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Thermal management of heating element in a ventilated enclosure☆

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ABSTRACT

Thermal management in a ventilated enclosure undergoing mixed convection is investigated by dividing the entire heating element into multiple equal segments and by positioning them appropriately on vertical side walls, namely at bottom, middle or top location. The performance analyses of segmental heating and whole heating are conducted for different Richardson number (0.01–100) and Reynolds number (50–200). Nine positional configurations of bi-segmental heating reveal the possibility of significant enhancement in heat transfer. The optimal locations of heater segments for maximum heat transfer depend upon Re and Ri. The study also indicates the increasing trend of heat transfer enhancement with more number of heater segments.

1. Introduction

The study of mixed convection receives a considerable attention since long past due to its dependency on flow geometry and boundary condition, and its applicability in different areas like cooling of electronic and electrical equipments, room cooling, air conditioning, drying, grains and food processing, solar heating, nuclear reactor, combustion chambers. Earlier, rectangular enclosures (or cavities) were used by researchers [1–7] for studying mixed convection, where the external flows (inflow and outflow) were provided directly through side openings [1–5] or through additional ports [6–9]. The isothermal heat source of fixed length was considered in [2,7,10] for a differentially heated [1,2] or partially heated [4,8] enclosure. The iso-flux heat source of fixed length was also investigated in [4,5,11–15] under mixed convection. The study was extended to address the case of discrete heating using two or more numbers of heat sources [3]. In other class of works, different obstructions were embedded within enclosures in the form of adiabatic baffle [18], conducting baffle [19] or partition [14,20], conducting block(s) [16,17] and adiabatic block(s) [21]. The effect of aspect ratio of the enclosure by keeping the length and position of heater fixed, was studied in [22] whereas, the effects of position of ports or vents [3], location of the same heater on different side of enclosure [7] and thermal condition from constant heating [4] to non-uniform [23] or periodic heating [24] are also available in the open literatures. The analysis of heat transfer is the primary focus of all these studies, which is generally investigated in terms of average Nusselt number by changing the flow governing parameters like, Reynolds number, Richardson number and Prandtl number.

The designing of a thermal system requires insight knowledge of the flow fields and heat transfer for proper management of the distribution of heating load in a given geometry. The information related to heat transfer enhancement is also very important to cater the ever-increasing demand on higher performance or size reduction of a system or device. In this context, the present work is formulated to distribute the entire heating load in a positive way so that the resulting heat transfer can be increased significantly utilizing the same enclosure geometry and external fluid flow. The investigation is made in terms of average Nusselt number, heat transfer enhancement, isotherms and streamlines.

2. Physical description and mathematical formulation

The details of ventilated enclosure of this work as depicted in Fig. 1 consists of a rectangular enclosure of height 1.2 L and width L, and two ventilation ducts of 0.2 $L \times$ 0.2 L. The inlet and outlet ducts, and top and bottom walls of the enclosure are taken adiabatic. Whereas, the side walls are utilized for the placement of heating element(s) assumed to have a fixed temperature T_h. Air (Pr = 0.71) is used as a working fluid. The temperature of surrounding air where the enclosure is located is taken constant as T_a. The overall length of all heating element(s) is L (Fig. 1a) and it could be accommodated as a whole heater either on the left wall or on the right wall (as shown in Fig. 1c) of the enclosure. On the other hand, the entire length of the heating element(s) if divided into two equal halves as shown in Fig. 1b, they could be placed on either side of the enclosure at different locations namely bottom, middle or top portion of side walls as shown in Fig. 1d. It yields nine possible combinations which are named (for brevity) using the positions of left and right heater segments as bottom–bottom (BB), bottom-middle (BM),

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Nomenclature

g	acceleration due to gravity
Gr	Grashof number
k	thermal conductivity
L	length of heating element/length scale
n	number of heater segments
Nu	Nusselt number
q	dimensionless heat transfer
p	pressure
p_a	ambient pressure
P	dimensionless pressure
Pr	Prandtl number
Re	Reynolds number
Ri	Richardson number
T	temperature
T_a	ambient temperature
T_h	heating element temperature
u, v	velocity components
u_i	velocity at inlet port
U, V	dimensionless velocity components
x, y	Cartesian coordinates
X, Y	dimensionless coordinates

Greek symbols

α	thermal diffusivity
β	thermal expansion coefficient
ψ	dimensionless stream function
θ	dimensionless temperature
ε	dimensionless heater segment length
ν	kinematic viscosity
ρ	density

Subscripts

avg	average
LH	left heater
RH	right heater
SH	segmented heaters
SL	left segment
SR	right segment

bottom–top (BT), middle-bottom (MB), middle–middle (MM), middle-top (MT), top–bottom (TB), top-middle (TM) and top–top (TT) (segmental/segmented) heater.

