Alexandria Engineering Journal (2015) 54, 917–927

HOSTED BY

ELSEVIER

Alexandria University

Alexandria Engineering Journal

www.elsevier.com/locate/aej
www.sciencedirect.com

ORIGINAL ARTICLE

Heat transfer in inclined air rectangular cavities with two localized heat sources

CrossMark

Samy M. Elsherbiny *, **Osama I. Ismail**

Mechanical Engineering Department, Faculty of Engineering, Alexandria University, Alexandria, Egypt

Received 3 May 2014; revised 26 March 2015; accepted 28 July 2015
Available online 14 September 2015

KEYWORDS

Natural convection;
Heat sources;
Rectangular cavities;
Numerical methods;
Laminar convection

Abstract The present work investigates numerically the effects of cavities' aspect ratio and tilt angle on laminar natural convection of air in inclined rectangular cavities with two localized heat sources. A mathematical model was constructed where the conservation equations governing the mass, momentum and thermal energy together with their boundary conditions were solved. The calculation grid is investigated to determine the best grid spacing, number of iterations, and other parameters which affect the accuracy of the solutions. The numerical method and computer program were tested for pure conduction and convection with full heating ($\varepsilon = 1$) to assure validity and accuracy of the numerical method.

The investigation used rectangular enclosures with position ratios of the heaters, $B_1 = 0.25$, $B_2 = 0.75$, size ratio, $\varepsilon = 0.25$, and covered Rayleigh numbers based on scale length, s/A ranging from 10^3 to 10^6. The tilt angle from the horizontal was changed from $\Phi = 0°$ to 180°, and the aspect ratio was taken as $A = 1$, 5, and 10. The results are presented graphically in the form of streamlines and isotherm contour plots. The heat transfer characteristics, and average Nusselt numbers were also presented. A correlation for Nu is also given.

1. Introduction

Natural convection in enclosures has been extensively studied in the past both analytically and experimentally. Chu et al. [1] reported an extensive survey on natural convection from a discrete heater in an enclosure. They examined the effect of heater location in the enclosure with $A = 0.4$–5, $Pr = 0.72$, $\Phi = 90°$, and for a range of Rayleigh number, Ra_H up to 10^5. They found that the Nusselt number, Nu_H was proportional to Ra_H for any location of the discrete heat source. Turner and Flack [2,3] have experimentally examined the heat transfer in geometry similar to that used by Chu et al. [1] for Grashof numbers, Gr_H up to 9×10^6. They obtained the same form of correlation as $Nu_H = C1 \cdot Gr_H^{c2}$, where $C1$ and $C2$ are functions of size ratio, ε. Yovanovich [4] reported an expression for the thermal constrictive resistance, r_c of a discrete heat source on a rectangular solid region with the heat sink on the opposite side and other sides are insulated. The expression is given as $r_c = (1/\pi\, k) \ln [(1/\sin(\pi\varepsilon/2) \cos \pi (B–0.5)]$.

Elsherbiny et al. [5] determined experimentally the effect of heater location using three heaters on the hot wall. They measured Nu for each heater and found that Nu for upper heater was decreased up to a certain Ra and then increased.

* Corresponding author.
E-mail address: samymelsherbiny@yahoo.com (S.M. Elsherbiny).

Peer review under responsibility of Faculty of Engineering, Alexandria University.

http://dx.doi.org/10.1016/j.aej.2015.07.013
1110-0168 © 2015 Faculty of Engineering, Alexandria University. Production and hosting by Elsevier B.V.

Nomenclature

A	aspect ratio of enclosure, $A = H/L$
b_1	distance from $x = 0$ to the center of the lower heater, m
b_2	distance from $x = 0$ to the center of the upper heater, m
B_1	position ratio of the lower heater, $B_1 = b_1/H$
B_2	position ratio of the upper heater, $B_2 = b_2/H$
C_p	specific heat at constant pressure, J/(kg K)
g	gravitational acceleration, m/s^2
Gr	Grashof number, $\mathrm{Gr} = g\beta\,(T_h - T_c)\dot{s}(s/A)^3/v^2$
H	width of enclosure, m
h	average heat transfer coefficient, W/m^2 K
h_x	local heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
L	height of enclosure, m
Nu	average Nusselt number, $h(s/A)/k$
Nu_x	local Nusselt number, $h_x(s/A)/k$
p_d	dynamic pressure, N/m^2
P_d	non-dimensional dynamic pressure, $P_d = p_d(s/A)^2/\rho\alpha^2$
Pr	Prandtl number, $\mathrm{Pr} = \mu\, C_p/k$
r_c	constriction resistance, K/W
Ra	Rayleigh number based on (s/A), $\mathrm{Ra} = g\beta(T_h - T_c)\cdot(s/A)^3/v\alpha$
s	length of heat source, m
T	local fluid temperature, K
T_c	temperature of cold surface, K
T_h	temperature of discrete heat source, K
ΔT	temperature difference, $\Delta T = T_h - T_c$, K
u	dimensional velocity component in x direction, m/s
U	non-dimensional velocity component in X direction, $U = u(s/A)/\alpha$
v	dimensional velocity component in y direction, m/s
V	non-dimensional velocity component in Y direction, $V = v(s/A)/\alpha$
x	dimensional coordinate, m
X	non-dimensional coordinate, $X = x/(s/A)$
y	dimensional coordinate, m
Y	non-dimensional coordinate, $Y = y/(s/A)$

