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Mixed convection heat transfer in a ventilated cavity with hot obstacle: Effect of nanofluid and outlet port location[☆]

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A B S T R A C T

In the present study, the lattice Boltzmann method is implemented to investigate the effect of suspension of nanoparticles on mixed convection in a square cavity with inlet and outlet ports and hot obstacle in the center of the cavity. The effect of outlet port location is examined on heat transfer rate then the effect of nanoparticles is inspected for volume fraction of nanoparticles in the range of 0 to 0.03 at the different position of outlet port. The study was carried out for different Richardson numbers ranging from 0.1 to 10. Grashof number is assumed to be constant (10^4) so that the Richardson number changes with Reynolds number. The isothermal boundary condition is assumed for obstacle walls and the cavity walls are adiabatic. The result is presented by isotherms, streamlines, and local and average Nusselt numbers. The maximum heat transfer rate occurs when the outlet port is located at P2 for Ri = 0.1 and P1 for Ri = 1, Ri = 10, respectively. Results show that by adding the nanoparticles to base fluid and increasing the volume concentration of nanoparticles the heat transfer rate is enhanced at different Richardson numbers and outlet port positions. But this phenomenon is not observed at Ri = 10 when the outlet port is located at P1.

1. Introduction

Mixed convection in ventilated cavity is applicable in many industrial transport processes such as, heat exchangers, pollution removal and solar systems. The effect of mixed convection is important in electronic chipset, where natural convection is not able to provide effective cooling [1]. Moallemi and Jang [2] studied the effect of Prandtl and Reynolds numbers on the flow field and thermal characteristics of a laminar mixed convection in a rectangular cavity. Prasad and Josef [3] have reported experimental results on mixed convection heat transfer process in a lid-driven cavity for different Richardson numbers ranging from 0.1 to 1000. They found that the heat transfer within the cavity is independent of Gr/Re^2 over the range study. Mixed convection was studied in a horizontal lid-driven cavity with an undulating base surface by Nasrin [4]. In the study the effect of different physical parameters such as cavity aspect ratio and amplitude of undulating was examined. Saeidi and Khodadadi [5] investigated the effect of the outlet port location, in the ventilated cavity. They showed that by placing the outlet port with one end at three corners, maximum overall Nusselt number of the cavity can be achieved. The effect of obstacle presence on mixed convection in vented cavities has not examined widely by researchers. Recently, a numerical study on the effect of a heated hollow cylinder on mixed convection in a ventilated cavity was investigated by Mamun et al. [6]. They examined the impact of Reynolds number, Richardson number, cylinder diameter and the solid-fluid thermal conductivity ratio in their studies. They presented that the cylinder diameter has significant effect on both the flow and thermal fields but the solid-fluid thermal conductivity ratio has significant effect only on the thermal field.

In recent years, nanofluids have attracted more attention for enhancement of heat transfer in various industrial applications because of remarkable increase in effective thermal conductivity of base fluid. Masuda et al. [7] reported on enhanced thermal conductivity of dispersed ultra-fine (nanosize) particles in liquids. Soon thereafter, Chol [8] was the first to coin the term "nanofluids" for this new class of fluids with superior thermal properties. Khanafer et al. [9] investigated the heat transfer enhancement in a two-dimensional enclosure utilizing nanofluids for a range of Grashof numbers and volume fractions. It was found that the heat transfer across the enclosure increases with the volumetric fraction of the copper nanoparticles in water for different Grashof numbers. Jou and Tzeng [10] did a numerical attempt to simulate the natural convection in a rectangular cavity with two different aspect ratios for just Cu–water nanofluid. They simulated the flow field in aspect ratios where the convection heat transfer is dominant respect to conduction heat transfer, but they didn't report the details of heat transfer enhancement. Talebi et al. [11] presented a numerical study of laminar mixed convection through copper–water nanofluid in square cavity for different Reynolds and Rayleigh numbers. They concluded that the effect of solid concentration increases by the augmentation of Rayleigh number. Mahmoodi et al. [12] presented numerical study of mixed convection flow and temperature fields in a vented square cavity subjected to an external copper–water nanofluid. Their results show

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Nomenclature

c_k	discrete lattice velocity in direction (k)
c_s	speed of sound in lattice scale
c_p	specific heat at constant pressure $kJkg^{-1}K^{-1}$
F_k	external force in direction of lattice velocity
f_k^{eq}	equilibrium distribution
g_y	acceleration due to gravity, (ms^{-2})
H	wide of inlet port (m)
k	thermal conductivity ($W/m.K$)
Nu	Nusselt number
Pr	Prandtl number (ν/α)
Ra	Rayleigh number ($g\beta\Delta TH^3/\alpha\nu$)
T	temperature (K)
Ri	Richardson number
w_k	weighting factor

Greek symbols

β	thermal expansion coefficient ($1/K$)
φ	solid volume fraction
ρ	density (kg/m^3)
τ	lattice relaxation time
Δt	lattice time step

Subscripts

c	cold
f	fluid
h	hot
l	local
m	mean
nf	nanofluid
s	solid
w	wall

that for higher Re and lower Ri numbers the presence of nanoparticles has more effects to increase the heat transfer performance.

