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Transition of free convection flow between two isothermal vertical plates

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ABSTRACT

Numerical simulations are performed to study the transition of the development of thermal boundary layer of air along isothermal heated plates in a large channel. In particular, the aim is to investigate the effects of the channel width on the transition of the flow under various plate temperatures. Realizable k–ε turbulence model with an enhanced wall function is employed to obtain the numerical simulations of flow and thermal fields in the channel. The channel width is varied from 0.04 m to 0.45 m and the numerical results of the maximum values of flow velocity, turbulent kinetic energy are recorded along the flow to examine the critical distance of the developing flow. Effects on the transition of the two different types of wall boundary conditions, isothermal and adiabatic, applied to the channel are also examined. The results particularly indicate that the flow transition in the isothermal cases takes later than that in the adiabatic cases.

1. Introduction

Over the recent years, more attention has been given to the natural convection as it naturally occurs in environment systems. Moreover, most practical and economical methods for developing a heating or cooling system use natural convection which is induced by buoyance forces with density gradient established thermally.

In 1942 the first experimental work on the buoyancy driven convection flow in a parallel walled vertical channel was done by Elenbaas [1] using air as a test fluid. Results were presented for a set of inclination angles of the plate varying from 0° to 90°, and particular attention was given to the prediction of heat transfer coefficient. A good agreement was obtained between the theoretical and experimental data. One of his key findings was that the solution on the single plate would have to be approached for large plate spacing. Bodia and Osterle [2] later performed the first numerical simulation of buoyancy induced flow developing on a vertical channel. The governing equations were expressed in finite difference form and the walls were treated as isothermal. The air flow velocity and heat transfer coefficient were provided and validated with those of the theoretical and experimental data of Elenbaas [1]. Later, the numerical techniques of Bodia and Osterle [2] were widely used by many researchers to solve the free convection in vertical channel with different boundary and operating conditions, e.g. see the work presented in Miyatake and Fujii [3], Aung et al. [4] and Oosthuzien [5].

Streamwise development of the turbulent free convection flow between two vertical plates was experimentally investigated by Miyamoto et al. [6] and Katoh et al. [7]. The test fluid in this case was also air and different width of the channel was examined. In Miyamoto et al. [6] the channel was installed at various height from floor, e.g. 10, 90, 170 and 465 mm. The experimental data demonstrated that the heat transfer coefficients in the vertical heated plate of over 2 m long are almost independent to the height of the channel when it is positioned 10 mm above the floor. While, in Katoh et al. [7] a bell mouth was installed at the bottom of the channel, and for the heat transfer the comparison between the channel with a bell mouth and without it was done. The results showed that the heat transfer becomes smaller in the two cases.

Two-dimensional numerical simulations of turbulent natural convection in a heated channel were performed by Said et al. [8] and Badr et al. [9]. The governing equations were solved by the finite volume discretisation method by assuming all the thermal properties of the air to be constant, except the density which was solved by the Boussinesq approximation.

In another recent study, thermal efficiency of flow in a vertical chimney (open ended channel) has experimentally been investigated by Burek and Habeb [10]. From the experimental data some correlations have been obtained to calculate the mass flow rate of air as well as temperature. The major finding has been that the

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Nomenclature

b	channel width
C_p	air specific heat capacity
g	gravitational acceleration
Gr	Grashof number = $g\beta(T_P - T_a)y^3/v^2$
h	heat transfer coefficient = $qp/(Tp - Ta)$
k	kinetic energy of turbulence
L	channel height
Nu	average of Nusselt number = hb/κ
n_y, n_x	number of nodes in the y and x directions, respectively
p	pressure
Pr	Prandtl number
q_p	heat flux of the plate = $\int \frac{\partial T}{\partial x}\big\vert_{x=0} dy$
T	temperature
u, v	velocity components in the x and y directions, respectively
x, y	Cartesian coordinates

Greek symbols

β	thermal expansion coefficient
Γ	exchange coefficient for general transport defined as μ/Pr
ρ	density
v	kinematic viscosity
ε	dissipation rate of turbulent kinetic energy
μ	dynamic viscosity coefficient
μ_t	turbulent molecular viscosity
σ_t	turbulent Prandtl number
κ	thermal conductivity

