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Analysis of entropy generation on mixed convection in square enclosures for various horizontal or vertical moving wall(s)☆

Monisha Roy [a], S. Roy [a], Tanmay Basak [b,*]

[a] Department of Mathematics, Indian Institute of Technology Madras, Chennai - 600036, India
[b] Department of Chemical Engineering, Indian Institute of Technology Madras, Chennai - 600036, India

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ABSTRACT

Entropy generation during the mixed convection process have been studied in a square enclosure for various moving horizontal (cases 1a–1d) or vertical wall(s) (cases 2a–2c) where the bottom wall of the cavity is isothermally hot, side walls are cold, and the top wall is adiabatic. Simulations have been performed for Prandtl number $Pr = 0.026$ and 7.2, Reynolds number $Re = 10 - 100$, and Grashof number $Gr = 10^3 - 10^5$. Results show that, in the case of the horizontally moving wall(s) (cases 1a–1d), the overall heat transfer rate ($\overline{Nu_b}$) and total entropy generation (S_{total}) are identical for cases 1a–1d and the cup-mixing temperature (θ_{cup}) is high for case 1b at $Pr = 0.026$, $Re = 100$, and $Gr = 10^5$. Similarly, in the case of the vertically moving wall(s) (cases 2a–2c), $\overline{Nu_b}$ and S_{total} are identical for cases 2a–2c with the maximum θ_{cup} occurring for the case 2a. At $Pr = 7.2$, $Gr = 10^5$, and $Re = 10$, case 1a and case 1c are preferable for horizontally moving wall(s) and either of case 2a–2c is preferable for vertically moving wall(s). At $Pr = 7.2$, $Gr = 10^5$, and $Re = 100$, case 1d may be preferable for the horizontally moving wall(s) and case 2a may be preferable for the vertically moving wall(s).

1. Introduction

Analysis of the fluid flow and heat transfer for the mixed convection in the closed enclosures received considerable attention due to the significant industrial applications such as the cooling of electronic components [1], solar collector [2], room ventilation [3], etc. Substantial research efforts involving the experimental as well as numerical studies have been performed to understand the fluid flow and heat transfer phenomena inside the enclosures for various velocity as well as thermal boundary conditions. A number of earlier studies also focused on the influence of the motion of various walls (horizontal or vertical) for the transport process within the square enclosure [4–8]. However, most of these studies are based on the first law of thermodynamics and only a few studies have been done on the entropy generation analysis.

The recent review [9] on the entropy generation for the convection process shows that the analysis of the entropy generation for the mixed convection process in various geometries involving the ducts or cavities is limited. Yilbas et al. [10] examined the effect of the protruding body aspect ratio on the total entropy generation for the mixed convection in a square cavity. Lioua et al. [11] analyzed the entropy generation during the mixed convection in a three-dimensional cavity for various directions of the motion of the lids. The mixed convection heat transfer and entropy generation in a square cavity with the discrete heat sources at the corners was investigated by Chacon et al. [12]. Mahmoudi and Hooman [13] investigated the effect of a discrete heat source location on the entropy generation in a ventilated cavity filled with the copper–water nanofluid during the mixed convective cooling. However, the role of various moving walls on the entropy generation during the mixed convection in the square cavities with the hot bottom wall, cold side walls, and adiabatic top wall has not yet been addressed in the literature. Therefore, it is necessary to investigate the efficiency of the system based on the minimal entropy generation versus the larger heat transfer rates for various moving wall(s) corresponding to practical applications involving the floating process [14], food processing [15], etc. Overall, the entropy generation analysis in this study will help to improve the efficiency of the system via choosing the proper physical and thermal parameters.

2. Modeling, simulation, and post-processing

2.1. Governing equations, boundary conditions, and numerical simulations

The three-dimensional physical domains are shown in Fig. 1(a) and (c). The two-dimensional computational domains are shown in Fig. 1(b) and (d) based on the semi-infinite approximation along Z direction. The fluid motion and heating patterns are studied for two different cases which are as follows: case 1: horizontal wall(s) are moving either in same or in opposite directions [Fig. 1(b)], and case 2: vertical wall(s) are moving in either same or opposite directions [Fig. 1(c)]. In

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* Corresponding author.
E-mail addresses: monisha.roy97@gmail.com (M. Roy), sjroy@iitm.ac.in (S. Roy), tanmay@iitm.ac.in (T. Basak).