Greek symbols

α	thermal diffusivity, $k/\rho C_p$, m^2/s
β	coefficient of volumetric thermal expansion, K^{-1}
ρ	local density, kg/m^3
ρ_c	cold density, kg/m^3
μ	dynamic viscosity, kg/m s
v	kinematic viscosity, $v = \mu/\rho$, m^2/s
Φ	angular coordinate, rad
Θ	dimensionless temperature, $\Theta = (T - T_c)/(T_h - T_c)$
ε	size ratio, s/H

Markatos and Pericleous [6] have experimentally examined the heat transfer in a square enclosure ($A = 1$), full contact heated wall ($\varepsilon = 1$), Pr = 0.71, $\Phi = 90°$, $B = 0.50$, and the range of Ra from 10^3 to 10^{12}. They obtained the velocity distribution and a correlation of Nusselt numbers as a function of Ra_H as $\mathrm{Nu}_H = C1.\,(\mathrm{Ra}_H)^{C2}$, where $C1$ and $C2$ depend on Ra_H. Keyhani et al. [7] have experimentally studied the heat transfer in an enclosure filled with ethylene glycol. The hot wall consisted of 11 discrete isoflux heaters where $A = 16.5$, Pr = 150, and the local modified Rayleigh number was in the range of 9.3×10^{11} to 1.9×10^{12}. They correlated the local Nusselt number as $\mathrm{Nu}_x = 1.009\ \mathrm{Ra}_x^{0.1805}$, where "$x$" is the local height, measured from bottom of cavity to mid-height of the heated section. Chadwick et al. [8] have experimentally examined the heat transfer in a rectangular enclosure with an isoflux heating mode with $A = 5$, Pr = 0.71, $\varepsilon = 0.133$, $\Phi = 90°$, and Gr^* ($\mathrm{Gr}^* = g\beta s^4/kv^2$) ranged from 10^4 to 5×10^5. They obtained the value of average Nusselt number as a function of Gr^* based on heater length. The average Nusselt number was obtained as $\mathrm{Nu} = \mathrm{C1}.\,(\mathrm{Gr}^*)^{C2}$, where $C1$ and $C2$ are functions of ε. Also, they obtained the value of local Nusselt number as a function of $\mathrm{Gr}_x{}^*$ and the local modified Grashof number based on distance from leading edge of heater ($\mathrm{Gr}_x{}^* = g\beta x^4/kv^2$) and "$x$" is the distance from the leading edge of the heater to any point at heater surface. The local Nusselt number, Nu_x was correlated as $\mathrm{Nu}_x = C1\dot{s}(\mathrm{Gr}_x{}^*)^{C2}$, where $C1$ and $C2$ are functions of B.

Ho and Chang [9] studied numerically and experimentally the influence of aspect ratio on heat transfer in an enclosure which has 4 isoflux heaters where A changed from 1 to 10, Pr = 0.71, $\Phi = 90°$ and a range of $\mathrm{Ra}_H{}^*$ from 10^4 to 10^6. They obtained the value of Nusselt number, Nu_H as $\mathrm{Nu}_H = C1\dot{s}\,(\mathrm{Ra}_H{}^*)^{C2}.\ A^{C3}$, where $C1$, $C2$ and $C3$ are constants depending on heater arrangement. Heindel et al. [10,11] have experimentally studied the heat transfer in a rectangular enclosure. The heat wall consisted of 3×3 array of heaters where $\Phi = 90°$, A ranged from 2.5 to 7.5, Pr ranged from 5 to 25, and Ra_L ranged from 10^5 to 10^8. They obtained the value of Nu for each row of heaters as $\mathrm{Nu} = C\dot{s}\mathrm{Ra}^{0.25}$ where C is a function of row arrangement.

Ahmed and Yovanovich [12] studied numerically the influence of discrete heat source location on natural convection heat transfer in a range of Ra^*, based on scale length, s/A from 0 to 10^6, Pr = 0.72, $A = 1$, $B = 0.5$ and $0 \leqslant \varepsilon \leqslant 1.0$. They obtained analytical correlations for this problem for either isothermal heat source or isoflux heat source.

Al-Bahi et al. [13] studied numerically the effect of heater location on local and average heat transfer rates in an enclosure which has a single isoflux heater where $A = 1$, Pr = 0.71, $\Phi = 90°$, $\varepsilon = 0.125$, Ra^* ranges from 10^3 to 10^6 and $B = 0.25$, 0.50 and 0.75. They obtained isotherms, streamlines and temperature distribution characteristics at high and low Ra^*. Al-Bahi et al. [14] studied numerically the effect of tilt angle from horizontal on local and average heat transfer rates in an enclosure which has a single isoflux heater where $A = 5$,