Mixed convection heat transfer in a lid-driven cavity was investigated by many researchers with different numerical methods such as finite volume method [13,14], Monte Carlo method [15], finite element method [16] and the lattice Boltzmann method. The lattice Boltzmann method is one of the suitable numerical methods that have been used in the recent years. It was used for simulation of the flow field and thermal characterization in wide ranges of the engineering applications such as natural and mixed convection [17–19], radiation heat transfer [20], laminar and turbulent flow [21], nanofluids [22], etc. Nemati et al. [23] investigated the effect of various nanofluids on mixed convection flows using lattice Boltzmann method. They achieved the effects of solid volume fraction grow stronger sequentially for Al_2O_3, Cuo and Cu. Kefayati et al. [24] studied natural convection in enclosure at different Rayleigh number and aspect ratios. They found that the effect of nanoparticles on heat transfer augments by increment of the enclosure aspect ratio.

By knowledge of authors, in previous studies the effect of nanoparticles on mixed convection in ventilated square cavity with central hot obstacle has not investigated. In the present study, the effect of various volume fractions of copper nanoparticles and outlet port locations on mixed convection flows in a vented cavity with a hot obstacle is investigated numerically. Richardson number changes from 0.1 to 10. The study is carried out for constant Grashof number of 10^4 so that Richardson number changes with Reynolds number. Results are provided as stream lines, isotherms, and local and average Nusselt number plots.

2. Numerical procedure

2.1. The lattice Boltzmann method

The thermal LB model utilizes two distribution functions, f and g, for the flow and temperature fields, respectively. It uses modeling of movement of fluid particles to capture macroscopic fluid quantities such as velocity, pressure and temperature. In this approach the fluid domain is discretized in uniform Cartesian cells. Each cell holds a fixed number of distribution functions, which represent the number of fluid particles moving in these discrete directions. For this work the D2Q9 model has been used. The values of $w_0=4/9$ for $|c_0|=0$ (for the static particle), $w_{1-4}=1/9$ for $|c_{1-4}|=1$ and $w_{5-9}=1/36$ for $|c_{5-9}|=\sqrt{2}$ are assigned in this model. The density and distribution functions i.e. the f and g, are calculated by solving the lattice Boltzmann equation (LBE), which is a special discretization of the kinetic Boltzmann equation. After introducing BGK approximation, the general form of lattice Boltzmann equation with external force can be written as:

for the flow field:

$$f_i(x+c_i\Delta t, t+\Delta t) = f_i(x,t) + \frac{\Delta t}{\tau_v}\left[f_i^{eq}(x,t) - f_i(x,t)\right] + \Delta t c_i F_k \tag{1}$$

for the temperature field:

$$g_i(x+c_i\Delta t, t+\Delta t) = g_i(x,t) + \frac{\Delta t}{\tau_D}\left[g_i^{eq}(x,t) - g_i(x,t)\right] \tag{2}$$

where Δt denotes lattice time step, c_k is the discrete lattice velocity in direction k, F_k is the external force in direction of lattice velocity, τ_v and τ_D denotes the lattice relaxation time for the flow and temperature fields. The kinetic viscosity υ and the thermal diffusivityα, are defined in terms of their respective relaxation times, i.e. $\upsilon=c_s^2(\tau_v-1/2)$ and $\alpha=c_s^2(\tau_D-1/2)$, respectively. Note that the limitation $0.5<\tau$ should be satisfied for both relaxation times to ensure that viscosity and thermal diffusivity are positive. Furthermore, the local equilibrium distribution function determines the type of problem that needs to be solved. It also models the equilibrium distribution functions, which are calculated with Eqs. (3) and (4) for flow and temperature fields respectively.

$$f_i^{eq} = w_i\rho\left[1 + \frac{c_i.u}{c_s^2} + \frac{1}{2}\frac{(c_i.u)^2}{c_s^4} - \frac{1}{2}\frac{u^2}{c_s^2}\right] \tag{3}$$

$$g_i^{eq} = w_i T\left[1 + \frac{c_i.u}{c_s^2}\right] \tag{4}$$

where w_k is a weighting factor, and ρ is the lattice fluid density. In order to incorporate buoyancy force in the model, the force term in Eq. (1) needs to be calculated as below in vertical direction (y):

$$F = 3w_i g_y \beta\theta \tag{5}$$

For natural convection the Boussinesq approximation is applied and radiation heat transfer is negligible. To ensure that the code works in near incompressible regime, the characteristic velocity of the flow for both natural $\left(V_{natural}\equiv\sqrt{\beta g_y \Delta TH}\right)$ and force$(V_{force}\equiv \mathrm{Re}\upsilon/H)$ regimes must be small compared with the fluid speed of sound. In the present study, the characteristic velocity was selected as 0.1 of sound speed.