Mixed Convective Heat Transfer with Surface Radiation in a Vertical Channel in Presence of Heat Spreader

S. K. Mandal, Arnab Deb and Dipak Sen

Abstract Numerical analysis of mixed convection with surface radiation on a vertical channel is conducted. Five protruding heat sources are mounted on the left wall of the channel, and copper heat spreader is attached upon each heat source. Governing equations are solved using SIMPLER algorithm in ANSYS 16.2 software. Results are presented to depict the effects of parameters like heat spreader width ($W_s = W - 2W$), emissivity of heat spreader ($\varepsilon_{sp} = 0.1$–0.9) and Reynolds number (*Re* 250–750) on the rate of heat transfer by fixing emissivity of heat source and substrate. It is found that with increasing spreader width and emissivity, heat transfer performance increases.

Keywords Mixed convection · Surface radiation · Heat spreader

Nomenclature

K	Thermal conductivity
ε	Emissivity
F_{jk}	View factor from jth element to the kth element of an enclosure
J_k	Radiosity of surface k
J_j	Radiosity of surface j
E_k	Emissive power of surface k
	Non-dimensional temperature $\theta = \frac{T - T_o}{\Delta T_{ref}}$,
ΔT_{ref}	Reference temperature difference, $\Delta T_{ref} = \frac{q''wh}{k}$

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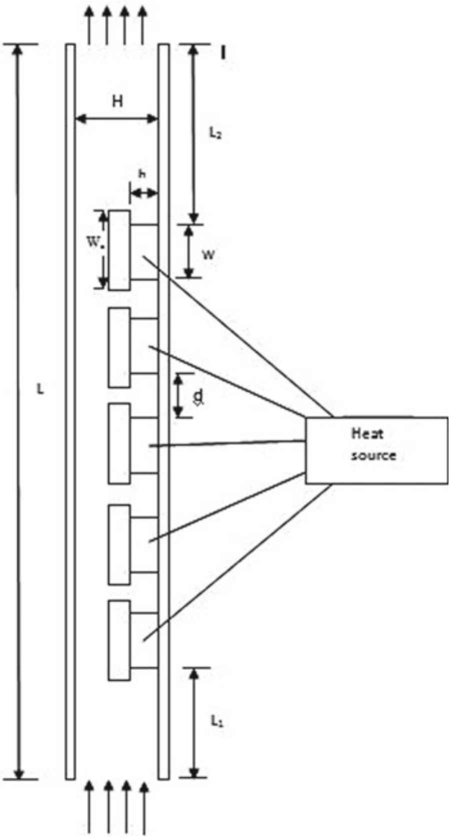
1 Introduction

In enhancing the reliability and to prevent the permanent failure of electronic devices, thermal management plays an important role. Improved heat generation together with different levels of electronic packaging creates a great challenge for the researcher. Each electronic package has heat source with different aspect ratio, which creates some vortex in the flow field. The rate of heat transfer by convection and radiation changes with changing the area of heat transfer surface. So, for efficient cooling, appropriate flow and mechanism of heat transfer must be analyzed, and accordingly, design must be made. As the present work is mixed convection with surface radiation, the significant literature briefly reviewed here. Smith et al. [1] numerically studied the effect of surface radiation with conjugate free convection considering diverse sizes of heat-generating component mounted on a printed circuit board. An analytical study carried out on a vertical cavity by Balaji and Venkateshan [2] on surface radiation with conjugate free convection considering conducting walls and isothermal bottom, whereas Bahlaoui et al. studied in a horizontal cavity on mixed convection with surface radiation. Premachandran and Balaji [3, 4] investigated surface radiation with combined free-forced convection from vertical and also in horizontal channel considering four numbers of protruding heat sources. Fahad et al. [5] analytically investigated the influence of surface radiation of a transparent gas between two asymmetrically heated vertical plates. Flow is considered to be laminar mixed convective and developing. Siddiqa et al. [6] numerically investigated on free convection in a vertically heated wavy surface. Investigation on natural convection with surface radiation in a vertical channel with copper heat source array which simulating electronic package performed both analytically and experimentally by Sarper et al. in 2018 [7]. Heat spreader also can be used to enhance the heat transfer from electronic devices. It also provides mechanical support to the devices to prevent physical damage during testing and handling.