Heat transfers from the whole heaters (Fig. 1c) are to be compared to that of bi-segmented heaters (Fig. 1d) to assess the improvement in performance through the division. More number of divisions (up to 10) of whole heating element (of length L) is investigated afterwards, after identifying the positional impact of bi-segmental heater on heat transfer.

For the purpose of thermal analysis, the length of whole heater and the distance between two side walls are taken same as L. The flow is assumed to be two-dimensional Cartesian, steady, incompressible, Newtonian and laminar, and governed by Boussinesq approximation. Insignificant viscous dissipation in energy equation is neglected [25]. The numerical simulations are carried out in non-dimensional form. So, corresponding governing equations for continuity, momentum and energy balances are considered in dimensionless form as:

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

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

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

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

Where, the scaled variables and dimensionless quantities are defined by

$$X=\frac{x}{L},\quad Y=\frac{y}{L},\quad U=\frac{u}{u_\mathrm{i}},\quad V=\frac{v}{u_\mathrm{i}},\quad P=\frac{p-p_a}{\rho u_\mathrm{i}^2},\quad \theta=\frac{T-T_\mathrm{a}}{T_\mathrm{h}-T_\mathrm{a}}$$

$$\mathrm{Re}=\frac{u_\mathrm{i}L}{\nu},\quad \mathrm{Pr}=\frac{\nu}{\alpha},\quad \mathrm{Gr}=\frac{g\beta(T_\mathrm{h}-T_\mathrm{a})L^3}{\nu^2}.$$

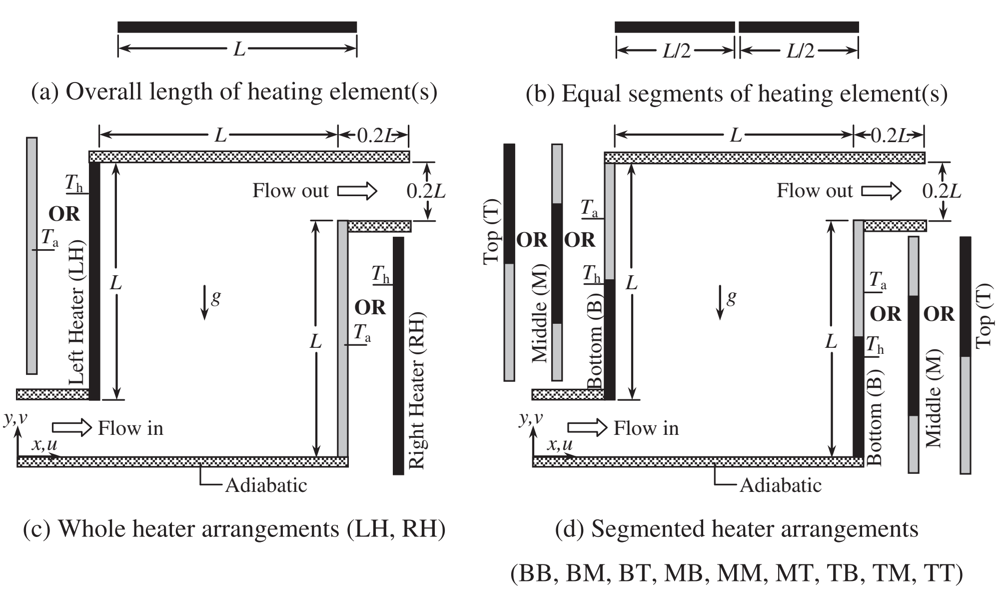

Fig. 1. Schematic arrangement of heating element in a ventilated enclosure.