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A new method of adjustment of inlet boundary for improving heat transfer of mixed convection in a vertical channel☆

Wu-Shung Fu *, Wei-Siang Chao, Pei-Jung Wu, Yu-Ming Lin

Department of Mechanical Engineering, National Chiao Tung University, Hsinchu 30010, Taiwan, ROC

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ABSTRACT

This article is dedicated to a new method for improving heat transfer of mixed convection in a vertical channel from the numerical perspective. An adjustable inlet boundary is proposed in the new method to substitute an inlet that is completely occupied by the amount of fluid provided by forced convection included in mixed convection. According to this new method, the flow reversal occurring in the large Richardson number situation can be eliminated significantly, which results in an improvement in the heat transfer rate of mixed convection. In order to investigate a compressible flow problem, this study focuses on four methods, i.e., Roe scheme, preconditioning, dual time stepping, and the LUSGS (Lower–Upper Symmetric-Gauss–Seidel), to solve the governing equations. Except for the aperture occupied by the mount of fluid provided by forced convection, this study simulates the non-reflecting boundary conditions in other open boundaries. Based on our previous study which has successfully explored a phenomenon of flow reversal, this study intends to enhance the heat transfer rate under this circumstance, and the result has shown a significant improvement, by 29 percentage, comparing with the previous study. Also, a criterion of estimation of the necessity of forced convection included in mixed convection is proposed for enhancing heat transfer rate. Present results and existing results have good agreement.

1. Introduction

The study of mixed convection in a vertical channel has been a focal issue for numerical and experimental studies in a long tradition; it is not only because of its complicated interactions between natural and forced convections, but also because of its wide applications in heat dissipation devices. Traditionally, an inlet of the channel is filled up with the flowing fluids provided by forced convection, which enforces other flowing fluids induced by natural convection to alter their flowing behaviors which include the fluids flowing into and out of the channel. This is the main reason to form structure of flow reversal. When the magnitude of the Richardson number is large, natural convection ought to be dominant. Due to the formation of the flow reversal, the flowing fluids induced by natural convection just passing through the outlet will flow reversely back into the channel and result in a phenomenon of convective heat transfer. This flow structure easily deteriorates the heat transfer rates between the channel surface and the flowing fluid. Oppositely, when a magnitude of the Richardson number is small, forced convection dominates behaviors of mixed convection means that the heat transfer rate of mixed convection is mainly contributed by forced convection, consequently, the influence of the flow reversal will be slight and negligible.

Many current researches have contributed to the relation between natural and forced convections and the formation of flow reversal of mixed convection, such as: Metais and Eckert [1] conducted an experimental work to draw a diagram indicating a flow relationship between Reynolds and Grashof numbers. In the diagram, regions of natural, forced and mixed convections divided into laminar and turbulent flows were clearly delimited. Ingham et al. [2] investigated phenomena of flow reversal of mixed convection in a two-dimensional constant temperature vertical duct with the Boussinesq assumption. In a range of $-300 \le Gr/Re \le 70$, the flow reversal easily appeared at the larger magnitude of $|Gr/Re|$. Tanaka et al. [3] conducted an experimental work like [1] to draw a diagram of Reynolds via Grashof numbers; the domain of Reynolds number was between 1000 and 5000. Regions of natural, forced and mixed convections were divided into the laminar and turbulent flows. Celata et al. [4] showed the experimental results of the distribution of the buoyancy parameter of Bo in a figure of Reynolds number via Grashof number with water. In the range of $Bo \le 1$, a laminar phenomenon was observed, and the corresponding Nusselt number was slightly smaller than that of pure forced convection. Zghal et al. [5] investigated aiding flow mixed convection in a two-dimensional vertical duct with the Boussinesq assumption. Effects of parameters of length,

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* Corresponding author at: Department of Mechanical Engineering, National Chiao Tung University, 1001 Ta Hsueh Road, Hsinchu 30056, Taiwan, ROC.
E-mail address: wsfu@mail.nctu.edu.tw (W.-S. Fu).

