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Conjugate laminar natural convection and surface radiation in enclosures: Effects of protrusion shape and position☆

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ABSTRACT

A 2-D numerical analysis has been carried out for conjugate heat transfer of laminar natural convection and surface radiation in square enclosure with broad protrusions. The air filled vertical enclosure has two horizontal adiabatic walls and two differentially heated isothermal walls. Investigations are carried out for Rayleigh number in the range of 10^3 to 10^6 and surface emissivity values 0 to 1. Protrusions of different shapes and at different positions are considered in this study. The governing equations of fluid flow and heat transfer with surface radiation are solved using finite volume method and SIMPLE algorithm. The convective and radiative Nusselt numbers are plotted against different key parameters. It has been observed that protrusion affects significantly the overall heat transfer and also modifies the temperature and flow fields in the enclosure. Surface radiation interaction leads to formation of secondary cell between the cold isothermal wall and protrusion. The intensity of vortex decreases gradually as protrusion moves from the hot wall to cold isothermal wall.

1. Introduction

Laminar natural convection with and without surface radiation in a square enclosure has been widely studied, experimentally as well as numerically due to its various engineering applications like solar energy collectors, heat exchangers, insulation panels and devices, electronic cooling, building design etc. Buoyancy driven natural convection is a strong function of temperature dependent density gradient and gravitational force. It is well known that surface radiation causes changes in convective heat transfer and flow field and it also enhances total heat transfer. Many studies have been reported for laminar natural convection with and without surface radiation. The fundamental study for natural convection in an enclosure was carried out by de Vahl Davis [1] for various Ra values. The square cavity problem was numerically analyzed and it has been considered as a benchmark solution for laminar natural convection in a square cavity. Frederick [2] numerically studied natural convection heat transfer in differentially heated cavity for various aspect ratios, to find out the aspect ratio for which the overall Nusselt number was maximum. Numerical study of laminar natural convection in square enclosure with baffles was carried out by Kalidasan et al. [3]. The effect of obstacles on the heat transfer is investigated for Ra range of 10^3 to 10^5. Significant influence of baffles has been noticed on natural convection flow. Mobedi [4] numerically studied the natural convection in air-filled square enclosure with steady heat source for different thermal boundary conditions for Ra range of 10^3 to10^7. Markatos and Pericleous [5] numerically studied the laminar and turbulent natural convection in a square cavity with differentially heated side walls for $\mathrm{Ra} = 10^3$ to 10^{16}. They proposed a correlation for Nu in terms of Ra up to 10^{16}. Acharya and Jetli [6–7] studied natural convection heat transfer in a partially divided square box. Analysis has been carried out for three different lengths and positions of the partition. It was found that, at high Ra, flow detaches from the cold wall and thermal stratification develops in between cold wall and partition. Vivek et al. [8] studied conjugate effect of natural convection and surface radiation in square and shallow cavities with different tilt angles ($\theta = -90°$ to $+90°$) and emissivity values varying from 0 to 0.9. The convective Nu was found to be maximum at an angle of $\theta = +15°$ and is independent of presence or absence of radiation.

The above literature survey shows that, partition or divider with shape and position parameters in a square cavity affects the overall Nusselt number and flow field. Also studies have been carried out for insulated as well as conductive baffles/partitions. However, the effect of surface radiation on convective and total Nusselt number in differentially heated enclosure with protrusion attached to wall is scarce. Therefore, the purpose of this study is to analyze the effect of protrusion with its shape and position parameters on total Nusselt number and flow field structure under the influence of surface radiation. Investigations are carried out for air filled enclosure with square and triangular shaped protrusions positioned at P(1) to P(4) on top wall and P(5) to P(8) on bottom wall with an interval distance of 0.25 L as shown in Fig. 1. The length/width of the enclosure and protrusion i.e., square and triangular are L and 0.25 L respectively.

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Nomenclature

B	radiosity, W/m^2
F_{i-j}	view factor from element i to j
g	acceleration due to gravity, 9.81 m/s^2
h	heat transfer coefficient, W/m^2 K
k	thermal conductivity of the fluid, W/m K
Nu	average Nusselt number, $q''L/k\Delta T$
Nu_r	radiative Nusselt number, $q_r''L/k\Delta T$
P(n)	protrusion at position n in square cavity, where n = 1 to 8
Pr	Prandtl number
q"	average convective heat flux, W/m^2
q_r"	average radiative heat flux, W/m^2
Ra	Rayleigh number, β g $(T_h - T_c)$ $L^3/(\nu\alpha)$
T	temperature, K
u, v	horizontal and vertical velocity components, m/s
x, y	horizontal and vertical coordinates, m