2 Problem Description

The schematic diagram of a rectangular vertical channel with five identical protruding heat sources and rectangular heat spreader pasted upon each heat source is shown in Fig. 1. Five protruding heat sources are located at the right wall of the channel maintaining spacing ‘d’ with successive heat source. Channel has a length ‘L’ and a width ‘H.’ Every heat source has a width ‘w’ and height ‘h.’ Each heat spreader has a width ‘W_s.’ Left face of the first heat source maintains a distance ‘L_1’ from the entrance plane. Right face of the 5th heat source is positioned at a distance ‘L_2’ before the outlet plane. The inlet fluid (air) temperature is assumed to be at 27 °C and non-participating media. Each heat source with volumetric heat generating capacity of 100,000 W/m^3 is chosen in the present case. Fluid properties are supposed to be constant.

Fig. 1 Schematic diagram of the problem



3 Governing Equations and Boundary Conditions

The governing equations for a 2D, steady, incompressible, laminar flow are given as follows:

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

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

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

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{RePr}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

Pressure outlet and velocity inlet boundary conditions are applied at the channel entrance and channel exit, respectively. No-slip boundary conditions are used at all surfaces. Surface-to-surface radiation model used assuming all interior surfaces are to be diffuse, opaque and gray. The surface-to-surface radiation model equations are given below

$$J_k = E_k + \rho k \sum_{j=1}^{n} F_{kj} J_j \tag{5}$$

Coupled boundary condition was used at wall-to-wall and wall-to-fluid boundaries. Emissivity of channel walls considered 0.9. Copper heat spreader emissivity varied from 0.1 to 0.9.

4 Grid Independence Test and Validation

The grid independency test is performed for $Re = 250$, heat generation rate (q_{v}) equal to 1×10^5 W/m^3 considering nonuniform grid throughout the domain. Result of grid independence test is shown in Table no. 1. It is found that the change in non-dimensional maximum temperature is less than 1% when number of nodes changes from 1,36,880 to 1,87,671. So, for present study, 136880 nodes are used.

In order to authenticate the present work, it was compared with the work of Premachandran and Balaji [4] maintaining identical test field and parameter used by them. The difference in result lies in less than 2% which has shown in Table no. 2. The present results show a good agreement with the literature.

Table 1 Grid independence test

S. no.	Nodes	Maximum non-dimensional temperature	Percentage of change (%)
1	69,687	0.16627	–
2	100,941	0.15468	6.9
3	161,265	0.14648	5.3
4	216,279	0.14517	0.89

Table 2 Validation

Emissivity	Reynolds number	$\theta_{\max}$ as per Premachandra and Balaji [4]	$\theta_{\max}$ as per present work
$\varepsilon_{\mathrm{p}} = 0.3$, $\varepsilon_{\mathrm{s}} = 0.55$	250	0.11373	0.11355
	500	0.09612	0.094886
	750	0.085512	0.08434

5 Results and Discussions

The study has been carried out for various heat spreader width (W, 1.5 W and 2 W), emissivity (0.1 to 0.9) in the channel with respect to various Reynolds number (*Re* = 250, 500 and 750) to create sufficient data of non-dimensional temperature (θ). An isotropic conduction was considered in channel walls and heat spreader.

5.1 Streamline and Temperature Contour

The flow field configuration is characterized by using streamline with uniform profile of temperature and uniform velocity of the fluid thrust within the channel. Temperature contour and streamline are shown in Figs. 2 and 3, respectively. Figure 2 shows that temperature of the first heat source is much lower than other as because cold air first came in contact with the first heat source. Maximum temperature arises at penultimate heat source as large circulation beyond last heat source carries away