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Nomenclature

a	sound speed [$m \cdot s^{-1}$]
A	area [m^2]
C_p	constant-pressure specific heat [$J \cdot kg^{-1} \cdot K^{-1}$]
C_v	constant-volume specific heat [$J \cdot kg^{-1} \cdot K^{-1}$]
e	internal energy [$J \cdot kg^{-1}$]
g	acceleration of gravity [$m \cdot s^{-2}$]
Gr	Grashof number, Eq. (12) $Gr = \frac{g\rho_0^2(T_h - T_0)l_1^3}{T_0\mu(T_0)^2}$
h	enthalpy [J]
k	thermal diffusivity [$W \cdot m^{-1} \cdot K^{-1}$]
k_0	surrounding thermal diffusivity [$W \cdot m^{-1} \cdot K^{-1}$]
l_0	length of the vertical square channel [m]
l_1	width of vertical square channel [m]
l_2	width of inlet occupied by forced convection [m]
l_3	width of gap [m]
l_4	width of surrounding walls [m]
$\overline{M}$	time averaged total mass flow rate, Eq. (16) $\overline{M} = \frac{\iiint \rho u dY dZ dt^*}{\rho_0 u_0 L_2^2 t^*}$
Nu_x	local Nusselt number, Eq. (13) $Nu_x = \frac{l_1}{k_0(T_h - T_0)}[k(T)\frac{\partial T}{\partial z}]$
$\overline{Nu}$	area averaged Nusselt number, Eq. (17) $\overline{Nu} = \frac{1}{A}\iint_A Nudxdy$
$(\overline{Nu})_{t^*}$	time and area averaged Nusselt number, Eq. (18) $(\overline{Nu})_{t^*} = \frac{1}{t^*}\int_{cycle} \overline{Nu}dt^*$
Pr	Prandtl number, Eq. (12) $Pr = \frac{C_p\mu}{k}$
R	gas constant [$J \cdot kg^{-1} \cdot K^{-1}$]
Ra	Rayleigh number, Eq. (12) $Ra = Pr \cdot Gr = Pr \cdot \frac{g\rho_0^2(T_h - T_0)l_1^3}{T_0\mu(T_0)^2}$
Re	Reynolds number, Eq. (10) $Re = \frac{\rho_0 u_f l_2}{\mu_0}$
Ri	Richardson number, Eq. (11) $Ri = \frac{Gr}{Re^2}$
T	Kelvin temperature [K]
T_0	surrounding temperature [K]
T_h	heat temperature of channel wall [K]
Δt	time interval [s]
t	time [s]
t^*	dimensionless time, Eq. (9)
u	velocity in x-direction [$m \cdot s^{-1}$]
u_f	velocity of forced convection at the inlet [$m \cdot s^{-1}$]
u_{in}	inlet velocity [$m \cdot s^{-1}$]
u_{max}	the maximum velocity [$m \cdot s^{-1}$]
U	dimensionless velocity in x-direction
U_{max}	the maximum dimensionless velocity
v	velocity in y-direction [$m \cdot s^{-1}$]
V	dimensionless velocity in y-direction
w	velocity in z-direction [$m \cdot s^{-1}$]
W	dimensionless velocity in z-direction
x, y, z	cartesian coordinates [m]
X, Y, Z	dimensionless cartesian coordinates

Greek symbols

α	ratio of $2l_3$ to l_1, Eq. (9)
γ	specific heat ratio
θ	dimensionless of temperature, Eq. (9) $\theta = \frac{T - T_0}{T_h - T_0}$
u	absolute viscosity [$kg \cdot m^{-1} \cdot s^{-1}$]
ρ	density [$kg \cdot m^{-3}$]
ρ_0	surrounding density [$kg \cdot m^{-3}$]
τ	artificial time [s]
υ	kinematic viscosity [$m^{-2} \cdot s^{-1}$]