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Nomenclature

g	Acceleration due to gravity (m s^{-2})
Gr	Grashof number
k	Thermal conductivity (W m^{-1} K^{-1})
L	Side of the square cavity (m)
n	Normal vector to the plane
Nu	Nusselt number
$\overline{Nu}$	Average Nusselt number
p	Pressure (Pa)
P	Dimensionless pressure
Pe	Peclet number
Pr	Prandtl number
Re	Reynolds number
Ri	Richardson number
S_ψ	Dimensionless entropy generation due to fluid friction
S_θ	Dimensionless entropy generation due to heat transfer
S_{total}	Dimensionless total entropy generation
T	Temperature of the fluid (K)
T_c	Temperature of cold wall (K)
T_h	Temperature of hot wall (K)
u	x Component of velocity (m s^{-1})
U	x Component of dimensionless velocity
U_0	Characteristic velocity
v	y Component of velocity (m s^{-1})
V	y Component of dimensionless velocity
x	Distance along x coordinate (m)
X	Dimensionless distance along x coordinate
y	Distance along y coordinate (m)
Y	Component of dimensionless velocity

Greek symbols

α	Thermal diffusivity (m^2 s^{-1})
β	Volume expansion coefficient (K^{-1})
γ	Penalty parameter
θ	Dimensionless temperature
ν	Kinematic viscosity (m^2 s^{-1})
ρ	Density (kg m^{-3})
ψ	Dimensionless streamfunction
μ	Dynamic viscosity (kg m^{-1} s^{-1})
Ω	Two dimensional domain

Subscripts

b	Bottom wall
cup	Cup-mixing
i	Local node number
l	Left wall
r	Right wall
s	Side wall

all the cases, the bottom wall is isothermally hot while side walls are isothermally cooled with the adiabatic top surface. The fluid is considered as incompressible and the flow is assumed to be two-dimensional and laminar. The Boussinesq approximation is employed to relate the changes in the density with temperature in the body force term. Based on these assumptions, the governing equations in non-dimensional forms are as follows:

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

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right), \tag{2}$$

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

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{1}{RePr}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right), \tag{4}$$

where

$$X=\frac{x}{L},Y=\frac{y}{L},U=\frac{u}{U_0},V=\frac{v}{U_0},\theta=\frac{T-T_c}{T_h-T_c}$$

$$P=\frac{p}{\rho U_0^2},Pr=\frac{\nu}{\alpha},Re=\frac{U_0L}{\nu},Gr=\frac{g\beta(T_h-T_c)L^3}{\nu^2}. \tag{5}$$

with the following boundary conditions

$$\begin{aligned}
&U(X,Y)=0\ \ or\ -1\ \ or\ \mp 0.5,\ V(X,Y)=0,\ \theta=1\ \forall Y=0,\ 0\le X\le 1\\
&U(X,Y)=0,\ V(X,Y)=0\ \ or\ \ 1\ \ or\ \ 0.5,\ \ \theta=0\ \forall\ X=0,\ \ 0\le Y\le 1\\
&U(X,Y)=0,\ V(X,Y)=0\ \ or\ \ \mp 0.5,\ \ \theta=0\ \forall\ X=1,\ \ 0\le Y\le 1\\
&U(X,Y)=0\ \ or\ \ 1\ \ or\ \ 0.5,\ V(X,Y)=0,\ \ \frac{\partial\theta}{\partial Y}=0\ \forall\ Y=1,\ \ 0\le X\le 1\\
&\theta(0,0)=\theta(1,0)=0.5
\end{aligned} \tag{6}$$

The momentum and energy balance equations [Eqs. (2)–(4)] are solved by using the Galerkin finite element method [16]. The continuity equation [Eq. (1)] has been used as a constraint due to the conservation of mass and this constraint may be used to obtain the pressure distribution. In order to solve Eqs. (2)–(3), we use the penalty finite element method where the pressure P is eliminated by a penalty parameter γ and the incompressibility criteria are given by Eq. (1), which results in

$$P=-\gamma\left(\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}\right). \tag{7}$$

For large values of γ, the continuity equation [Eq. (1)] is automatically satisfied and based on the constraint optimization, $\gamma=10^7$, found to give the optimal solution. Applying Eq. (7) in Eqs. (2) and (3), we get the modified momentum equations. These two modified momentum equations and the energy balance equation [Eq. (4)] associated with boundary conditions [Eq. (6)] are solved by the Galerkin finite element method [16]. As the solution procedure is discussed in an earlier work [17], the detailed description is not included in this article.

2.2. Streamfunction, Nusselt number, and entropy generation

The fluid flow pattern can be visualized via the streamfunction (ψ) and the relationship between the streamfunction (ψ) and velocity components (U and V) is as follows:

$$\frac{\partial^2\psi}{\partial X^2}+\frac{\partial^2\psi}{\partial Y^2}=\frac{\partial U}{\partial Y}-\frac{\partial V}{\partial X}. \tag{8}$$

Here, the positive sign of ψ corresponds to the anticlockwise circulation and the negative sign of ψ corresponds to the clockwise circulation. The no-slip condition is applicable for all the boundaries as there is no cross-flow of the fluid. Therefore, $\psi=0$ is used for all the boundaries. The heat transfer rate in terms of the local Nusselt number (Nu) is defined as

$$Nu=-\frac{\partial\theta}{\partial n}\,, \tag{9}$$