Subscripts

a	air
P	plate
c	critical

mass flow rate is affected by both the channel depth and heat input to the system with the thermal efficiency remaining unaffected by the depth of the channel. Taofeek et al. [11], on the other hand, have used a Particle Image Velocimetry method to record turbulent characteristics of free convection in a channel with anti-symmetric heating. The data has been provided for two values of Rayleigh number, 1×10^8 and 2×10^8. The results indicated that the distributions of the velocity and temperature of the flow in this experimental model of the vertical channel are similar to those in a closed cavity. And, with increasing the Rayleigh number by 50%, the location of peak velocity moves close to the surface wall, which has close agreement with data recorded in a closed cavity.

As seen, all these experimental or numerical studies mainly focused on the investigation of the behaviour and characteristics of free convection flow inside a channel under different boundary and operating conditions. And most of them were carried out either on laminar or turbulent flow and hence neglected transition. But most recently Alzwayi and Paul [12,13] provided numerical results on the flow transition inside a channel where one of the heated plates in the channel was kept isothermal and the other one, placed opposite to it, was adiabatic. However, very little is known about the transition stage in a vertical channel when the two heated plates are kept under an isothermal condition. Moreover, when free convection occurs relatively in a large scale, little reliable information can be obtained from an experimental/laboratory based experiment, and in some cases undertaking various experimental tests e.g. with varying the channel width and operating as well as boundary conditions may be difficult and a costly process. The key objective of this work is therefore to perform a series of numerical examination to investigate the natural convection flow developing between two isothermal plates and then examine the transition of this flow with a particular attention paid to the interaction which will occur between the thermal boundary layers separating from each heated plate. Furthermore, effects on the transition stage occurring between the two-flow conditions of a channel (isothermal and adiabatic) are also investigated.

2. Model geometry

The channel is formed by two vertical plates with length L, and the distance separating them is denoted by b. Both the plates (left and right) are kept at an isothermal condition. The numerical simulations are considered to be two-dimensional, incompressible, Newtonian, free-convection and steady state. The air with a Prandtl number of 0.7 is chosen to be the test fluid. The model geometry along with the Cartesian coordinate system is shown in Fig. 1 which has the same schematic drawing as in Alzwayi and Paul [12].

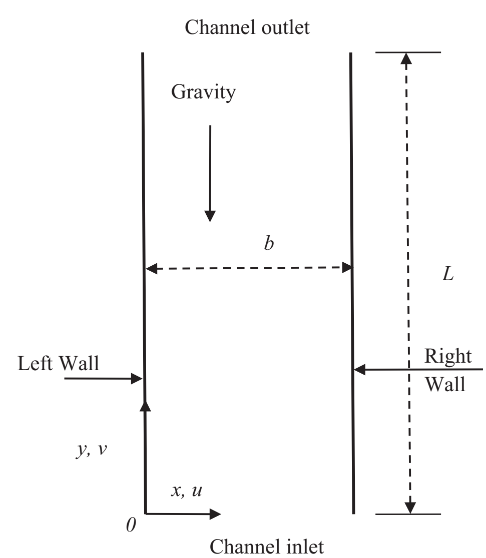

Fig. 1. Schematic of a vertical parallel plate channel.

3. Mathematical formulation

3.1. Governing equation

The conservation equations of mass, momentum and energy for a two-dimensional incompressible fluid flow are, respectively written in the following forms:

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

$$\frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right), \tag{2}$$

$$\frac{\partial(\rho vu)}{\partial x} + \frac{\partial(\rho vv)}{\partial y} = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x}\left(\mu\frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial v}{\partial y}\right) + g(\rho - \rho_0), \tag{3}$$

$$\frac{\partial(\rho uT)}{\partial x} + \frac{\partial(\rho vT)}{\partial y} = \frac{\partial}{\partial x}\left(\Gamma\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma\frac{\partial T}{\partial y}\right). \tag{4}$$