Greek symbols

α	thermal diffusivity, m^2/s
δ	Kronecker delta
β	coefficient of volumetric expansion, K^{-1}
ε	emissivity of the enclosure surfaces
μ	dynamic viscosity, Ns/m^2
ρ	density of air (kg/m^3)
θ	normalized air temperature, $\{(T - T_c) / (T_h - T_c)\}$
ψ	normalized stream function, $\psi/(\rho\alpha L)$

Subscript

h	hot
c	cold

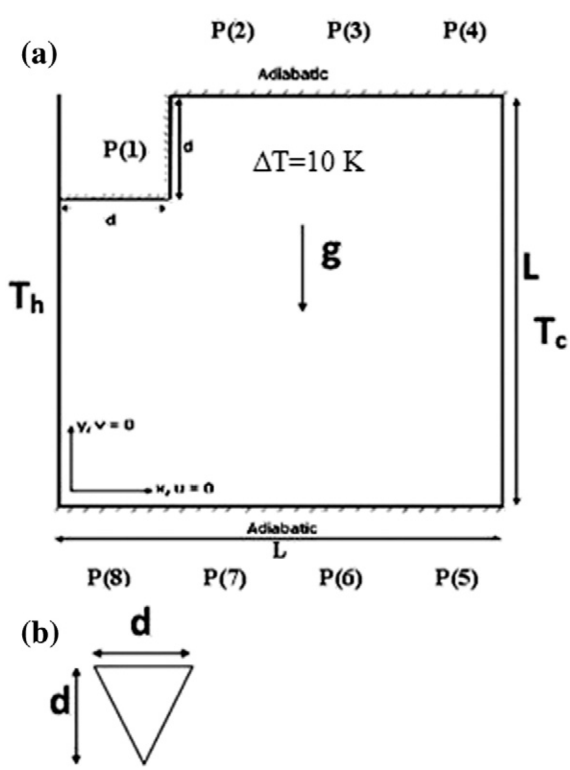

Fig. 1. Schematic of the square enclosure with (a) square protrusion positioned at P(1) and (b) triangular protrusion.

2. Governing equations

2.1. Natural convection fluid flow

Focus of the present study is on conjugate natural convection and surface radiation in broad protrusions placed on horizontal walls in an enclosure filled with air (Pr = 0.7). Fig. 1 depicts the physical model of the problem analyzed. Investigations are carried out for Rayleigh number ranging from 10^3 to 10^6. Vertical walls of the enclosure are considered as isothermal at different temperatures. For buoyancy simplification Boussinesq approximation is applied. The set of conservation equations of mass, momentum and energy is (Bejan, [9]):

Continuity

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

x-Momentum

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right] \tag{2}$$

y-Momentum

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right]+\rho\beta g\Delta T \tag{3}$$

Energy

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=k\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right] \tag{4}$$

2.2. Coupling between natural convection and surface radiation

Due to enclosed geometry, multiple radiation reflections take place from all the surfaces. For the calculation of wall temperatures from the radiation model, local values of heat transfer coefficients and fluid temperature are required from the flow calculations. The heat transfer coefficient h is evaluated as $k/\Delta s$, where k is the thermal conductivity of the fluid adjacent to the wall and Δs is the distance of the first fluid node from the wall surface. It has been assumed that the temperature variation across the wall in the thickness direction is negligible. The surfaces of the walls are assumed to be gray, opaque and diffuse with constant emissivity. Therefore, generalized energy balance equation for any wall has been used and it can be derived as given below (Siegel and Howell [10]):

$$h_1(T_1-T_w)+h_2(T_2-T_w)+\varepsilon_1 I_1+\varepsilon_2 I_2=(\varepsilon_1+\varepsilon_2)\sigma T_w^4 \tag{5}$$

where T_1 and T_2 are local values of fluid temperature, T_w is wall temperature, and I is the irradiation falling on the wall surface. Subscripts 2 and 1 refer to conditions on inner side and outer side of the enclosure. For the isothermal hot wall, h_1 is set to a large value with $T_1 = T_{h,}$ and $\varepsilon_1 = 0$. The irradiation is related to radiosity (B),

$$B_i=\varepsilon_i\sigma T_{wi}^4+(1-\varepsilon_i)I_i \tag{6}$$

where subscript i is the index of the elemental segment forming the enclosure. There are N (equal to $N_x \times N_y$) elemental segments in the enclosure, where N_x and N_y are the number of elements in the x and y directions. Substituting for I in terms of B and using the inverse relationship,

$$I_i=\sum_{j=1}^{N}B_jF_{i-j} \tag{7}$$

and

$$\sum_{j=1}^{N}\left[\frac{\delta_{i,j}-(1-\varepsilon_i)F_{i-j}}{\varepsilon_i}\right]B_j=\sigma T_i^4 \tag{8}$$