Reynolds number and Richardson number on Nusselt number were examined. Appearance of flow reversal was mainly determined by a relationship of Peclet and Richardson numbers. Behzadmehr et al. [6] investigated aiding flow mixed convection with the Boussinesq assumption under conditions of uniform heat flux and low Reynolds numbers in a vertical circular duct. Relationships of Grashof and Nusselt numbers were yielded to distinguish the laminar and turbulent regions. The results showed that the regions of Re = 1000, $8 \times 10^5 < Gr < 5 \times 10^7$ and Re = 1500, $2 \times 10^6 < Gr < 10^8$ were included in the turbulent region. Boulama and Galanis [7] studied aiding flow mixed convection in two-dimensional vertical parallel plates with a condition of fully developed flow at the outlet, and the analytical solutions were dependent on the parameters which combined the effects of thermal and solutal buoyancy. The results revealed that buoyancy effects significantly improved heat and momentum transfer rates near the heated walls. Desrayaud and Lauriat [8] investigated aiding flow mixed convection of a two-dimensional vertical duct with a high wall temperature. The results showed that when the magnitude of Gr/Re^2 was larger than 1, phenomena of flow reversal were observed. Under a condition of a constant Grashof number, the flow reversal was more difficult to be found when the Reynolds number was large. Barletta [9] investigated viscous dissipation effect of mixed convection in a two-dimensional vertical duct. The phenomenon of flow reversal in opposing flow mixed convection was more apparent than that in aiding flow mixed convection under the Boussinesq assumption. Barletta [10–11] adopted an analytical method and the Boussinesq assumption to investigate mixed convection in a rectangular cross-section duct with a fully developed flow condition in the z axis. Thermal conditions of walls were composed of different combinations of high and low temperatures and constant heat flux. Phenomena of flow reversal were examined under different shapes of rectangular cross section. Nguyen et al. [12] investigated transient mixed convection in a high heat flux circular duct with the Boussinesq assumption numerically. Boundary conditions at the inlet and outlet were a uniform velocity and a fully developed flow, respectively. In an opposite situation, the flow reversal appeared near the outlet region at $Gr = 3 \times 10^5$, and in an aiding situation the flow reversal appeared near the center of the axis at $Gr = 10^6$. Yang et al. [13] adopted numerical and analytical methods to investigate the flow reversal and heat transfer in a vertical duct. The analytical method could predict the penetration depth of the reversed flow. Comparing the experimental data, the present results had good agreement. Oulaid et al. [14] investigated the flow reversal in combined laminar mixed convection with phase change and the working fluid was humid air. The results showed that the flow reversal was induced by buoyancy forces for high air temperatures and mass fractions at the channel entrance, and heat transfer associated with phase change was sometimes more significant than sensible heat transfer. Fu et al. [15] firstly adopted the non-reflecting boundary on the outlet to investigate the formation process of unsteady flow reversal. That the formation of flow reversal was mainly caused by the mass flow rate induced by natural convection being much larger than that provided by forced convection was clarified. The larger the Richardson number was, the more drastic flow reversal was observed. Results revealed different phenomena between pure natural convection and mixed convection under the same thermal and geometric conditions. From the above literature, the flow reversal seems to play a villain to reduce heat transfer rates. Based upon this point, a method, which is able to extinguish the flow reversal, is expected to be proposed for improvement of heat transfer rate of mixed convection and examination of necessity of forced convection included in mixed convection.

Therefore, the aim of the study proposes an effective method, which adopts an adjustable inlet boundary instead of a fixed inlet boundary traditionally used, to destroy the structure of the flow reversal and enhance heat transfer rates of mixed convection further. The adjustable inlet is composed of three parts of an inlet duct, surrounding walls and a gap. The inlet duct, of which the size is assigned, is smaller than the aperture of the channel and also filled with flowing fluids provided by forced convection. The surrounding walls are around the inlet duct with the same width. The width is adjustable based on the demand of natural convection included in mixed convection. According to the