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Thermo-dynamic irreversibility induced by natural convection in square enclosure with inner cylinder. Part-II: Effect of vertical position of inner cylinder

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A B S T R A C T

Two-dimensional numerical simulations were conducted to investigate the natural convection heat transfer induced by the temperature difference between cold walls of the square enclosure and a hot inner circular cylinder in the Prandtl number range of $10^{-2} \leqslant Pr \leqslant 10^{2}$ and Rayleigh number range of $10^{3} \leqslant Ra \leqslant 10^{6}$. In this paper as a sequent research of Mun et al. (2015), the additional geometrical configuration of the system, which is the variation in the vertical position of the inner cylinder, was considered. The change in the structure of convection cells and corresponding heat transfer characteristics were analyzed, which are induced by the variation in the vertical position of the inner cylinder located in the enclosure. And the characteristics of the entropy generation due to heat transfer and fluid friction associated with the flow and thermal structures were addressed in this study.

1. Introduction

Buoyancy-induced flow in an enclosure attracts academic attention due to a variety of its applications including solar collector receivers, heat exchangers, nuclear reactors and electronic packaging. Many researchers have investigated parameters that affect the natural convection characteristics in an enclosure and reported that the characteristics of the natural convection in an enclosure is influenced by the geometry and position of an inner body, the configuration of the enclosure and thermo-physical properties of working fluids [1–23].

The accompanying study presented in [24] undertook two-dimensional numerical study on the natural convection characteristics in the square enclosure having an inner circular cylinder in the Rayleigh number range of $10^{3} \leqslant Ra \leqslant 10^{6}$ and the Prandtl number range of $10^{-2} \leqslant Pr \leqslant 10^{2}$, and the natural convection flow is driven by the temperature difference between a hot cylinder surface and cold walls of the enclosure. In the accompanying study [24], authors have focused on the effect of the variation in the tilted angle of the enclosure ($0^\circ \leqslant \gamma \leqslant 45^\circ$) on the natural convection characteristics and the production of the thermo-dynamic irreversibility induced by the natural convection was addressed through the analysis on the local entropy generation due to heat transfer and fluid friction. In addition, the heat transfer characteristics on the cylinder surface and walls of the enclosure associated with the structure of the convection cell formed in the enclosure were properly analyzed. As a summary of the accompanying study [24], the transition of the flow regime between the steady state to the unsteady one occurs from $Ra = 10^5$ and $Pr = 0.1$ except cases in the tilted angle range of $15^\circ \leqslant \gamma \leqslant 30^\circ$. As reported previously, the state of the system is directly dependent on the thermal boundary condition and fluid property. Also, the state of the system is influenced by the geometrical configuration of the system, which means that the system at $Ra = 10^5$ and $Pr = 0.1$ shows the quasi-state between the steady state and the unsteady one depending on the geometrical configuration of the system.

In this paper as a sequent research of [24], therefore, authors focus on the additional geometrical configuration of the system, which is the variation in the vertical position of the inner cylinder, in the same ranges of the Rayleigh number and Prandtl number as those considered in [24]. The change in the structure of convection cells and corresponding heat transfer characteristics were carefully analyzed according to the variation in the vertical position of the inner cylinder located in the enclosure. And the characteristics of the entropy generation due to heat transfer and fluid friction associated with the flow and thermal structures were addressed in this study.

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Nomenclature

Be	Bejan number
f_i	momentum forcing
g	gravitational acceleration
k	thermal conductivity
L	length of the enclosure
n	normal direction to the wall
Nu	local Nusselt number
P	dimensionless pressure
Pr	Prandtl number
q	mass source or sink
R	radius of circular cylinder
Ra	Rayleigh number
S	distance along the enclosure
t	dimensionless time
T	temperature
u_i	dimensionless velocity vector
x_i	Cartesian coordinates system

Greek symbols

α	thermal diffusivity
β	thermal expansion coefficient
γ	tilted angle of the enclosure
δ_{i2}	Kronecker delta
ρ	density
μ	dynamic viscosity
ν	kinematic viscosity
φ	angle of circular cylinder
θ	dimensionless temperature

Sub/superscripts

$*$	dimensional value
cyl	cylinder
en	enclosure

Abbreviations

FFI	fluid friction irreversibility
HTI	heat transfer irreversibility

Mathematical symbol

$-$	surface-averaged quantity
$\langle\rangle$	volume-averaged quantity

2. Computational details

2.1. Numerical methods

In this study, the numerical method is exactly the same as those used in the accompanying study [24]. The immersed boundary method was used to capture the virtual boundary of the inner circular cylinder in the Cartesian coordinate system.

