as shown in Eq. (6) [18, 19]:

$$\eta_{\mathrm{eff}} = 1 + \frac{\mathrm{Bn}}{\sqrt{\frac{1}{2}I_{\dot\gamma}}}\left\{1 - \exp\left(-M\sqrt{\frac{1}{2}I_{\dot\gamma}}\right)\right\}, \tag{6}$$

where M is a reduced parameter representing the exponential growth ($M = mV_{\mathrm{lid}}/H$). Mitsoulis and Zisis [18] as well as Mitsoulis [19] advice to take $m = 1000$.

The heat transfer across the active cavity walls can be quantified using a wall surface averaged Nusselt number based on the cavity length scale (H) which is given as

$$\left|\overline{\mathrm{Nu}}_{\text{left hot wall}}\right| = \left|\overline{\mathrm{Nu}}_{\text{right cold wall}}\right| = \int_0^1 \left.\left(\frac{\partial\theta}{\partial X}\right)\right|_{\text{wall}} \mathrm{d}Y. \tag{7}$$