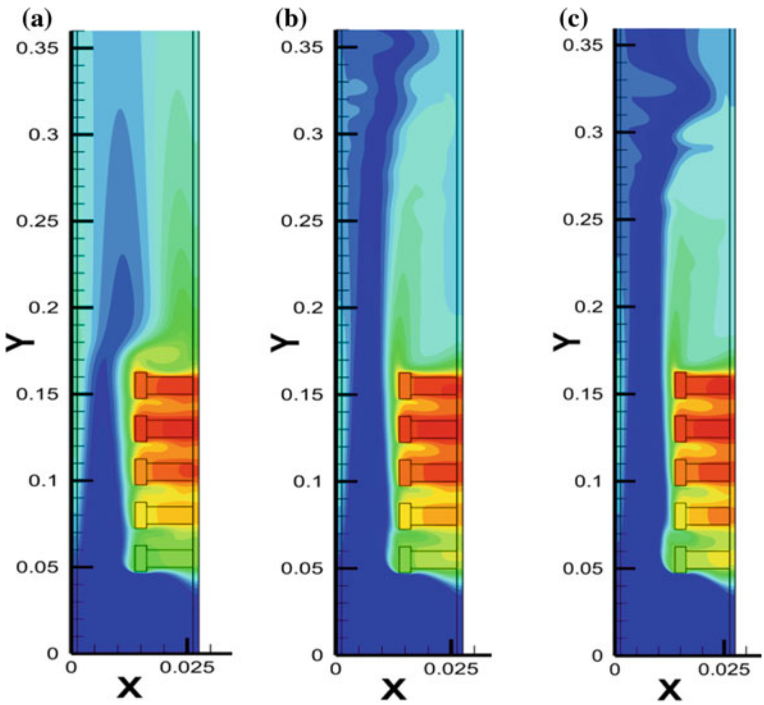


Fig. 2 Temperature contour for $W_s = 1.5$ W, $\varepsilon_c = 0.5$, $\varepsilon_{sp} = 0.5$. **a** $Re = 250$, **b** $Re = 500$ and **c** $Re = 750$

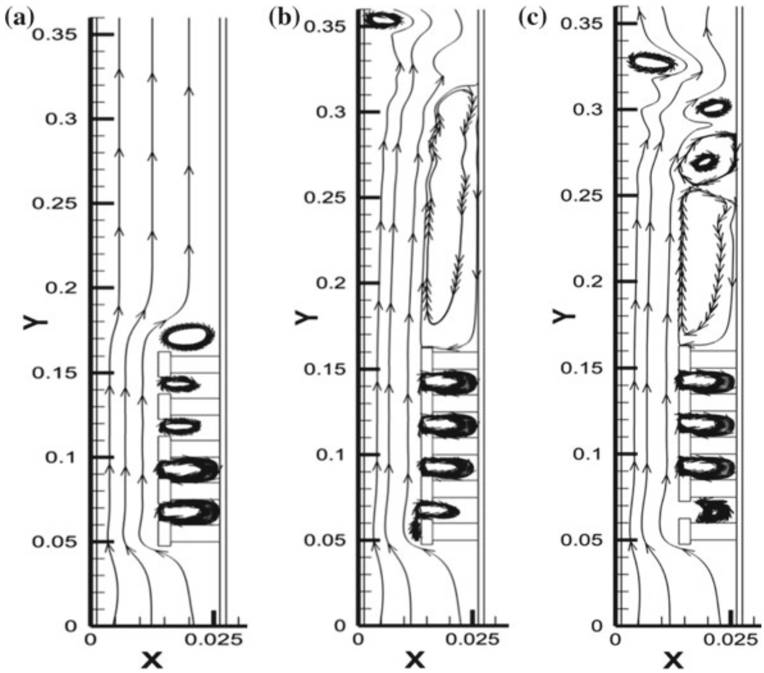


Fig. 3 Streamline for $W_s = 1.5$ W, $\varepsilon_c = 0.5$, $\varepsilon_{sp} = 0.5$. **a** $Re = 250$, **b** $Re = 500$ and **c** $Re = 750$

heat to the core flow as shown in Fig. 3. As Reynolds number increases, circulation strength also increases. As the first heat source temperature is lowest compared to other, radiative heat transfer is insignificant compared to the rest. Heat source mounted on right wall substrate which carries heat by conduction, and its temperature increases which again involved in radiation. Left wall substrate temperature increases due to the radiative interaction with heat source and right wall substrate. A thermal boundary layer was developed over left wall substrate due to radiation. With increasing Reynolds number, thickness of that layer decreases.