Fluid considered in this study is incompressible and Newtonian. Fluid flow is assumed to be laminar in the absence of heat generation, chemical reactions and thermal radiation. And also viscous dissipation in the energy equation has been neglected. Based on these assumptions, the dimensional governing equations of mass, momentum and energy for unsteady incompressible viscous flow and thermal fields are expressed as follows:

$$\frac{\partial u_i^*}{\partial x_i^*} = 0 \quad (1)$$

$$\rho\left[\frac{\partial u_i^*}{\partial t} + u_j^*\frac{\partial u_i^*}{\partial x_j^*}\right] = -\frac{\partial P^*}{\partial x_i^*} + \mu\frac{\partial^2 u_i^*}{\partial x_j^*\partial x_j^*} + \rho g\beta(T - T_c)(\sin\varphi\delta_{i1} + \cos\varphi\delta_{i2}) \quad (2)$$

$$\frac{\partial T}{\partial t} + u_j^*\frac{\partial T}{\partial x_j^*} = \alpha\frac{\partial^2 T}{\partial x_j^*\partial x_j^*} \quad (3)$$

The governing equations to which the immersed boundary method is applied are the continuity, momentum and energy equations in their non-dimensional forms, which are expressed as follows:

$$\frac{\partial u_i}{\partial x_i} - q = 0 \quad (4)$$

$$\frac{\partial u_i}{\partial t} + u_j\frac{\partial u_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + Pr\frac{\partial^2 u_i}{\partial x_j\partial x_j} + RaPr\theta(\sin\varphi\delta_{i1} + \cos\varphi\delta_{i2}) + f_i \quad (5)$$

$$\frac{\partial\theta}{\partial t} + u_j\frac{\partial\theta}{\partial x_j} = \frac{\partial^2\theta}{\partial x_j\partial x_j} + h \quad (6)$$

Here, the dimensionless governing equations (4)–(6) are obtained from the dimensional governing equations (1)–(3) using the dimensionless variables defined as follows:

$$t = \frac{t^*\alpha}{L^2},\quad x_i = \frac{x_i^*}{L},\quad u_i = \frac{u_i^*L}{\alpha},\quad P = \frac{P^*L^2}{\rho\alpha^2},\quad \theta = \frac{T - T_c}{T_h - T_c} \quad (7)$$

In the above equations, ρ, T, and α represent the density, dimensional temperature and thermal diffusivity, respectively. The superscript * in Eq. (7) represents the dimensional variables. x_i is the dimensionless Cartesian coordinate, u_i is the corresponding dimensionless velocity component, t is the dimensionless time, P is the dimensionless pressure and θ is the dimensionless temperature. The above non-dimensionalization results in two dimensionless parameters are $Pr = \nu/\alpha$ and $Ra = g\beta L^3(T_h - T_c)/\nu\alpha$, where ν, g, and β are the kinematic viscosity, gravitational acceleration, and volume expansion coefficient, respectively. The terms of q, f_i, and h in Eqs. (4)–(6) are related to the immersed boundary method. The mass source/sink q in Eq. (4) and momentum forcing f_i in Eq. (5) are imposed on the body surface and inside the body to satisfy the no-slip condition and mass conservation in the cell containing the virtual boundary. In Eq. (6), the heat source/sink h is applied to satisfy the iso-thermal boundary condition at the virtual boundary. The momentum forcing term f_i and heat source/sink term h in Eqs. (5) and (6) were obtained by

$$f_i = \frac{U_i - u_i^n}{\Delta t} + \frac{3}{2}NL(u_i^n) - \frac{1}{2}NL(u_i^{n-1}) + \frac{\partial P^n}{\partial x_i} - \frac{1}{2}\sqrt{\frac{Pr}{Ra}}[DIF(u_i^{n+1}) + DIF(u_i^n)] - \theta^n\delta_{i2} \quad (8)$$

$$h = \frac{\Theta - \theta^n}{\Delta t} + \frac{3}{2}NL(\theta^n) - \frac{1}{2}NL(\theta^{n-1}) - \frac{1}{2\sqrt{RaPr}}[DIF(\theta^{n+1}) + DIF(\theta^n)] - \theta^n\delta_{i2} \quad (9)$$

where U_i and Θ represented the desired velocity and temperature to realize the viscous and thermal boundary conditions on the surface of the immersed object, and superscripts n and $n+1$ in Eq. (9) represent the time levels.

The nonlinear term $NL(\varphi)$ and diffusion term $DIF(\varphi)$ in Eqs. (8) and (9) are defined as: