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Numerical simulation of turbulent plume spread in ceiling vented enclosure

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ABSTRACT

The buoyancy-induced turbulent flow generated by a heat source in a square enclosure with single and multiple ceiling vents has been studied numerically. A two-dimensional, turbulent natural convection flow is investigated in stream function and vorticity formulation approach. A low Reynolds number k–ϵ turbulence model of Lam Bremhorst is used to solve the governing equations using high accuracy compact finite difference schemes. Results are reported for different Grashof numbers varied from 10^8 to 10^{10}. The effects of heat source location, vent location and multiple vents on flow characteristics in enclosure are presented. The heat transfer characteristics, ambient entrainment flow rate and the oscillatory nature of the penetrative and recirculating flow inside the vented enclosure are reported. The results indicate significant change in the flow behavior by varying the location of heat source and vent for fixed Grashof number. The effect of entrainment of ambient air is significant with increase in Grashof number. The volume flow rates through the two ceiling vents showed a significant variation depending on the location of vent. Present results are matching very well with the experimental and numerical results available from the literature.

1. Introduction

The buoyancy driven flow patterns inside vented enclosures have received considerable attention because of their wide applications in thermal design of buildings, solar energy receivers with open cavities, fire propagation in rooms and cooling of electronic devices. The mass and energy transfer through openings in buildings by natural ventilation has significant impact on the indoor air quality and in the case of fire scenario the flow of air and combustion products across vents governs the growth and spread of fires in buildings. Moreover the transport phenomena due to fire in buildings are usually buoyancy dominated and hence can be modeled as buoyancy-induced turbulent flow in partial enclosures. The study of buoyancy induced flows in enclosures have received considerable attention in the literature [1,2]. The transport phenomena due to fire in rooms and other compartments are reported in [3,4]. One of the earliest studies on flow through opening was reported by Rockett [5] and emphasized that the inflow of ambient air into the compartment was mainly dependent on the shape of the opening rather than on the temperatures. A study on buoyancy driven flow through the vertical opening in enclosure was reported in [6,7]. The flow through doors and windows has been studied in detail and incorporated mathematical models to predict the growth of room fires [8–10]. Smoke and hot gas movement inside vertical open enclosures have been studied experimentally by Mercier and Jaluria [11]. Smoke and hot gases are injected into the enclosure from lower opening and the resulting downstream flow and temperature fields are reported. They have observed the wall plume generated along the enclosure wall from the inlet to the outlet.

The horizontal vents are openings in ceilings, floors, and stairwells and could be a broken window at the top of atrium roof. However, not much work has been done on the counter current exchange flow through the horizontal openings. Atkinson [12] experimentally studied the smoke movement driven by fire under a ceiling and found rapidly rotating smoke rolls near the ceiling. Tan and Jaluria [13] experimentally studied the mass flow rate through a horizontal vent in an enclosure due to pressure and density differences. They found that in the absence of a pressure difference but with heavier fluid overlying lighter fluid, a bidirectional flow arises across the vent due to buoyancy effects. The critical pressure has been identified at which transition from bidirectional to unidirectional flow occurs across the vent. Recently Venkatasubbaiah and Jaluria [14] numerically investigated the enclosure fire with single and multiple horizontal vents. Their studies are limited within the laminar flow regime and the critical Grashof number is 10^6, above this flow becomes chaotic nature in the enclosure.

Most of the numerical works presented in partial open enclosures were in the laminar regime and the works related to turbulent flows are quite limited. Turbulent penetrative and recirculating flow in a compartment with vertical opening has been

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Nomenclature

H	Height of the enclosure
D	Width of the ceiling vent
T_s	Heat source temperature
T_∞	Initial fluid temperature
g	Gravitational force per unit mass
β	Coefficient of thermal expansion
α	Thermal diffusivity
ν	Kinematic viscosity
ν_t	Turbulent eddy viscosity
Pr	Prandtl number
Pr_t	Turbulent Prandtl number
k	Turbulent Kinetic energy
ϵ	Turbulent dissipation rate
ψ	Stream function
ω	Vorticity
Ψ	Dimensionless stream function
Ω	Dimensionless vorticity
θ	Dimensionless temperature
U, V	Dimensionless velocity components along x and y directions
τ	Dimensionless time
K	Dimensionless turbulent kinetic energy
E	Dimensionless turbulent dissipation rate
Gr	Grashof number
Ra	Rayleigh number
Nu	Nusselt number

studied numerically by Abib and Jaluria [15]. They have used the k–ϵ low Reynolds number model of Lam Bremhorst and found that the flow field has multicellular patterns i.e. strong main convective cell at the bottom and weak counter cell at the top. Buoyancy-induced turbulent flows have been studied numerically using low Reynolds number turbulence model of Lam Bremhorst and reported in [16,17]. They found that this model has higher capability for predicting turbulent quantities reasonably well in regions near the wall and away from the walls. The turbulent natural convection flow in a rectangular enclosure with finite thickness walls has been studied numerically using k–ϵ turbulence model by Kuznetsov and Sheremet [18]. Turbulent natural convection flow with surface radiation in a square enclosure has been studied numerically and reported in [19,20]. Flow field model of enclosure fires using CFD with different turbulence models were reported in [21–24].

Ventilation performance predictions in buildings using different models has been discussed by Qingyan [25] and it is reported that CFD model has better predicting capabilities compared to analytical methods. The capability and accuracy of different RANS turbulence models to predict particle deposition in vertical and horizontal turbulent channel flows are reported by Gao et al. [26]. Numerical simulation on thermal plumes with different k–ϵ turbulence models has been studied and reported by Chow and Li [27]. It is found that a more complicated form of k–ϵ models might not give better results.

Very few numerical studies on thermal plume propagation inside enclosures with ceiling vents were reported earlier. The present investigation is an extension of the previous work on ceiling vented enclosure [14], where studies are limited within the laminar flow regime. The present study investigates the turbulent effects and the effects of heat source and vent locations are studied in detail.

Here, the buoyancy driven turbulent thermal plume flow characteristics in a square enclosure with ceiling vent is studied numerically. The stream function and vorticity formulation with k–ϵ low Reynolds number turbulence model of Lam Bremhorst is used to solve the governing equations. Results are reported with different Grashof numbers. The effects of heat source location, vent location and multiple vents on the flow characteristics in enclosure are reported.

2. Governing equations and boundary conditions

The buoyancy induced turbulent flow in a square enclosure with ceiling vent is shown in Fig. 1. There is a finite-size heat source at constant temperature T_s located at the bottom wall of enclosure and ceiling vent of width D above to the ambient media at temperature T_∞. The flow phenomena is modeled as two-dimensional unsteady state incompressible buoyancy-induced turbulent flow in a long partial square enclosure. The governing equations for turbulent natural convection flows is described mathematically by the Reynolds averaged Navier–Stokes equations (RANS), including the time averaged energy equation for the mean temperature field that drives the flow by buoyancy force. The buoyancy term is modeled by Boussinesq approximation that treats density as a constant value in all equations, except for the buoyancy term in the momentum equation. Turbulence is modeled with a low Reynolds number k–ϵ model of Lam Bremhorst including the contribution of buoyancy force in the turbulent kinetic energy generation and dissipation. The RANS equations for the velocity and temperature fields are as follows.

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$\begin{aligned}\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y} &= -\frac{1}{\rho}\frac{\partial p}{\partial x}+\nu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right]\\ &\quad+2\frac{\partial}{\partial x}\left[\nu_t\,\frac{\partial u}{\partial x}\right]\\ &\quad+\frac{\partial}{\partial y}\left[\nu_t\,\frac{\partial u}{\partial y}\right]+\frac{\partial}{\partial y}\left[\nu_t\,\frac{\partial v}{\partial x}\right]\end{aligned} \tag{2}$$

$$\begin{aligned}\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y} &= -\frac{1}{\rho}\frac{\partial p}{\partial y}+\nu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right]\\ &\quad+\frac{\partial}{\partial x}\left[\nu_t\,\frac{\partial u}{\partial y}\right]+\frac{\partial}{\partial x}\left[\nu_t\,\frac{\partial v}{\partial x}\right]\\ &\quad+2\frac{\partial}{\partial y}\left[\nu_t\,\frac{\partial v}{\partial y}\right]+g\beta(T-T_\infty)\end{aligned} \tag{3}$$

$$\begin{aligned}\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y} &= \frac{\partial}{\partial x}\left[\left(\alpha+\frac{\nu_t}{\mathrm{Pr}_t}\right)\frac{\partial T}{\partial x}\right]\\ &\quad+\frac{\partial}{\partial y}\left[\left(\alpha+\frac{\nu_t}{\mathrm{Pr}_t}\right)\frac{\partial T}{\partial y}\right]\end{aligned} \tag{4}$$

$$\begin{aligned}&\frac{\partial k}{\partial t}+u\frac{\partial k}{\partial x}+v\frac{\partial k}{\partial y}=\frac{\partial}{\partial x}\left[\left(\nu+\frac{\nu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]\\ &\quad+\frac{\partial}{\partial y}\left[\left(\nu+\frac{\nu_t}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]-\frac{g\beta\nu_t}{\mathrm{Pr}_t}\frac{\partial T}{\partial y}-\epsilon\\ &\quad+\nu_t\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2\right]\end{aligned} \tag{5}$$

$$\begin{aligned}&\frac{\partial \epsilon}{\partial t}+u\frac{\partial \epsilon}{\partial x}+v\frac{\partial \epsilon}{\partial y}=\frac{\partial}{\partial x}\left[\left(\nu+\frac{\nu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x}\right]\\ &\quad+\frac{\partial}{\partial y}\left[\left(\nu+\frac{\nu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial y}\right]-C_{2\epsilon}f_2\frac{\epsilon^2}{k}\\ &\quad+C_{1\epsilon}f_1\left[\nu_t\left\{2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2\right.\right.\end{aligned}$$