5.2 *Influence of Heat Spreader*

Heat spreader creates an additional heat surface area like extended surface. Whenever heat spreader attached with heat source, heat transfer takes place from heat source to heat spreader by conduction, and temperature of heat source decreases. So, it is important to check the influence of heat spreader on overall heat transfer within the channel. Figure 4a shows that, after introducing heat spreader over the heat source, non-dimensional maximum temperature (θ_m) within the channel decreases.

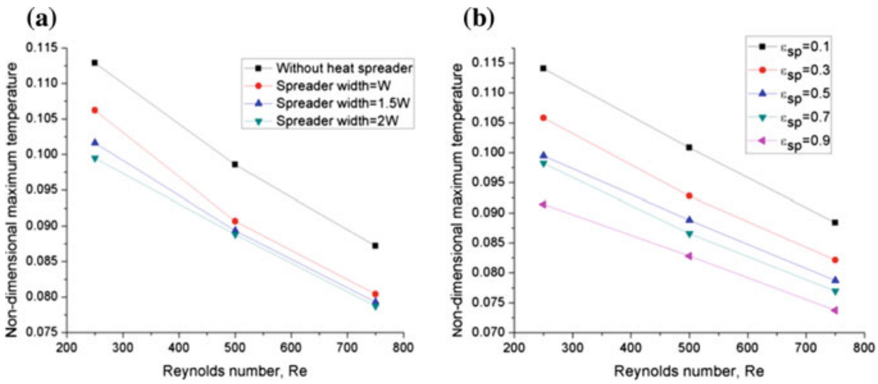


Fig. 4 Graphs **a** variation of non-dimensional maximum temperature with Re for different spreader width at $\varepsilon_c = 0.5$, $\varepsilon_{sp} = 0.5$, **b** variation of non-dimensional maximum temperature with *Re* for different spreader emissivity at $\varepsilon_c = 0.5$

For spreader width of W, 1.5 W and 2 W, θ_m decreases by 6%, 9% and 12%, respectively, at *Re* 250 with comparison to without spreader. Temperature distribution for different spreader width at *Re* 250 is shown in Fig. 5. Figure 4b depicts that when

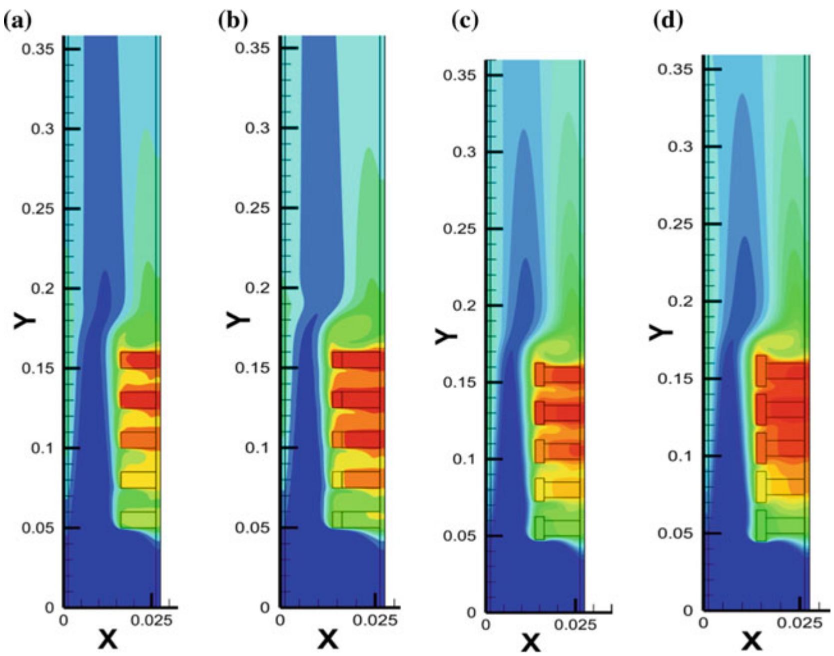


Fig. 5 Temperature distribution at $Re = 250$ for **a** without spreader, **b** $W_s = W$, **c** $W_s = 1.5$ W, **d** $W_s = 2$ W

