International Journal of Heat and Mass Transfer 68 (2014) 194–202

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Transition of free convection flow inside an inclined parallel walled channel: Effects of inclination angle and width of the channel

Ali S. Alzwayi [1], Manosh C. Paul *

Systems, Power & Energy Research Division, School of Engineering, University of Glasgow, Glasgow G12 8QQ, UK

ARTICLE INFO

Article history:
Received 3 May 2013
Received in revised form 6 September 2013
Accepted 9 September 2013
Available online 8 October 2013

Keywords:
Free convection
Channel inclination
Laminar flow
Transition
Critical distance
Turbulent flow
Heat transfer

ABSTRACT

Transition of free convection flow in an inclined parallel walled channel has been investigated numerically by employing k–ε turbulent model. Particular attention is paid on how the inclination angle and width of the channel affect the transition process of the flow developing in the channel. The upper plate of the channel is heated isothermally and facing downward, while the lower one is kept under the adiabatic condition. The inclination angle of the channel is varied from 0° to 85° with respect to its vertical position while the distance separating the two plates is systematically reduced from 0.45 to 0.06 m. Distributions of velocity, turbulent kinetic energy and local heat flux are presented to examine the critical distance and the results obtained show good agreement with experimental data available in the literature.

1. Introduction

Natural convection in inclined parallel plates has received great attention in the recent years due to a wide range of engineering applications including solar energy systems. Particularly, in modern buildings sloped roofs, multi-glazed windows and skylights are frequently installed, and they are exposed to both free and forced convections. As far as the flow of free convection on a tilted plate is concerned, the first experiment was carried out by Rich [1] in 1953. Average heat transfer rate along the heated plate at various angles up to 40° was measured. The experimental data indicated that the heat transfer drops when the heated plate is moved from its vertical position.

After about three decades, Azevedo and Sparrow [2] performed a comprehensive experimental study of laminar free convection in an inclined isothermal channel. They investigated three heating models: heating from above, heating from below, and symmetric heating, where the channel was closed from both sides. The average of Nusselt number was evaluated in all the experimental conditions in the range 0°–45° from the vertical position, and they reported the data in a global correlation with an average error of ±10%. Later, Manca et al. [3] followed this work and concerned the investigation of the Nusselt number with a uniform heat flux condition.

Onur et al. [4] and Onur and Aktas [5] provided experimental data of the effect of the inclination angle and width on developing heat transfer between inclined parallel plates with different heating conditions. In particular, Onur et al. [4] considered the lower plate isothermally heated and the upper plate insulated, while in [5] the upper plate was heated isothermally whereas the lower was insulated. The channel inclination was 0°, 30° and 45° with a range of Rayleigh number between 3×10^7 and 9×10^7. The measurement was carried out for different temperatures of air and plate and the results indicated that both the inclination and width influence the heat transfer rate.

In the late 90s and more recently, numerous numerical papers on free convection have been published. Numerical investigation of the free convention in inclined channel carried out by Baskaya et al. [6] seems relevant to the work presented in this paper. However, they particularly focused on the laminar convection flow. Recently, two-dimensional turbulent natural convection flow inside an inclined parallel walled channel has been studied numerically by Said et al. [7,8], where both the upper and lower plates were isothermal. A k–ε turbulent model (FLUENT code) was employed in the simulation and the results satisfactory agreed with [9–11]. Their results further indicated that the average of Nusselt number is reduced with an increase in the plate angle. However, the process of transition that occurs from laminar to turbulence was not considered in the study. In fact, all the papers cited above either considered

* Corresponding author. Tel.: +44 (0)141 330 8466; fax: +44 (0)141 330 4343.
E-mail address: Manosh.Paul@glasgow.ac.uk (M.C. Paul).

[1] On study leave from the Department of Mechanical Engineering, Faculty of Engineering and Technology, Sabha University, P.O. Box 53808, Brak, Libya.

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.09.015

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 = $g\beta(T_p - T_a)$
k	kinetic energy of turbulence
L	channel height
$\dot{m}$	mass flow rate
Nu	average of Nusselt number = hb/k
n_y, n_x	number of nodes in the y and x directions respectively
p, p_o	static and ambient pressure respectively
Pr	Prandtl number
q_p	heat flux of the plate = $\int \frac{\partial T}{\partial x}\big\|_{x=0}\, dy$
Ra	Rayleigh number = $Gr\,Pr$
T	temperature
u, v	velocity components in the x and y directions respectively
x, y	Cartesian coordinates

Greek symbols

θ	angle of inclination with respect to the vertical direction
β	thermal expansion coefficient
Γ	exchange coefficient for general transport defined as μ/Pr
p	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
e	experimental
n	numerical
c	critical
in	inlet

laminar or turbulent flow and neglected the transition of developing free convection in an inclined channel. Most recently, Alzwayi and Paul [12] have investigated numerically the transition of thermal boundary layer in a vertical parallel plate channel with an affect of its width and temperature.

In the context of a heated surface facing upward, several researchers [13–17] studied the transition stage under different kinds of fluid and heating conditions. Recently, Paul et al. [18] also looked into the effects of angle on the thermal boundary layer stability. However, very little information is available on the transition of flow on a heated surface facing downward. Hassan and Mohamed [15] tried to investigate the transition of free convection flow for both the downward and upward facing heated surfaces but they were unable to get the transition point for the heated surface facing downward due to the short length of the plate. Even they could not get the transition stage in the vertical case because of the same reason. Whereas, Lloyd and Sparrow [14] mainly focused on the transition on the heated plate facing upwards, but they provided experimental data of the critical Grashof number only at 10° for the heated plate facing downward.

Motivation of this study is therefore on the transition of free convection flow developing in a channel with the heated plate facing downward. Attention is restricted to the flow of air ($Pr = 0.7$) with potential application linked to the convection occurring under a photovoltaic panel (e.g. see [19]). Effects of the width and angle of the channel on the transition stage will also be examined under various air and plate temperatures by using the local values of velocity, turbulent kinetic energy and heat transfer.

2. Geometry and numerical modelling

The channel is formed by two inclined plates each with length L, and the distance between the plates is denoted by b. The wall on the top side is isothermal and heated below, while the other one is adiabatic. The numerical simulation is considered to be two-dimensional free convection and steady state. Air is chosen to be the test fluid. The model geometry along with the Cartesian coordinate system is shown in Fig. 1.

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

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

$$\frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right) + g \times \sin\theta(\rho - \rho_0), \tag{2}$$

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

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

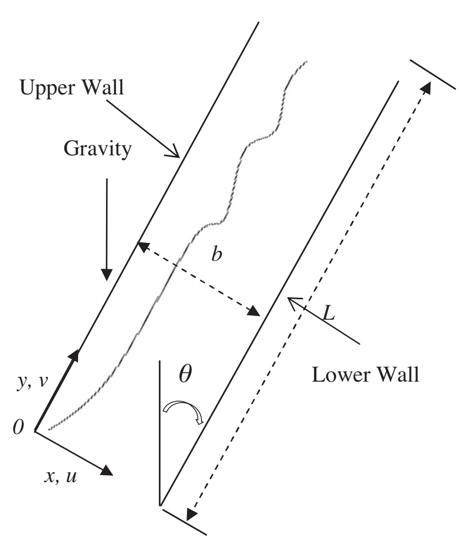

Fig. 1. Flow geometry with coordinate directions.