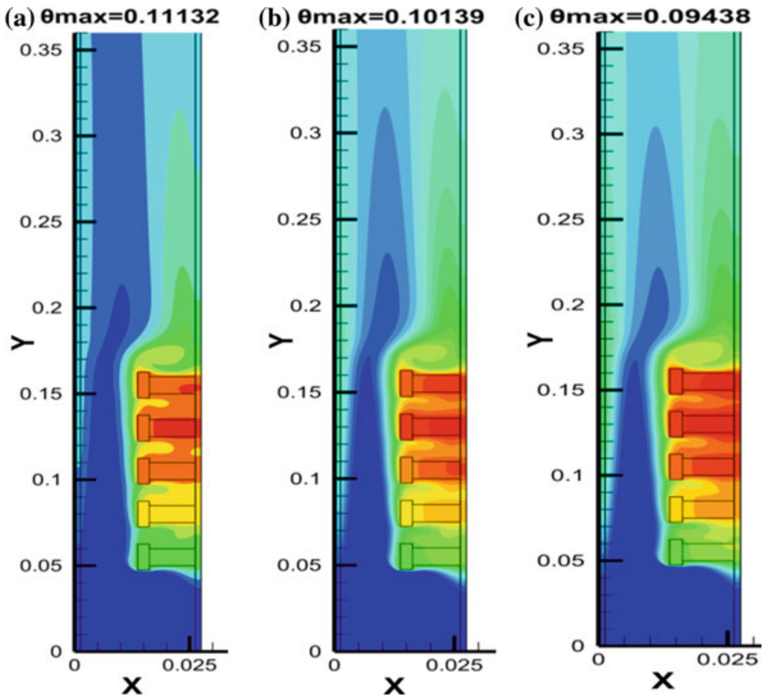


Fig. 6 Temperature distribution for $Re = 250$. **a** $\varepsilon_{sp} = 0.1$, **b** $\varepsilon = 0.5$ and **c** $\varepsilon_{sp} = 0$

heat spreader emissivity, ε_{sp}, varies from 0.1 to 0.9, non-dimensional maximum temperature decreases by 20, 18 and 16.5% at $Re = 250$, 500 and 750, respectively, and as with increasing emissivity, radiative interaction between surfaces increases. Temperature distribution for different emissivity is shown in Fig. 6.

6 Conclusions

Based on numerical study in a vertical channel, the following conclusions are found

- With increasing Reynolds number, maximum non-dimensional temperature decreases, and radiation heat transfer decreases.
- As heat spreader width increases from W to 2 W, non-dimensional maximum temperature decreases by 6–12% in comparison with the value obtained without spreader.
- As the emissivity of the heat spreader increases from 0.1 to 0.9, maximum non-dimensional temperature decreases by 20% at $Re = 250$.

References

1. Smith TF, Beckermnn C, Weber SW (1991) Combined conduction, natural convection, and radiation heat transfer in an electronic chassis. ASME J Electron Packag 113:382–391
2. Balaji C, Venkateshan SP (1995) Combined conduction, convection and radiation in a slot, cavity. Int J Heat Fluid Flow 14:260–267
3. Premachandran B, Balaji C (2011) Conjugate mixed convection with surface radiation from a vertical channel with protruding heat sources. Numer Heat Transfer Part A Appl Int J Comput Methodol 60(2):171–196
4. Premachandran B, Balaji C (2006) ‘Conjugate mixed convection with surface radiation from a horizontal channel with protruding heat sources. Int J Heat Mass Transf 49:3568–3582
5. Al-Amri FG, El-Shaarawi MAI (2012) Mixed convection with surface radiation between two asymmetrically heated vertical parallel plates. Int J Thermal Sci 58:70–78
6. Siddiqa S, Hossain MA, Saha SC (2013) Natural convection flow with surface radiation along a vertical wavy surface. Numer Heat Transfer Part A 64:400–415
7. Sharper B, Saglam M, Aydin O (2018) Experimental and numerical investigation of natural convection in a discretely heated vertical channel: effect of the blockage ratio of the heat sources. Int J Heat Mass Transfer 126:894–910